

# **TESTING AND APPLICATIONS**

# Second Volume Graves/Wysocki, editors (T) STP 1116

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# Insulation Materials: Testing and Applications, 2nd Volume

Ronald S. Graves and Donald C. Wysocki, editors



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The quality of the papers in this publication reflect not only the obvious efforts of the authors and the technical editor(s), but also the work of these peer reviewers. The ASTM Committee on Publications acknowledges with appreciation their dedication and contribution to time and effort on behalf of ASTM.

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### Foreword

This publication, Insulation Materials: Testing and Applications, 2nd Volume, contains papers presented at the symposium of the same name, held in Gatlinburg, TN on 10–12 Oct. 1991. The symposium was sponsored by ASTM Committee C-16 on Thermal Insulation. Ronald S. Graves of Oak Ridge National Laboratory, Oak Ridge, TN and Donald C. Wysocki of the Mobay Corporation, Pittsburgh, PA, presided as symposium co-chairmen and are editors of the resulting publication.

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#### Overview

ASTM Committee C-16 on Thermal Insulation enters its second halfcentury with the 1991 Symposium on Insulation Materials: Testing and Applications. STP 1116 contains the proceedings of the Symposium that brought together the latest information on thermal insulations, test methods applicable to thermal insulations, and applications. In keeping with its scope, ASTM Committee C-16 has held many programs to present the latest information on insulation materials and their test methods. A tangible result of these programs has been the publication of over a dozen ASTM Special Technical Publications covering the technology of thermal insulation.

This STP reflects the interest in this decade of conserving energy and protecting the environment. The Symposium and the resulting STP clearly demonstrate the Committee's objective of providing timely information on the testing, characterization, use, and development of thermal insulations. In the 1970s and 1980s, interest and activity in thermal insulation was spurred by the growing recognition of the need for energy conservation. The value of thermal insulation in combating the impact of rising energy costs and in sustaining economic growth was widely recognized. While these economic factors are still important, society's attention today is shifting toward the effects which the production and consumption of energy have on our environment. The role of thermal insulation in mitigating these effects until new, more environmentally sound energy technologies are developed becomes even more important.

International agreements to phase-out the use of CFCs has impacted the field of thermal insulations. This impact is demonstrated by a significant fraction of the Symposium and the STP devoted to the long-term performance for cellular plastics and foamboard insulations. Fundamental questions about aging, stability, testing, and overall acceptability of a new generation of insulating foams are addressed. Increasing interest in high-R innovative insulations is demonstrated in this STP by reports of research on evacuated panel insulations, non-CFC high performance insulations; aerogels, and thinwall ceramic spheres.

As the need for energy conservation increases, so does the need for environmentally sound insulation materials, the need for better understanding of the factors which affect insulation performance, and the need for better test methodology.

Research in the area of thermal insulations and the testing of products has matured. Heat transfer mechanisms are being given more attention than in the past, and the effects of moisture and positioning of insulation in the building envelope are being studied. A wide range of factors that affect performance were presented and discussed at the 1991 Symposium and presented in this STP. An understanding of mechanisms and performance factors is crucial to the design and use of thermal insulations to obtain optimum

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results. Control of thermal radiation and a quantitative evaluation of convective effects in relatively low-density insulations are examples of the detailed research that was presented.

The manufacture, testing, and use of insulations in the building industry has high priority with C 16 and received significant attention at the Symposium. Economic evaluations and regulation activities are becoming more sophisticated. Inspections of materials in actual use configurations continues to be an important issue.

The 1991 Symposium was international in scope with participation from seven nations including the United States. The papers presented represented input from the industrial sector, the National Laboratories in the United States, Canada, and Australia, and Universities in the United States, the Netherlands, and Portugal. The STP will be useful to a wide range of professionals dealing with thermal insulations by presenting research on currently used materials and thermal insulations under development.

The significance of the STP is directly related to the efforts of the contributing authors, the technical reviewers, and session chairpersons. Session chairpersons and reviewers are identified on the following page and appreciation is expressed here. The editors also wish to acknowledge the contributions of the Steering Committee: R. L. Baumgardner, W. M. Edmunds, and D. L. McElroy.

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Reflectives/Radiant Barriers/Radiation Control Coatings Robert W. Anderson, David W. Yarbrough, Ronald S. Graves, and Robert L. Wendt

PRELIMINARY ASSESSMENT OF RADIATION CONTROL COATINGS FOR BUILDINGS

**REFERENCE:** Anderson, R. W., Yarbrough, D. W., Graves, R. S., and Wendt, R. L., "Preliminary Assessment of Radiation Control Coatings for Buildings," <u>Insulation Materials: Testing and</u> <u>Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Radiation Control Coatings (RCCs) applied to external building surfaces can reflect about 85 percent of the solar heating from the surfaces of buildings. Since in warm climates, solar heating is the primary source of heat gain through walls and roofs, RCC technology represents an alternative or adjunct to conventional thermal control methods (e.g. thermal insulation) for opaque building components.

The primary objectives of this project were to: (1) obtain solar and infrared reflectance data for representative RCC products, (2) evaluate test methods for measurement of the radiative properties of RCCs; (3) calculate the changes in heat flow attributed to RCCs in flat roof applications in several geographic locations; and (4)compare field tests and calculated thermal performance of an RCC in a flat roof configuration.

Data are presented for the reflectance properties of five commercially available RCC products as determined by several test methods. The potential energy benefits of RCCs are presented for flat roofs in both warm and cold climates.

KEYWORDS: radiation control coatings, building heat loads, building cooling loads, thermal insulation, solar reflectance, emittance, infrared reflectance

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#### INTRODUCTION

Coating materials that have the combined properties of high solar reflectance and high long wavelength emittance can reduce heat gains through roof and wall sections. Such coatings, defined as Radiation Control Coatings (RCCs), have potential as an energy conservation material when applied to the exterior of buildings in locations where cooling loads predominate. Under certain applications, RCCs represent an alternative or adjunct to conventional thermal insulations for thermal control. Although the potential benefits of RCCs have been discussed in the literature [1-3], there are only limited data regarding the properties of RCC materials and their thermal performance in building applications.

Under this project, integrated research was conducted to first characterize RCC products; secondly, to calculate the thermal performances of RCCs in different climatic areas by means of computer modeling; and thirdly, to measure the thermal performance of a RCC product under field conditions. The primary objectives of this project were:

1. To obtain valid radiative property data for typical RCCs for the evaluation of the thermal performance of RCCs in building applications.

2. To assess the energy conservation potential of RCCs in building applications.

3. To increase the awareness of the building industry of the potential applications of RCCs.

This paper summarizes the major results of the project. More detailed data will be presented in the final project report.

The type of RCC products selected for testing were limited to commercially available white elastomeric coatings formulated with acrylic latex resin binders. It should be noted that in addition to this class of RCC products, other products formulated with different resin bases, such as epoxies or polyurethanes, are also commercially available. Since the radiative properties of these other products were not tested, characteristics of such other products cannot be made from the test data described in this project.

#### EXPERIMENTAL APPROACH

The thermal performance of RCCs is dependent on their radiative properties. Only limited radiative property data are available from manufacturers. Although there are some standardized testing procedures available for measuring radiative properties, such testing methods have not been evaluated for RCC materials.

To meet the objectives of the project, a three step approach was taken to evaluate the potential thermal performance of RCCs in building applications:

1. Characterize the application and radiative properties of five commercially available RCC products.

2. Calculate the thermal performance of RCCs in different flat roof configurations and in different climatic areas with the BLAST computer simulation program.

3. Measure the thermal performance of one RCC product in a flat roof configuration under limited summer and winter conditions in the Oak Ridge National Laboratory (ORNL) Roof Testing Research Apparatus (RTRA).

Five commercially available RCC products were obtained for testing; three products were obtained directly from the manufacturers and two were purchased from commercial retail stores. According to product label or other literature, each product was white in color, was an "elastomeric acrylic" coating material, had a water-based formulation, and made claims as to the product's ability to reflect solar heat or otherwise conserve energy in buildings.

Testing of solar and infrared reflectance properties of the products was performed in accordance to established American Society for Testing and Materials (ASTM) testing procedures whenever possible. For the preparation of test samples for measurement of radiative properties, material was brush-applied at three thicknesses on both aluminum foil and black rubber roofing membrane (ethylene propylene diene monomer or EPDM) substrates. The aluminum substrate had high reflectance and low emittance and the black rubber had low reflectance and high emittance; these differences in radiative properties assisted in the interpretation of the test data.

Calculations of the thermal performance of RCCs were performed with the aid of BLAST, a computer simulation program developed by the U.S. Army Corps of Engineers [4]. BLAST allows the input of detailed construction parameters and calculates design day and annual heat loads based on hourly weather data. BLAST was used to evaluate the thermal performances of RCCs in different flat roof construction configurations and in different climatic areas. BLAST was also used for the comparison of the calculated thermal performance of roof sections to values measured in the RTRA for limited summer and winter field testing. Results of the summer test cycle are summarized in this paper.

For the RTRA tests, the heat fluxes and surface temperatures were compared for side-by-side flat roof sections of the following construction: Section 1 - Black 1.1 mm EPDM over fir plywood 3.8 cm thick; and Section 2 - Same as Section 1, except that the black EPDM was coated with 0.9 mm of a RCC. Detailed description of the RTRA test configuration is given in [5].

#### CHARACTERIZATION OF RCC PRODUCTS

The objectives of the characterization of commercially available products were: to obtain valid radiative property data, to identify a representative product for field testing, and to evaluate different methods for measurement of radiative properties. Characterization of five commercial RCC products included the following properties:

- application characteristics,

- effect of coating thickness on apparent solar and infrared reflectances,

- solar and infrared reflectances of the RCC products,

- effect of angle of incidence on the solar reflectance, and

- surface gloss.

Application characteristics shown in Table 1 are based on visual observations during preparation of samples of the five RCC products.

Product code	Bond to rubber	Flexibility (bending)	Surface	Texture
S	good	good	ripple	smooth
Н	good	poor	smooth	granular
Ε	good	poor	smooth	granular
Р	good	good	ripple	smooth
ĸ	good	good	ripple	smooth

TABLE 1 -- RCC coating characteristics.

The effect of thickness on the apparent solar reflectance of RCC product "S" was tested at coating thicknesses from 0.08 to 1.14 mm using a solar spectrum reflectometer [6]. As summarized in Table 2, coating thicknesses in excess of 0.5 mm exhibited the maximum reflectance values.

Additional tests of the five RCC products applied to both bright aluminum and black rubber substrates, demonstrated that coating thicknesses in excess of 0.5 mm resulted in maximum solar reflectance values and thicknesses in excess of 0.35 mm resulted in maximum infrared reflectance values (infrared reflectance was not tested for coating thicknesses less than 0.35 mm).

TABLE 2 -- Effect of thickness on solar reflectance.

Coating thickness (mm)	Reflectance (TTU <sup>a</sup> )
0.08	0.59
0.13	0.68
0.15	0.68
0.19	0.72
0.36	0.77
0.51	0.80
0.66	0.79
0.89	0.80
1.14	0.80
ATopponento Technological Independent	

Tennessee Technological University, Cookeville, TN.

Solar and infrared reflectance data were obtained for the five RCC products plus two samples of single-ply rubber roofing membrane materials -- a black rubber (EPDM) membrane manufactured by Firestone and a white rubber membrane manufactured by Goodyear. The following tests were conducted:

(a) solar reflectance in accordance with ASTM E-903; tests conducted by DSET Laboratories, Inc., Phoenix, AZ;

(b) solar reflectance in accordance with ASTM E-1175; tests conducted by DSET Laboratories, Inc.;

(c) solar reflectance as measured with a solar reflectometer manufactured by D&S, Inc.; tests conducted by Tennessee Technological University (TTU), Cookeville, TN;

(d) infrared reflectance in accordance with ASTM E-408, method A; tests conducted by DSET Laboratories, Inc.; and

(e) infrared reflectance in accordance with ASTM E-408, method B; tests conducted by TTU.

The solar and infrared reflectance data, at near normal angle of incidence, are summarized in Tables 3 and 4, respectively, and solar data for different angles of incidence are given in Table 5.

Product ASTM E-903 code DSET <sup>a</sup>		ASTM E-903 DSET <sup>a</sup> TTU	
S	0.766	0.797	- 4
Н	0.723	0.815	-13
Ε	0.770	0.787	- 2
P	0.816	0.817	0
K	0.789	0.827	- 5
Black rubber	0.067	0.071	- 6
White rubber	0.769	0.783	-2

TABLE 3 -- Solar reflectance (normal).

<sup>a</sup>DSET Laboratories, Phoenix, AZ

<sup>b</sup>(DSET - TTU) x 100/DSET

#### TABLE 4 -- Infrared reflectance (normal).

Product code	ASTM E-408, A DSET	ASTM E-408, B TTU
S	0.10	0.15
н	0.09	0.11
Е	0.10	0.19
Р	0.11	0.14/0.20
К	0.10	0.18
Black rubber	0.15	0.17
White rubber	0.13	0.20
Black rubber White rubber	0.15 0.13	0.17 0.20

Product	ASTM E-1175, 7°	ASTM E-1175, 45°	%
code	DSET	DSET	difference
S	0.85	0.82	3 3
H	0.86	0.84	

TABLE 5 -- Solar reflectance at 7° and 45° angles of incidence.

Measurements of surface gloss reported in Table 6 were made for each of the RCC products and the black and white rubber membranes. The purpose of these tests was to assess the amount of specular reflection that may occur from the surfaces of the products.

Product code	ASTM D-523, at 60° DSET
S	4-5
Н	2
E	3
Р	6 - 8
K	4
Black rubber	6
White rubber	4
Aluminum foil	>100

TABLE 6 -- Surface gloss (gloss units).

There were observable application differences between the five RCC products. Whereas all products exhibited good adhesion to rubber and aluminum substrates, the products fell into two groups when considering flexibility and surface qualities. Products S, P and K were flexible and did not crack under stretching or bending conditions but products E and H easily cracked when bent and stretched. Likewise, products S, P and K retained brush marks from application but otherwise had smooth surfaces similar to a semi-gloss paint. In contrast, products E and H showed no brush marks from application (e.g. was "self-leveling") and had granular surface textures.

Coating thicknesses in excess of 0.5 mm were necessary to achieve maximum solar and infrared reflectance values; thinner coatings allow measurable amounts of solar transmission through the coating. All RCC samples and the white rubber membrane showed similar solar and infrared reflectances and gloss values notwithstanding differences in surface conditions and texture. Small differences in the measured reflectance values are probably related to differences in composition and surface conditions.

Differences in measured solar and infrared reflectance values were obtained from the different testing methods. The causes for such differences need to be evaluated and standardized testing procedures established. Changes in the angle of incidence from  $7^{\circ}$  to  $45^{\circ}$  caused a 3% change in the solar reflectance values in the two RCC products. The data indicate that solar reflectances of RCCs are insensitive to angle of solar incidence. RCC products exhibited low gloss values, thereby indicating that the specular reflection component is low and that almost all reflected energy is diffuse.

#### COMPUTER SIMULATION OF RCC THERMAL PERFORMANCE

The objectives of the simulations with BLAST were to: assess the maximum effects due to RCC applications by comparing the thermal performance of RCCs with black roofs, e.g. EPDM rubber membranes; assess the effects due to RCC applications to roofs in different climatic areas, and assess the effects of RCCs applied in conjunction with thermal insulation.

For the simulations, flat roofs were identified having the following construction layers:

Top layer - black rubber membrane or white RCC; the solar reflectance values for the black and white materials were 0.10 and 0.85 respectively. The infrared absorbance values were 0.9 for both materials.

Second layer - thermal insulation; three levels of R-value. Third layer - 1.9 cm thick plywood deck.

The thermal performances of the roofs were calculated for cities in both warm and cold climates. For all simulations, a dead-band control system was used for temperature control; the heating and cooling setpoints were 18.3°C and 25.5°C respectively. All reported loads represent only sensible loads, since no air infiltration was assumed for the roof sections, there were no latent loads. The listed Rvalues represent the total surface-to-surface R-values for the roofs.

The calculated thermal performances for the roofs in different climatic areas are summarized in Tables 7-10.

Summer design day profiles of cooling loads, roof surface temperatures and ceiling temperatures for the roofs located in Las Vegas, Nevada, are shown in Fig. 1 for the white rubber roof and Fig. 2 for the black RCC roof.

The simulation results show that RCC roofs had lower cooling loads than the corresponding black roofs having similar levels of thermal insulation; however, the magnitude of reductions of cooling loads were greater for the warm climates (see Table 7). Compared to black roofs with low levels of insulation, the reduction of cooling loads by the RCCs is "equivalent" in effect to thermal insulation levels in the R-value range of 2.3-4.4 m<sup>2</sup>K/W. However, the equivalency of RCCs to thermal insulation varied for different climates.

RCCs can effectively reduce the annual cooling loads of roofs having a wide range of level of thermal insulation. The benefits from reductions of cooling loads are balanced by increases in heating loads; the greatest net benefits are to be gained in climates where



ROOF SURFACE TEMPERATURES



CEILING TEMPERATURES CEILING TENP., DC ġ. zi HOUR а 

FIG. 1 Roof with white RCC surface, Las Vegas summer design day.



FIG. 2 Roof with black rubber surface, Las Vegas summer design day.

City	Black rubber		White RCC		;	
	R-0.9	R-2.3	R-4.4	R-0.9	R-2.3	R-4.4
Las Vegas						
heating	118.4	60.0	41.0	175.6	89.9	61.3
cooling	289.6	115.2	52.4	98.5	34.3	12.1
Tampa						
heating	48.3	28.6	24.2	70.0	42.6	36.9
cooling	206.2	75.9	31.0	62.2	16.4	3.4
Fort Worth						
heating	141.6	72.2	49.9	185.6	95.4	65.9
cooling	179.0	67.3	28.3	62.6	19.5	5,6
Minneapolis						
heating	402.3	190.3	118.0	468.3	222.2	137.9
cooling	77.2	28.0	11.2	16.3	4.2	0.8

TABLE 7 -- Annual heating and cooling loads, 1000  $kJ/m^2$  (R-values in  $m^2\cdot K/watt)$ 

TABLE 8 -- Maximum hourly demands,  $kJ/m^2$  (R-values in  $m^2\cdot K/watt)$ 

City	В	lack rubb	er	<u> </u>	White RCC	;
	R-0.9	R-2.3	R-4.4	R-0.9	R-2.3	R-4.4
Las Vegas			_	[		
heating	88.9	41.0	23.9	88.9	41.0	23.9
cooling	242.8	106.0	54.7	75.2	30.8	14.8
Tampa				Į		
heating	66.1	30.8	18.2	66.1	30.8	18.2
cooling	159.6	68.4	34.2	43.3	16.0	6.8
Fort Worth				}		
heating	104.9	47.9	27.4	104.9	41.9	27.4
cooling	144.8	60.4	30.8	55.9	21.7	9.1
Minneapolis						
heating	180.1	79.8	45.6	180.1	79.8	45.6
cooling	130.0	51.3	26.2	39.9	14.8	5.7

City	Max ODB <sup>a</sup>	B1	ack rubb	er		White RC	c
	°C	R-0.9	R-2.3	R-4.4	R-0.9	R-2.3	R-4.4
Las Vegas	42	84	- 85	86	46	47	47
Tampa	33	69	70	70	38	38	38
Minneapolis	32	50	51	51	33	34	34

#### TABLE 9 -- Maximum roof surface temperature, °C Summer design day, July 21 $(R-values in m^2 \cdot K/watt)$

#### TABLE 10 -- Maximum ceiling temperature, °C Summer design day, July 21 (R-values in $m^2 \cdot K/watt$ )

City	B	lack rubb	er		White RCC	
	R-0.9	R-2.3	R-4.4	R-0.9	R-2.3	R-4.4
Las Vegas	33	27	25	26	24	23
Tampa	30	26	24	25	23	22
Minneapolis	28	25	24	24	23	22

cooling loads are predominate. For each climate, optimized levels of thermal insulation in conjunction with RCCs need to be determined based on relative costs for heating and cooling energy sources and/or energy use policies.

For all levels of thermal insulation, the application of RCC significantly lowers maximum hourly cooling demands for the black roofs in all climates. In contrast, the RCC roofs did not increase the maximum hourly heating demands compared to black roofs since maximum heating demands occur during night time hours (see Table 8). Hence, RCC application reduces cooling equipment capacity requirements without corresponding increases in heating equipment capacity requirements.

For black roofs, the maximum roof surface temperatures calculated for summer design days ranged from 51°C for Minneapolis to 86°C in Las Vegas; the higher levels of thermal insulation increased the maximum surface temperatures only slightly (see Table 9). In contrast, for RCC roofs, maximum roof surface temperatures for summer design days ranged from 34°C for Minneapolis to 47°C for Las Vegas. The maximum roof surface temperatures for RCC roofs were 3-5°C above the maximum outside air temperatures for all locations. The high surface temperatures on black roofs indicate maximum temperatures of thermal

insulation placed below the roof membrane can reach similar temperatures. Also, average temperatures within the thermal insulation can reach 55°C or more. High insulation temperatures have several important energy related implications:

(1) High roof temperatures can exceed some insulation manufacturers' recommended service conditions for their products, and premature loss of thermal and mechanical performance can be expected.

(2) Building thermal design is generally based on insulation R-values determined at an average temperature of  $24^{\circ}C$ . Since the R-values of all insulation materials decrease with increasing temperatures, the designed thermal performance overstates the actual summer thermal performance, that is, greater than expected heat flow will be experienced. It should be noted that building thermal insulation R-value data are generally not available for average temperatures other than  $24^{\circ}C$ .

(3) All computerized heat load calculation methods, including BLAST, assume a constant R-value for insulation materials, typically the R-value determined at 24°C. Consequently, calculated cooling loads are understated.

The RCC roofs had lower ceiling temperatures during summer periods than the black roofs of corresponding thermal insulation levels (see Table 10). High ceiling temperatures cause radiation heating of occupants and, for a given inside setpoint temperature, the occupants tend to lower the setpoint temperature; the result is greater cooling energy consumption.

#### PRELIMINARY FIELD TESTING OF THE THERMAL PERFORMANCE OF RCCs

Preliminary field tests of the thermal performance of RCCs were conducted on the ORNL RTRA from July 24, 1990 to October 15, 1990; additional testing was planned through the remainder of the year and beyond to obtain cold weather data and long term characteristics of the RCC material.

The objectives of this test were: to demonstrate the thermal performance of RCCs during summer conditions, to compare the measured thermal performance of RCCs to those calculated by the BLAST computer simulation program and, to assess changes in solar reflectance and mechanical properties due to solar and atmospheric exposure.

The RTRA is a small free-standing building with instrumented roof sections exposed to full solar and atmospheric conditions with the inside surface facing a climatically controlled space. The roof sections were instrumented to continuously measure inside and outside surface temperatures and heat flux through the roof sections. The construction and operation of the RTRA is described in Ref. [5].

Two test sections, each 1.2 m by 1.2 m, were used for testing. Each section consisted of three sheets of 1.27 cm thick fir plywood covered on the top surface with a 0.11 cm thick black EPDM single-ply membrane. The top surface of one section was coated with 0.11 cm of a white RCC product (product code S). The surface-to-surface R-values of the test sections were 0.40  $m^2K/W$ . The air temperature at the

underside of the test sections was controlled at a nominal  $24^{\circ}$ C, however there were uncontrolled temperature fluctuations of about  $\pm 3^{\circ}$ C.

A preliminary comparison was made of the measured thermal performance of the two test sections with those calculated by the BLAST program. The test day of August 18, 1990, was selected as being typical of a clear summer day. For the computations, hourly outside air temperatures for the test day were modeled using the minimum and maximum day temperatures recorded by the ORNL weather station. The solar heat flux was determined by the BLAST program for the Knoxville, Tennessee location using a clearness factor of 1.0. The inside air temperature was set at 24°C.

The measured and calculated outside air temperature profiles for August 18, are shown in Fig. 3. The measured and calculated hourly heat loads, roof surface temperatures and ceiling temperatures for the two test sections are shown in Figs. 4, 5, and 6 respectively.

Both the measured and calculated results show that the application of RCCs to black roof sections can significantly reduce the heat flow through the roofs during summer conditions. The measured heat flows however, were greater than those calculated. Although there was good agreement between the maximum measured and calculated black roof surface temperatures, the calculated ceiling temperatures were higher than those measured. Causes for the differences in the measured and calculated heat flow and ceiling temperatures are as yet unknown; possible causes for the differences are:

- The solar reflectance value of the RCC on the RTRA roof section was lower than it's original value due to three weeks of atmospheric exposure; a lower solar reflectance would increase the solar heating.

- The R-value of the wood substrate in the RTRA decreased when exposed to high temperatures and lower R-values would increase heat flow; a constant R-value was used in the BLAST model.

- The air temperatures inside the RTRA fluctuated during the test period and the air was moved at high velocities; the inside air boundary conditions of the RTRA were variable and could not be modeled by BLAST.

- The radiative properties of the RTRA internal surfaces were assumed; differences between actual RTRA values and those assumed for the BLAST model could affect the respective heat flow and ceiling temperature values.

- Although the algorithms used in BLAST are generally considered to be accurate, there has not been any experimental verification of the calculated results at this level of detail.

#### SUMMARY

The commercially available RCC products tested have solar reflectances to 0.85 and infrared reflectances of 0.1 (emittances of 0.9). The high solar reflectance of RCCs is the key property for reducing solar heat loads in buildings. The low infrared reflectances of RCCs indicate they have little value for controlling radiation heat transfer within buildings. There are differences in the mechanical and



Fig 3. Measured and BLAST outside air temperatures, August 18.



Fig. 4 Measured (RTRA) and calculated (BLAST) cooling loads for RTRA roof configuration, August 18.

HOURLY COOLING LOADS, kJ/SQ METER AUGUST 18, 1998



Fig. 5 Measured (RTRA) and calculated (BLAST) roof surface temperatures for RTRA roof configuration, August 18.



Fig. 6 Measured (RTRA) and calculated (BLAST) ceiling temperatures for RTRA roof configuration, August 18.

surface properties of commercially available RCC products that could affect the long term performance of RCCs in field applications.

As compared to black flat roofs, the application of RCCs significantly reduces both the annual and peak cooling loads for roofs with insulation level up to at least  $6.1 \text{ kJ/m}^2$ . The reductions in annual cooling loads are offset to varying degrees by increases in heating loads depending on climatic conditions; the greatest benefits of RCC application are in areas where cooling loads are predominate. Whereas the application of RCCs does reduce the hourly cooling demand requirements, its application has no effect on the hourly heating demand requirements.

The data indicate that for each climatic area, there is an optimum combination of RCC and thermal insulation that will provide minimum combined heating and cooling loads or total energy costs.

Compared to black roofs, RCCs are effective for reducing summer roof surface temperatures and temperatures of insulation placed below roofing membranes. Reduced roof and insulation temperatures provide additional benefits, such as increased roof life and improved thermal performance of the insulation.

Compared to black roofs, RCCs are effective for reducing summer ceiling temperatures, especially for roofs having low levels of thermal insulation. Lower ceiling temperatures reduces the levels of radiation heating of interior building space.

Whereas the measured and calculated summer roof temperatures were in good agreement, differences in summer heat flow and ceiling temperatures were found. Additional research is needed to identify and correct the causes of the differences between the measured and calculated results. Also, computer simulation algorithms are needed that are able to adjust the R-values of thermal insulation and other building materials in accordance with the temperatures experienced.

There is a need for evaluation of the effects of extended solar and atmospheric exposure on the radiative and mechanical properties and thermal performance of RCCs.

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#### Andre O. Desjarlais and David W. Yarbrough

### PREDICTION OF THE THERMAL PERFORMANCE OF SINGLE AND MULTI-AIRSPACE REFLECTIVE INSULATION MATERIALS

REFERENCE: Desjarlais, A. O. and Yarbrough, D. W., "Prediction of the Thermal Performance of Single and Multi-Airspace Reflective Insulation Materials," <u>Insulation Materials: Testing and</u> <u>Applications, 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A recent test program to determine the thermal performance of reflective thermal insulation materials and systems was carried out on behalf of the Department of Energy. This program indicated that the thermal performance of multi-airspace reflective cavities (multi-layer reflective insulations) could not be accurately predicted by a onedimensional analysis applied in combination with the thermal performance data for single airspaces.

A new correlation has been developed in an effort to improve the accuracy of predicted Rvalues for multi-airspace reflective systems. This correlation is described and compared to the thermal test data generated by the recent test program.

Generally good agreement between calculated and measured R-Values is achieved for single airspaces. However, calculated values for multiple airspaces are up to 60 percent greater than the measured values. This disagreement indicates that the commercial multifoil product

Mr. Desjarlais is the Manager of Thermal Insulation Evaluations at Holometrix, Inc., 99 Erie Street, Cambridge, MA 02139, and Dr. Yarbrough is the Chairman of the Chemical Engineering Department at Tennessee Technological University, Cookeville, TN 38505. that was tested may not achieve the idealized one-dimensional heat flow that is true of the correlation.

KEYWORDS: reflective insulations, R-Value, multi-airspace cavities, thermal performance

#### INTRODUCTION

A reflective insulation is defined as a thermal insulation consisting of one or more low emittance surfaces bounding one or more enclosed airspaces. Unlike mass insulations, the thermal resistance of a reflective insulation is not a material property which can be defined as the ratio of its thickness and apparent thermal conductivity. Since the total heat transfer through a reflective insulation can include a convective component, factors such as heat flow direction and temperature difference have a major impact on its thermal resistance.

The American Society for Testing in Materials, through the efforts of Subcommittee C16.21 on Reflective Insulations, has been developing a concensus standard specification for reflective insulations. This standard specification includes guidelines for testing these materials for thermal resistance using ASTM C 236, Standard Test Method for "Steady-State Thermal Performance of Building Assemblies by Means of a Guarded Hot Box.<sup>[1]</sup>"

To experimentally verify these proposed guidelines, Oak Ridge National Laboratory contracted the Thermatest Division of Holometrix through the sponsorship of the Department of Energy Building Thermal Envelope Systems and Materials (BTSEM) Program to perform a series of guarded hot box experiments on several test panels containing reflective insulation materials. These findings have been previously reported<sup>[2]</sup>. The test panel results were treated to extract the cavity (or reflective insulation material) thermal resistance and these values were compared to the data listings in the ASHRAE Handbook of Fundamentals<sup>[3]</sup>. When multi-airspace cavities were analyzed, the ASHRAE data for the individual cavities was determined and summed.

The test program demonstrated that there was excellent agreement between the test results and the listings in the ASHRAE Handbook of Fundamentals for most single airspace cavities. However, multi-airspace cavities did not perform in a manner that could be predicted by summing nominally equivalent single airspace data. The literature significantly overestimated the cavity thermal resistance of multi-airspace cavities. This paper describes a FORTRAN program that has been developed to improve the calculation of the thermal resistance of single and multi-airspace reflective cavities. Results from this program can be obtained as a function of the number of airspaces, heat flow direction, effective airspace emittance, and temperature difference. The numerical procedure is described and compared to the previously referenced experimental and analytical findings.

DESCRIPTION OF NEW CORRELATION TO CALCULATE THE R-VALUE OF REFLECTIVE AIRSPACES

The R-value calculation described in this section is based on Equations (1-4) and the experimental data of Robinson and Powell<sup>[4,5,6]</sup>. The equations used to quantify heat transfer across an air space by conductionconvection are low-order polynomials for the dimensionless Nusselt Number (Nu) in terms of the dimensionless Grashoff Number (Gr). The expressions become linear as Gr increases thus making limited extrapolation a possibility. The previously published correlations <sup>[7]</sup> were polynomials that should not be extrapolated to large thicknesses or temperature differences. Refer to the Table of nomenclature for the definitions of these symbols.

$$R = (E * h_r + h_c)^{-1}$$
(1)

 $E = (1/\epsilon_1 + 1/\epsilon_2 - 1)^{-1}$  (2)

$$h_r = 0.00686 \left[ \left( \overline{T} + 460 \right) / 100 \right]^3$$
 (3)

$$h_{c} = f (L, \overline{T}, \Delta T, heat flow direction)$$
 (4)

A previously published polynomial for  $h_c^{[7]}$  describes the experimental data but is strictly limited to the domain of the data. The present correlation gives the Nusselt Number (Nu) in terms of the Grashoff Number (Gr) such that a smooth transition exists for small temperature differences or airspace width. The correlation is represented by Equations (5) and (6).

$$log_{10} (Nu) = a + b log_{10} (Gr) + c log_{10} (Gr)^{2} (5)$$

$$(Gr)_{1} \leq (Gr) \leq (Gr)_{2}$$

$$log_{10} (Nu) = a_{2} [log_{10} (Gr) - log_{10} (Gr)_{0}]^{2} + a_{3} [log_{10} (Gr) - log_{10} (Gr)_{0}]^{3} (6)$$

$$(Gr)_{0} \leq (Gr) < (Gr)_{1}$$

The coefficients a, b, and c in Equation (5) were

obtained by applying the Method of Least Squares to the data set that forms the basis of the tables for reflective airspace R-values in the ASHRAE Handbook of Fundamentals<sup>[3]</sup>. Equation (6) is used to provide a smooth transition to Nu = 1.0 at Gr =  $(Gr)_0$ . The coefficients (5) and (6) were chosen to make Equations (5) and (6) intersect and have the same slope at  $(Gr)_1$ . Equation (6) also gives Nu = 1.0 at  $(Gr)_0$ . The limiting value  $(Gr)_1$  is the point at which Equation (5) begins to deviate from the experimental data. The combination of Equations (5) and (6) provides a smooth curve for Nu.

 $log_{10}(Nu) = Eq (5) log_{10}(Gr)_{1} \le log_{10}(Gr) \le log_{10}(Gr)_{2}$ = Eq (6) log\_{10}(Gr)\_{0} \le log\_{10}(Gr) \le log\_{10}(Gr)\_{1} = 0 log\_{10}(Gr) < log\_{10} (Gr)\_{0} (7)

The constants needed to calculate Nu are given in Table 1 for five heat-flow directions. The Gr for an airspace of width L, mean temperature T, and temperature difference  $\Delta T$  can be obtained from Equation (8).

$$log_{10}(Gr) = log_{10}(\Delta T * L^3) + 3.4146 - 0.4359 \times 10^{-2}\overline{T} + 0.36441 \times 10^{-5} \overline{T}^2$$
(8)

The temperature dependence in Equation (8) is due to the temperature dependence of the properties of air.

The constants in Table 1 have been used to reproduce the R-values shown in the ASHRAE Handbook<sup>[3]</sup>. The agreement is excellent except for the case of heat flow down at 45°. In this case, the scatter in the data is greater than in the other directions. The curve for heatflow down at 45° was forced to be parallel to the curves for the other heat-flow directions in the present correlation.

The R-value for airspaces in series can be calculated from the requirement that the steady-state heat flux be the same across each individual airspace. This requirement leads to an expression for the temperature difference,  $\Delta T_i^{(J)}$ , and thermal resistance,  $R_i^{(J)}$ , across airspace i after j iterations.

$$\Delta T_{i}^{(j+1)} = (\Sigma_{i} \Delta T_{i}^{(j)}) * R_{i}^{(j)} / \Sigma_{i} R_{i}^{(j)}$$
(9)

The  $\Delta T^{(j)}$  are constrained by the overall temperature difference across the air spaces. An estimate for  $\Delta T_i^{(1)}$  is obtained from Equation (10) and  $R_i^{(1)}$  is then obtained from Equation (7).

$$\Delta T_{i}^{(1)} = l_{i} \Delta T_{\text{Total}} / L_{\text{Total}}$$
(10)

The iteration shown by Equation (9) continues until Equation (11) is satisfied for each air space.

Constants for the calculation of R-Values for reflective airspaces. ł TABLE 1.

Heat Flow Direction	(Gr)0	(Gr) 1	(Gr) 3	Q	٩	υ	a2	a3
Down	2,50	5.09	7.50	1.4959	-0.6080	6.3590E-02	6.5572E-03	2.7129E-04
Down, 45 deg.	3.50	4.68	7.82	-1.3771	0.2989	5.9300E-03	1.2676E-03	8.7056E-02
Horizontal	2.50	4.35	7.20	-2.2234	0.6784	-3.0280E-02	1.6481E-03	2.3528E-02
Up, 45 deg.	2.50	4.14	6.90	-0.3000	0.0381	2.4090E-02	1.5679E-02	-3.4369E-02
Up	2.50	3.49	6.80	-0.8620	0.2912	-3.8630E-04	1.6645E-01	-1.3970E-02

$$|\Delta T_{i}^{(j+1)} - \Delta T_{i}^{(j)}| < 1.0 \times 10^{-4}$$
 (11)

Steady-state R-values,  $R_i$ , and the total R-value for the series of airspaces,  $\Sigma R_i$ , are calculated with Equations (1-6) once the temperature distribution is known. The input for the calculation consists of the exterior bounding temperatures, the width of each airspace, the emittance of the bounding surfaces for each airspace, and the heat-flow direction.

#### REFLECTIVE INSULATION DATABASE

To determine the accuracy of the predictive program, the model was used to determine the thermal resistance of a series of reflective insulation systems that had been recently tested and reported<sup>[2]</sup>. The results from this series of ninety-four ASTM C 236 guarded hot box experiments on thirteen different reflective insulation systems were used to validate the model.

Table 2 gives construction details of the thirteen reflective insulation systems used to test the model. These systems are shown schematically in Figure 1. The test panels were constructed with either 2 x 4 or 2 x 6 studs placed 16 inches on center forming a panel that was 94 inches square for horizontal heat flow experiments or 75 by 78 inches for vertical heat flow experiments. All the test panels were sheathed on both sides with a 1/4inch thick plywood material. The studs were fabricated from either extruded polystyrene foam (XEPS) or kiln-dried wood.

To create a single reflective airspace, having an effective emittance of approximately 0.05, a single sheet of an aluminum foil paper laminate reflective insulation material was adhered to the interior side of the hot side sheathing. A single cavity having an effective emittance of 0.03 was constructed by applying the reflective insulation to the interior sides of both sheathings. Two airspace cavities with each airspace having an effective emittance of 0.05 were constructed by installing a sheet of the double-sided reflective insulation material in the center of the cavity. To decrease the effective emittance of each airspace to approximately 0.03 in the two airspace cavities, the reflective insulation material was added to the interior side of both sheathings. Four airspace cavities were created by utilizing a commercial threelayer reflective insulation material.

The thermal resistance of each of the materials used to construct the test panels was determined in accordance with ASTM C 518-85, Standard Test Method for "Steady-State
	Keat	No. 16	Airspa	ace Thio	ckness,	Inches	Ca	wity Surface	Emittances, e			
Test Panel	Flow Direct.	No. of AirSps	#1	#2	#3	#4	#1	#2	#3	#4		
1A	Horiz.	1	3.50				0.052/0.77					
1B	Horiz.	2	1.75	1.75	••••		0.77 /0.052	0.052/0.77				
2A	Horiz.	1	3.50				0.052/0.052					
2в	Horiz.	2	1.75	1.75			0.052/0.052	0.052/0.052				
2C	Horiz.	4	0.88	0.88	0.88	0.88	0.77 /0.027	0.85 /0.029	0.029/0.85	0.027/0.77		
3A	Horiz.	1	3.50	•••			0.052/0.052					
3B	Horiz.	2	1.75	1.75			0.052/0.052	0.052/0.052				
3C	Horiz.	4	0.88	0.88	0.88	0.88	0.77 /0.027	0.85 /0.029	0.029/0.85	0.027/0.77		
11A	Up	1	3.50				0.052/0.77					
11A	Down	1	3.50				0.052/0.77					
12A	Up	1	5.50				0.052/0.77					
12A	Down	1	5.50				0.052/0.77					
12B	Down	4	1.88	0.88	0.88	1.88	0.77 /0.027	0.85 /0.029	0.029/0.85	0.027/0.77		
13A	Up	1	5.50				0.052/0.77					
13A	Down	1	5.50			<i>.</i>	0.052/0.77					
13B	Down	4	1.88	0 <b>.88</b>	0.88	1.88	0.77 /0.027	0.85 /0.029	0.029/0.85	0.027/0.77		

Table 2. -- The description of thirteen test cavities containing reflective insulation material

Notes: 1. Airspace thicknesses and cavity emittances are listed from the cold side to the hot side, respectīvely. Airspace 1 is the first airspace when the test panel is viewed from the cold surface.

 The cavity surface emittances are listed from the cold side. The first emittance listed for each cavity is the cold surface of that airspace.





Heat Flux Measurements and Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus.<sup>[8]</sup>" The emittances of all of the materials which would bound an airspace were determined using a Devices and Services Emissometer<sup>[9]</sup>.

In addition to the temperature instrumentation normally used when performing guarded hot box experiments, thermocouples were installed on each major surface within the cavity and in the central stud. This temperature data, in conjunction with the thermal resistance data for the building materials, was used to calculate the cavity thermal resistance following the procedure outlined in the proposed ASTM Standard Specification for "Reflective Insulation for Building Applications<sup>[10]</sup>."

At least five experiments were performed on each panel with varying cavity temperature differences. No experimental replication was performed. These data are presented in Table 3. Table 3 also includes predictions of the cavity thermal resistance based on the work of Robinson and Powell<sup>[4,5]</sup>, R<sub>(NBS)</sub>, and the present model,  $R_{(PRED)}$ . The cavity resistance,  $R_{(NBS)}$ , was computed utilizing the curve-fitting coefficients developed by Yarbrough when he applied the Method of Least Squares to the data generated by Robinson and Powell<sup>[7]</sup>. When there was more than a single airspace forming the cavity,  $R_{(NBS)}$ was determined for each airspace and summed.

The measured cavity thermal resistances,  $R_{(MEAS)}$  were fitted as a function of cavity temperature difference; the fit coefficients were used to compute the cavity thermal resistance for a temperature difference of 30°F. Uncertainty in the temperature difference measurement does not significantly impact the curve-fitting. Similar to the treatment of the experimental data,  $R_{(NBS)}$  and  $R_{(PRED)}$ were calculated for each of the experiments on each panel, were curve fit as a function of cavity temperature difference, and were computed for a temperature difference of 30°F. These results are compiled in Table 4, along with the differences between these results and the measured cavity thermal resistance,  $R_{(MEAS)}$ .

#### SINGLE AIRSPACE CAVITIES

The comparisons between the measured and calculated cavity thermal resistances for single airspace systems are shown in Figure 2. For single airspace cavities 3.5 inches deep,  $R_{(MEAS)}$  for heat flow horizontal (Test Panels 1A, 2A, and 3A), up (Test Panel 11A), and down (Test Panel 11A) averaged 2.59, 2.00, and 7.85 hr ft<sup>2</sup> F/Btu respectively. There was excellent agreement between

			Tempera	ture, F	R·Valu	ue, hr ft	:^2 F/Btu	% Diff Pred. V	<sup>f</sup> erence, /s. Meas.
Test Panel	No. of AirSps	Test Number	Tmean	dī	R(Meas)	R(NBS)	R(Pred)	R(NBS)	R(Pred)
				(0 <b>7</b>			2.24		
18	1	H-1	50.0	49.7	2.17	2.21	2.21	4.4	1.8
		H-2	50.1	40.5	2.27	2.30	2.33	5.8	2.7
		н-5	49.8	30.0	2.3/	2.52	2.52	5./	).y
		H-4	50.8	20.0	2.03	2.78	2.78	5.4	5.5
		H-2	49.9	10.5	3.04	3.41	3.35	10.8	9.2
1B	2	H-6	70.7	92.5	3.26	4.25	4.38	23.3	25.6
		H-7	71.1	80.9	3.42	4.39	4.57	21.9	25.0
		H-8	64.6	59.9	3.70	4.82	4.99	23.2	25.9
		H•9	59.6	40.0	4.24	5.53	5.65	23.4	24.9
		H-10	54.7	20.3	5.14	6.92	6.97	25.8	26.3
2A	1	H-11	74.3	50.7	2.37	2.31	2.34	-2.6	-1.2
		H•12	73.7	40.3	2.48	2.42	2.49	-2.7	0.5
		H · 13	73.2	31.6	2.57	2.55	2.67	•0.7	3.6
		H•14	74.0	20.1	2.81	2.89	3.03	3.0	7.4
		H-15	74.2	10.6	3.23	3.53	3.65	8.5	11.4
		H•16	50.0	50.4	2.34	2.39	2.32	1.9	-1.0
2в	2	H-17	71.9	92.2	3.52	4,50	4.65	21.8	24.4
		H-18	55.0	79.3	3.60	4.77	4,80	24.6	25.1
		H-19	59.8	59.4	4.02	5.17	5.31	22.2	24.2
		H-20	65.9	39.7	4.69	5.89	6.11	20.4	23.2
		H-21	70.1	20.3	5.92	7.36	7.70	19.5	23.1
		H-22	73.2	10.0	7.15	8.83	9.87	19.0	27.6
20	4	H-23	73.6	49.8	5.27	14.60	14.32	63.9	63.2
		H•24	74.7	39.4	5.57	15.30	14.81	63.6	62.4
		H-25	75.1	29.9	6.04	16.07	15.32	62.4	60.5
		H-26	75.3	21.0	6.31	16.90	15.86	62.6	60.2
		H-27	73.7	9.7	7 18	18.21	16.73	60.6	57.1
		H-28	51.4	50,4	5.54	15.06	14.47	63.2	61.7
34	1	H- 20	75 4	<b>48</b> /	2 47	2 22	2 37	-6.2	-6 2
w71	•	H-30	76.6	40.1	2 60	2.41	2.50	-7.7	.3 0
		H-31	75.6	28.8	2 80	2.60	2.74	-7.7	-2.4
		H-32	75 0	10.6	3 05	2 02	3 04	-4 5	0.4
		H-33	74.8	11.2	3.40	3.47	3.50	1.0	5.2
		H-34	50.1	49.1	2.51	2.40	2.33	-4.5	-7.3

Table 3. - The measured and the predicted thermal resistances of thirteen test cavities containing reflective insulation materials.

Note: The percentage differences were computed from % Difference = ([R(NBS) · R(Meas)] / R(NBS)) \* 100 or % Difference = ([R(Pred) · R(Meas)] / R(Pred)) \* 100.

			Tempera	ture, F	R-Valu	ue, hr ft	-2 F/Btu	% Diff Pred. v	erence, /s. Meas.
Test Panel	No. of AirSps	Test Number	Tmean	dT	R(Meas)	R(NBS)	R(Pred)	R(NBS)	R(Pred)
3в	2	H-35	73.0	96.4	3.96	4.44	4.59	10.9	13.7
		H-36	55.2	80.1	4.04	4.75	4.79	15.1	15.7
		H-37	60.2	60.4	4.45	5.13	5.28	13.4	15.8
		H-38	65.2	40.2	5.12	5.87	6.08	12.7	15.7
		H-39	69.9	20.5	6.27	7.34	7.68	14.6	18.3
		H-40	72.5	10.4	7.60	8.77	9.73	13.3	21.9
3C	4	H-41	74.7	50.7	5.28	14.53	14.27	63.7	63.0
		H-42	75.7	39.8	5.57	15.26	14.77	63.5	62.3
		H-43	74.4	29.2	6.09	16.15	15.37	62.3	60.3
		H-44	74.8	20.2	6.92	16.99	15,93	59.3	56.5
		H-45	74.8	10.0	7.46	18.14	16.67	58.9	55.3
		H-46	49.9	49.4	5.66	15.20	14.53	62.8	61.0
11A	1	V-1U	50.1	49.2	1.76	1.78	1.74	0.8	-1.0
		V-2U	50.9	40.6	1.85	1.86	1.84	0.5	-0.9
		V-3U	50.5	31.3	1.99	1.99	1.96	-0.2	•1.5
		V-4U	49.9	20.4	2.17	2.23	2.20	2.8	1.3
		V-5U	50.3	10.5	2.40	2.62	2.60	8.5	7.7
		V-60	75.7	49.2	1.62	1.71	1.71	5.5	5.2
		V-70	49.8	50.2	7.53	7.55	7.59	0.2	0.8
		V-8D	50.5	40.9	7.64	7.72	7.80	1.1	2.1
		V-9D	50.0	29.0	7.92	8.09	8.16	2.1	2.9
		V-10D	50.3	20.3	8.32	8.44	8.49	1.4	1.9
		V-11D	50.1	9.4	9.47	9.05	9.10	-4.6	-4.1
		V-12D	74.8	50.9	6.84	7.04	7.22	2.9	5.3
12A	1	V•13U	74.9	49.4	1.61	1.56	1.86	-2.8	13.6
		V-14U	74.5	40.6	1.73	1.64	1.96	·5.2	11.5
		V-15U	74.3	29.0	1.85	1.92	2.13	3.4	13.1
		V-16U	74.6	20.4	2.03	2.12	2.33	4.4	13.1
		V-17U	69.9	8.6	2.79	2.04	2.89	-36.5	3.4
		V•18U	50.3	49.9	1.73	1.62	1.84	-7.2	5.9
		V-19D	75.2	50.2	6.95	4.53	7.74	+53.4	10.1
		V-20D	75.3	39.9	7.12	4.67	8.03	-52.4	11.3
		V-21D	74.6	29.4	7.41	5.12	8.41	-44.7	12.0
		V-22D	75.8	20.3	7.61	5.86	8.81	-29.9	13.6
		V-23D	75.0	12.3	8.21	7.10	9.34	-15.5	12.2
		V-24D	50.8	49.5	7.56	4.77	8.09	•58.3	6.6
12в	4	V-25D	75.8	49.1	16.56	23.88	23.07	30.7	28.2
		V-26D	76.1	40.0	17.32	24.02	23.19	27.9	25.3
		V-27D	75.1	31.2	17.17	24.22	23.38	29.1	26.6
		V-28D	74.8	18.9	18.07	24.46	23.66	26.1	23.6
		V-29D	75.0	9.8	18,42	24.62	23.95	25.2	23.1
		V-30D	51.0	50.4	17,04	24.94	24.25	31.7	29.7

Table 3. -- (Cont.) The measured and the predicted thermal resistances of thirteen test cavities containing reflective materials.

			Tempera	ture, F	^2 F/Btu	% Difference, Pred. vs. Meas.			
Test	No. of	Test	•••••				•••••	••••	• • • • • • • • •
Panel	AirSps	Number	Tmean	đĭ	R(Meas)	R(NBS)	R(Pred)	R(NBS)	R(Pred)
13A	1	V-32U	74.8	49.8	1.72	1.56	1.86	-9.9	7.5
		V-33U	74.8	39.9	1.81	1.66	1.97	-9.3	7.9
		V-34U	75.0	30.1	1.94	1.88	2.11	-3.0	8.1
		V-35U	74.7	20.2	2.12	2.12	2.34	0.0	9.2
		V-36U	75.0	9.9	2.55	2.07	2.79	-23.3	8.6
		V-37U	59.1	49.7	1.69	1.60	1.85	-5.9	8.6
		V-38D	75.7	50.1	7.06	4.53	7.74	-55.9	8.7
		V-39D	75.8	41.5	7.29	4.63	7.97	-57.4	8.6
		V-40D	74.9	30.8	7.41	5.04	8.35	-47.2	11.3
		V-41D	76.8	20.1	7.90	5.87	8.80	•34_6	10.2
		V-42D	75.3	10.8	8.59	7.43	9.46	-15.7	9.2
		V-43D	60.7	49.9	7.62	4.67	7.95	-63.0	4.1
13B	4	V-44D	76.2	50.7	14.87	23.84	23.03	37.6	35.4
		V-45D	74.4	38.7	15.63	24.12	23.29	35.2	32.9
		V-46D	75.6	29.9	16.30	24.22	23.38	32.7	30.3
		V-47D	76.2	19.7	16.90	24.38	23.57	30.7	28.3
		V-48D	74.3	10.8	20.31	24.63	23.95	17.5	15.2
		V-49D	50.0	48.6	16.65	25.01	24.33	33.4	31.6

Table 3. -- (Cont.) The measured and the predicted thermal resistances of thirteen test cavities containing reflective materials.

			R-Value	e, hr ft^2	F/Btu	Pred. v	s. Meas.
Test Panel	No. of AirSps	Test Numbers	R(Meas)	R(NBS)	R(Pred)	R(NBS)	R(Pred)
1A	1	H-1 to H-5	2.39	2.51	2.51	4.8	4.8
18	2	H-6 to H-10	4.68	6.21	6.30	24.6	25.7
2A	1	H-11 to H-15	2.60	2.58	2.70	-0.8	3.7
28	2	H-17 to H-22	5.37	6.73	7.11	20.2	24.5
2C	4	H-23 to H-27	5.95	16.05	15.31	62.9	61.1
3A	1	H-29 to H-33	2.77	2.56	2.69	-8.2	-3.0
38	2	H-35 to H-40	5.82	6.76	7.13	13.9	18.4
3C	4	H-41 to H-45	6.13	16.06	15.31	61.8	60.0
11A Up	1	V-10 to V-50	2.00	2.01	1.98	0.5	-1.0
11A Down	1	V-7D to V-11D	7.85	8.04	8.11	2.4	3.2
12A Up	1	V-13U to V-17U	1.80	1.92	2.10	6.2	14.3
12A Down	1	V-19D to V-23D	7.34	5.07	8.38	-44.8	12.4
128 Down	4	V-25D to V-29D	17.48	24.23	23.39	27.9	25.3
13A Up	1	V-32U to V-36U	1.93	1.90	2.11	-1.6	8.5
13A Down	1	V-38D to V-42D	7.48	5.05	8.36	-48.1	10.5
13B Down	4	V-44D to V-48D	16.03	24.23	23.39	33.8	31.5

Table 4. -- The measured and the predicted thermal resistances of thirteen test cavities containing reflective insulation materials calculated from curve-fit coefficients for a temperature difference of 30F.

Notes: 1. The thermal resistances for the measured cavity, R(Meas), calculated from the curve-fit coeffients developed by Yarbrough, R(NBS), and calculated from the present model, R(Pred), are listed in this Table. They were determined by curve fitting the thermal resistance data as a function of temperature difference presented in Table 3 and computing the thermal resistance for a temperature difference of 30F from these coefficients.

> 2. The percentage differences were computed from % Difference = {[R(NBS) - R(Meas)] / R(NBS)} \* 100 or % Difference = {[R(Pred) - R(Meas)] / R(Pred)} \* 100.



Figure 2. The Percentage Difference Between the Thermal Resistances of Six Single-Airspace Test Cavities.



Figure 3. The Percentage Difference Between the Thermal Resistances of Seven Multi-Airspace Test Cavities.

 $R_{(MEAS)}$  and both models;  $R_{(NBS)}$  differed from the measured results by -1.4, 0.5, and 2.4 percent while  $R_{(PRED)}$  varied by 1.8, -1.0, and 3.2 percent for the three heat flow directions, respectively.

For single airspace cavities 5.5 inches deep,  $R_{(MEAS)}$  for heat flow up (Test Panels 12A and 13A) and down (Test Panels 12A and 13A) and 13A) were 1.86 and 7.41 hr ft<sup>2</sup> F/Btu respectively.  $R_{(NBS)}$  differed from the measured results by 2.3 and -44.8 percent while  $R_{(PRED)}$  varied by 11.4 and 11.4 percent for the two heat flow directions, respectively. Since the original database used to develop both models was limited to airspaces with a maximum depth of 3.5 inches, both models are extrapolating to larger depths for their predictions of the 5.5-inch cavities. Although the original model is more accurate for the low resistance heat flow up direction, the new model is more precise in its treatment of the depth extrapolation with respect to the heat flow direction. Experimental verification of these extrapolations is recommended.

#### MULTIPLE AIRSPACE CAVITIES

The comparisons between the measured and calculated cavity thermal resistances for multiple airspace systems are shown graphically in Figure 3. No appreciable improvement in the calculation of the multiple airspace cavities were noted when the new correlation was utilized.

Measurements on two-airspace cavities (Test Panels 1B, 2B, and 3B) were limited to the heat flow horizontal direction.  $R_{(MEAS)}$  for these cavities averaged 5.29 hr ft<sup>2</sup> F/Btu.  $R_{(NBS)}$  and  $R_{(PRED)}$  differed from the measured results by 19.6 and 22.9 percent, averaging 6.57 and 6.85 hr ft<sup>2</sup> F/Btu, respectively.

Measurements on four-airspace cavities were performed in the heat flow horizontal (Test Panels 2C and 3C) and down (Test Panels 12B and 13B) directions. The average  $R_{(MEAS)}$  for the two heat flow directions were 6.04 and 16.76 hr ft<sup>2</sup> F/Btu, respectively.  $R_{(NBS)}$  differed from the measured results by 62.3 and 30.8 percent while  $R_{(PRED)}$  varied by 60.6 and 28.4 percent for the two heat flow directions, respectively. No appreciable improvement in the calculation of the multiple airspace cavities were noted when the new correlation was utilized.

#### CALCULATION OF THE INTERNAL TEMPERATURES

To access the algorithm used to compute the internal temperatures within a multiple airspace cavity (see equations 9 through 11), the calculated internal surface temperatures were compared with actual test results on two four-airspace cavities (Test Panels 2C and 3C). Both of these test panels had been evaluated with horizontal heat flow.

Table 5 lists the actual surface temperatures for the twelve experiments performed on Test Panels 2C and 3C, the calculated equivalent temperatures, and the difference between these sets. Figure 4 graphically depicts the two sets of temperatures for Test Numbers H-23 and H-41. The average differences between the actual and calculated internal cold side, middle, and internal hot side surface temperatures are 0.4, -1.3, and -1.7°F, respectively. correlation underestimates the temperature differences The (and therefore the thermal resistances) of the two outside airspaces while overestimating the temperature differences of the two inside airspaces. However, these differences in interior temperature are not large enough to fully account for the large differences between the calculated and measured R-Values for the multifoil product that was tested.

#### SUMMARY

The correlation for airspace R-Value based on NBS data and presented in this paper reproduces the measurements obtained at NBS and Holometrix for single reflective airspaces. The correlation forms the basis for a program that computes R-Values for an "idealized" assembly of reflective airspaces. Satisfactory agreement is not achieved between calculated and measured R-Values for heat flow through a commercial multifoil product. A clear explanation for the lack of agreement is not available. The measured heat transfer does not conform with that calculated for a series of parallel airspaces with one-dimensional heat flow. This may be a result of product design and/or the one-dimensional convective heat transfer assumptions.

The correlation in this paper agrees more closely with the measured R-Values for the relatively wide airspaces than previously proposed polynomials. The present correlation also provides for smooth transition to small airspace widths and temperature differences.

			Cavity	Surface		Actual		с	alculate	ed	Difference,				
_		_	Temper	ature,F	Internal	Tempera	tures,F	Internal	Tempera	atures,F	,F Actual-Calculated				
Test Panel	No. of Air\$ps	No.	cs	нs	cs	Mid	нs	cs	Mid	HS	cs	Mid	KS		
20	4	ਸ਼-23	48.7	98.5	61.5	71.6	84.0	61.3	73.8	86.2	0.2	-2.2	-2.2		
		H-24	55.0	94.4	65.0	73.0	83.3	65.0	74.9	84.6	0.0	-1.8	-1.3		
		∦-25	60.1	90.0	67.6	73.7	81.5	67.7	75.2	82.6	-0.1	-1.5	-1.1		
		H-26	64.8	85.8	69.9	74.2	79.9	70.1	75.4	80.6	-0.2	-1.2	-0.7		
		₩-27	68.8	78.5	71.1	73.1	75.8	71.3	73.7	76.1	-0.2	-0.6	-0.3		
		H-28	26.2	76.6	40.0	49.9	60.8	38.9	51.5	64.1	1.1	-1.6	-3.3		
3C	4	H-41	49.3	100.0	63.5	73.4	84.4	62.1	74.8	87.4	1.4	-1.4	-3.0		
		H-42	55.8	95.6	66.6	74.5	83.5	65.9	75.9	85.7	0.7	-1.3	-2.2		
		H-43	59.8	89.0	67.5	73.4	80.2	67.2	74.5	81.8	0.3	-1.1	-1.5		
		H-44	64.7	84.9	69.9	74.1	79.0	69.8	74.9	79.9	0.1	-0.8	-0.9		
		H-45	69.8	79.8	72.2	74.4	77.0	72.3	74.8	77.3	-0.1	-0.4	-0.3		
		H-46	25.2	74.6	39.6	48.2	58.1	37.7	50.0	62.3	1.9	-1.8	-4.2		

Table	5.	• •	The	measu	٠ed	and	the	calculated	internal	temperatures	of	two	test	cavities	containing
			refl	ective	e i	nsula	atio	n materials.							

Notes: CS = Cold Surface of Cavity, HS = Hot Surface of Cavity, and Mid = Middle of Cavity.



Notes: CS = Cold Surface ICS = Internal Layer on Cold Side of Midplane MID = Midplane IHS = Internal Layer on Hot Side of Midplane HS = Hot Surface

Figure 4. The Measured and the Calculated Internal Temperatures of Two Test Cavities Containing Reflective Insulation Materials.

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#### TABLE OF NOMENCLATURE

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E = Effective emittance, Equation (2)
\epsilon = Surface emittance
               \epsilon_i, for surface i
g = Gravitational acceleration, 4.1783 x 10^8 ft/hr<sup>2</sup>
Gr = Grashof Number, \Delta T 1^3 \beta g / 1728 v^2
         Heat transfer coefficient, Btu/hr ft<sup>2</sup> F
h =
               h_c , conduction - connection
         h_r^c , radiation
Thermal conductivity, Btu-in/ft<sup>2</sup> hr F
k =
         Distance across air space, inches
L =
         l, ith air space
Nusselt Number, h<sub>c</sub>l/k
Nu =
         Thermal resistance, hr ft<sup>2</sup> F/Btu
R =
T = Temperature, F
               \Delta T, temperature difference T_i^{(j)}, jth value for space i T_i^{(j)}, average value for space i
\beta = Expansion coefficient, 1/(T + 460)
v = Kinematic viscosity, ft<sup>2</sup>/hr
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# Danny S. Parker, Philip W. Fairey, and Lixing Gu

# A STRATIFIED AIR MODEL FOR SIMULATION OF ATTIC THERMAL PERFORMANCE

**REFERENCE:** Parker, D. S., Fairey, P. W. and Gu, L., "A Stratified Air Model for Simulation of Attic Thermal Performance," <u>Insulation Materials: Testing and Applications.</u> <u>2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: An improved computer simulation model for attics has been developed. The model accounts for detailed radiation, buoyancy and wind driven air flows and thermal stratification within the attic airspace. Through comparison with measured data, the paper examines the relative importance of the various modeling parameters and assumptions upon simulation accuracy. Stratification of the attic air was found to be a critical modeling characteristic which is typically ignored in building energy simulations. Both model and measurement results show that air stratification increases significantly when low-emittance surfaces are included.

KEYWORDS: Attics, thermal performance, attic ventilation, simulation models, radiant barriers.

#### INTRODUCTION

The critical importance of accurate modeling of thermal processes in residential attics has been acknowledged for some time. Monitored performance of Twin Rivers town-houses showed that 35% of winter heat losses in New Jersey could be attributed to attic heat transfer mechanisms [1]. Recent research at the Florida Solar Energy Center suggests that attic thermal conditions may have critical importance to air conditioning demand when air distribution supply ducts are located within the attic space [2].

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The existing design data for attic spaces in the <u>ASHRAE Handbook of</u> <u>Fundamentals</u> is based on early experimental work performed by Joy [3,4]. This research and that of others has shown that radiation accounts for the majority of the total heat transfer from an insulated attic to the house interior. Attic ventilation was determined to play a relatively minor role in heat gain across the attic airspace [5,6]. However, when at least one surface in the attic has low emissivity characteristics, such as with radiant barrier systems (RBS), attic ventilation rates and ventilation air flow paths become relatively more important to performance [4]. Recent studies have evidenced increased stratification of air temperatures within attic spaces when radiant barriers are installed [7,8].

## PREVIOUS RESEARCH

A number of past investigators have developed models to simulate attic thermal performance. Joy created a steady state attic heat balance model in order to generalize his experimental study of attic thermal performance. As described by Wilkes, the model solves a series of simultaneous differential equations which considers inside and outside boundary conditions, direct radiation exchange between the inside roof surface and floor and ventilation [9]. The model did not accommodate capacitance, temperature dependent convective heat transfer coefficients or heat transfer through non-roof/non-floor surfaces such as gable roof ends. Blancett et al., created a transient attic simulation for the Electric Power Research Institute (EPRI) which remedied some of these faults and allowed time-related calculation of attic performance [10]. However, this model did not explicitly model radiative transfer, assumed the attic air was well mixed and treated heat transfer coefficients as non-temperature dependent functions. Three hourly computer simulation models, TRNSYS [11] and NBSLD [12] and BETEH [13] have subroutines that calculate dynamic attic thermal performance. Although attic air temperatures estimated by NBSLD showed reasonable agreement with a monitored Houston, Texas attic, all of these calculation schemes have similar limitations to the EPRI model.

A more comprehensive transient attic model was developed by Peavy [14]. Radiative exchanges within the attic space were explicitly modeled as was thermal capacitance and temperature dependent convective heat transfer coefficients at the various surfaces. As with Joy's model, the ventilation air is not assumed to be well mixed in the horizontal dimension. This results in a more realistic description of the attic's horizontal thermal regime. However, Peavy's model ignored the effects of gabled roof ends and the effects of thermal buoyancy on attic ventilation rates. Wilkes developed a modification of Peavy's model for use by Owens Corning Fiberglas (OCF) which included gabled roof ends and improvements in the correlations for convective heat transfer coefficients [15]. The algorithm did not, however, include buoyancy in the estimation of attic ventilation rates or vertical thermal gradients. Comparison of the various models with a calibrated thermal research facility showed the OCF model to give the best prediction of actual attic thermal performance of the previous models.

Recently, Fairey and Swami created a simplified steady-state attic model

[16]. A key difference in the model was that the attic air space was split into two nodes to simulate thermal stratification. The model showed good agreement with Joy's measured steady-state data. The investigators also studied the effects of moisture absorption and desorption within an experimental attic using a very detailed attic simulation incorporating combined heat and moisture transfer. These simulations showed moisture phenomenon to be important to adequate thermal modeling of attics. This agrees with previous findings by Cleary [17] and Ford [18].

# DESCRIPTION OF A TWO ZONE ATTIC THERMAL MODEL

In order to more accurately characterize the thermal regime of residential attics, a two zone model of the attic domain was developed. The model allows explicit treatment of thermal stratification within the attic air space. A predictive model of attic ventilation rates was also created using very detailed building simulation, FSEC 2.1 [19]. This finite element computer program has been used to study complex interactions within the building; it is possible to characterize thermal as well as moisture absorption and desorption properties within the attic zone. The model's predictions were later validated using monitored data. These data were taken from measurements made in three attic test cells at FSEC's Passive Cooling Laboratory (PCL) in Cape Canaveral, Florida. The geometry and measurement locations of these test cells is shown in Figure 1.



Figure 1. Geometry of Passive Cooling Laboratory Attic Test Cells

# Radiant Heat Transfer

Radiative heat transfer is the dominant heat transfer mechanism in residential attics, particularly during summer conditions. The radiation component of attic heat transfer is governed by radiative exchange between pairs of surfaces of a 'N' surface enclosure in accordance with the surface temperature differences, their emissivities, view of the other surfaces and the Stefan-Boltzmann constant. For a pair of surfaces the net radiative heat transfer per unit area is:

$$Q_{rad} = \frac{\sigma (T_2^4 - T_1^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} * F$$
(1)

Where

 $\sigma$  = Stefan-Boltzmann constant (5.67 x 10<sup>-8</sup>) T = temperatures of surfaces (°K)

 $\varepsilon$  = the emissivities of the attic surfaces (dimensionless)

F = view factor (dimensionless)

For the five major surfaces in a simple residential attic (two roof decks, insulation surface, two gable roof ends), the problem is complex. This general equation is transformed to use script-F factors [20] to account for total enclosure emissivity and view factors with respect to each surface:

$$Q_{i} = \bigwedge_{j=1}^{n} \sum \sigma F_{ij} (T_{i}^{*4} - T_{j}^{4})$$
(2)

Where

 $A_i$  = areas of surfaces  $F_{ii}$  = script - F factor (dimensionless)

The emissivities of the various surfaces within the attic are taken from ASHRAE [3] and Cess and Sparrow [21].

#### Convective Heat Transfer

Convective heat transfer coefficients for inside surfaces in the attic enclosure were estimated based on the correlations adopted by ASHRAE [3]. These relationships account for surface orientation and instantaneous temperature differences between the surface and the air node. Turbulent natural convection was assumed since measured air velocities over attic surfaces, even under windy conditions with combined ridge and soffit vents, are well below 3 meters per second [8,9,18]. The values for the convective heat transfer coefficients are taken from McAdams [22]:

for vertical surfaces:

$$h_{n,v} = 1.31 * (T_{surf} - T_{air})^{0.33}$$
(3)

for energy flow down:

$$h_{n,d} \approx 0.76 (T_{surf} - T_{air})^{0.33}$$
 (4)

and for energy flow up:

 $h_{n,d} \approx 1.52 (T_{surf} - T_{air})^{0.33}$  (5)

for roof surfaces which are neither vertical or horizontal with energy flow down:

$$h_{n,rd,d} = \frac{1.805 (T_{surf} - T_{air})^{0.33}}{1.375 + \cos(\beta)}$$
(6)

and with energy flow up:

$$h_{n,rd,u} = \frac{9.624 (T_{surf} - T_{air})^{0.33}}{7.333 - \cos(\beta)}$$
(7)

Where

 $h_n =$  natural convective heat transfer coefficients (W/m<sup>2</sup>-K)  $T_{surf} =$  temperature of component surface (K)  $T_{air} =$  temperature of air node (K)  $\beta =$  roof slope pitch angle (degrees)

Although air movement in attics is generally assumed to be dominated by natural convection, attic ventilation may increase surface heat transfer due to forced convection. For instance, Metais and Eckert [23] demonstrated that a mixed convection regime between turbulent and laminar flow regimes will tend to increase free convection over the assumption of uniform turbulent flow.

In order to accommodate potential forced convection, a forced convection term was added for attic surfaces to estimate its magnitude. Peavy recommends the following relationship for turbulent flow over smooth flat plates at low air velocities (V < 3 m/s) [14]:

(8)

$$h_f = 7.176 (V_a)^{0.8} L^{-0.2}$$

Where

 $h_f$  = the forced air convection coefficient (W/m<sup>2</sup>·K)  $V_a$  = attic air velocity over surfaces (m/s) L= the air flow path length (m)

The air flow path length is taken as the sum of the soffit to ridge vent distance and the attic height. The calculation of average air velocity over attic surfaces is problematic since interior attic air flows are complex. Ford and

(9)

others have estimated attic air flow velocity based on the volumetric rate of ventilation over the cross sectional area of the attic perpendicular to the direction of flow [18]:

$$V_a = (Vol_a)$$

A<sub>fp</sub>

Where

 $AC_s$ = attic air changes per second  $Vol_a$  = attic volume (m<sup>3</sup>)  $A_{fp}$  = cross sectional area of air flow path (m<sup>2</sup>)

Unfortunately, such a calculation greatly underestimates the actual air velocities proximate to the attic heat transfer surfaces, particularly for the roof deck, since air flow through the attic is non-uniform. Figure 2 shows the relationship between measured roof deck and insulation surface air velocities inside the PCL over the course of a summer day. Other data shows air velocities over attic surfaces at least as great as those measured at the PCL. Ober [7] used smoke pencils to visualize air flow patterns in two test attics in Florida which had relatively low overall ventilation rates (~1.5 ACH). The tests showed that a considerable fraction of the air flowed along the bottom of the roof deck in all tests with two flow layers; a thin fast moving boundary layer with measured velocities of 0.30 - 0.46 m/s and slower boundary region with a measured air velocities of 0.10 - 0.20 m/s. Air movement over the insulation surface was lower, averaging 0.03 to 0.09 m/s.

A simple empirical model was developed based on measured air velocity measurements made in the PCL. Average attic roof deck and insulation surface air velocities were approximated as:



Figure 2. Measured Surface Air Velocities

$$V_{a,surf} = \left\{ \frac{v^2}{hb^2} + 4 hb g \frac{\Delta T}{(T_{sum})} \right\}^{0.5} - \frac{v}{hb}$$
(10)

Where

 $\begin{array}{l} \Delta T = \mbox{temperature difference between air node and surface (K)} \\ V_{a,surf} = \mbox{air velocity over deck or insulation surface (m/s)} \\ hb = \mbox{boundary layer thickness (0.05 m)} \\ v = \mbox{kinematic air viscosity (kg/ m/s)} \\ g = \mbox{gravitational acceleration (9.81 m/s^2)} \\ T_{sum} = (T_{surf} + T_{air}) \end{array}$ 

Peavy's heuristic method for combining forced and natural convection was adopted:

$$h_{c,i} = h_f + \frac{(9 - V_a^2)}{9} h_n$$
 (11)

Where

 $h_n$  = natural convection coefficients calculated for surfaces.

#### Potential Uncertainties Associated with Convective Heat Transfer

The empirical data which produced the heat transfer correlations in Equations 3 - 7 were based on laboratory measurements using 0.3 m square smooth plates which were parallel to each other with unrestrained air flow at the edges. Much larger actual surfaces in attics feature a number of irregular surfaces and are restrained at the edges.

Past research provides reason to question the appropriateness of laboratory estimates of heat transfer for buildings. As an example, ElSherbiny et. al [24] performed experiments which showed that convective heat transfer coefficients were increased by up to 50% for corrugated compared to smooth surfaces. Similarly, Anderson and Bohn [25] examined the effect of heat transfer of roughness elements with the same length as the thermal boundary layer. They found that roughness increased the heat transfer rate of a uniformly isothermal wall by 10 - 15% with local increases of 40%. Finally, analysis of measured convective heat transfer by McAdams [22] reveals that heat transfer rates are approximately 10% higher with rough plates versus smooth ones. In lieu of the very uneven nature of attic surfaces a first order approximation is made:

$$h_{c'} = h_{c,i} * 1.15$$
 (12)

Of perhaps greatest importance in this regard, one notes that the ASHRAE still air heat transfer coefficients for low emissivity materials [3] are considerably higher than the values commonly computed based on the relationships presented by Holman [26] or McAdams [22] as reported in the <u>ASHRAE</u> <u>Handbook</u>. The ASHRAE still air values are based on a 5.5 °K temperature difference between air and surfaces when evaluated at a temperature of 294 °K. Figure 3 shows a plot of the computed convective heat transfer coefficients for the PCL attic insulation surface and roof deck surfaces for October 19th, 1987. Equations 3 - 12 were used to make the estimates. The resulting time varying values are compared to the ASHRAE still air coefficients for those same surfaces at the respective tilts for each surface. A large discrepancy in the convective value for the insulation surface is obvious. Due to their larger magnitude, use of the ASHRAE still air values result in a much better calculation of the attic lower zone air and insulation surface temperatures. This research found that heat transfer coefficients calculated using the above equations were satisfactory if the value for the insulation surface was increased by 100%. This resulted in very accurate predictions of the various attic surface temperatures (see Figures 9 and 10).



In general, it is concluded that convective heat transfer coefficients associated with building elements, such as commonly found in residential attics may have relatively higher heat transfer rates that the common simplified assumptions arising within most building simulations. Certainly one must stretch the analogy to propose a fibrous insulation surface to possess the convective heat flow characteristics of a small smooth flat plate with unrestrained air flow at the edges.

# Forced Convection Coefficient of the Outside Roof Surface

The convective heat transfer coefficient of the outside roof surface was approximated by assuming fully developed turbulent flow across the entire roof section. Relatively few studies have been completed on measured convection heat transfer for building exterior surfaces [27] although the various assessments are in reasonable agreement [18]. A simple correlation is taken from Burch and Luna [28]:

$$h_r = 2.8 + 4.8 (V')$$
 (13)

Where

 $h_r = roof$  convective heat transfer coefficient (W/m<sup>2</sup>·K)

V' = wind speed parallel to the roof surface (m/s).

The reduced wind speed at the roof surface was determined using the correlation recommended by Sherman and Modera [29] to translate weather tower wind speeds to those on site:

$$V' = \frac{\alpha_{,w} (H_w/10)^{\gamma,w}}{\alpha_{,s} (H_s/10)^{\gamma,s}} * V$$
(14)

 $\begin{array}{ll} \alpha_w &= \mbox{alpha terrain parameter for weather tower (0.67)} \\ H_w &= \mbox{height of wind tower (m) (10)} \\ \gamma,w &= \mbox{gamma terrain parameter for weather tower (0.15)} \\ \alpha_s &= \mbox{alpha terrain parameter for site (1.0)} \\ H_s &= \mbox{height of site (m) (3.0)} \\ \gamma,s &= \mbox{gamma terrain parameter for site (0.25)} \end{array}$ 

i = wind speed at measurement site (m/s)

The predicted roof surface temperature is estimated as an energy balance taking into account the roof surface absorptance and emittance properties, the solar flux, the sky temperature and the associated radiative, conductive and convective heat transfer coefficients.

#### Measured Attic Ventilation Rates

Although the adequacy of attic ventilation rates to reduce moisture accumulation have received considerable attention [30,31,32], the actual determination of typical **in situ** rates in residential attics is less complete. Grot and Siu [33] took test data on three houses in Houston, Texas which had soffit vents. Measured ventilation rates using sulfur hexafluoride (SF<sub>6</sub>) tracer gas tests for the attics were 1.7 to 2.3 air changes per hour during the month of August, 1976.

Cleary and Sonderegger [32] made several measurements using SF<sub>6</sub> tracer gas to measure attic air change rates at various wind speeds in a house in Oroville, California. They found rates of 0.023 m<sup>3</sup>/s per m/s wind speed in an attic with 3,000 cm<sup>2</sup> of soffit vents. Given the volume of the residential attic, this equates to an approximate air change rate of 4.6 air changes per hour (ACH) at a 7 m/s windspeed. Using similar SF<sub>6</sub> equipment Ford [18] measured attic air change rates of 3 - 4 ACH under moderate wind conditions in Princeton, New Jersey. Dietz et al., measured a rate of 2.9 ACH in a long term tracer gas test on an attic in an Illinois house [34]. In a number of experiments using SF<sub>6</sub> tracer gas in two attics in Ocala, Florida, Ober measured average air change rates of 0.9 to 1.8 ACH in two attics in test periods ranging from 2 to 27 days [7].

During the summer and fall of 1987 attic ventilation rates were measured at the PCL in Cape Canaveral. The data was taken from attic test cells under normal wind and thermal conditions. A calibrated anemometer and a bulk air flow duct measured ventilation rates during this period while on-site weather data was collected.

Measured average air change rates were approximately 2.70 ACH over the three day data collection period in a normal attic configuration. Figure 4 shows the pronounced relationship between the attic test cell ventilation rate and site measured wind speeds. Although some scatter is evident, a nearly linear relationship is observed. A similar plot of attic air change rate against inside to outside temperature difference (Figure 5) shows considerably greater scatter, but a definite trend of higher rates at higher temperature differences. As a statistical check, the residual errors from the regression between attic air change rate and wind speed were regressed against temperature difference itself. Although wind speed is the primary driver of attic ventilation, thermal buoyancy appears to be of significant secondary importance. This conclusion was statistically significant at better than a 99% confidence level.

#### Attic Ventilation Model

There are few existing attic ventilation models. Peavy used results from  $SF_6$  tests from three test attics in Houston, Texas to formulate an empirical model of attic ventilation [14]:



Fig. 4--Relationship of natural attic ventilation rates to local wind speed.



$$V_{a} = 0.45 V (0.087 + 0.131 | sin D|^{5/2})$$
 (15)

Where

 $V_a$  = the volume air flow rate per ceiling area (cuft min/sqft)

V = wind speed (mph)

D = wind direction measured from south (degrees)

Blancett et al. [10] created a similar model based on the measurements of Hinrichs [6]:

I = 0.5 + 0.245 (V)(16)

Where

I = attic air change rate

Both attic ventilation models ignore the effects of thermal buoyancy on ventilation. Each calculation also relates attic air infiltration rates to empirically fitted parameters that are unlikely to be generalizable to a variety of attic geometries and climatic locations.

Rather than develop another empirical model with only limited application, a more general calculation of attic ventilation was created based on readily available engineering data. Attic ventilation is defined as a function of two primary driving forces, wind and thermal buoyancy. The air flow rate then depends on inlet and outlet areas, local wind speeds and temperature differences and attic geometry. Attic ventilation due to buoyancy is estimated according to the relative air densities of the ambient air and the upper attic zone.

Based on observations by Ober [5] and of PCL data, we assume that the attic roof deck to ambient temperature difference drives the buoyancy phenomenon for the overall attic:

$$\dot{m}_{buo} = L_o [gH_s | T_d T_\alpha | /T]^{0.5}$$

$$\dot{m}_{buo} = mass flow rate in lower attic caused by buoyancy (m3/s)$$

$$L_o = free inlet area (m2; soffit inlet area)$$

$$g = acceleration due to gravity (9.81 m/s2)$$

$$H_s = height to neutral pressure plane (m)$$

$$T_\alpha = ambient temperature (K)$$

$$T_d = deck surface temperature (K)$$

$$T = min \{T_\alpha, T_d\}$$

$$(17)$$

The vertical distance to the neutral pressure plane in the attic is determined according to the calculation suggested by ASHRAE [3].

$$H_{s} = \frac{H}{1 + (A_{1}/A_{2})^{2} (T_{o}/T_{i})}$$
(18)

where

H = the attic inlet to outlet height (m)  $A_1$  = the lower soffit inlet opening area (m<sup>2</sup>)  $A_2$  = the upper ridge vent outlet opening area (m<sup>2</sup>)  $T_i$  = inlet air temperature (K)  $T_o$  = outlet air temperature (K)

The buoyancy driven air flow is assumed to move from the soffit vents to the lower attic air zone and then to the upper attic air zone before exiting through the ridge vents. Calculation of wind driven ventilation requires knowledge of the attic inlet and outlet resistance to air flow. The soffit-ridge vent discharge coefficient is taken from experimental measurements made by Burch and Treado [35]:

(19)

$$\dot{m}_{wind} = L_o C V'$$

Where

 $L_{n} = \text{free soffit inlet area } (m^2)$ 

$$C =$$
soffit discharge coefficient (0.38)

V' = reduced wind speed (m/s)

## Attic Air Stratification

Monitored data show that attic air temperatures on hot summer days may be quite stratified and use of mid-zone attic temperatures can lead to errors in calculation of ceiling heat flux [36]. The relative degree of stratification is greater when radiant barriers are in place [4,8]. Figure 6 shows the air temperature distribution in an insulated attic test cell at noon on October 3, 1987. The air temperature near the ridge vent is approximately 5°C warmer than the air temperature directly above the insulation surface. The lower panel shows data



Figure 6-1. Temperature distribution within PCL attic test cell at noon on October 3, 1978: Normal configuration.



Figure 6-2. Temperature distribution within PCL attic test cell at noon on October 6, 1978: Roof deck mounted radiant barrier.

taken in the same test cell three days later when an attic radiant barrier was in place. The graph shows a stratification between the upper and lower attic temperatures of over 8 °C. Ober found similar levels of stratification in his investigation using attics above a duplex in Ocala, Florida [7].

The two attic air nodes are thermally linked by temperature related buoyancy within the attic. The buoyancy term calculated above computes a mass air flow rate from the lower to upper zone. The air flow to the lower attic zone from outside is assumed to equal the buoyancy driven ventilation between the upper and lower air nodes, with all of the additional air mass flow to the upper zone node. The overall attic ventilation is determined by adding the combined wind and stack terms in quadrature [3]:

$$\dot{m}_{inf,tot} = R' (\dot{m}_{buo}^2 + \dot{m}_{wind}^2)^{0.5}$$
 (20)

 $\dot{m}_{inf,tot}$  = total attic mass flow rate in upper attic caused by buoyancy and wind (m<sup>3</sup>/s)

The estimated mass flow rate is modified based on the difference in the inlet and outlet data. This relationship is based on regressed values from data given in ASHRAE [3]:

$$R' = 0.3733 + 1.0171 [1 - exp(-\phi)]$$
(21)

- R' = increase in flow caused by excess of one opening over another (dimensionless)
- $\phi$  = outlet area to inlet area ratio (ratio  $\geq$  1) (2.0)

Based on our simplified description of the attic ventilation air flow path, the lower attic zone ventilation is assumed to be equal to the buoyancy mass flow rate:

$$\dot{m}_{inf,l} = \dot{m}_{buo} \tag{22}$$

where

 $\dot{m}_{inf,l}$  = ventilation mass flow rate to lower attic zone from outside (m<sup>3</sup>/s)

The remainder to the ventilation air flows to the upper attic node:

$$\dot{m}_{inf,u} = \dot{m}_{inf,tot} - \dot{m}_{inf,l} \tag{23}$$

where

 $\dot{m}_{inf,u}$  = ventilation mass flow rate to upper attic zone from the outside (m³/s)

#### Description of the Overall Attic Heat and Moisture Balances

A graphical representation of our attic model is shown in the schematic in Figure 7. The attic is bisected to form two distinct air zones. Each attic surface is radiatively coupled with the other four attic surfaces. Convective heat transfer is estimated from each of the surfaces within the two zones. The upper and lower air zones are linked by buoyancy driven mass flows.



Figure 7. Schematic of two-zone attic model.

A general description of the lower and upper attic energy balances can be summarized mathematically as:

Lower Attic Energy Balance

$$\rho VC_{\rho} \frac{\partial T_{L}}{\partial \tau} = \dot{m}_{buo}C_{\rho} (T_{\alpha} - T_{L}) + \Sigma h_{i}A_{i} (T^{*} - T_{L})$$
(24)

Lower Attic Moisture Balance

$$\rho V - \frac{\partial W_{L}}{\partial \tau} = \dot{m}_{buo} (W_{\alpha} - W_{L}) + \Sigma h_{m,i} A_{i} (W^{*} - W_{L})$$
(25)

Upper Attic Energy Balance

$$\rho VC_{\rho} \frac{\partial T_{U}}{\partial \tau} = \dot{m}_{buo}C_{\rho} (T_{L} - T_{U}) + \Sigma h_{i}A_{i} (T' - T_{U})$$

$$+ \dot{m}_{inf,U}C_{\rho}(T_{\alpha} - T_{U})$$
(26)

# Upper Attic Moisture Balance

$$\rho V - \frac{\partial W_{U}}{\partial \tau} = \dot{m}_{buo} (W_{L} - W_{U}) + \Sigma h_{m,i} A_{i} (W^{*} - W_{U})$$

$$+ \dot{m}_{int,u} (W_{\alpha} - W_{U})$$
(27)

Where

A <sub>i</sub>	= area of i-th surface (m <sup>2</sup> )
Ċ,	= specific heat (J/kg·K)
h	= convective heat transfer coefficient for i-th surface $(W/m^2 \cdot K)$
h <sub>m,i</sub>	<ul> <li>convective moisture transfer coefficient for i-th surface (kg/m<sup>2</sup>·s)</li> </ul>
m <sub>buo</sub>	= mass flow rate in lower attic caused by buoyancy (kg/s)
m <sub>inf.u</sub>	= mass flow rate to the upper attic zone from the outside $(m^3/s)$
T <sub>L</sub>	= lower attic temperature (K)
Т,	= zone temperature (K)
T <sub>u</sub>	= upper attic temperature (K)
T <sub>a</sub>	= ambient temperature (K)
T	= inside surface temperature (K)
Wi	= lower attic humidity ratio (kg/kg)
Wi	= upper attic humidity ratio (kg/kg)
W	= ambient humidity ratio (kg/kg)
w	= inside surface humidity ratio (kg/kg)
V	= volume of each zone $(m^3)$
ρ	= air density (kg/m <sup>3</sup> )
τ	= time (s)

# MODEL COMPARISON WITH MEASURED DATA

The accuracy of the described attic model was compared with monitored data collected from the PCL attic test cells. As shown in Figure 1, the test cells are highly instrumented 2 x 3 meter sections that approximate one half of a typical attic cross section. The 8.7 m<sup>3</sup> test cells have a 28.5 degree roof slope with light gray asphalt shingles and continuous soffit and ridge vents. The free area of soffit vents was estimated for the test cell to be 15.2 cm<sup>2</sup>. A bulk air flow duct with a calibrated hot wire anemometer is located at the top of the test cells which is used to measure attic ventilation rates. Figure 8 shows a comparison of the measured attic test cell ventilation rates and those predicted by the ventilation model. The predictions are excellent with a standard error of only 0.07 ACH for the three day period.

Figure 9 shows a simplified version of our model which uses only a single zone to represent the air node within the attic space and ignores thermal stratification of attic air. The measured attic deck temperature is a prescribed boundary condition for the simulation. The insulation surface temperature is slightly over-predicted in the upper graph. Moreover, the overall thermal regime is very poorly predicted for the radiant barrier case relative with the one-zone model as shown in the lower graph. The single zone model over predicts both the temperature of the insulation surface and the attic air temperature.



Figure 8. Comparison of measured and predicted attic test cell ventilation rate.





The two-zone model, however, exhibits none of these problems, and accurately predicts the various temperatures within the attic regime for both attic configurations. Figure 10-1 predicts attic temperatures in the PCL test cell, both as monitored and predicted by the described model for the upper and lower air zones and the insulation surface.



Figure 10-2 shows the predictions for the temperatures when a roof deck mounted radiant barrier is in place on a different day. Thermal stratification of the attic air is clearly visible in both cases. The predictions of the model are quite good for both cases.



The two graphs in Figure 11 show the overall success of our two-zone model in predicting attic thermal performance.





Identical calculations were performed with the roof deck temperature estimated depending on measured solar flux and sky temperature. Figure 11-1 shows the PCL attic test cell in its standard configuration; Figure 11-2 shows the prediction when a radiant barrier was installed. In both cases, the deck, internal attic air and surface temperatures all compare well to the measured data.

#### **Roof Moisture Accumulation**

In performing the comparisons of the model results with the PCL attic data, it was noted that the roof deck temperatures were often depressed between 5 A.M. and 8 A.M. relative to the model predictions. Subsequent early morning examination of the PCL attic roof shingles revealed moisture deposition on the roof surfaces due to dew formation at night. During mornings, the attic roof deck was being evaporatively cooled as the moisture was driven off by the solar flux. This phenomenon was incorporated into the attic model in a simplified fashion. A film of water is deposited upon the roof shingles when the shingle temperature reaches that of the ambient dew point. For this purpose, a calculation by Sissons [37] was adopted to estimate the rate of evaporation. This is a function of the vapor pressure of water and its temperature, the vapor pressure of water in the ambient air and its dewpoint and the velocity of air flow with respect to the surface:

$$W_{p} = 9.315 \times 10^{-4} 0.2(95 + 83.66 V')$$
 (Pws-Pw) A/Lat (28)

where

 $W_n$  = water condensation (evaporation) rate (kg/sec)

V' = roof level air velocity (m/s)

Pws = saturation vapor pressure of water at temperature (Pa)

- Pw = vapor pressure of the moisture in the air at its temperature and relative humidity (Pa)
- A = exposed surface area (m<sup>2</sup>)
- Lat = latent heat of vaporization (J/kg)

Figure 12 shows the improvement in the simulation prediction for the early morning hours with the above evaporation equations incorporated in the attic model.



#### Example Analysis for a Typical Residential Attic

A prototype residential attic is used to illustrate our attic simulation model. The attic is conventional with a simple rectangular shape  $(9.1 \times 15.2 \text{ m})$  with asphalt shingles on a gabled roof with a 30 degree slope. The attic floor is covered with RSI 3.34 m<sup>2</sup> K/W fibrous insulation. Orlando typical meteorological year (TMY) data is used to produce two simulations. Figure 13 shows the temperature distribution within the attic on August 1st for the prototype configuration. Figure 13-2 shows the temperature distributions in the same attic with a roof mounted radiant barrier installed. The resulting reduction in insulation surface temperature from the radiant barrier and the increase in attic air stratification is clearly evident from comparison of the two plots.



## SUMMARY

A stratified air model of a residential attic has been described. The two zone model allows improved estimation of attic ventilation, ceiling heat flux and influence of radiant barrier systems on overall building performance. The ventilation and air node temperature prediction of the model has been compared to monitored attic data. Agreement between the model and the monitored data was very good.
Analysis revealed that characterization of moisture condensation on attic roof assemblies as well as treatment of moisture dynamics within the attic itself is useful to improving the prediction of attic thermal performance. Comparison with measured data also indicated that convective heat transfer coefficients for interior attic surfaces may be significantly higher than commonly supposed in the literature. Air flows within attic structures appear to be quite complex. Future research on attic air flow distributions and interaction with attic surface heat transfer rates is recommended.

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**Economics and Energy Impact** 

Merle F. McBride

THE USE OF ECONOMIC ANALYSIS IN DEVELOPING AN ENERGY STANDARD: LESSONS LEARNED

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ABSTRACT: Development of energy standards in the early 1970's was initiated and accelerated by the "energy crisis". Criteria were proposed based on "professional judgement" and accepted when "consensus" was reached. That process performed a valued service but the conditions have since changed. Today, energy conservation is not a single focus point. It is constantly being challenged as to whether it is cost-effective. In that environment development of new energy standards has incorporated economic analysis as a basis for setting the criteria. The objective of this paper is to present the economic basis used to develop the proposed ASHRAE Standard 90.2P. traditional life-cycle The approach analysis was simplified by development of scalar ratios as a new A national energy model was developed which technique. integrated scalar ratios to illustrate the interdependence between economic assumptions and energy savings. This entire process was new and innovative, which challenged prior concepts and conventional procedures. Many lessons were learned and their influence on the standard development will be highlighted.

KEYWORDS: energy, standard, economics, scalar ratio

## INTRODUCTION

The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) proposed Standard 90.2P "Energy Efficient

Dr. McBride is a Research Associate at Owens-Corning Fiberglas Corporation, Technical Center, 2790 Columbus Road, Rt. 16, Granville, Ohio 43023-1200. Design of New Low-Rise Residential Buildings" [1] represents a significant departure from previous energy standards [2] in approach as well as format. The purpose of this paper is to present the economic procedures used to develop this proposed standard and the lessons learned in implementing those procedures.

## BACKGROUND

Historically, ASHRAE developed their residential energy standard using "professional judgement" as the basis for setting the criteria. The criteria was accepted when "consensus" was reached. The criteria was presented as U-values versus heating degree days (HDD). This format explicitly ignored cooling. Trade-offs were allowed assuming equal energy performance was achieved. This approach was challenged as subsequent addenda were proposed because the criteria was not cost-effective. Consequently, when the standard was due for a total revision, the basis was changed to incorporate economics directly to ensure the criteria was cost-effective.

## ECONOMICS

Implementation of economics as the basis of the proposed standard began by deciding on certain basic principles. The specific methodology then grew out of these principles. Next, detailed cost data was developed and utilized in the methodology. However, the economics could not stand alone and the energy consequences of the economic assumptions had to be understood. This was resolved with a national energy model. Then specific economic data were available and utilized in the methodology. This led to specific criteria as the basis for the standard, but the results were too voluminous. Many simplifications and compromises were necessary to reduce those results to a form that was acceptable as a standard.

# Basic Principles

There were three basic principles underlying the economics used to develop the standard. Each will be briefly reviewed. The first principle was that the economics would be representative of the consumer as opposed to societal economics. Societal economics would have significantly increased the criteria stringency and encountered heavy resistance for adoption. The second principle was that marginal or incremental analysis would be used. This meant that each step. upgrade, or improvement must satisfy the economic criteria in order to be called cost effective. The third principle was that the energy savings attributed to the criteria must be greater than or equal to the first costs required to achieve the specific construction. Simply stated, this means that the incremental energy savings (over some time period) associated with an increased level of conservation would be equal to or greater than the incremental first cost of materials necessary to achieve that level of conservation. Once these basic principles were defined, a specific methodology was developed.

## Specific Methodology

The specific economic methodology can be stated as:

Incremental Energy Cost Savings Over Time ≿ (1) Incremental First Cost of Materials

In equation form, it is:

$$FYS_{h} \cdot P_{h} \cdot S_{h} + FYS_{c} \cdot P_{c} \cdot S_{c} \ge \Delta FC \cdot S_{2}$$
(2)

where:

FYS	=	First Year Energy Savings for Heating, kJ/m <sup>2</sup>
Р. "	=	Price of Fuel for Heating, \$/kJ
Sh	=	Economic Scalar for Heating, dimensionless
FŸS	=	First Year Energy Savings for Cooling, kJ/m <sup>2</sup>
P	=	Price of Fuel for Cooling, \$/kJ
Sč	=	Economic Scalar for Cooling, dimensionless
∆řc	=	Incremental First Cost of Materials, \$/m <sup>2</sup>
S <sub>2</sub>	=	Economic Scalar for Materials, dimensionless

Economic scalars,  $S_h$  and  $S_c$ , are simply the "modified" uniform present worth factors corresponding to the time horizon (life), discount rate and fuel escalation rates assumed. (Modified means that the dollar value of energy savings may not be constant from year to year, but increases over time at some projected rate of change.) The economic scalar,  $S_2$ , is the ratio of the down payment and present value of all mortgage payments (adjusted for income tax savings) to the first cost of the modification to which these payments apply. The distinct advantage of the economic scalar methodology is the ability to represent various cost-effectiveness criteria in a consistent set of equations.

On a unit area basis the first year heating savings is:

$$FYS_{h} = \frac{\Delta U H LF H DD}{AFUE H DE}$$
(3)

where:

ΔU	=	Incremental Change in U-value for Materials, W/m <sup>2</sup> ·K
HLF	=	Heating Load Factor, hr/day
HDD	=	Heating Degree Days Base 18.3°C, °C·day
AFUE	=	Annual Fuel Utilization Efficiency, dimensionless
HDE	=	Heating Distribution Efficiency, dimensionless

This equation can be modified to apply to other types of heating systems by substituting the appropriate overall coefficient of performance for the AFUE variable. For example, a heat pump could be analyzed by substituting the heating season performance factor (HSPF) for the AFUE. One would also have to incorporate the appropriate conversion factors at the same time.

On a unit area basis the first year cooling savings is:

$$FYS_{c} = \frac{\Delta U \cdot CLF \cdot CDH}{SEER \cdot CDE}$$
(4)

where:

CLF	=	Cooling Load Factor, dimensionless
CDH	=	Cooling Degree Hours Base 23.3°C, °C hr
SEER	=	Seasonal Energy Efficiency Ratio, dimensionless
CDE	=	Cooling Distribution Efficiency, dimensionless

Combining Eqs 2, 3 and 4 produces:

$$\frac{\Delta U \cdot HLF \cdot HDD \cdot P_h \cdot S_h}{AFUE \cdot HDE} + \frac{\Delta U \cdot CLF \cdot CDH \cdot P_c \cdot S_c}{SEER \cdot CDE} \geq \Delta FC \cdot S_2$$
(5)

Eq 5 represents the complete equation used to set the criteria. Conversion factors have been intentionally omitted from all equations.

#### Criteria Equations

Starting with a specified increase in energy conservation, for example, a change in insulation levels, the first costs were fixed and the unknowns became the climate, both HDD and CDH, that were necessary to satisfy Eq 5. These heating and cooling values were easily determined by solving for each variable independently. The calculated values of HDD and CDH are called intercepts and appeared on the criteria curves (CDH vs HDD) as the points where the lines intersect the horizontal and vertical axes. The heating equation is:

$$HDD = \frac{\Delta FC \cdot AFUE \cdot HDE}{\Delta U \cdot HLF \cdot P_h \cdot \frac{S_h}{S_2}}$$
(6)

where:

and the cooling equation is:

$$CDH = \frac{\Delta FC \cdot SEER \cdot CDE}{\Delta U \cdot CLF \cdot P_c \cdot \frac{S_c}{S_2}}$$
(7)

where:

 $S_c/S_2 =$  Economic Scalar Ratio (SR<sub>c</sub>) for Cooling, dimensionless

## <u>Implementation</u>

In order to calculate the criteria lines, specific values had to be substituted into Eqs 6 and 7. Key issues relating to the development and application of the variables used in these equations is presented below.

<u>Incremental first costs</u>: National average costs for materials were used. An extensive cost data base was prepared for single family, multi-family and manufactured (mobile) homes [3]. The data base consisted of costs by eleven regions across the United States plus the national average for each material. Over time, it was revised and expanded by committee members and trade organizations. For purposes of the standard's development, national average incremental costs for materials were used, see Table 1 for ceiling insulation and truss prices.

Table 1 -- Statistics on incremental ceiling prices

Material	Average \$/m <sup>2</sup>	Std. Dev. \$/m <sup>2</sup>
Ceiling Insulation (Blown): R-0 to R-1.94 m <sup>2</sup> ·K/W R-1.94 to R-3.35 R-3.35 to R-5.28 ·R-5.28 to R-6.69	2.88 1.19 1.70 1.51	0.49 0.43 0.72 0.49
Ceiling Truss Framing Standard to Raised	2.03	1.29

<u>Envelope constructions</u>: Each envelope component of a residence (ceiling, walls, floors, fenestration, foundations, etc.) was identified along with the various forms (flat vs. cathedral ceilings; wood, metal and masonry walls; etc.) and the typical insulation levels. Every combination was calculated using Eqs 6 and 7 and then an optimization technique selected the specific combinations that met the economic criteria [4,5].

A standard truss was assumed to be the base case for a flat ceiling. The standard truss was changed to a raised truss when the insulation resistance exceeded a value of  $5.28 \text{ m}^2$ K/W. The increased truss price was then included in the economic analysis.

<u>Heating and cooling equipment efficiencies</u>: The efficiencies used in Eqs 6 and 7 were the minimums defined by the 1987 Federal law [6]. The heating system was a central, forced-air gas furnace with an annual fuel utilization efficiency (AFUE) of 0.68 and the cooling system was a central, forced-air, electric split-system with a seasonal energy efficiency ratio (SEER) of 8.5. These efficiencies were the minimums that could be manufactured and sold in 1987 according to the federal law.

Distribution efficiencies: The energy losses due to warm-air distribution systems are significant. The losses occur by conduction through the duct walls and air leaks at joints and seams. These losses vary depending upon their location (attic, basement, crawl space, garage, roof, etc.), the level of insulation, their size and the type of heating system. The ranges presented in the proposed standard for heating are 0.78 to 0.91 while cooling ranges from 0.74 to 0.92 [7,8]. These ranges reflect the air distribution in different locations, with different insulation levels and in various building types. Complete statistics are presented in Table 2. For purposes of developing the criteria in the standard, the air distribution system was assumed to be in an unconditioned basement and the heating distribution efficiency (HDE) was set at 0.75 while the cooling distribution efficiency (CDE) was set at 0.80.

Table 2 -- Statistics on distribution efficiencies

Item	Average	Std. Dev.
Heating Distribution Efficiency	0.82	0.029
Cooling Distribution Efficiency	0.84	0.043

Since it is common for single family residences to have their distribution systems located completely within conditioned spaces, alternate envelope criteria were developed with the heating and cooling distribution efficiencies set equal to one.

This doubled the number of criteria figures for single family residences.

Load factors: The heating and cooling load factors (HLF and CLF) were developed as a simplified methodology to estimate heating and cooling load savings through envelope components [9]. Heating and cooling loads refers to the heat loss or cooling gains through the envelope components on a seasonal basis. Load factors for above-grade envelope components were developed in a two step process. First, load savings were determine for specific construction options by analyzing seven different house types in 45 climates [10,11]. The load savings analyses were completed by the Lawrence Berkeley Laboratories utilizing their Residential Energy Analysis Program (PEAR) [12]. The heating load savings were expressed as the savings per degree day for a specific construction change. Next, the load factors were calculated as the slope of the line relating the savings per degree day to the incremental change in U-values. An identical approach was also utilized to develop the cooling load factors except degree hours were used instead of degree A similar two step approach was used to develop the load days. factors for all foundation systems. Load factors for all foundations were developed by Christian and Strzepek [13] utilizing the results of a two-dimensional finite difference analysis completed by Shipp [14]. The distinct advantage of the load factor approach was the ability to analyze any incremental construction change instead of being limited to just the specific constructions originally analyzed. Once the load factors were calculated they were assumed to be constant for each component across all climates.

Fuel prices: The fuel prices used were marginal fuel prices which correspond to distinct steps in the rate schedules. Natural gas prices for 132 locations were identified in five steps or blocks of usage so the correct incremental value for winter could be used [3]. The gas prices range from 25.2 to 208 \$/Million kJ. Electric prices for 104 locations were identified in six steps of usage so the correct incremental values for summer could be used [3]. The electric prices range from 0.0164 to 0.154 \$/kWh. Publicly and privately owned electric utilities were included. These data were used to calculate a national average. Next individual state averages were calculated and then the average for all the states was determined. The statistics are presented in Table 3. When the data were reviewed questions arose on selected cities so an independent survey was completed for 73 locations and those results were weighted by housing starts [15], see Table 3. The difference between the national average and the housing start weighted average gas price was 7.2% while the electrical difference was 2.6%. These differences in national averages were not statistically significant using a rejection criteria at the 2.5% level. The weighted results were subsequently used to develop the proposed standard.

Fuel Type	Average	Std. Dev.
Natural Gas: \$/Million kJ		
National	5.99	1.22
States	5.96	0.61
Weighted by Housing Starts	5.56	1.30
Electricity: \$/kWh		
National	0.0766	0.0229
States	0.0732	0.0174
Weighted by Housing Starts	0.0786	0.0220

Table 3 -- Statistics on fuel prices

Scalar ratios: Scalar ratios (SR) are simply the scalars for heating  $(S_h)$  and cooling  $(S_c)$  energy savings divided by the scalar for the materials  $(S_2)$ . Development of the scalar ratio concept significantly simplified the use of economics in preparing the proposed standard [4]. The scalar ratio is a combination of life, discount rates, fuel escalation rates. federal and state tax rates, down payment amount, and loan placement fee ("points"). Three distinct cost-effectiveness criteria were investigated and they were; (1) two years to positive annual cash flow, (2) seven years to break even, and (3) full life-cycle cost (LCC) analysis with discounting of cash flows. The LCC approach was evaluated for economic lives of 20 and 30 years. The results of sensitivity analyses on the models and key variables are presented in Table 4 and vary depending upon the fuel type. The advantage of the scalar ratio concept was that many combinations could lead to the same result. This was a tremendous advantage because each variable did not have to / defended. However, there was also a Scalar ratios did not indicate directly the be individually defended. disadvantage. relative energy savings that would occur. Thus, a relationship was needed between scalar ratios and energy savings. This was resolved with the national energy model.

		Env	velopes			
Fuel	Cross Over	Cash Flow	Life	Cycle <sup>1</sup>	Mort- gage	Fuel Escal.
Туре	2 yr	7 yr <sup>2</sup>	20 yr	30 yr	Rate	Rates
Elec.	12.61	13.05	14.26	19.01	12%	Varv <sup>3</sup>
Gas	12.27	13.60	17.20	23.74	1.2%	Vary <sup>4</sup>
Elec.		15.49	15.95	21.94	10%	Vary <sup>3</sup>
Gas		16.45	19.23	27.40	10%	Vary <sup>4</sup>
Elec.	16.09	16.39	16.91	23.69	9%	Vary <sup>3</sup>
Gas	15.65	17.08	20.39	29.59	<b>9</b> %	Vary <sup>4</sup>
A11		10.64	10.07	10.13	10%	Zero⁵
A11		11.52	11.75	14.18	10%	2.00%
A11		12.48	13.83	17.51	10%	4.00%
A11			14.12	18.00	10%	4.25%)
A11			14.43	18.51	10%	4.50%°
A11			15.05	19.60	10%	<b>5.00%</b> )
A11		13.53	16.41	22.06	10%	6.00%°
Elec.			13.86		11%	Vary <sup>3</sup>
Gas			16.61		11%	Vary <sup>4</sup>
A11			12.34		11%	4.25%°
			HVAC	Equipme	ent	_
			1	5 yr		
Elec.	10.28	10.49	ī	1.40	12%	Vary <sup>3</sup>
Gas	10.00	10.93	1	3.18	12%	Vary4
			Domesti	c Hot W	later	
				10 yr		
Elec.	8.54	8.26		8.14	12%	Varv <sup>3</sup>
Gas	8.31	8.61		8.87	12%	Vary <sup>4</sup>
Notes:			-			
1- Dis	count r	ate = 1	0% (aft	er-tax	equiva	lent)
2- Zer	o disco	unt rat	e and 3	0 year	mortga	ge Ø 7 10%

Table	4		Scalar	Ratio	Sensitivity	Analysis
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2- Zero discount rate and 30 year mortgage 3- 5 yrs @ 4.90%, 10 yrs @ 6.26%, 15 yrs @ 7.13% 4- 5 yrs @ 6.18%, 10 yrs @ 9.38%, 15 yrs @ 6.77% 5- No fuel escalation

6- Constant fuel escalation rates

## National Energy Model

The national energy model was a computer simulation of a typical residence located in 73 cities. The residence was a single story floor plan with the ceiling and floor equal to  $139.7 \text{ m}^2$ , the opaque wall area was 107.3 m<sup>2</sup> and the fenestration was 13.1% of the floor area. The fenestration was uniformly distributed on all four orientations and there were no overhangs or external shading. The house contained a warm-air gas furnace and central air conditioning. The analysis consisted of selecting a specific scalar ratio, developing an energy standard using Eqs 6 and 7 and national average prices, calculating the annual energy consumption, and weighting the results by the housing This process was repeated by incrementing the scalar ratio starts. from 2 to 30. The results are presented in Fig. 1. For comparisons, the Department of Housing and Urban Development (HUD) Minimum Property Standards (MPS) [16] for gas and electric resistance were superimposed. Also, the current ASHRAE Standard 90A-1980 was superimposed for comparisons. They appear as horizontal lines since they do not change with the scalar ratio. This analysis established a relationship between the national energy consumption, the scalar ratio and current standards.

Prior to this analysis the ASHRAE 90.2 committee had proposed that the standard should be 25 percent more stringent than the current standard [2]. This percentage of stringency was reached by consensus. Various percentages were discussed. Those below 25 percent seemed too lenient and not worth the effort to revise the standard if the impact was below that level. Levels above 25 percent seemed too stringent since the standard is revised on a five year cycle.

Using the 25 percent reduction, a horizontal line was drawn on Fig. 1 that incidently coincided with the HUD electric resistance criteria. This line intersected the proposed standard curve at a scalar ratio of 18. This value and national average prices were used to set the final envelope criteria for the proposed standard.

#### Envelope Criteria Figures

The format for the envelope criteria is presented in Fig. 2. The horizontal axis is heating degree days and the vertical axis is cooling degree hours. The diagonal lines represent regions or bands with a constant U-value as the criteria. To use the figure one simply locates the city of interest by the two weather variables and determines which band applies.

## <u>Uncertainty Analysis</u>

Although national average prices were used in Eqs 6 and 7 to develop the proposed standard there were many challenges to that approach. One



means to address the variation that arose was to complete an uncertainty analysis on the basic equations according to the root-sum square formula [17,18]. The basis for the uncertainty analysis used

was one standard deviation (sd) from the average. In equation form this becomes:

$$HDD_{sd} = \sqrt{\left(FC_{sd}\frac{\partial HDD}{\partial FC}\right)^2 + \left(P_{h_{sd}}\frac{\partial HDD}{\partial P_h}\right)^2 + \left(SR_{sd}\frac{\partial HDD}{\partial SR}\right)^2 + \left(HDE_{sd}\frac{\partial HDD}{\partial HDE}\right)^2}$$
(8)

Standard deviations on all the variables are presented in Tables 1-3. Standard deviations on the scalar ratios (SR) were approximated as onefourth of the ranges [19] from Table 4. These statistics were used to determine the standard deviation that occur in calculating the heating and cooling intercepts (HDD and CDH) for ceilings with attics assuming that the equipment efficiencies, incremental U-values and load factors were constant. The results are presented in Table 5 for first costs, scalar ratios, fuel prices and distribution efficiencies plus their combinations. In general, ranking these items shows that the first costs contribute the most uncertainty, followed by scalar ratios, then fuel prices and last was distribution efficiencies. When items are combined their impact is not simply additive so the combined standard deviations are not significantly larger than the single largest one.

Table 5 -- Uncertainty analysis of one standard deviation on heating intercepts for ceilings with attics

Item	U=0.460	U=0.290	U=0.204	U=0.148
HDD Intercept	223	725	2014	5342
First Costs	52	365	1158	2960
Scalar Ratio	74	244	665	780
Natural Gas	31	104	282	331
Distribution Efficiency	11	35	96	112
First Costs and Gas	61	380	1193	2978
First Costs, Gas and SR	96	451	1366	3079
First Costs, Gas, SR, and Dist. Eff.	97	453	1369	3081

# DISCUSSION OF RESULTS

A new and innovative economic methodology called scalar ratios has been developed and utilized to prepare the proposed ASHRAE residential energy standard. The underlying principles were based on consumer economics, incremental or marginal costs and a cost effective criteria that required all energy savings to be equal to or greater than the first costs. This was a departure from previous energy standards which did not incorporate economics directly into their development.

Implementation of the economic methodology required extensive cost data bases on construction materials and fuel prices. Using national average prices allowed the standard to be simplified. Heating and cooling load factors were developed as a simplification to predict the heating and cooling energy savings. Distribution efficiencies were set as representative values and equipment efficiencies followed the federal law requirements.

Controversy surrounded the use of national average fuel prices because the range was large. Alternate methods were used to calculate a national average and the results are presented in Table 3. One approach was to use housing starts as weighting factors. This lowered the gas price by seven percent while the electric price increased by three percent. Another approach was to average the individual states. This led to a one percent reduction for gas and a four percent reduction for electric in the national averages. These were not considered major changes.

There was a strong argument for using national average fuel prices in the prescriptive portion of the standard because local fuel prices were used in an alternative compliance path. The proposed standard included a trade-off procedure which was called the annual energy cost (AEC) method. Compliance in this path was achieved when the AEC of a proposed design was less than or equal to a residence that meets the prescriptive criteria. In the AEC method local fuel prices were mandatory in order to demonstrate compliance so a user of the standard does have an option to use local fuel costs.

Scalar ratios were developed for three economic models. The advantage of the scalar ratio concept was that it avoided the problem of having to defend the value of each variable that defines the economics. Instead, a single value was used which could be justified under many different assumptions. Significant confusion surrounded the scalar ratio of 18. Challenges were initiated that it was too stringent economically. A value of ten was thought to be more realistic. Others thought that 12-13 should be an upper limit. What people failed to recognize was that the value of 18 was the economic criteria for a cutoff. It was not the operational value. This is best explained by reviewing Fig. 3 where the scalar ratios for four ceiling constructions are presented. The first curve, U=0.460, begins at a scalar ratio of 30 and continues to decrease. At a scalar ratio of 18 that construction meets the economic criteria and that defines the heating intercept at 223 heating degree days. Between 223 and 725 the scalar

ratio continues to decrease to a value of about four until the next construction meets the criteria of 18. What this means is that the average or effective or operational scalar ratio is not 18 but must be something much lower. Similar trends exist for all envelope components as shown in Figs. 4-6. In order to actually determine what the operational scalar ratio was for the United States, 3349 locations were investigated and the average was 9.8 when ceilings, walls, floors and fenestrations were combined. The absolute maximum combined value (a simple average of all four components) for any location was 13.9. This clearly demonstrated that the economics were not too stringent but follow almost exactly the expectations of those who initially challenged the use of a scalar ratio of 18.









A national energy model was integrated with the economic methodology to ensure the proposed standard would reduce the national energy consumption of new residences by 25 percent relative to the current standard. This integration set the final scalar ratio at a value of 18. Then all the criteria were calculated and defined in figures that explicitly incorporated cooling as well as heating for the first time in an ASHRAE residential energy standard.

An uncertainty analysis was completed which quantified the variability that arises when using national average values. One standard deviation in each key variable was investigated along with their combinations. The variability was greatest with first costs, followed by the scalar ratio, the fuel price and finally the distribution efficiency. The variability that arose when using national averages was shown to be acceptable for setting a national standard. This was a key point of discussion in developing the standard and may appear to be a broad generalization but there were no options. If national averages were not used, then regional, state or local values would be used which would not lead to a national standard.

Application of the entire methodology led to the development of 128 figures. Every envelope component had multiple figures. For example, ceilings had figures for flat and cathedral while walls had figures for wood frame, log, and masonry constructions. The masonry was further divided into three insulation locations: exterior, interior and integral. Similar figures were defined for below-grade constructions. Basement and crawl space insulation locations were on the interior,

integral or exterior to the structural wall. Fenestration was separated into windows, sliding glass doors and skylights.

Although this approach strictly adhered to the prescribed economic methodology, it was considered too voluminous to be proposed as a standard. Many of the figures were consolidated. Assumptions and compromises were made to significantly reduce the number of criteria figures. Some of the compromises were straight-forward while others were complex and required more effort to combine and simplify. For example, all fenestration criteria were defined to be equal and could then by presented on a single figure. Foundation criteria were simplified by combining all weather wood foundations (AWWF) with masonry. The argument for this consolidation was unique. Since AWWF have large open cavities, the insulation levels were significantly higher than those for masonry because it is very cost effective to justify filling the cavity. However, masonry walls are expensive to insulate so the insulation levels are significantly less. The perception was that people would think AWWF are less energy efficient than masonry because AWWF required more insulation. The cost effective economic criteria were satisfied in both cases but the energy saving and insulation levels were not equal. The final criteria were a combination of AWWF and masonry with exterior or integral insulation, the specific criteria are presented in Ref. 13. Masonry with interior insulation remained a separate figure.

The format of the criteria figures also raised objections. Critics challenged it as stifling innovation. The discrete bands or regions that define the criteria could all be met using existing technology. Furthermore, the bands were considered either too broad on some figures or too narrow on others. These objections failed to consider the simplified trade-off procedure contained in the standard. Trade-offs of all envelope components are permitted provided their combined performance does not increase the overall energy consumption relative to the prescriptive criteria. This allows virtually an infinite number of options to demonstrate compliance and by no means limits innovation.

## CONCLUSIONS

The economic concept of scalar ratios was developed and shown to be a viable approach to develop a national energy standard. It was a new and innovative technique that can be applied to future standards provided the prices that change with time are updated. However, scalar ratios can not stand alone as the basis for the criteria. It was shown through the national energy model that the energy consequences of the economic assumptions must be thoroughly understood before the criteria can be actually finalized. Even then, implementing the basic methodology produced too many criteria figures so that simplifications had to be made in order to produce a standard that was reasonable in size and complexity. The impact of key variables on the final criteria was evaluated through an uncertainty analysis. It showed that the variability was still acceptable to use in setting the criteria for a national standard.

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Bruce D. Nelson

ENERGY CODE MEASURES TO ASSURE THE EFFECTIVENESS OF THERMAL INSULATION INSTALLED IN BUILDINGS

REFERENCE: Nelson, B. D., "Energy Code Measures to Assure the Effectiveness of Thermal Insulation Installed in Buildings," <u>Insulation Materials: Testing and Applications.2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Several common residential construction details are detrimental to the performance of installed thermal insulation. Degradation of insulation performance is caused by air exfiltration (convective losses and moisture gain in insulated cavities) and wind wash (the passage of outdoor air through insulation).

Existing model energy codes in the United States address the exfiltration issue, although enforcement appears to be only partially effective. U.S. model energy codes do not address wind wash. New provisions of the Minnesota Energy Code for residential buildings are intended to require construction details that would reduce exfiltration and wind wash. The language addressing these construction details is generally more prescriptive than in existing model energy codes.

The inclusion of language in residential energy codes to improve the performance of thermal insulation follows the lead of special programs' building standards for energy efficient construction.

KEYWORDS: air leakage, code, energy conservation, moisture control, thermal insulation, residential

The primary purpose for an energy code is to assure that buildings are constructed so that they can be operated in an energy efficient manner. Model energy codes used in the United States stipulate maximum permitted thermal transmittance through exterior envelope components. Enforcement of these codes is done under the assumption that installed insulation will perform as indicated by laboratory measurement. Model energy codes also require control of air leakage by stipulating that exterior joints in the building envelope be sealed.

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Studies of the effectiveness of installed thermal insulation and air leakage control in residential buildings indicate that enforcement of energy codes has neither assured effectiveness of thermal insulation nor controlled air leakage. Standards and regulations are lacking on the subjects of residential ventilation and air tightness [1]. Uncontrolled air leakage results in a significant amount of energy loss as well as building structural damage [2]. Wind wash (the passage of outdoor air through insulation) reduces the effectiveness of thermal insulation, resulting in excess heat transfer and the potential for condensation during the heating season on interior cold surfaces [3].

Modifications to energy codes may be able to change construction practices to reduce exfiltration and wind wash. The effect of the code on building energy performance would thereby be improved. Air leakage measurements of buildings constructed under more stringent energy standards do indicate these buildings are tighter [4]. However, air tightness has not been correlated with specific requirements of the more stringent standards.

## Building Practices that Allow Air Flow through Insulation

Thermal insulation functions in part by restraining air (or other low conductivity gas) within its cells or fibers. Thermal insulation effectiveness is reduced if air from external sources replaces air held by the insulation [5]. This is not a failure of the insulation per se, but failure of the building system in which the insulation material is installed.

## <u>Exfiltration</u>

<u>Problems</u>: Exfiltration of conditioned air into insulated cavities can cause several problems. The loss of the energy embodied in the conditioned air is most obvious. Convective transport of water vapor during the heating season results in energy loss and may also cause problems due to moisture accumulation.

Sherwood [6] identified exfiltration induced water vapor air leakage as a major contributor to moisture condensation in walls. Sherwood found that puncturing the vapor retarder, such as with an electrical outlet, resulted in considerable condensation on the sheathing in the region of the puncture.

In case studies of three Canadian homes, Scott [2] identified water vapor transport into bathroom walls by exfiltration as the cause of enormous moisture damage. The source of the problems was identified as the absence of a barrier in concealed areas under the tub to prevent moist air movement from the house into the walls.

Exfiltration into attics was also a problem in one of Scott's case study houses. Field studies at Princeton University in the late 1970's [7] and in Minnesota in the early 1980's [8] revealed that passages for air convection into attics are common. All three of the previously cited studies traced moisture accumulation in attic spaces to air convection from living spaces. Air passages into attics were found at oversized holes for plumbing stacks, at oversized electrical feed holes, at dropped ceilings, at ceiling light fixtures, and at unsealed duct work. Air passages were also found where heating system and exhaust vent duct work enter the attic and where interior partition walls intersect the ceiling.

<u>Solutions</u>: Preventing air exfiltration into insulated cavities in new residential construction is neither difficult nor expensive. Although Scott [2] concludes, "Without adequate and timely inspection, the construction process soon conceals the evidence (of passages allowing for exfiltration), making future discovery and correction difficult and costly." Several builders' manuals for energy efficient residential construction provide several simple recommendations for preventing exfiltration [1, 3, 9, 10]. The vapor retarder should be continuous behind bathtubs and showers on exterior walls. A barrier should be installed at this location to prevent air movement from the house into the wall. An interior barrier should be installed to prevent air from entering insulated cavities at plumbing, electrical and air duct penetrations of the ceiling and wall. All air leakage pathways into the attic should be sealed.

#### <u>Wind Wash</u>

<u>Problems:</u> The passage of outdoor air through thermal insulation diminishes the effectiveness of the insulation by replacing entrapped air with unconditioned air. This effect has been referred to as "wind wash," "blow-through," [3] or "wind-driven air" [9]; we have chosen to use the first term. Lstiburek [3] has explained how wind wash commonly occurs where exterior perimeter walls meet insulated ceilings and at exterior corners (see Fig. 1). Lstiburek asserts that wind wash is most often responsible for excessive heat loss during the heating season at exterior wall/ceiling corners. This results in cool interior surfaces creating conditions susceptible to condensation and mold growth.

Recent Canadian research [11] indicated that wind wash was responsible for large decreases in ceiling perimeter R-value with a strong dependance on wind speed and direction. When cardboard inserts were placed to prevent wind wash there was an immediate increase in the effective thermal resistance. The study of Minnesota homes built



FIG. 1 - Wind wash at the edge of attic insulation



FIG. 2 - Extra sheathing provides a barrier to prevent wind wash

as part of an energy efficient demonstration project in 1980 [8] identified wind wash as the reason for cold ceiling/exterior wall corners.

<u>Solutions:</u> The Canadian research concluded that wind wash in ceiling/exterior wall corners can be minimized by installing an impermeable barrier at the edge of the attic insulation [10] (see Fig. 2). Recommendations for builders of energy efficient homes [1, 3, 9, 10] include attic exterior perimeter baffles and placement of an air barrier material at exterior corners to prevent wind wash of thermal insulation.

#### Treatment by U.S. Model Energy Codes

Most energy codes in the Unites States today are based on the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard 90A-1980 [12], or the Council of American Building Officials (CABO) Model Energy Code [13] which is itself based on ASHRAE Standard 90A-1980.

Both the current ASHRAE Standard 90A-1980 and the CABO Model Energy Code include the following paragraph pertaining to air leakage reduction:

"Exterior joints in the building envelope that are sources of air leakage, such as around window and door frames; between wall cavities and window or door frames; between walls and foundations, between walls and roof/ceilings and between wall panels; openings at penetrations of utility services through walls, floors and roofs; and all other such openings in the building envelope shall be caulked, gasketed, weatherstripped or otherwise sealed in an approved manner."

This same paragraph appears in the first U.S. model energy standard published in 1975, ASHRAE Standard 90-75. Neither current ASHRAE Standard 90A-1980, nor the CABO Model

Neither current ASHRAE Standard 90A-1980, nor the CABO Model Energy Code contain requirements addressing the problem of wind wash.

#### Building Standards that Address Effectiveness of Insulation

#### <u>Exfiltration</u>

<u>Residential Standards Demonstration Program (RSDP)</u>: The single family design specifications of the RSDP [14] contain three, progressively more stringent infiltration control packages. Infiltration control package A (the least stringent) includes all of the specific requirements for air leakage reduction that are included in ASHRAE Standard 90A-1980. Additionally, package A prohibits recessed lights in exterior ceilings, requires gaskets on all electrical boxes on both exterior and interior walls, and has requirements to prevent exfiltration with fireplaces, wood stoves and exhaust systems. Infiltration control package B requires, in addition to the elements of package A, a continuous air infiltration barrier with specific criteria. Infiltration reduction package C further tightens requirements for windows.

<u>Super Good Cents</u>: The Super Good Cents specifications adopted by the Bonneville Power Administration [15] also addresses air leakage more extensively than the ASHRAE Standard 90A-1980. The basic air tightness criteria for this standard requires attainment of certain tightness as measured by a fan depressurization test (ASTM E 779-81) or a tracer gas test (ASTM E 741). Alternatively, the house must comply with a prescriptive requirement similar to the RSDP infiltration control package A. An exception is that recessed light fixtures are permitted, provided that they are much less leaky than standard fixtures.

An advanced air leakage control criteria specified by the Super Good Cents requires attaining a tighter structure as measured by the fan depressurization or tracer gas test. Following prescriptive criteria similar to RSDP infiltration reduction package B is also permitted.

The States of Washington and Oregon have adopted the Super Good Cents air leakage criteria as part of their state energy codes.

<u>Minnesota Energy Code</u>: In 1991, the Minnesota Energy Code [16] was amended to address exfiltration and wind wash in residential buildings. The Minnesota code adopts and amends the CABO Model Energy Code, 1989 edition.

Since its inception in 1976, the Minnesota code has required a vapor retarder in low-rise residential buildings "to maintain the thermal performance of required insulation and the integrity of building materials against cold weather vapor condensation." The 1991 amendments added two requirements designed to prevent exfiltration:

"A barrier against air leakage must be installed to prevent the leakage of moisture-laden air from the house into the building envelope. An air barrier must be continuous at all plumbing and heating penetrations of the interior surface of the building exterior envelope. If a tub or shower is located on an exterior wall, an air barrier must be provided at the interior surface of the building exterior envelope behind the tub or shower."

and

"Fire stops must be installed in accordance with the State Building Code. When mineral fiber or glass fiber materials are used as fire stop construction at ceilings and wall cavities separating conditioned and unconditioned spaces, the fire stop must be installed to block air movement."

The new Minnesota requirements specifically address many of the problems identified by research cited earlier.

At first reading, it may seem inappropriate for the energy code to add to the requirements for fire stops. This phrase was chosen because, in discussions with Minnesota builders and building officials, it was indicated that they consider many areas where exfiltration occurs as being the same as a fire stop. Thus, the code uses this familiar term.

The 1991 Minnesota Energy Code also added a performance alternative for air leakage based on ASHRAE Standard 119-1988. This standard, titled "Air Leakage Performance for Detached Single-Family Residential Buildings," prescribes air tightness performance tests and classifications of test results. The classifications are given letter identifiers, with class "A" being the tightest.

Also incorporated into the Minnesota Energy Code is ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality." This standard prescribes a ventilation rate for residential buildings, but notes "(this) ventilation is normally satisfied by infiltration and natural ventilation." Since the intention of the code requirement is to limit exfiltration, it was felt that infiltration and natural ventilation may not provide adequate ventilation in the houses built under this alternative approach. Some assurance, we felt, should be provided so that homes built under the provisions of Standard 119-1988 would be ventilated.

Since the use of Standard 119-1988 is an optional alternative for compliance with the code, builders will not necessarily be burdened with the addition of language to assure that homes built under these provisions are ventilated:

"As an alternative to the prescriptive requirements of (the code) for detached single-family residential buildings, air tightness must comply with air leakage class A, B, C, or D of ANSI/ASHRAE Standard 119-1988. In addition, this alternative requires that the ventilation system must provide a ventilation rate of not less than 0.35 air changes per hour (determined in accordance with ASHRAE Standard 62-1989, Table 2.3) or 15 cfm per person, whichever is greater. This ventilation rate must be verified by measurement."

#### Wind Wash

<u>Residential Standards Demonstration Program (RSDP):</u> The RSDP single-family design specifications [14] addresses wind wash in attic spaces with the inclusion of the sentence, "Adequate baffling of the vent openings shall be provided so as to deflect the incoming air above the surface of blown or poured insulation." The States of Washington and Oregon have recently amended their state energy codes to include language addressing wind wash that is identical to the RSDP.

<u>Minnesota Energy Code</u>: The 1991 Minnesota Energy Code [16] defines wind wash as "the passage of unconditioned air through thermal insulation of the building envelope." It goes on to address wind wash at two locations within residential buildings:

"A barrier must be provided at the following locations to mitigate wind wash:

A. the exterior edge of attic insulation; and

B. cantilevered floors and bay windows, including corners with adjoining walls above and below."

Both the RSDP and Minnesota requirements are quite specific about what must be done to prevent wind wash. This is to assure that building officials will have clear requirements available to them in their efforts to enforce this measure.

#### **Conclusions and Recommendations**

The effectiveness of installed thermal insulation in new residential construction could be improved with construction practices that reduce exfiltration and wind wash. To be truly effective, an energy code must not only prescribe the maximum permitted U-value, but also measures that will assure the effectiveness of the installed thermal insulation.

Buildings constructed under the air leakage requirements of current model energy codes have been found to have significant air leakage problems. Energy standards associated with special energy efficiency programs (RSDP and Super Good Cents) prescribe measures to reduce exfiltration and reduce wind wash in considerably more detail than do other model energy codes used in the United States today. Recent amendments to the Minnesota Energy Code are an attempt to specifically address exfiltration and wind wash. These requirements use very prescriptive language to assure that builders will understand what is expected of them.

An attempt has begun to assure the effectiveness of thermal insulation performance in residential buildings by adding provisions to energy codes that will prevent exfiltration and wind wash. If success is demonstrated, these requirements should be incorporated into national model energy codes.

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# Robert A. Penney and David W. Yarbrough

# A SURVEY OF LOOSE-FILL INSULATIONS INSTALLED IN RESIDENTIAL ATTICS

REFERENCE: Penney, R. A. and Yarbrough, D. W., "A Survey of Loose-Fill Insulations Installed in Residential Attics," <u>Insulation Materials: Testing</u> <u>and Applications, 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

**ABSTRACT:** A project to survey the quality of installed loose-fill insulation in Georgia residences resulted in several hundred attic inspections to measure insulation thickness and density. Thermal resistance (R-values) for various types of attic insulation have been calculated using generalized correlations for apparent thermal conductivity as a function of density. The calculated R-values were compared with contracted R-values in an effort to assess the adequacy of current quality control measures. An analysis is presented of the measurements and calculations along with general observations about the installation in the attics that have been inspected.

**KEYWORDS:** insulation, fiberglass, R-value, density, loose-fill insulation, rock wool, attics

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## INTRODUCTION

A concern regarding the quality of installed loose-fill insulation installations in residential attics in Georgia was brought to the attention of the Georgia Governor's Office of Consumer Affairs by the Georgia Chapter of the Insulation Contractors Association of America (ICAA) in 1988 [1]. A study completed in 1988 revealed that a number of builders, insulation contractors, and utility inspectors had been negligent [2]. The result was that these companies were required to add insulation to the deficient attics and to verify, using an independent testing agency, that all the other houses in the area had adequate attic insulation installed. In January, 1990, Southface Energy Institute was contracted to perform a second study to ascertain whether installation practices have improved.

# SELECTION AND INSPECTION PROCEDURE

To obtain a base population of houses to inspect, a flier (developed by the Georgia Governor's Office of Consumer Affairs) to explain the study and solicit voluntary cooperation from homeowners was distributed at new subdivisions. The initial focus of this distribution was new subdivisions identified by the local ICAA officers as candidates for inspection. The distribution area was then expanded to include the entire Metro-Atlanta area and will soon be further expanded to include additional Georgia cities. The subdivisions targeted were those constructed within the past three years, although since some subdivisions have different construction phases, some older homes were included in the study.

The initial phase of the study involved measuring insulation properties at six sites within each attic. The inspector mentally divided the attic into three equal regions, and from each region selected two sites representative of that area. The area within eight feet of the attic entrance was avoided because that area could have more insulation than the average due to installers compacting the insulation as they leave the attic.

Five thickness measurements were taken at each of the six attic sites, using a long metal skewer and a metal measurement scale readable to one millimeter. At one site in each of the three regions, a specimen was removed using a cylindrical metal "cookie cutter" approximately one foot in diameter, with a sharp, scalloped lower cutting edge [3]. Insulation fibers partially within the cylinder were cut with a utility knife along the interior of the cutter perimeter. The insulation within this cutter was placed in a clear plastic specimen bag, sealed with evidence tape, and labeled using special chain-of-possession labels [4]. A fiberglass batt approximately 0.093 m<sup>2</sup> (1.0 ft<sup>2</sup>) in area with an R-value of 5.28 K-m<sup>2</sup>/W (30 °F-ft<sup>2</sup>-h/Btu) was then placed into the cutter to fill the void and the cutter was removed.

Although not directly relevant to the investigation, notes were taken on other aspects of the attic insulation and ventilation to maximize the opportunity to inspect hundreds of attics. These aspects included the type of attic ventilation provided, any hazards encountered such as buried recessed lights or combustion appliance flues in direct contact with insulation, as well as notes regarding the uniformity of the insulation installation. The attic was sketched, showing approximate locations of all measurements and sample locations, the attic entrance, and other specific observations such as hazards or uninsulated areas. The information on an attic card, if present, was recorded.

The specimen bags were returned to Southface Energy Institute for identification of insulation type and weighing. Identification of the insulation was accomplished through a careful comparison of the collected specimens and specimens of the most commonly used loose-fill insulation products in the Metro-Atlanta area. If there was any question of identification, the specimen was identified as the product having the higher R-value per inch; only a small percent of the specimens fell into this category.

The inspection consisted of three measurements of density. The determination of density involved five thickness measurements and a loading that was determined from the weight of material inside the cutter and the area of the cutter. In addition to the 15 thicknesses associated with the density determination, there were 15 additional thickness measurements were taken to bring the total to 30. The data from a single attic was used to calculate average values for thickness, T, loading (weight per unit area), L, and density, D. The standard deviation,  $S_x$ , for each for these loading and density was calculated for each attic using Equation (1).

$$S_{x} = \sqrt{\sum_{i=1}^{n} (X_{i} - \overline{X})^{2} / n}$$
(1)

A mean value for the set of standard deviations was obtained from attics containing a specific type of insulation. This was simply the arithmetic average of the  $S_x$  values. A coefficient of variation, CV, defined for variable X by Equation (2) provides a measure of the significance of the standard deviation. The CV directly compares the standard deviation of the variable with the mean value of the variable.

$$CV_{x} = \overline{S}_{x} / \overline{X}$$
<sup>(2)</sup>

The data obtained from the attic inspections was used to calculate densities, D, and R-values,  $R_i$ . A BASIC program that uses Equations (3) and (4) with constants listed in Table 1 was used for these calculations.

$$\mathsf{D} = \mathsf{L}/\mathsf{T} \tag{3}$$

$$R_{i} = T_{i} / (A_{n} + B_{n} * D_{i} + C_{n} / D_{i})$$
(4)

Insulation type	An	$\mathbf{B}_n$	Cn
Fiberglass A	1.5862 E-2 <sup>a</sup>	1.7914 E-4	3.4532 E-1
	(0.1100) <sup>b</sup>	(0.0199)	(0.14595)
Fiberglass B	2.5956 E-2	4.5010 E-5	2.8504 E-1
	(0.1800)	(0.0050)	(0.1234)
Fiberglass C	2.5956 E-2	4.5010 E-5	4.1254 E-1
	(0.1800)	(0.0050)	(0.1786)
Fiberglass D	2.5956 E-2	4.5010 E-5	4.2062 E-1
	(0.1800)	(0.0050)	(0.1821)
Rock wool	7.5561 E-3	2.2145 E-4	9.4473 E-1
	(0.0524)	(0.0246)	(0.4090)

 TABLE 1 -- Constants used in Equation (4) to calculate thermal resistance

**a.** For density in kg /  $m^3$ , T<sub>i</sub> in meters, and R-value in  $m^2$ -K / W; E-n indicates a power of 10, 10<sup>-n</sup>.

**b.** Numbers in parentheses are for density in  $lb / ft^3$ ,  $T_i$  in inches, and R-values in  $ft^2$ -h-°F / Btu.

A single R-value was calculated for attic *i* containing product type *n*. The  $D_i$  used in Equation (4) was the average value for the insulation density in each attic. The denominator on the right side of Equation (4) is the apparent thermal conductivity. Values for the constants  $A_n$  and  $B_n$  were obtained from published correlations for rock wool and fiberglass type A, B, C, and D. The value of constant  $C_n$  for rock wool was also obtained from published correlations [5,6]. Values of  $C_n$  for fiberglass types A, B, C, and D were calculated to produce R-values equal to the product label information.

The calculated R-values were entered into an information database along with all vital data concerning the attic, the house address and subdivision, the homeowner, and other notes. The claimed R-values were also entered, and a percentage of claimed R-value was calculated.

# **DISCUSSION OF THE DATA**

The first result of interest was the types of insulation materials encountered in the attics inspected. After 500 investigations, no cellulose was found, and rock wool was found in less than five percent of the residences visited. The dominant product was loose-fill (blown) fiberglass, of which four varieties were observed. A
few of the houses visited were insulated with fiberglass batt insulation, but these house were not included in this study. Table 2 shows the distribution of insulation types encountered in this study.

Туре	3	$\overline{\mathbf{S}}_{\mathbf{x}}$		CV		
	Density	Loading	Density	Loading	of sample	
Fiberglass-A	0.07	0.08	0.11	0.16	31	
Fiberglass-B	0.07	0.08	0.11	0.16	34	
Fiberglass-C	0.08	0.09	0.08	0.11	11	
Fiberglass-D	0.06	0.11	0.08	0.17	21	
Rock wool	0.12	0.13	0.07	0.11	3	

**TABLE 2 -- Insulation characteristics** 

The second item of interest was the percentage of contracted R-value determined to be present in the attics that were inspected. Figure 1 summarizes the Rvalue data that were collected. The claimed (or contracted) R-values varied from 3.34 to 5.28 K-m<sup>2</sup>/W (19 to 30 °F-ft<sup>2</sup>-h/Btu). The distribution of percent contracted R-values that were calculated for the insulation examined is shown in Figure 1. Approximately 23 percent of the attics inspected had measured R-values 20 to 40 percent below contracted R-value, with an additional 5 percent below 40 percent of that contracted. It is also interesting to note that 48.4 percent of attics had more than the contracted amount of insulation. Attic blowing is not an exact science, and it is possible that the better contractors over-insulate to ensure meeting contracted values. The calculated R-values for the installed insulation varied from 35 percent to 145 percent of contracted R-values. The mean value for the ratio of R-actual / R-contracted was 0.96. Table 2 shows the mean standard deviations for the density and loading of each insulation type as well as the coefficients of variation for the density and loading of each insulation type.

The mean standard deviation for each insulation type is calculated from the number of samples of that insulation type and the average value of that parameter. Rock wool specimens, having a significantly greater density and loading than fiberglass specimens, have a considerably higher standard deviation than the fiberglass sample. Because of this, a coefficient of variation was calculated for both the density and loading of each insulation type.

50 houses where the calculated average R-value was less than 80 percent of the contracted R-value were selected at random from the data collected for fur-





ther analysis. The average R-value was 66 percent of the contracted R-value, the average loading was 73 percent of the labeled loading for the contracted R-value, and the average density was 123 percent of the labeled density for the contracted R-value. There was an apparent correlation between measured R-value and the amount of insulation delivered. The average density of installed material was consistently higher than labeled values, indicating a difficulty in achieving the manufacturer's specifications.

Data from approximately four percent of the attics sampled were removed from the calculations of mean standard deviation and coefficient of variation due to large magnitudes of the variations. These occurrences could have been caused by unusual variations in installation, such as specimens taken at sites with markedly different thicknesses or where the installer compacted blown insulation with the hose nozzle pointed toward the attic floor. It may also have been through technician error in measuring thickness or weight. The criterion used for the removal of data points was a coefficient of variation over 0.25, which was greater than typical values for this coefficient calculated from an earlier study [5]. This refinement of the data is included in the results that are reported.

Some difficulties were encountered in the study. Chief among these was a lack of documentation of claimed R-value for loose-fill insulation installations. By Federal Trade Commission regulations, this information should be stated clearly on a receipt for the homeowner. This is typically an attic card secured to a framing member adjacent to the attic entrance. We found, however, that this receipt was absent in 40 percent of the attics investigated. The R-value should also be stated in the closing documents from the sale of the house, but in a number of houses either this was not the case or the homeowner was not cooperative in thoroughly checking these documents [7]. This occurred in approximately 25 percent of the houses inspected. Another method to determine contracted R-value was to ascertain the financing source for the house. If this source was the U.S. Department of Housing and Urban Development, the U.S. Federal Housing Administration, or the U.S. Veterans Administration, then the contracted R-value should have been up to the standards set by those agencies, 4.23 K-m<sup>2</sup>/W (24 °F $ft^2$ -h/Btu) [8]. Similarly, if the house was certified by a local utility as part of an energy conservation certification program, such as "Good Cents," the contracted Rvalue can be assumed to be at least the minimum required under that program,  $5.28 \text{ K-m}^2/\text{W}$  (30 °F-ft<sup>2</sup>-h/Btu) for Good Cents [9].

A key to successful completion of an investigation of this magnitude was to provide rigorous monitoring to ensure quality control at each step of the investigation. This is accomplished by providing extensive training and periodic oversight of inspectors, carefully examining the field specimens and data forms, checking weights and identities of specimens, and reviewing the data for inconsistencies.

In addition to checking each procedure for technical accuracy, the legal integrity of the investigation must be maintained. Though this investigation provides interesting building insulation data, its primary function is to provide legal evidence for court proceedings. Some examples of how this affects the investigation include proper use of evidence tape to fully seal each bag, proper use of chain-of-possession labels on the sample bags, and maintaining security of the sample bags and the data forms so that there can be no opportunity for unauthorized alteration of the evidence.

# SUMMARY

In the state of Georgia, loose-fill fiberglass insulation is by far the most commonly used product for attic insulation in new construction. Of the attics inspected, approximately 28 percent were under-insulated by at least 20 percent of the contracted R-value. Approximately 48 percent of the attics were over-insulated. The percentage of contracted R-value ranged from 35 to 145 percent, with a mean value of 96 percent. A detailed analysis of 50 houses with an installed Rvalue of less than 80 percent of contracted R-value showed that the average loading was 73 percent of label values and that the average density was 123 percent of label values. The shortfall in loading means that the proper amount of insulation was not installed. The higher than label densities indicate a contractor difficulty in meeting the manufacturer's specifications. The key to success in a study of the type being reported is effective methods for discovering contracted R-value and comprehensive quality control of both technical and legal procedures.

# ACKNOWLEDGEMENTS

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ZIP 2.0: THE ENHANCED ZIP-CODE INSULATION PROGRAM

**REFERENCE**: Petersen S. R., "Zip 2.0: The Enhanced Zip-Code Insulation Program," <u>Insulation Materials</u>: <u>Testing and Applications. 2nd Volume</u>, <u>ASTM STF 1116</u>, R. S. Graves and D. C. Wysocki, Eds., <u>American Society</u> for Testing and Materials, Phil#delphia. 1991.

ABSTRACT: ZIP 2.0, the enhanced Zip Code Insulation Program, determines economic levels of thermal insulation for new and existing houses in any location in the United States, given the first three digits of its Zip code. Economic insulation levels are calculated for attics, cathedral ceilings, exterior walls, floors over unheated areas, slab floors, basement and crawlspace walls, ductwork in unconditioned spaces (attics and crawlspaces), and water heaters. Local climate data and default insulation costs and energy prices are retrieved from data files on the ZIP disk. The user can enter site-specific insulation costs, energy prices, heating and cooling system types and approximate efficiencies, and other parameters to customize the calculations for a specific house. ZIP was developed at the National Institute of Standards and Technology, under the sponsorship of the U.S. Department of Energy. ZIP and its supporting data files are provided on diskette for use with microcomputers having an MS-DOS operating system (typical IBM-PC/ XT/AT and compatible systems).

KEYWORDS: thermal insulation, engineering economics, energy conservation, building economics, life-cycle cost analysis, information software

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#### INTRODUCTION

The purpose of this paper is to demonstrate the computational capabilities of ZIP 2.0, the enhanced ZIP-Code Insulation Program; to familiarize the reader with its input requirements and output report; and provide an example of its use. This paper offers ZIP as an example of a program for the personal computer that can provide site-specific applications of timely and complex building research results to an increasingly large number of potential users. ZIP 2.0 is in the public domain, and is available, along with a user's guide, on a diskette for use with IBM PC and compatible microcomputers.

The increased use of thermal insulation as a means of permanently reducing energy bills in new and existing houses became a subject of considerable interest after energy prices began to rise sharply in the early 1970's. Since that time numerous guidelines for recommended (or, in some cases, mandatory) insulation R-values have been published by government agencies, research organizations, utilities, professional associations, and private authors. Early guidelines were often based on practical considerations, reflecting what the industry had already recognized, if often overlooked, as good design practice. Most guidelines since the mid 1970's have been based on more detailed technical and economic analysis, usually balancing long-term savings against first costs to determine insulation levels that are economically justified.

Published insulation guidelines usually reflect typical values for such parameters as heating and cooling degree days, energy prices, seasonal efficiencies of space heating and cooling systems, insulation costs, and related design and operating parameters, often generalized for wide areas of the country. Providing for sitespecific parameter values requires numerous worksheets and tables, making the guidelines too lengthy for most publications and too complex for most users.

Paralleling the proliferation of insulation guidelines, at least since 1980, has been the proliferation of personal computers (PCs). PCs have become increasingly available to the U.S. population, at home, at school, and in the workplace. This requires researchers to reconsider the means of reporting research results which are driven by numerous, often interactive, parametric assumptions. Computer software designed for "user-friendly" applications allows interested consumers (or others acting in their behalf) to enter data and make choices that reflect site-specific conditions. This software can then display customized, concise, recommendations that put timely research results to their best use.

This is the purpose of ZIP, the Zip-Code Insulation Program [1], developed at the National Institute of Standards and Technology under the sponsorship of the U.S. Department of Energy. ZIP 1.0, the original version of this program released in 1988, provides users with custom calculations of economic insulation levels for residential applications. ZIP 1.0 employs accepted engineeringeconomic methods for calculating economic levels of insulation in attics, exterior walls, floors, and crawlspace and basements walls of new and existing houses, based on data and assumptions entered by the user. Climate data appropriate to the first three digits of the user's ZIP Code are retrieved from a disk file, along with default energy and insulation prices. Local energy prices, insulation prices, and information on the type and efficiency of the heating and cooling equipment can be entered by the user. A concise report showing the economic levels of insulation specific to the user's input data and only for the building components designated by the user is displayed to the screen and can then be printed for permanent reference.

ZIP 1.0 has had widespread distribution throughout the United States. It has been used by homeowners, utilities, builders, insulation manufacturers and installers, universities, and government agencies. It was also the basis for updating the recommended insulation levels in the DOE "Insulation Fact Sheet" [2].

Version 2.0 [3], released in January 1991, represents a major enhancement of the original ZIP program, both in terms of capability and functionality. New insulation applications are available for cathedral ceilings, ducts in unheated spaces (both supply and return), exterior walls in unheated basements, and water heaters. Separate specifications for rigid foam sheathing applications are included, recognizing that typical insulation R-values for sheathing are usually different than those for most mineral fiber and cellulosic insulation. Improved climate data (cooling degree hours) are included in the data base. Expanded default energy prices are available to reflect state and local factors. Screen displays and data entry formats are improved. A more comprehensive output report has been created (including the user's input data). This report can be displayed on the screen, sent to a printer, or saved to a disk file for later retrieval by word processor. User's input data can also be saved to a file for retrieval as default values for a later ZIP session, if desired.

#### INPUT DATA REQUIREMENTS

ZIP input requirements can be divided into four parts: (1) setting up the general computational environment for evaluating insulation savings, (2) specifying insulation systems to be evaluated, (3) entering existing levels of insulation (existing houses only), and (4) entering insulation costs and related parameters specific to the insulation systems of interest.

#### (1) General computational environment

A ZIP analysis begins by entering the first three digits of the local ZIP Code for the house in which insulation systems are to be evaluated. Climate data (heating degree days base 18.3C (65F) and cooling degree hours base 23.3C (74F)) for that location are retrieved from the ZIP climate file. These data are displayed on the screen, along with the principal city for that ZIP Code, for the user's approval to proceed. The ZIP Code is also used in retrieving default values for local energy prices, as well as regional DOE energy price escalation rates and insulation cost adjustment factors, from supporting files.

The user then indicates whether the analysis will be for a new or an existing house, selects the primary heating system and cooling system types, and indicates their approximate seasonal efficiencies. The heating and cooling system type and efficiency options available for selection are shown in Table 1. Note that the Heating Seasonal Performance Factors (HSPF) shown for heat pumps correspond to those published in the Air Conditioning and Refrigeration Institute's Directory of Unitary Air-Source Heat Pumps [4], and are adjusted to reflect local climate conditions when energy savings are computed later in the ZIP analysis. If a central heating or cooling system has been designated, the user must indicate whether or not there is ductwork in unconditioned spaces, such as attics and crawlspaces.

TABLE 1 -- Heating and Cooling Systems and Efficiency Options

Heating System Types

Cooling System Types

gas furnace oil furnace LPG furnace electric furnace (central) electric heat pump electric baseboard electric heat pump

none central air conditioning room air conditioner

Efficiency designators and corresponding parameters used in ZIP calculations:

	Corresponding Efficiency Parameter				
Efficiency	Gas/Oil/LPG	Heat	Pumps	AirConds	
Designator	Systems	SCOP <sup>a</sup> (HSPF <sup>b</sup> )	SCOP(SEER <sup>c</sup> )	SCOP(SEER)	
		<u>Heating</u>	<u>Cooling</u>	Cooling	
low	50%	1.6(5.5)	2.1(7.25)	1.8(6.0)	
medium	65%	1.9(6.5)	2.6(8.75)	2.3(8.0)	
high	80%	2.2(7.5)	3.0(10.25)	2.9(10.0)	
very high	90%	2.5(8.5)	3.4(11.75)	3.5(12.0)	

<sup>a</sup> SCOP = Seasonal Coefficient of Performance (kWh/kWh)

<sup>b</sup> HSPF = Heating Seasonal Performance Factor (Btu/kWh)

<sup>c</sup> SEER = Seasonal Energy Efficiency Ratio (Btu/kWh)

Default energy prices for heating and cooling systems are then displayed. The user is encouraged to overwrite these with current prices from the local utility. If electricity is used for space heating or cooling, the user is instructed to enter winter rates for heating and summer rates for cooling. The user is also advised that the electricity prices used in the ZIP analysis should be for last

kWhs used in the peak winter and summer months, rather than an average price for those months. (Most electric rate schedules have different rates for different levels of usage. The kWhs saved by increasing insulation levels are the last kWhs consumed each month, and thus it is the cost of those last kWhs that is needed in the economic analysis.) Default oil, gas and LPG prices retrieved from the disk are based on state averages. For electricity, default kWh prices are based on rate schedules for the dominant electric utility serving that location or a neighboring location in the same state. All default energy prices are representative of 1990 cost levels.

#### (2) Insulation systems to be evaluated

The user then selects insulation applications for analysis from the screen display shown in Figure 1. Only the insulation applications actually selected by the user are evaluated and reported. Note that the user must indicate whether rigid foam sheathing or batt/blanket insulation is to be evaluated for some components, including cathedral ceilings, masonry exterior walls, and basement and crawlspace walls. However, both can be selected if desired. For an evaluation of an existing house, only blown-in insulation can be evaluated in wood-frame exterior walls.

	Selec	ct Ins	Sula	ation	System	s fo	or	Anal	Lys:	ĹS		
Press	<space< th=""><th>Bar&gt;</th><th>to</th><th>Toggl</th><th>e Syste</th><th>em (</th><th>Dn</th><th>[X]</th><th>or</th><th>Off</th><th>[</th><th>].</th></space<>	Bar>	to	Toggl	e Syste	em (	Dn	[X]	or	Off	[	].

ATTIC:	CRAWLSPACE WALLS:
[ ] 1. batt/blanket/loose fill	[ ] 1. batt/blanket/loose fill
CATHEDRAL CEILINGS:	[ ] 2. rigid foam sheathing
[ ] 1. batt/blanket/loose fill	BASEMENT WALLS (HEATED SPACES):
[ ] 2. rigid foam sheathing	[ ] 1. Exterior Insul.(rigid foam)
ÉXTERIOR WALLS (ABOVE GRADE):	2. Interior Insulation:
[ ] 1. Wood Frame (batt/blanket)	[] a. batt/blanket
<ol><li>Concrete-Masonry:</li></ol>	b. rigid foam sheathing
[ ] a. batt/blanket/loose fill	BASEMENT WALLS (UNHEATED SPACES):
[ ] b. rigid foam sheathing	[ ] 1. Exterior Insul. (batt/blkt)
FLOORS:	2. Interior Insulation:
[ ] 1. Over unheated spaces (batt)	[] a. batt/blanket
[ ] 2. Concrete slab (rigid foam)	[ ] b. rigid foam sheathing
[ ] DUCT INSULATION	[ ] WATER HEATER INSULATION

FIG. 1 -- Menu for selecting insulation systems of interest.

#### (3) Existing insulation R-values

If economic insulation levels for an existing house are to be calculated, the user must indicate the approximate level of insulation currently existing in the attic, floors over unheated areas, crawlspace walls, and ductwork if these areas have been selected for analysis. The user chooses from several typical levels of insulation displayed on a screen, based on R-values and the corresponding thickness for the most common insulation materials. If blown-in insulation for existing wood-frame walls is selected for analysis, the user must indicate whether or not there is already some insulation in these walls. If this is answered in the affirmative, ZIP advises that it is unlikely that blown-in insulation will be cost effective, and will eliminate this component from further evaluation. This same procedure is repeated for the addition of insulation to previously insulated cathedral ceilings, exterior masonry walls, and basement walls.

#### (4) Insulation costs and related data

Finally, insulation costs are entered for each insulation system selected for analysis. Typically, three to eight levels of insulation are displayed for each system, by R-value, along with default prices (1990) retrieved from the ZIP disk and adjusted by a regional cost factor. An example of the display for attic insulation costs is shown in Figure 2. Insulation costs should be representative of current installed costs, at the building site, for each R-value shown, on a per square foot or per linear foot basis (as indicated on the screen). The user is encouraged to change the default costs to reflect local conditions. Insulation costs should include the cost of any structural modifications (e.g., expanded wall thicknesses) or protective coverings (e.g., exterior siding), to the extent that these are needed solely to accommodate the insulation levels to be evaluated. Any R-value can be eliminated from the analysis by entering a cost of \$99.99. This allows the user to avoid finding prices for insulation levels that are impractical, available, or unlikely to be cost effective locally (e.g., low levels in a cold climate with high fuel prices).

Attic Insulation Costs (Batt/Blanket/Loose Fill): (Cost per square foot of gross attic area)				
R-11 @ \$ 0.34 per square foot R-19 @ \$ 0.47 per square foot R-22 @ \$ 0.54 per square foot R-30 @ \$ 0.67 per square foot R-38 @ \$ 0.85 per square foot R-49 @ \$ 1.05 per square foot				
Do you want to change any of these costs? Yes No				
<pre><esc>=back up <fl>=review input <f7>=Quit <f10>=start over</f10></f7></fl></esc></pre>				

FIG. 2 -- R-values and default prices for attic insulation.

If a water heater blanket is selected for analysis, ZIP asks the user to identify the type of water heater (gas, oil, LPG, or electric resistance) and whether the internal tank insulation is foam or mineral fiber. (Water heaters with foam insulation are assumed to have lower stand-by losses than those insulated with mineral fiber.) The approximate capacity (in gallons), its approximate remaining life, and current fuel prices are also requested. If the fuel type is the same for heating or cooling, the default fuel price is the same as previously entered for the space heating or cooling system. The economic calculations for the water heater blanket are based on the remaining life of the water heater, with no salvage value assumed.

If insulation on supply and return ducts in unconditioned spaces is to be analyzed, the approximate cross-section dimensions must be identified for the most representative sizes. The approximate size for both supply and return ducts is selected from a menu of choices, as shown in Figure 3. Economic R-values are reported separately for supply and return ducts in attics and in both vented and unvented crawlspaces.

Supply ducts in unconditioned spaces:					
What is most typical	duct size (approxima	te size in inches)?			
Rc	ound Recta	ingular			
(1)	) 4 (6)	3 x 12 5 x 12			
(2)	) 8 (8)	8 x 14			
(5)	(9) 10 (9) 1 ) 12 (10) 1	$2 \times 20$			
	Enter by code (1-10	)):			
Rc (1) (2) (3) (4) (5)	ound Recta   4 (6)   6 (7)   8 (8)   10 (9)   12 (10)   Enter by code (1-10)	ngular 3 x 12 6 x 12 8 x 14 .0 x 16 .2 x 20 			

FIG. 3 -- Selection of supply duct size.

#### THE ZIP REPORT OF ECONOMIC INSULATION LEVELS

ZIP displays the results of the economic analysis directly to the screen. An example of the ZIP report is shown in Figure 4. Results for all of the insulation systems that can be evaluated by ZIP are shown in this example. For a custom analysis, only the insulation systems selected for analysis are displayed. The ZIP Code and reference location, and the number of annual heating degree days base 18.33C (65F) and cooling degree hours over 23.33C (74F) are shown at the top of the report. Also included are the types of heating and cooling equipment, seasonal energy efficiencies, and fuel prices used in the analysis. For existing houses, the existing insulation level, amount to be added, and total level for each application are shown. Note that the economic levels of insulation for both floors over crawlspaces and for crawlspace walls are shown if both were selected for analysis. However, only one of these methods would typically be used.

Additional information is also provided to the user, based on the insulation systems evaluated. This information provides caveats regarding installation rather than a guide to installation, per se. For example, the user is advised to specify insulation by R-value, not by thickness; to keep insulation away from heat sources; to

place a vapor barrier on the ground if a crawlspace is to be closed off; to freeze-protect water pipes if the floor over a vented crawlspace is to be insulated; and to tape all duct joints before insulating. The report also contains all of the insulation costs used in the analysis, in order to complete the documentation of the input data used in generating the report.

The ZIP report can also be sent to the printer or saved as a flat (ASCII) file for later retrieval by word processor. The user can also save the current input data to the disk for use as default values in a later ZIP analysis.

Economic Insulation Levels for Zip 111XX -- New House Reference Location: Queens, NY Heating Degree Days (65F) = approx. 5000 Cooling Degree Hours(74F) = approx. 9000

Heating Equipment: Natural GasCooling Equipment: Central-ElectricSeasonal Efficiency: 65%SEER (seasonal eff.): 8.0 Btu/WhUnit Fuel Cost: \$0.650/thermUnit Fuel Cost: \$0.140/kWh

Attic Insulation	R-38
Cathedral Ceiling Insulation	
Batt/Blanket	R-30
Rigid Foam	R-16
Wood-Frame Wall Inculation	R-10 P 22
Maconry Wall Insulation	K-25
Patt /Planhot	B 10
Dall/DIANKel	R-19 D 10
Rigid Foam	R-1Z
Floors over Grawispace (if crawispace walls up	ninsulated)
Batt/Blanket	R-19
Crawlspace Wall Insulation (If floor above is	uninsulated)
Batt/Blanket	R-11
Crawlspace Wall Insulation (If floor above is	uninsulated)
Rigid Foam	R- 5
Slab-Edge Insulation (Rigid Foam)	R- 5
Insulation of Basement Walls:	
Exterior Insul. for Heated Basement (RF)	R-10
Interior Insul. for Heated Basement	
Batt/Blanket	R-19
Rigid Foam	R-12
Exterior Insul for Unheated Basement(RF)	R- 0
Interior Insul for Unheated Basement(KI)	R- 0
Batt/Blanket	D 2
Digid Form	
Traulation of Supply Ducto in Unconditioned St	K- 0
Ducts in Unconditioned Sp	aces:
Ducts in Attics:	R- 8
Ducts in vented Grawispaces:	R- 8
Ducts in Unvented Crawlspaces:	R- 6
Insulation of Return Ducts in Unconditioned Sp	aces:
Ducts in Attics:	R- 6
Ducts in Vented Crawlspaces:	R- 4
Ducts in Unvented Crawlspaces:	R- 4
Water Heater Insulation	Yes

FIG. 4 -- ZIP report for all insulation types in new house (example).

#### COMPUTATIONAL METHODS

In determining the economic level of insulation for any given component, ZIP uses an economic methodology consistent with ASTM Standards E 917 (Practice for Measuring Life-Cycle Costs of Buildings and Building Systems) [5] and E 1074 (Practice for Measuring Net Benefits for Investments in Buildings and Building Systems) [6]. The level of insulation with the greatest present-value net savings (total savings minus total costs) is designated as the "economic" level of insulation for each component. This method of economic optimization is equivalent to selecting the level with the lowest present-value life-cycle cost.

Energy savings are based on methodologies developed to support the ASHRAE 90.2P energy standard for new low-rise residential buildings [7]; a similar methodology developed at Oak Ridge National Laboratory for the analysis of foundation, basement, and floor insulation [8]; the methodology used by the California Energy Commission for the analysis of duct losses in one and two story houses [9]; and a research report on water heater insulation from the National Bureau of Standards [10].

The general computational methodology consists of six parts:

(1) Calculate the reductions in annual heating requirements (AHR) and annual cooling requirements (ACR) for each level of insulation, relative to the base case (no insulation in new houses, existing insulation in existing houses), on a square foot or linear foot basis as appropriate. This is repeated for each insulation system evaluated. The actual equations used to compute reductions in AHR and ACR are reported in the ZIP reference manual.

(2) Calculate the corresponding reductions in annual energy purchases for heating and cooling  $(AEP_h \text{ and } AEP_c)$  by dividing reductions in AHR and ACR by the seasonal energy efficiencies of the heating and cooling systems.

$$\Delta AEP_{h} = \Delta AHR / EFF_{h}$$
  
 $\Delta AEP_{c} = \Delta ACR / EFF_{c}$ 

where  $EFF_h$  = seasonal efficiency of the space heating system, and  $EFF_c$  = seasonal efficiency of the space cooling system.

For air conditioners,  $\text{EFF}_{c}$  is derived by dividing the Seasonal Energy Efficiency Ratio for cooling (SEER) by 3.412 Btu/Wh. For heat pumps,  $\text{EFF}_{h}$  is derived from the Heating Seasonal Performance Factor (HSPF) to reflect the severity of the winter climate (heating degree days 65F, HDD65F):

 $EFF_h = HSPF/6.5 * 1.06 * [2.3 - 0.1(HDD65F/1000)]$ 

(3) Calculate corresponding savings in annual heating and annual cooling costs ( $AES_h$  and  $AES_c$ ) by multiplying reductions in purchased energy requirements by unit cost of energy and the units per million Btu.

$$\begin{split} AES_{h} &= \Delta AEP_{h} * (\$/unit_{h}) \times (unit_{h}/million \; Btu) \\ AES_{c} &= \Delta AEP_{c} * (\$/unit_{c}) \times (unit_{c}/million \; Btu) \end{split}$$

where:

- \$/unit<sub>h</sub> = the current cost per unit of energy used for heating (e.g., \$.08/kWh),
  - - unit<sub>h</sub> = the number of units of heating energy per million Btu (e.g., 293 kWh = one million Btu), and
    - $unit_c$  = the number of units of cooling energy per million Btu.

(4) Calculate the total present-value savings (PVS) in energy costs over the designated insulation life (30 years for new houses, 20 years for existing houses). Annual savings for heating and cooling are multiplied by the modified uniform present worth factor (UPW\*) corresponding to the life, discount rate (7% real, i.e., net of general inflation), and appropriate DOE energy price escalation rates, and summed. (DOE projections of price escalation rates (in real terms) for the specified energy type and region of the country are retrieved from a ZIP data file.)

 $PVS = AES_h * UPW*_h + AES_c * UPW*_c$ 

- where UPW\*h = the modified uniform present worth factor for heating energy, given a study period, the discount rate, and the annual rate of heating energy price escalation over the study period,
  - and UPW\*<sub>c</sub> = the modified uniform present worth factor for cooling energy, given a study period, the discount rate, and the annual rate of cooling energy price escalation over the study period.

(5) Calculate the present-value net savings (NS) for each insulation level by subtracting the total cost of the insulation at that level from its present-value savings.

NS = PVS - C

where

C = initial cost (installed) for the insulation level of interest.

(6) Determine the insulation R-value for each application of interest that has the greatest present-value net savings. This is designated as the economic R-value for that application. The rate of return for the higher R-values falls below the 7 percent discount rate (over and above general inflation) that is used in the ZIP analysis.

#### LIMITATIONS

ZIP 2.0 provides a powerful means of getting site-specific estimates of economic R-values to homeowners, utilities, homebuilders, insulation manufacturers and installers, and Federal, state and local government agencies. However, some present limitations on its analytical capabilities should be noted.

ZIP does not report the expected savings from the economic insulation levels. The data needed to adequately estimate economic R-values is not sufficient to provide reliable estimates of actual energy savings for each component or for the overall house. To do so would require more information on the size, design, and orientation of the house, base-case U-values for all components, thermostat settings, solar loads, internal loads, and related factors. Instead, an energy audit for the specific house that can take these additional factors into consideration is suggested.

The energy estimating relationships used to calculate reductions in space heating and cooling requirements are based on uniform component surface areas (e.g., per square foot of net wall area), independent of the building size. Ideally, more sophisticated relationships could be developed that would be more sensitive to this and other design parameters. A balance must be struck between the additional benefits of such improvements and the additional data input needed to run the program successfully.

The three to eight R-value levels displayed and evaluated for most insulation applications are fixed in the program data base (although they can be changed without recompiling the program). Ideally, the user could enter other values of interest during the ZIP analysis.

ZIP is an insulation information program, and as such does not consider other important energy conservation design options for buildings, such as alternative window treatments and equipment choices. Perhaps future versions will venture into these areas as well.

The user cannot change the discount rate, insulation life, or energy price escalation rates used in the ZIP analysis. These parameters have been fixed to promote uniformity in estimating economic R-values, consistent with other Federal government guidelines relating to energy conservation in buildings.

#### CONCLUSIONS

ZIP provides a good example of the potential for providing timely and complex building research results to large segments of the U.S. population via the personal computer. ZIP provides a sitespecific analysis of economic insulation R-values for use by homebuilders, architects, and homeowners. It is sensitive to local climate data, heating and cooling system parameters, energy costs, insulation costs, and related input data provided by the user or retrieved from data files based on the local ZIP Code. It can also be used by utilities, extension services, insulation manufacturers and vendors, and others to provide economically sound insulation guidelines to interested individuals who may not have ready access to a PC or the current data needed to use the program properly.

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# Long-Term Thermal Performance of Foams

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EVALUATION OF LONG-TERM THERMAL PERFORMANCE OF CELLULAR PLASTICS REVISITED

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ABSTRACT: Since the Montreal Protocol has established restrictions on the use of substances, that deplete the ozone layer, evaluation of performance of foams manufactured with alternative blowing agents, partially halogenated hydrochlorofluorocarbons (HCFC) in particular, became a high priority issue.

This paper reviews the issues involved in the evaluation of long-term thermal performance under service conditions and presents an agenda for further research.

KEYWORDS: Cellular plastics, foams, aging, thermal resistance, long-term performance, R-value, k-factor

# INTRODUCTION

The National Research Council of Canada has recently joined forces with the Society of Plastics Industry to develop a methodology for predicting long-term thermal performance of foams manufactured with any blend of blowing agents and any type of permeable or impermeable facers.

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This paper, in its first part, reviews long-term thermal performance of the foams. Initially, we analyze invariable components of heat transfer, viz. solid phase conduction and radiative heat transfer, then we analyze heat transfer due to cell-gas conduction. Since the latter depends on cell-gas composition, factors such as initial pressure of blowing agent, its solubility in and permeation through the polymer matrix are examined. This review aims at the development of the evaluation methodology.

This evaluation methodology must involve rapid testing. If the changes in cell-gas composition can be described by a gas diffusion theory, then, duration of the tests may be reduced in two ways: either by testing foams exposed to elevated temperature or by using thin layers of the material. Conversely, the same relation may be used to predict aging of thick boards from experimental data on thin slices. This relation is normally expressed as a complex model of heat and mass transfer. In some cases, however, this relation may be expressed as a simple arithmetic operation, denoted as the scaling technique. This paper reviews the use and limitations of the "slicing and scaling" approach.

This evaluation methodology is to be used for various foams: extruded polystyrene, phenolic, polyisocanurate laminated with gas barriers, or sprayed polyurethanes. In all these cases, experiments are performed on thin material layers and models are used to derive material characteristics from these experiments. The models are also necessary to predict long-term thermal performance of full boards and to analyze the effect of statistical variation in material properties, e.g. variations in the efficiency of gas barriers.

Much of the information necessary for evaluation of field performance of different foams is, however, missing. Therefore, the second part of this paper presents the immediate research needs identified with a view to establishing a comprehensive evaluation methodology.

#### PART I: STATE OF THE ART

#### Understanding Of The Aging Process

Initially, cells of fresh foams contain the highest fraction of blowing agent (heavy molecular gas) and the lowest fraction of air components. During the service life, the cell-gas composition changes primarily because of air ingress, solubility of blowing agent in the polymer and loss of blowing agent to the environment. Ultimately all the blowing agent is replaced by air and the foam insulating efficiency attains its final steady level. Since the duration of the aging process may vary from a few years to a few hundred years, the understanding how different factors affect the aging process is very important for any foam manufacturer. A review of this knowledge with a stress on recent developments is given below.

## Mechanisms of heat transfer through foams

Similar to other thermal insulations, due to the small cells in the foam, the convective gas movements within the insulation are prevented and only the following heat transfer mechanisms occur in the foam:

- (1) conduction through the cell-gas,
- (2) conduction through the solid phase, i.e., transfer of heat around the cell-gas spaces,
- (3) radiation across the cell-gas spaces,
- (4) interactions between the radiation and solid phase conduction.

Interaction between conductive and radiative modes of heat transfer was shown [1] for a mineral fibre insulation with density of 51 kg/m<sup>3</sup>, to be approximately 6 percent of the heat flux. Nevertheless, the contribution of this interaction was included in the solid conduction to simplify modeling of heat transfer [2]. An identical approach is used in modelling of heat transfer through the foams.

Upon these conditions, one may express the apparent thermal conductivity,  $k_{app}$ , by adding thermal conductivity of the gas,  $k_g$ , and that of solid matrix,  $k_s$ , to the contribution of radiation,  $k_r$ :

$$k_{app} = k_g + k_s + k_r \tag{1}$$

Thermal conductivity coefficient is an apparent property because it depends on the test conditions, for instance it may vary with temperature gradient applied to the specimen. Even though term "apparent" is omitted in the further text, one must not forget about the apparent nature of this material characteristic.

Today, we have a good understanding of all components of thermal conductivity. Thermal conductivity of the cellgas, may be calculated from an equation for a mixture of polyatomic gases [3]. Thermal conductivity of the solid phase, following the research at the Massachusetts Institute of Technology [4] is divided into contribution of struts and membranes (cell windows). The radiative heat transfer for differet directions of heat flow was also analysed separately for struts and cell windows [5]. The authors, first estimated thermal conductivity of solid phase, Table 1, and then, by subtracting gas and solid phase conductivity from the experimental values, Table 2, estimated radiative heat transfer.

TABLE 1 Foam thermal conductivity and estimated radiative heat transfer for a free-rise laminate polyurethane foam. Quoted from Cunningham and Sparrow [5] with permission.

Heat flow mechanism	Rise direction (W/mK)	Perpendicular to rise (W/mK)	<pre>% Contribution in the rise direction</pre>
Experimental tot	al 0.0	18 0.014	-
Gas conductivity	0.0	08 0.008	44
solid phase	0.0	04 0.002	23
radiative transf	er 0.0	06 0.004	33

TABLE 2 Estimated directional contributions of thermal conductivity for the studied polyurethane foam. Quoted from Cunningham and Sparrow [5] with permission.

	Rise	transverse	"isotropic
	direction	direction	foam"
Struts	0.0037	0.0013	0.0021
Windows	0.0005	0.0010	0.0008
Total	0.0042	0.0023	0.0029

Radiative heat transfer through foams was carefully examined at MIT. Glicksman and Torpey [6] have, as a first approximation, disregarded the effect of cell-walls and considered only the attenuation of radiation as affected by cylindrical, opaque struts. They correlated the extinction coefficient with the mean cell diameter. This approximation agreed well with the experimental data [5,6] for foams with small cell size. More recently, Glicksman et al [7], included also the extinction coefficient for optically thin cell walls, extending the agreement between calculated and measured values of radiative heat transfer.

It appears that the radiative heat transfer is related to a spacing factor which originally was assumed to be the distance between cell walls and more recently the distance between struts. This spacing factor has been correlated with the mean cell size. But the spacing factor for radiation is different in each direction. Therefore, the "mean cell diameter" in the model calculations need not to be the same as the geometrical mean cell diameter.

The extinction coefficient for radiation not only vary in each direction but may also vary between different locations in the foam, for instance, when the cell morphology varies in the cross-section of the foam. Therefore, the contribution of radiative heat transfer must always be experimentally verified to check if the estimated solid, gas and radiative contributions add to the measured thermal conductivity.

## Factors Affecting Cell-gas Composition

During the manufacture, the temperature of the foam is always higher than that in the environment. For instance, for the sprayed polyurethane foam manufactured with different blowing agents the maximum temperature varied between 124 and  $178^{\circ}C$  [8]. When the exothermal reactions are completed and the foam cools to the ambient temperature, the cell gas pressure becomes reduced. Furthermore, a fraction of blowing agent becomes attracted to molecules of the polymeric matrix and absorbed on the cell wall surface or dissolved within the polymeric matrix.

In addition to processes taking place within the foam, gas diffusion will strive to equalize concentration differences between the foam and the environment. Diffusion of oxygen and nitrogen has been extensively studied on polymeric films [9], and on different foams [10]. Shankland [11] established a relation between the effective foam diffusion coefficient to that through the polymeric film. Booth [12] uses the slope of thermal conductivity plot against time (see later discussion on normalized thermal resistivity i.e. inverse of conductivity ) to derrive effective diffusion coefficient for the blowing agent. This work also supports the findings of Shankland [11] that the ratio between film and foam diffusion coefficient is constant.

Another approach to rapid measurements of gas diffusion is based on the sorption technique [13,14]. Progress was made in methods to calculate diffusion coefficient [15] and the sorption technique is now used for routine measurements of diffusion for oxygen, nitrogen and carbon dioxide through foams. Its use for blowing agents that may condense within the studied temperature range is, however, uncertain. This uncertainty is caused by the following factors:

- experimental errors are increased when period of testing becomes 20-100 times longer tan that for nitrogen,
- if more than one BA is used the rate of each BA diffusion may be affected by the presence of BAs,

3) conditions of exposure prior to testing are usually unknown, making also unknown the initial distribution of BA in the specimen.

Progress was also made in understanding long-term solubility of the blowing agents. A method based on the beginning of condensation was proposed [16] and applied to study long-term changes in the cell-gas composition [17]. To eliminate the exchange between the specimen and its environment, the experiments were performed on encapsulated specimens. Dependence of thermal resistivity on temperature was measured at specified time intervals. All tested specimens showed shift between curves determined at different time. Using a model of aging and comparing various calculated curves with those experimentally obtained, two reasons for these shifts were established:

- (i) redistribution of the gases in the enclosed space, and
- (ii) sorption of various gases by the polymer matrix, which affects the composition of the cell-gas.

The extent to which the latter process comprises a surface phenomenon and penetration of the cell-gas into the polymer matrix is not known. While these mechanisms may yet to be understood, their effect on long-term thermal performance of the foam must be recognized.

The significance of blowing agent solubility was demonstrated in the research on sprayed polyurethane foams. For a variety of these foams, immediately after the foaming and cooling, the BA partial pressure ranged between 0.65 and 0.70 atmosphere [17]. Measurements performed on a fresh foam manufactured with CFC 11 (Base 88), using indirect test method [18], indicated partial pressure of CFC 11 in the range of 0.60 to 0.65 atmosphere. The blowing agent pressure was also determined on the same material using the beginning of condensation method. CFC 11 pressure determined on a few week old Base 88 foam was 0.53 atmosphere [16], which is close to the values reported for polyurethane boardstock [19]).

One year later, however, the CFC 11 pressure determined on some encapsulated specimens of the same material was found to be much lower, namely 0.43-0.44 atmosphere. It appears that the long-term equilibrium between the blowing agent and the polymer (solubility) and a short-term equilibrium, may differ.

Two kinds of specimens were tested: sealed (encapsulated with epoxy resin) and open (exposed to air on all sides). It was found, that after one year time the concentration of blowing agent in 10 mm thick specimen was the same, even though air ingress caused their thermal resistivity to differ

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for approximately 10%. The fact that the results obtained on encapsulated and non-encapsulated specimens were practically identical indicates that the BA pressure reduction cannot be fully attributed to the outward diffusion of BA and that the total solubility of the blowing agent (either BA absorption on the surface or BA dissolution in a polymer matrix) must also be taken into consideration. However, as all these processes are slow and require long testing time, the need to reduce testing time becomes apparent.

#### HOW TO REDUCE TIME OF AGING TESTS

As discussed in previous section both the solubility and permeation of the blowing gas through the polymeric matrix are very slow and accelerating these processes in some form becomes a key to the successful evaluation methodology. Two such techniques are discussed in the following sections.

### Effect Of Elevated Temperature.

Traditionally, an acceleration of the aging process was thought to be achieved by exposing the foam to an elevated temperature. Indeed, as proved by experiments [10,20], permeation of cell gases follows the Ahrenius relation, and performing test at elevated temperature will undoubtedly increase the rate of permeation.

Since change of temperature from 24 to  $90^{\circ}$ C increases permeance of nitrogen and CFC 11 by factor of 20 and 50 respectively, one could infer that exposing 25 mm thick specimen for six months at  $60^{\circ}$ C should be comparable to at least 10 years of room aging. This observation led to the so called "accelerated aging" procedures introduced to some material standards [21]. However, a few cases of prolonged exposure of polyurethane foams at elevated temperature reported by the SPI k-factor task force [22] did not show the anticipated reduction of thermal performance.

To highlight limitations of the "accelerated aging" approach for prediction of long-term thermal performance, a series of comparative tests were performed on 25 mm thick polyisocyanurate foam specimens. Three exposure conditions were used: constant temperature oven at 100 and 60°C and 21°C room for the control specimens. The exposure to elevated temperature lasted for one year only. Then, the specimens exposed to elevated temperature were placed in the room. All measurements of thermal resistivity, independent of the conditions with 24°C mean temperature. Figure 1 shows measurements performed during exposure to elevated temperature, solid line those performed in the following period of laboratory storage. Measurements of control specimens are also shown.



Figure 1. Thermal resistivity of polyisocyanurate specimens exposed for 12 month to elevated temperature as compared with control specimens kept in the room.

If 90°C is supposed to give 50 times increase in CFC diffusion, one year exposure to 100°C should have given a reduction in CFC content in excess of the 50-year laboratory exposure. The foam was later stored in room to reach equilibrium between different components of blowing agents and allow diffusion of air components. Assuming that 9 month is sufficient for most of these equalization processes [17], the reduced CFC content cause the thermal resistance of these specimens to be lower than that of control specimens kept in the room. Figure 1 shows no such a difference. Except for the initial period of exposure, when a higher rate of thermal resistivity change may be observed, there is no lasting effect of elevated temperature.

One or two years later, one can not observe any significant difference between the specimens exposed for one year to 100°C and control specimens kept in the laboratory

room. Specimens exposed to  $60^{\circ}$ C for some time showed reduction in thermal performance slightly larger than those exposed at  $100^{\circ}$ C. Both cases, however, the reductions seen in Figure 1 are dramatically lower than those anticipated from the so called "accelerated" aging.

One may therefore postulate that while the elevated temperature increases the apparent rate of gas transport, it does so as a combined effect of a few factors. For instance, elevated temperature may change the equilibrium between gas and adsorbed phase; it also may affect the solubility of various gases in the solid polymer matrix, however, each to a different degree. As reaching these equilibria may require significant time elapse, see [17], a strict interpretation of results obtained from thermal testing requires that the specimens were subjected to invariable temperature for a long period time.

The analysis of gas permeation alone is not sufficient to describe the aging process. Furthermore, since no correlation between the "accelerating effect" of the elevated temperature and the rate of aging under field conditions can be established, other means to reduce the duration of experiments such as testing thin material layers must be considered.

#### Testing Thin Material Layers

The theory of mechanical similitude, when applied to a process of diffusion through an infinite slab exposed to constant boundary conditions, defines the transfer phenomena with the help of dimensionless quantities: Fourier and Biot numbers and the relative distance (distance from the surface, x, divided by a characteristic dimension, L). When an average property (integrated from 0 to L) is sought under conditions of very high Biot number (negligible surface resistance) then the only requirement necessary to maintain the similitude of transport processes is that Fourier number is constant [23].

Assuming constant diffusion coefficient, the requirement of Fourier number being constant relates the characteristic dimension to the period needed to achieve a specific degree of aging. This knowledge is the basis of so called scaling concept [24-26]. Precise determination of the characteristic dimension (see later text), permits testing very thin material layers, and therefore results in a significant reduction of testing time, see Figure 2. This figure, derived from model calculations [27], shows how the time needed to achieve the secondary stage of aging (when ingress of air components is mainly completed) depends on the specimen characteristic dimension. One may see that, depending on the specimen thickness, the same stage of aging will be reached after few hours, few days or many years.



Figure 2. Time needed to reach the secondary stage of aging, when the ingress of air is essentially completed, on sprayed polyurethane specimen.

## SLICING AND SCALING TECHNIQUE

As discussed in previous section, testing thin material layers has definite advantages over other techniques for "acceleration" of aging phenomena. Generally speaking, to relate these results for full board thickness, one needs an aging model. But then, in some specific cases one can use the simplest form of modelling, namely the scaling factors. The following section reviews the use and limitations of scaling factors.

#### Normalized Thermal Resistivity-time Curve

For a homogeneous material the scaling factor is:

$$\frac{t_2}{t_1} = \frac{L_2^2}{L_1^2}$$
(1)

where:  $t_1$  and  $t_2$  are periods to arrive at the same stage of aging and  $L_1$  and  $l_2$  are the characteristic dimensions that represents a diffusion path in the respective slab.

Note that the diffusion path may differ from the geometric thickness. While there is a small fraction of open cells in each "closed cell foam", this fraction may be increased on the surface by cutting and preparing thin layer for testing, and the actual path of diffusion becomes shorter than the geometric thickness. Therefore, an equivalent thickness of the open cells at the specimen surfaces was introduced as "the thickness of destroyed surface layer" (TDSL) [26,28]. Subtracting two values of TDSL from the geometrical thickness one obtains the effective thickness of the layer.

Some researchers claim that for 10 mm thick layers effect of TDSL may be neglected. Using TDSL results from literature [28] for 50 mm thick extruded polystyrene with TDSL of 0.22 mm one finds error in the time estimate equal to 9 percent (negligible), but for 50 mm thick layer cut from the core of a phenolic foam, TDSL is 1.06 mm and error of 61 percent is determined. Thus measurement of TDSL is necessary before a decision can be made on the magnitude of the error.

Statistical considerations require that the aging curve is established on the basis of thermal resistivity - time curves determined for a number of specimens. These specimens may have different thickness. While equation (1) may be used to recalculate the aging curves from one thickness to another, it is more convenient to use the same basis of comparisons, for instance the reference thickness of 10 mm. Furthermore, a prolonged exposure involving temperature and humidity variations may affect thickness of the specimen, (e.g. differences in specimen thickness of the specimens should be used in these calculation of thermal resistivity.

Though it may be assumed that the foam samples are homogeneous, in practice this is never achieved. Hence, the aging curves, calculated for the reference thickness from experimental data on several test specimens from the same sample, instead of being unique and well defined curve, will be a family of curves, with some displacement from one another. From this family of curves, one may select a curve

in the middle of the band (average) and normalize it by dividing each value by its initial thermal resistivity. The process of normalization also reduces errors associated with the so-called thickness effect (thermal resistivity varies with the specimen thickness).

The normalized aging curve together with the initial thermal resistivity data may be used to predict long-term thermal performance of the foam. One of the methods employed for such a prediction is called scaling.

#### <u>Use Of Scaling For Prediction Of Long-term Performance</u>

Measurements are performed with two different objectives:

- to establish mean value and variability in the initial thermal resistance,
- (2) to establish the normalized aging curve.

The measurements of the first type are performed using the standard ASTM test methods C518 or C177 on full boards (or thick material layers) to reduce effects of the material variability. Care must be taken to ensure that the period and conditions of aging were similar, (e.g. specimens between 7 and 14 days old) as well as that the appropriate statistical techniques were applied to ensure that these specimens represent average performance of the product. The measurements of the second type involve cutting four thin specimens, one from each surface and two from the material core, and testing them over extended period of time. If the difference between these four separate aging curves is small and one may establish a normalized thermal resistivity as a function of time the scaling approach may be used. Otherwise, one must use other techniques to predict long-term thermal performance of this material.

# Limitations In The Use Of Scaling Factors

While scaling approach may be used for all homogeneous foams, there appears to be little ground for direct application of the scaling factors to non-uniform boardstock products. For instance, a difficulty in separating primary and secondary aging stages may prevent use of the scaling factors [31]. Furthermore, scaling factors can not be considered for materials faced with impermeable barriers [31]. One must use computational procedures (models) capable of addressing statistical variations in the retardation effect of skins and facers.

### PART II: AGENDA FOR FUTURE RESEARCH

#### Introduction

The use of thin layers increases precision of thermal test by increasing uniformity of the test specimen and reduces duration of the tests, but increases the number of measurements to be performed. To reduce number of tests, one needs improved methods for material characterization and computational models. Models could estimate the effect of variation in material characteristics and would permit testing of a small number of specimens if one knew that their properties represent either the mean or the extreme case, as required. Models are also needed for evaluation of efficiency of gas barriers in retarding aging processes.

Environmental factors were shown to have marked effects on thermal resistance of foams irrespective of the blowing agent used [32]. Without better understanding of the environmental effects one can not compare results obtained under different climatic conditions nor can one interpret the results from field measurements.

Thus, research must address the following issues:

- characterization of cellular structures, (1)
- (2) the concentration of blowing agent in the cells as affected by time and temperature,
- (3) effective diffusion through a polymer matrix when a blend of two or more blowing agents is used,
- (4) effect of environmental factors on aging process,
- (5) efficiency of gas barriers.

# Methods For Characterization Of Cellular Structure

Most likely methods of material characterization are those creating an image of the cellular network from measurements of the visible light (or other electromagnetic radiation) transmission through a foam slice. Otherwise, the measurements of effective gas diffusion may also be used to characterize the foams.

The following aspects should be investigated:

- \* Examine the potential of image analysis systems with a view to developing the following techniques:
  - (a) to determine cell-size distribution and cell orientation, see progress report [33], to determine the fraction of imperfections
  - (b) (e.g. broken cells),
  - (c) to determine distribution of struts and window thickness,

- \* Prediction of the effective gas diffusion from the film permeability and foam characterization.
- \* Use of rapid gas diffusion measurement with helium (carbon dioxide) to characterize the foam structure.

# Concentration Of Blowing Agent In The Cell-gas

This is a key information for both modelling and product development. This information can be obtained from the following:

- maximum foaming temperature and BA solubility measurements,
- 2. beginning of condensation (BC) method.

The first approach comprises determination of the maximum temperature, which, assuming that cell-gas pressure during the foaming is equal to approximately one atmosphere, permit calculating BA pressure after foam cools to the ambient temperature. Further reduction in the BA pressure occurs when the gas penetrates into the polymer matrix. This reduction could be estimated from BA content in dissolved or crushed polymer [17,34].

The second approach comprises determination of thermal resistivity as a function of temperature, to find out when BA begins to condense, [16,17]. This approach, called BC method, permits estimating the actual BA concentration without destruction of the specimen and therefore may be used in the study of long-term BA solubility as well as effects of the environmental factors on BA solubility.

The following aspects should be investigated:

- \* Develop a set of equations describing BA surface absorption and BA penetration into the polymer matrix as a function of time and temperature.
- \* Measure maximum foaming temperature and, using equations discussed above, predict blowing agent concentration as a function of time.

# Diffusion Coefficients For Captive Blowing Agents

The assumption used in the sorption technique measurements that the two components of the effective diffusion (permeability and storage) remain constant, may not hold true when such measurements are performed over an extended period of time or more than one blowing agent is used. Furthermore, determination of the effective diffusion coefficient requires knowledge of the initial BA pressure distribution in the specimen. Incorrectly assumed pressure distribution may cause significant errors. This research comprises two different approaches:

- 1. Using sorption technique, examine :
- \* Calculational procedures (models) that permit a time-dependent storage (solubility component).
- \* Application of the sorption technique to measure diffusion of two blowing agents.
- Using normalized thermal resistivity (conductivity) vs time examine: calculational procedures (models) that permit simultaneous use of two blowing agents as well as a time-dependent storage (solubility). component.

#### Effect Of Environmental Factors

Exposure to elevated temperature was often discussed in terms of cell breakage [21]. Other environmental effects, particularly those associated with moisture and temperature gradient, or freezing should also be considered in the context of cell integrity.

These considerations lead to selecting three environmental exposures:

- (a) thermal gradient alone,
- (b) thermal gradient with moisture on the hot side,
- (c) thermal gradient with freezing on the cold side.
- \* Effect of these environmental exposures on thermal resistivity, stress-strain relation, friability and effective gas diffusion through the foam.

## Efficiency Of Gas Barriers

There is a need for developing a test to control the efficiency of gas barriers and the following research projects are proposed:

- \* A method to evaluate efficiency of gas barrier based on laboratory testing (HFM apparatus).
- \* A model to relate the gas barrier efficiency established in the HFM tests to efficiency of facers on full boards.

## CONCLUDING REMARKS

Measurements of thermal resistance alone, may not provide good basis for evaluation of long-term thermal performance,

because of a large uncertainty associated with these results. This uncertainty does not originate from the test methods, since ASTM test methods C177 and C518 have high precision, but is due to the spatial variability of specimens. Material variability introduces such a large uncertainty into the results of the standard ASTM measurements [35] that the number of tests necessary for the requisite precision [36] becomes impractical. Furthermore, a few years of aging would be necessary to predict long-term performance of materials with alternative blowing agents.

Another approach to evaluation of long-term thermal performance is being developed. It involves use of thin material layers cut from different boards and across the board thickness. Use of the thin layer is effective in reducing duration of the experiment, however, it increases number of tests and causes need for model application. Use of the model, is however, necessary to include effect of selected environmental conditions. Research agenda for development of material characterization as well as further development of the aging model is proposed in this paper.

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THERMAL MEASUREMENT OF IN-SITU AND THIN-SPECIMEN AGING OF EXPERIMENTAL POLYISOCYANURATE ROOF INSULATION FOAMED WITH ALTERNATIVE BLOWING AGENTS

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ABSTRACT: This paper reports apparent thermal conductivity (k) values from field and laboratory aging tests on a set of industry-produced, experimental polyisocyanurate (PIR) laminated boardstock foamed with hydrochlorofluorocarbons (HCFCs) as alternatives to chlorofluorocarbons (CFCs). The PIR boards were blown with five gases: CFC-11, HCFC-123, HCFC-141b, and 50/50 and 65/35 blends of HCFC-123/ HCFC-141b.

The k-values were determined at mean temperatures from 0 to 50°C (30 to 120°F) using techniques that meet ASTM C 1114 (Thin Heater Apparatus) and ASTM C 518 (Heat Flow Meter Apparatus) test methods. Results on laminate boards with facers provide an independent laboratory check on the increase in k observed for field exposure in the ORNL Roof Thermal Research Apparatus (RTRA). The observed laboratory increase in k was between 8% and 11% for all three blowing agent foams for a 240 day field exposure in the RTRA.

A thin-specimen aging procedure established the long-term thermal resistance of gas-filled foams. Thin specimens were planed from the industry-produced boardstock foams and aged at 24 and 65°C for up to 300 days.

An exponential dependency of k with the quantity (diffusion coefficient X time)<sup> $\frac{1}{2}$ </sup>/thickness, provided effective diffusion coefficients for air components into the foams and blowing agent out of the foams. The foams blown with alternative blowing agents exhibited k-values 8 to 16% (average 12.7%) above CFC-11 foams under similar conditions.

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Field exposures were conducted on specimens under single ply EPDM membranes in the RTRA for over 400 days. Hourly averages of panel temperature and heat flux were analyzed to obtain k as a function of mean temperature on a week by week basis. The k-values derived from the field data provided effective diffusion coefficients for air components in the foam, which were greater than those obtained from the thinspecimen aging procedure at 24°C by 20 to 70%, but were less than the 65°C aging values by 20 to 80%.

The relative performance of test specimens of HCFC-141b under a black and under a white membrane is reported. The field data suggests that the percent increase in k over that of the foam blown with CFC-11 is, after one year of aging, 4.3% for HCFC-123 and 10.2% for HCFC-141b. This leads to the same ordering of foams as derived from the thin-specimen analysis.

KEYWORDS: polyisocyanurate foams, alternative blowing agents, thermal conductivity, thin-specimen aging, diffusion coefficients, field exposure aging, roof insulation, in-situ performance, gas-filled foams, CFC-11 and HCFC blown foams, thin heater apparatus, heat flow meter apparatus, aging, roof thermal research apparatus

#### INTRODUCTION

In the mid-1980's, it was recognized that further increases in chlorofluorocarbon (CFC) concentrations in the upper atmosphere could lead to long-term damage to the ozone layer. International recognition of this mechanism culminated in the signing of the Montreal Protocol in 1987 by more than 50 industrialized and developing countries [1]. Domestic legislation [2,3] and the Protocol address the global impact of CFCs and outline a timetable for reduction of CFC consumption. If environmentally acceptable alternative gases and foams are not available, the estimated energy impact for building applications alone is between 0.65 and 1.5 quad/year (10<sup>15</sup> Btu/year) [4].

The current effort is a cooperative industry/government program to establish the viability of alternative blowing agents that resulted from two workshops [5]. The project is directed by a Steering Committee of representatives of the sponsors and Oak Ridge National Laboratory. The purpose of the project is to determine if the thermal insulation performance of polyisocyanurate roof insulation foam boards blown with alternate agents differs from boards blown with CFC-11.

This paper describes apparent thermal conductivity (k) results obtained from field and laboratory tests during 1989 and 1990 on a set of experimental polyisocyanurate (PIR) laminate boardstock produced to evaluate the viability of alternative hydrochlorofluorocarbons (HCFCs) as blowing agents. All boardstock was manufactured from similar formulations that were not optimized for thermal performance. Commercial boardstock made in the future may differ in performance from this set. The PIR boards were prepared with CFC-11, HCFC-123, HCFC-141b, and 50/50 and 65/35 blends of HCFC-123/HCFC-141b.

#### OBJECTIVES

The cooperative project has two field tasks that are supported by two laboratory tasks. One field task is to monitor thermal performance of roof test panels of the boardstock installed in the ORNL RTRA and exposed to local seasonal weather cycles. A second field task is to examine the behavior of roof panels for a range of installation conditions in the ORNL Roof Mechanical Properties and Foundations Research Apparatus (RMPFRA). The objective of the first laboratory task (Task A) is to establish the k of boardstock from mean temperatures of 0 to 50°C prior to installation and as a function of exposure time to field conditions in the RTRA. The objective of the second laboratory task (Task B) is to establish k at a mean temperature of 24°C as a function of aging time at 24 and 65°C for specimens of three thicknesses sliced from the original boardstock.

Task A used two apparatuses that meet ASTM standards: the ORNL Unguarded Thin-Heater Apparatus (UTHA) (ASTM C 1114) [6] and the ORNL Heat Flow Meter Apparatus (HFMA) (ASTM C 518) [7]. The tests were conducted on rigid boardstock foam specimens that form the test panels to be exposed in the RTRA. Each RTRA panel was nominally 76 mm thick and consisted of two boards,  $0.6 \text{ m} \times 0.6 \text{ m}$ , each nominally 38 mm thick, with an embedded heat flux transducer (HFT) at the board interface. The embedded HFTs were calibrated in-place to allow analysis of RTRA data.

For Task B, specimen aging at  $24^{\circ}$ C was conducted under normal laboratory conditions and the  $65^{\circ}$ C aging treatment was conducted in an environmental chamber. The ORNL HFMA was used to determine k ( $24^{\circ}$ C) of the specimens as a function of aging time. The goals of Task B are to evaluate thin-specimen testing as an accelerated aging procedure and to provide a database to compare to RTRA results with predictions of an aging model. Specimen characterization tests [8] were an integral part of Task B. Characterizations included cell size, preferred rise dimensions, cell wall thickness, fraction solid in the cell wall, and foam permeability to  $0_2$ ,  $N_2$ , and blowing agent. These properties are needed for a model to predict the increase of k over the life of the rigid foam [9].

#### EQUIPMENT

# ORNL Unguarded Thin-Heater Apparatus

The k of the RTRA panels and the calibrations of the embedded HFTs (Task A) were determined from 24 to  $50^{\circ}$ C in the ORNL UTHA [6, 10, 11]. Initial tests were performed in a one-sided heat flow mode on specimens with GAF black-facers (0.64 mm. thick) in place, as recommended in ASTM C 1013 [12, 13]. The UTHA tests were performed in a two-sided heat flow mode operation for the RMPFRA panels.

The UTHA meets the requirements of ASTM C 1114-89 [6]. The UTHA is an absolute, longitudinal heat flow technique with less than 2% uncertainty for steady-state thermal resistance tests at mean temperatures from 20 to 50°C. This technique has been described in the literature [6, 10, 11]. In 1983 [11], tests were conducted on two standards from the National Institute of Standards and Technology (NIST). The ORNL results on the NIST Certified Transfer Standard were within 0.6% of NIST values at 303 K and 313 K. ORNL measurements from 22 to 60°C on SRM 1450b yielded a maximum difference of 0.9% between measurements by the two laboratories at 297.13 K. These SRMs were retested in the UTHA in 1990 (see Table 1) and agreement with the NIST values was 0.3% for SRM 1451 and 1.1% for SRM 1450b. The UTHA k-values for the SRMs were fit to better than 0.3% by a linear function of temperature. Since all of these comparisons are within the most probable uncertainty of 1.2%, the UTHA k-values reported in this paper provide an accurate description of the temperature dependency of k of several materials.

TABLE 1 -- A comparison of ORNL UTHA and NIST results (1990)

Specimen	Mean Sample Temperature (K)	Sample Density (kg/m <sup>3</sup> )	k-ORNL (₩/m·K)	k-NIST (₩/m·K)	Percent Difference <sup>a</sup> (%)
Certified Tranfer Standard, SRM 1451					
Two-Sided	303.14 313.14	9.255 9.270	0.04829 0.05180	0.04827 0.05166	0.04 0.27
SRM 1450b					
Two-sided (4 points)	297.13	127.0	0.03445	0.03485	-1.14

#### ORNL Heat Flow Meter Apparatus

The thermal conductivity of the RTRA panels (Task A) were determined from 0 to  $50^{\circ}$ C in the ORNL HFMA [14]. This apparatus was also used to determine k (24°C) of the sliced, aging specimens (Task B). This comparative heat flow meter technique is designed to meet ASTM C 518, Configuration B: two transducers, both faces [7]. The ORNL HFMA is described in Ref. 15.

As specified by ASTM C 518, the HFT is calibrated with specimens of Standard Reference Materials, SRM 1450b and SRM 1451, to establish calibration factors as a function of specimen thickness and temperature prior to a measurement campaign. The apparatus uncertainty has been established to be less than  $\pm 5$ % by tests on identical specimens in the UTHA and in other C 518 apparatuses. The two standard deviation value ( $2\sigma$ ) from a comparison to other C 518 apparatuses was 2.2% for planed polyisocyanurate boards [17]. This apparatus was used to test the RTRA panels as a two-board sandwich and to test the top and bottom boards at mean temperatures of 0, 15, 24, 32, and 50°C. All tests of RTRA panels were performed on specimens with the GAF facer in place, as recommended in ASTM C 1013 [12].

#### ORNL Roof Thermal Research Apparatus

Field comparisons were conducted on test specimens installed in the ORNL RTRA [18, 19]. It is a 3 m x 8.5 m conditioned building that can accommodate four 1.2 m x 2.4 m test specimens as sections on its roof. A cross-section of a test specimen is shown in Fig. 1. Each specimen is a 1.2 m x 1.2 m panel of two layers of 38 mm thick PIR foam insulation and a 1.1 mm black ethylene propylene diene monomer (EPDM) single-ply membrane. In addition, an HCFC-141b specimen was exposed under a white EPDM. The central 0.6 m x 0.6 m area of each specimen is used for thermal measurements and the 0.3 m perimeter is a passive thermal guard. A 50 mm square HFT [20] is mounted between the two layers of insulation in the center of the measuring section. Copper-Constantan thermocouples, all from the same spools to reduce relative errors, are placed at each boundary. Prior to installation, the initial values of k of the central area were measured and the HFTs were calibrated using the ORNL UTHA. The RTRA has a complete weather station for continuous monitoring of ambient conditions as well as solar and infrared radiation. This information and data from the thermocouples and HFTs are recorded at one-minute





FIG. 1 -- Temperature and heat flux transducer placement in one of the test specimens of the RTRA.

intervals and averaged over an hour. Analysis of RTRA data is on a week-by-week basis using the computer program PROPOR (Properties - Oak Ridge) [21, 22].

For lightweight roof systems an adequate set size is 168 hourly data points for each input parameter. In this study, the PROPOR data output consists of the k and dk/dT at the mean temperature of the specimen. For each specimen, independent calculations are carried out for the top board, the bottom board, and both boards combined (since each of these configurations have a different mean temperature each week).

#### SPECIMENS

Thermal performance tests have been conducted on two types of test specimens produced from the laminate boardstock manufactured by industry [23]: (1) panels for the RTRA and RMPFRA, and (2) thin specimens for aging at 24 and 65°C. The boardstock, blown with CFC-11, HCFC-123, or HCFC-141b, was produced in June 1989 and boardstock blown with 50/50 and 65/35 blends of HCFC-123/HCFC-141b was produced in December 1989.

The Task A test specimens were nominally  $0.6 \text{ m} \times 0.6 \text{ m} \times 38 \text{ mm}$ , with GAF black facers (0.6 mm thick) on each face for each type of blowing agent. Two of these specimens formed the central area of each 1.2 m x 1.2 m panels for the RTRA tests. A 50 mm x 50 mm x 3 mm slot was routed into the lower board to position the embedded HFT at the interface of the two boards.

Task B required 0.6 m x 0.6 m specimens of three thicknesses for each type of blowing agent. Three thicknesses, nominally 33, 19, and 10 mm were produced by planing the facer and foam from boardstock to produce one, two, and four specimens, respectively, for tests in the HFMA. The 33 and 19 mm thick specimens contained the boardstock centerline and the 10 mm thick specimen had the boardstock centerline as one face for the CFC-11, HCFC-123, and HCFC-141b boardstock. All of the blend specimens contain the boardstock centerline.

#### RESULTS

#### **Characterization**

Table 2 contains structural results obtained on the three boardstock foams produced in June 1989. The cells are elongated in the direction of boardstock production; cell wall thicknesses are between 0.3 and 0.5  $\mu$ m; and the fraction solid in the cell wall is greater for the HCFC gases. Previous studies at MIT on foams blown with HCFC-123 and HCFC-141b show a similar increase in the fraction solid in the cell wall [8].

	CFC-11	HCFC-123	HCFC-141b
Average distance between cell walls (mm) Parallel to facer Perpendicular to facer	0.24 0.16	0.20 0.15	0.27 0.16
Cell wall thickness ( $\mu$ m)	0.30	0.40	0.53
Percent solid in cell wall	17	30	38

TABLE 2 -- Structural features of boardstock blown with CFC-11, HCFC-123, or HCFC-141b

## Task A. RTRA and RMPFRA Panel Thermal Conductivity

Figure 2 shows the temperature dependence of k as measured in the ORNL UTHA and the ORNL HFMA apparatuses for the Task A specimen blown with CFC-11. The panels for the other blowing agents showed a similar temperature dependence for k, i.e., a minimum k below  $15^{\circ}$ C, a nearly linear temperature dependence above  $27^{\circ}$ C, but a displacement in k that



FIG. 2 -- The temperature dependency of the thermal conductivity of boardstock blown with CFC-11. Tested 65 days after manufacture.

depended on blowing agent and age at the time of testing. ORNL/ TM-11645 [15] provides additional results on all of the test materials. The k-values determined with the UTHA are lower than the k-values determined with the HFMA in the temperature range of overlap, but are within the experimental uncertainties for the two apparatuses. Because the UTHA is more accurate, our data analysis is weighted toward the UTHA k values. A least-squares fit was produced for both data sets. Each curve for the HFMA data showed a minimum and was displaced to lower k-values to produce agreement with the UTHA data from 27°C to 50°C. The resulting curve is shown in Fig. 2. Table 3 contains the k values (including the facers) as a function of temperature, specimen density, and specimen age.

TABLE 3 -- The thermal conductivity<sup>a</sup> of a boardstock blown with CFC-11, HCFC-123, HCFC-141b, and two blends 50/50 and 65/35 HCFC-123/HCFC-141b

Mean Temperature	Thermal Conductivity (W/m·K)					
°C	CFC-11	HCFC-123	HCFC-141b	50/50	65/35	
1. RTRA Panels Age when tested <sup>b</sup>	65 d	71 d	78 d	14 d	19 d	
-4 4 10 24 38 43	0.0179 0.0169 0.0171 0.0185 0.0205 0.0212	0.0191 0.0180 0.0181 0.0195 0.0215 0.0222	0.0211 0.0195 0.0194 0.0206 0.0227 0.0234	0.0195 0.0211 0.0217	0.0196 0.0212 0.0218	
2. RTRA Panels Age when tested <sup>c</sup> after exposure 24	334 d 0.0200	336 d 0.0216	340 d 0.0225	 		
3. RTRA Panel Density, Kg/m <sup>3</sup>						
Panel Core	44.5 32.4	44.5 32.4	43.5 32.0	46.3 34.4	44.5 33.6	
4. RMPFRA Panels Age when tested <sup>b</sup>	365 d	365 d	365 d			
24	0.0221	0.0219	0.0238			

<sup>a</sup>Includes GAF Facer.

 $^{\mathrm{b}}\mathsf{Time}$  since production when tested prior to installation in the RTRA.

<sup>c</sup>Includes 241 days of exposure in RTRA under black EPDM membranes.

The results shown in Table 3 describe the thermal performance of the respective panels just prior to their installation in the RTRA on August 28,1989. The RTRA panels blown with blends had nearly equal k  $(24^{\circ}C)$  values at an age of 14 to 19 days. The other panels at an age of about 75 days show:

$$k(CFC-11) < k (HCFC-123) < k (HCFC-141b)$$

After exposure in the RTRA for 241 days, 330 days after manufacture, the order relation remained, but the average k had increased 8.6% for CFC-11, 11.1% for HCFC-123, and 9% for HCFC-141b. Table 3 also

contains k-values for the RMPFRA panels that were one year old when tested, and these values are greater than the RTRA panels by 12 to 20%.

#### Task B: Thin Specimens Aging at 24°C and 65°C

The ORNL HFMA was used to obtain k ( $24^{\circ}$ C) values for planed specimens of three thicknesses being aged at  $24^{\circ}$ C and at  $65^{\circ}$ C. The tests were conducted periodically on stacks of one, two, or four thin specimens for aging times up to 300 days.

This provides a means to measure the two-stage diffusion process that causes foams without a transport barrier to lose some of their insulating value. Without a barrier, air diffuses into the foam cells and the blowing agent diffuses out of the foam cells or goes into solution in the plastic. This transport process changes the cell gas composition, which changes the cell gas thermal conductivity and this changes the product thermal resistance. The thinner specimens show a more rapid change in k because of the shorter diffusion distance to the specimen centerline.

The results given in Table 4 for  $24^{\circ}C$  aging confirm the premise of the test procedure. Reference 15 provides results for aging at 65°C. Except for the initial values, the 54 k-values for the 24°C aging study (3 materials, 3 thicknesses, and 6 test times) show that for each thickness and time the order of the material k-values is:

$$k$$
 (CFC-11) <  $k$  (HCFC-123) <  $k$  (HCFC-141b)

and for each material and time:

k (33 mm) < k (19 mm) < k (10 mm).

The material order is the same as observed for the tests on the RTRA panels. These 54 k-values are smooth, monotonic functions of time divided by (thickness)<sup>2</sup>,  $t/h^2$  [15, 16].

We have found that the non-linear behavior of the increase in k with time/(thickness)<sup>2</sup> can be described by two linear regions if one plots ln 100 k versus t<sup>k</sup>/h. Empirically, k can be described by an exponential dependence on effective diffusion coefficient (D), time (t), and thickness (h):

$$k = k_o \exp\{(Dt)^{\frac{1}{2}}/h\} , \qquad (1)$$

where ko is the initial thermal conductivity, then

$$ln \ k = ln \ k_o + (Dt)^{\frac{1}{2}}/h$$
if: 
$$A = ln \ k_o$$

$$Y = ln \ k$$

$$X = t^{\frac{1}{2}}/h$$

$$B = D^{\frac{1}{2}}.$$
then: 
$$Y = A + B \ X$$
(3)

		Thermal Conductivity, $(W/m \cdot K) \cdot 10^3$				
Age (days)	Boards	CFC-11	HCFC-123	HCFC-141b		
3	1	18.29	19.78	20.52		
	2	17.40	19.36	20.28		
	4	19.07	20.35	21.77		
17	1	17.87	19.82	20.94		
	2	18.55	20.73	22.13		
	4	20.05	21.85	22.79		
51.5	1	18.97	20.52	21.81		
	2	20.14	22.16	23.67		
	4	21.67	23.40	24.45		
106.5	1	19.07	21.12	22.21		
	2	21.29	23.71	24.48		
	4	22.50	24.25	25.01		
190	1	20.45	21.95	23.16		
	2	22.21	24.82	24.94		
	4	22.96	24.65	25.49		
290	1	20.77	24.44	23.44		
	2	23.11	24.71	25.79		
	4	23.48	25.41	26.19		

TABLE 4 -- k (24°C) values for planed specimens aging at 24°C

Table 4 (cont'd)

		Thermal Conductivity,	$(W/m \cdot K) \cdot 10^3$
Age	Boards	50/50	65/35
2	1	19.76	19.92
	2	19.62	19.50
	4	21.17	20.94
42.5	1	20.97	21.43
	2	23.02	22.77
	4	25.36	25.31
74.5	1	21.95	22.76
	2	24.69	24.64
	4	26.14	26.02
127	1	21.74	22.63
	2	25.24	24.62
	4	26.66	26.12

Thus, if one measures the k of a foam product of thickness (h) as a function of aging time (t), then a plot of Y versus X should yield a straight line with slope B. A least square fitting of the data to Eq. 3 yields an intercept of  $ln \ k_o$  and a slope of  $D^{\frac{1}{4}}$ .

Figure 3 shows the increase of k (24°C) (plotted as  $ln \ 100 \ k$  for convenience) as a function of  $t^{*}/h$ , (days<sup>\*</sup>/mm) for specimens of three





thicknesses of foam blown with CFC-ll and aged at 24°C. The test data for the specimens of three thicknesses describe two linear regions of behavior, with an intermediate transition zone. The thin specimens reach larger values of t<sup>\*</sup>/h than the thick specimens. We believe the first linear region should be associated with the increase in k due to the influx of air components and the second, lower slope, linear region should be associated with the loss of CFC-ll from the foam. The results of six tests up to 290 days after planing are similar to the MIT model predictions [9, 24] for 50.8 mm thick specimens aged for 15 days (t<sup>\*</sup>/h of 0.76) at 24°C. The predictions are higher in k due to the model assumptions, but the behavior of k with t<sup>\*</sup>/h is supportive of the test results.

Figure 4 includes the model predictions and the test data on three thicknesses of foam blown with CFC-11 aged at 65°C. The two linear region behavior occurs for aging at 65°C with larger values of k (24°C) that are closer to the model predictions. For example, at 65°C, the linear region extrapolations intersect near a value of  $t^{\frac{1}{2}}/h$  of 0.25 days  $\frac{1}{2}/mm$ , but at 24°C this intersection is near 0.55 days  $\frac{1}{2}/mm$ . This result shows that, as expected, the diffusion of air components into the foam is faster at 65°C than at 24°C.

We treated the model predictions for  $24^{\circ}$ C aging and the available test data for the foams blown with five gases aging at  $24^{\circ}$ C and at 65°C, as suggested by Eqs. 4 and 5 [15, 16]. The specific equations used were

$$ln k (Region 1, Air) = ln k_1 + (D_1 t)^{\frac{1}{2}}/h$$
 (4)

$$ln k (Region 2, Blowing Agent) = ln k_2 + (D_2 t)^{\frac{1}{2}}/h$$
 (5)

- where  $k_1$  is the projected initial k of the foam (Region 1), W/m·K,  $k_2$  is the intercept for Region 2, W/m·K,
  - $D_1$  is the effective diffusion coefficient for air components into the foam,  $\mbox{cm}^2/\mbox{s},$  and
  - $D_2$  is the effective diffusion coefficient of the blowing agent out of the foam,  $\rm cm^2/s.$

Table 5 is a summary of the data fits obtained by the method of least-squares. The average deviation is less than 1% for all of the results, but is 2% for that of the blends in Region 2 and aging at 65°C. The square root of the B coefficients for the data fits provides the effective diffusion coefficients for Region 1 and Region 2. The resulting values for  $D_1$ ,  $D_2$ , and the ratio,  $D_1/D_2$ , are given in Table 6. The effective diffusion coefficients derived from the aging tests are the same order of magnitude as the MIT program. The results for aging at 24°C show  $D_1$  values near 1.5 x  $10^{-8}$  cm<sup>2</sup>/sec for the foams blown with individual gases and  $D_1$  values near 2.5 x  $10^{-8}$  cm<sup>2</sup>/sec



FIG. 4 -- Increase in k (24°C) for thin specimens of rigid board foamed with CFC-11 aging at 65°C compared to MIT model predictions.

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Specimens aging at 24°						
Blowing Agent	Region	¥	B × 10 <sup>4</sup>	Number of Data Points	Average Percent Deviation %	Intercept <u>k</u> (W/m·K)·10 <sup>3</sup>
CFC-11	2	0.5533 0.7528	1.230 0.2063	12 6	0.65 0.39	17.39 21.23
HCFC-123	- 0	0.6445 0.8454	1.218 0.1643	12 6	0.41 0.57	19.05 23.29
HCFC-141b	- 0	0.7021 0.8904	1.1437 0.1329	12 6	0.52 0.54	20.18 24.36
50/50 Blend	- 0	0.6622 0.8599	1.5692 0.3582	60 -4	0.63 0.11	19.39 23.63
65/35 Blend	- 2	0.6575 0.8548	1.6070 0.3462	8 4	0.56	19.30 23.51
Specimens Aging at 65°C						
CFC-11	1 2	0.5972 0.8025	3.2829 0.3015	9 9	0.64 0.51	18.17 22.31
HCFC-123	1	0.6841 0.8463	2.6102 0.3087	ŶŶ	0.57 0.40	19.82 23.31
HCFC-141b	- 2	0.7429 0.9087	2.7556 0.2641	96	0.49 0.83	21.02 24.81
50/50 Blend	2 7	0.6419 0.8320	2.9327 0.7231	or or	0.08 1.75	19.00 22.98
65/35 Blend	- 0	0.6699 0.8333	2.7907 0.7013	<b>6</b> 6	0.64 1.88	19.54 23.01

	$D_1 \times 10^8$	$D_2 \times 10^{10}$	D <sub>1</sub> /D <sub>2</sub>
MIT Model (CFC-11)	18.78	9.23	203.5
<u>24°C</u>			
CFC-11 HCFC-123 HCFC-141b 50/50 65/35	1.51 1.48 1.31 2.46 2.48	4.26 2.70 1.77 12.83 11.99	35.4 54.8 74 19.2 21.5
<u>65°C</u>			
CFC-11 HCFC-123 HCFC-141b 50/50 65/35	10.78 6.81 7.59 8.60 7.79	9.09 9.53 6.98 52.3 49.2	118 71 109 16.4 15.8

TABLE 6 -- Effective diffusion coefficients derived from aging tests,  $cm^2/sec D_1$  (air components),  $D_2$  (blowing gas)

for the blends. The D<sub>1</sub> values for aging at 65°C are three to seven times larger than the D<sub>1</sub> (24°C aging) as would be expected for temperature dependent diffusion processes. The results for aging at 24°C show D<sub>2</sub> values significantly lower than the D<sub>1</sub> values. The D<sub>2</sub> values range from 1.8 to 4.3 x 10<sup>-10</sup> cm<sup>2</sup>/sec for the individual gases and are above 12 x 10<sup>-10</sup> cm<sup>2</sup>/sec for the blends. The D<sub>2</sub> values for aging at 65°C are 2 to 4 times larger than the D<sub>2</sub> values for aging at 24°C. Some evidence exists that the D<sub>2</sub> values for the foams aging at 24°C are lower for the more complex alternative blowing agents, i.e., D<sub>2</sub> (HCFC-123 and HCFC-141b) < D<sub>2</sub> (CFC-11). The D values are a clear reflection of the foam structural properties and the diffusing species and may be a guide to optimizing boardstock. If one plots the D<sub>1</sub> and D<sub>2</sub> values obtained at 24°C and 65°C as a function of 1/T (K), then the activation energy for Region 1 is near 8000 cal/mol and for Region 2 is near 6000 cal/mol [15].

The constants, A, given in Table 5 can be used to obtain the intercept values  $k_1$  and  $k_2$  for Regions 1 and 2 given in Table 5. The values of  $k_1$ , the initial k of the foam, can be used to compare the impact of the blowing agents before any aging occurred. The 24°C aging results show that the order of  $k_1$  values from low to high are: CFC-11, HCFC-123 (9%), 50/50 (11%), 65/35 (11%), and HCFC-141b (15%), where the value in parentheses is the percent increase in k over that of the foam blown with CFC-11. The 65°C aging results suggest a slightly different order.

It has been suggested [25, 26] that the k of a 32 kg/m<sup>3</sup> fresh (unaged) foam can be calculated by adding the blowing gas k and a constant term, 0.01053 W/m·K, to represent the solid and radiation conduction contribution to k. Table 7 shows that this type calculation overestimates  $k_1$ , but the percent difference between k (calculated) and  $k_1$  is less than 10%. An alternative calculation is to subtract k(gas) from  $k_1$  and associate the difference with k(solid) + k(radiation). Table 7 shows that the result of doing so yields an average value of 0.0092  $\mathbb{W}/m\cdot K$  for the prototypical foams being tested in this project.

Gas	Gas k (W/m·K)·10 <sup>3</sup>	Foam k Calculated (W/m·K)·10 <sup>3</sup>	* Difference <sup>a</sup>	k <sub>1</sub> -k(g) (W/m·K)·10 <sup>3</sup>	Е m <sup>-1</sup>
CFC-11 HCFC-123 HCFC-141b	8.22 10.48 10.10	18.75 20.91 20.63	-7.8 -9.8 -2.2	9.17 8.67 10.08	1716 1831 1575
50/50 65/35	10.24 10.28	20.77 20.81	-7.1 -7.8	9.15 <u>9.02</u> 9.22	1772 <u>1745</u> 1728

TABLE 7 -- Calculated k (24) for unaged foams for various gases

<sup>a</sup>100 x  $[k_1 - k (calculated)]/k_1$ .

If half of the solid and radiation contribution is assigned to the radiation contribution [25], and if this is described by the Rosseland approximation [27, 28],

$$k(Rad) = \frac{16}{3} \times \frac{\sigma n^2 T^3}{E}$$
(6)

where  $\sigma$  is the Stefan-Boltzmann constant,

n is the index of refraction, and

E is the extinction coefficient,  $m^{-1}$ ,

then the E-values in Table 7 can be computed from Eq. 6. The average E-value is 1728 m<sup>-1</sup>. Theory predicts a value of about 2950 m<sup>-1</sup> for a foam density of 31.2 kg/m<sup>3</sup> and a cell size of 0.2 mm [24]. This E-value would be obtained if 30% of k(radiation) + k(solid) had been associated with k(radiation) instead of 50%.

The accelerated aging test results from this study may be used to predict the thermal resistivity (r, where r=1/k) at 24°C as a function of aging time at 24°C or 65°C for 38.1 mm thick unfaced prototypical foam boards. Table 8 shows initial and predicted r-values for one, two, and five years, which correspond to values of t<sup>1/2</sup>/h, (days)<sup>1/2</sup>/mm, of 0.50, 0.71, and 1.12, respectively, for a 38.1 mm thickness. The predicted r-values decrease with time at 24°C, but exceed 39.5  $m \cdot K/W$ for the individual gas blowing agents tested. This value exceeds the minimum stabilized r-value of 38.8 for unfaced PUR or PIR foams stated by the Society of the Plastics Industry assessment [29]. All of the values reported in Table 8 are interpolated values. The results given in Table 8 show that thin specimen aging is a promising accelerated aging procedure that is well worth pursuing. This yields a positive response to the key question of this program: the blowing agents tested in these prototypical experimental boards exhibit a long-term thermal performance at 24°C that is within 8 to 16% (average 12.7%) of that obtained by CFC-11 under similar conditions.

	Init	tial	A	ging Time, Yea	irs
Blowing Agent	k <sub>1</sub> <sup>a</sup>	r <sup>b</sup>	1 r(1) <sup>c</sup>	2 r(2)	5 r(5)
<u>At 24°C</u>					
CFC-11 HCFC-123 HCFC-141b 50/50 65/35	17.45 19.05 20.18 19.18 19.30	57.30 52.49 49.55 51.71 51.81	47.84 43.75 41.74 41.11 40.70	45.13 41.46 39.93 39.31 39.59	44.02 40.70 39.31 37.65 37.92
<u>At 65°C</u>					
CFC-11 HCFC-123 HCFC-141b 50/50 65/35	18.17 19.82 21.01 19.00 19.54	55.04 50.45 47.60 52.63 51.18	42.85 40.97 40.84 39.03 39.10	42.08 40.21 38.13 37.44 37.58	40.56 38.76 36.95 34.32 35.53

TABLE 8 -- Predicted thermal resistivity at 24°C for unfaced 38.1 mm thick prototypical foam boards aged at 24°C and at 65°C. Prediction used laboratory data.

 $^{a}_{b}$ (W/m·K) 10<sup>3</sup>.

m·K/W.

Number in parentheses indicates years.

# ROOF FIELD EXPOSURE

Weekly k-values collected for a 67-week period from September 1989 until December 1990 are shown in Fig. 5 for the two-board test systems containing CFC-11, HCFC-123, and HCFC-141b boards under the black EPDM. These k-values are reported at the combined mean board insulation temperatures experienced during that week and varied from  $5^{\circ}C$  (41°F) in the winter of 1989-90 to 32°C in the summer of 1990. The average annual temperature during the first complete annual cycle of field exposure was 21°C. After 538 days from production the relative ordering of performance of all three boards remains the same. Although the performance difference between the CFC-11 and HCFC-123 specimen was insignificant for several summer months in 1990.

Weekly k-values collected for a 43-week period from January 1990 until December 1990 are shown in Fig. 6 for the HCFC blends. These boards were not exposed to the same (temperature/time) conditions as the other three boards, so the relative comparison to CFC-11 is a bit misleading. The slight dip in k-value, shown for the CFC-11 specimen does not occur for the blends because these boards were installed after the period of cold temperatures, which occurred in winter of 1989-90. The mean temperatures of the 50/50 and 65/35 blend (HCFC-123/HCFC-141b) boards were higher during their initial aging period than that of the exposure for the CFC-11 boards. There is a very small thermal performance difference between the two blends for most of the summer which occurred between days 150 and 300.



Field k-values at Weekly Mean

FIG. 5 -- Field k-values for boards under black EPDM facers as a function of time since production using various blowing agents.

A second objective of the field testing of HCFC blown roof insulation boards is to determine if roof application practices might be identified which would reduce the rate of thermal aging. The testing of HCFC-141b under both a black and white membrane was designed to determine if a reduction of in-service mean insulation temperature would reduce the rate of thermal aging. Figure 7 shows the k-value of the HCFC-141b test specimens under a white and a black EPDM membrane for a 445 day field exposure. The k-value is displayed for a mean insulation temperature of 24°C. The temperature effect on RTRA k-values was adjusted using the k-value temperature relationship developed by steady state laboratory measurements of the test specimens prior to installation on the RTRA shown in Fig. 2. The equation used to fit the laboratory data was used to derive an adjustment factor as a function of mean insulation temperature. This relationship was then used to convert the weekly measured k-values at actual experienced mean weekly insulation temperature to k-values at 24°C. This procedure assumes that the shape of the k-value versus mean insulation temperature remains constant for the full field testing period. Prior to field insulation the two stacks of HCFC-14lb boards were tested in the laboratory. The two stacks were measured at an age of 77 days.



Field k-values at Weekly Mean

FIG. 6 -- Field k-values as a function of time since production for boards foamed with various blowing agents between black EPDM facers.

There was no significant difference in k-value as shown in Fig. 7. Throughout the first winter season (1989-90) it appeared that the HCFC-141b boards under the white membrane did age less than the HCFC-141b under the black membrane. However, in the spring the field k-values measured the same. This was supported by the laboratory measurements shown in Fig. 7 around 336 days of age, which also showed an insignificant to small difference between these two sets of test specimens. Throughout the summer of 1990, there appears to be no difference in thermal performance between the specimens under the black and white membranes. For one complete annual cycle the mean roof insulation temperature for both test boards combined, under the white membrane was  $18^{\circ}$ C,  $3.3^{\circ}$ C less than the insulation under the black membrane 21.2°C. The maximum weekly mean temperature of the top board only under the black membrane was 34°C compared to 26.7°C for the top board under the white membrane. Surprisingly after the summer cooling season a difference in field thermal performance measurements reappears. In one week the boards under the black membrane rise in kvalue and in the opposite direction of about the same magnitude the boards under the white membrane appears to have improved. Just after this unusual behavior, which replicates that which occurred in the transition from cooling to heating in 1989, the test specimens were checked in the laboratory. The laboratory measurements are remarkably consistent with the field data throughout the summer but do not suggest a difference in k-value at the time of measurement. Τt appears as if the field measurement procedure is sensing a phenomena



FIG. 7 -- Field k-value, normalized to 24°C, for boards blown with HCFC-141b under black and white EPDM.

which is occurring only during heat flow primarily in the upward direction that is not reflected when removed from the dynamic roof environment. The field measured difference between the k-value shown for the last month between the HCFC-141b test boards under the black and white membrane is less than 4%.

The data sets shown in Fig. 7 illustrate a linear relationship between k-value (24°C) and age for the first 8-10 months of field exposure. This observed relationship can be used to derive the effective air component diffusion coefficients for field-installed foams. Using the empirical equation described in Eq. 3.

Figures 8, 9, and 10 show the increase of field measured k  $(24^{\circ}C)$ (plotted in the same form as the laboratory slicing data) as a function of time<sup>0.5</sup>/thickness ( $days^{1/2}/mm$ ) for the two-board combination of the CFC-11, HCFC-123, HCFC-141b black, and HCFC-141b white. The linear fit leads to reasonably high  $R^2$  for several cases (0.85, 0.92; 0.97, 0.97) for CFC-11, HCFC-123, HCFC-141b black, and HCFC-141b white. After about 9-10 months of field exposure, the k-values do suggest a leveling out for 38 mm thick specimens just as predicted from the aging model and slicing analysis.



FIG. 8 -- Increase in k for CFC-11. increase k (24 °C [75 °F) for CFC-123



FIG. 9 -- Increase in k for HCFC-123.

Ln 100 k



FIG. 10 -- Increase in k for HCFC-141b.

The resulting effective air component diffusion coefficients obtained from the field data are shown in Table 9 and compared with the values derived from the slicing analysis, aged at 24°C and 65°C.

The effective air diffusion coefficients appear to be higher in the faced field specimens than laboratory measurements on the unfaced sliced specimens conditioned at  $24^{\circ}$ C by 20 to 71% for those specimens under the black membrane. The same HCFC-141b boards under a white membrane lead to a much higher value than that under the black membrane. The air diffusion appears to be lower in the field specimens under the black membrane than in laboratory specimens condition at  $65^{\circ}$ C by 20 to 80%. However, the diffusion coefficient derived from the HCFC-141b boards under the white membrane is actually 21% higher than even the diffusion coefficient derived at  $65^{\circ}$ C conditioned samples.

Table 10 contains the field data at one year of age for all four faced 38 mm thick test specimens, the k-values at 24°C are compared to predictions based on laboratory slicing made after less than 200 days of actual laboratory aging at 24°C and 65°C. The slicing predictions were within  $\pm 1.5$ % of the field data normalized to 24°C for laboratory specimen conditioned at 24°C and were 10% above the field data when the sliced specimens were conditioned at 65°C. The comparison suggests that conditioning the boards at 24°C is reasonable for predicting aging behavior for roof application for at least one year aging time.

	CFC-11 Black EPDM	HCFC-123 Black EPDM	HCFC-141b Black EPDM	HCFC-141b White EPDM
RTRA Data	2.21	1.96	2.58	4.02
Steady-State Lab (24°C)	1.52	1.64	1.51	1.51
<pre>% Difference</pre>	+45%	+20%	+71%	+166%
Steady State Lab (65°C)	10.78	6.81	3.33	3.33
<pre>% Difference</pre>	-80%	-71%	-23%	21%

TABLE 9 -- Effective diffusion coefficients derived from RTRA data and slicing tests in laboratory,  $10^{-8}\ \rm cm^2/sec$  for the air components,  $D_1$ 

TABLE 10 -- Field k-value at  $24^{\circ}$ C for faced 38 mm thick prototypical foam boards using field data compared to unfaced slicing analysis prediction values conditioned at  $24^{\circ}$ C and at  $65^{\circ}$ C, for an age of one year (365 days)

		k-valu	ıe W∕m∙k	
	CFC-11 Black EPDM	HCFC-123 Black EPDM	HCFC-141b Black EPDM	HCFC-141b White EPDM
Field data	0.02127	0.02219	0.02338	0.02345
Steady state (24°C)	0.02118	0.02189	0.02372	0.02372
<pre>% Difference*</pre>	-0.4%	-1.3%	+1.5%	+1.19%
Steady state (65°C)	0.02357	0.02457	0.02557	0.02557
*% Difference*	+10.8%	+10.7%	+9.4%	+9.0%

# $\frac{100 \times [R (Lab) - R (Field)]}{R (Field)}$

#### CONCLUSIONS

This paper reports apparent k-values on a set of industryproduced, experimental polyisocyanurate laminated boardstock foams blown with five gases: CFC-11, HCFC-123, HCFC-141b, and 50/50 and 65/35 blends of HCFC-123/ HCFC-141b. The k-values were determined from 0 to 50°C using the ORNL Unguarded Thin-Heater Apparatus and the ORNL Heat Flow Meter Apparatus. The test results on panels with facers provide an independent laboratory check on the increase in k observed for a 241-day field exposure in the ORNL Roof Thermal Research Apparatus. The observed laboratory increase in k was between 8% and 11%.

Test results are reported on a thin-specimen accelerated aging procedure to establish the long-term thermal resistance of gas-filled cellular foams. These thin specimens were planed from the industryproduced boardstock foams and aged at 24°C and 65°C for up to 290 days. The resulting k-values were correlated with an exponential dependency on (diffusion coefficient x time)<sup>k</sup>/thickness and provided effective diffusion coefficients for air components into the foam and blowing agent out of the foam. This accelerated aging procedure is a promising technique and was used to predict the five-year thermal resistivity of the foams. Aging at 24°C and at 65°C showed the foams blown with alternative blowing agents exhibited a thermal performance within 8 to 16% (average 12.7%) of that obtained by CFC-11 under similar conditions.

Field thermal performance at one year of age was within  $\pm 1.5$ % of predictions from laboratory aging. Field tests in the RTRA for up to 67 weeks yielded effective diffusion coefficients for air components that were higher than those of the thin aging tests. After about 40 weeks of field exposure, air component diffusion has neared completion for specimens blown with individual blowing agents. The field performance of specimens blown with two blends were similar to each other after 25 weeks of exposure. Field exposure of HCFC-141b specimens under black and white EPDMs yielded no difference in relative thermal performance.

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FUNDAMENTAL ASPECTS OF THERMAL CONDUCTIVITY AGING AND DIMENSIONAL STABILITY OF RIGID POLYURETHANE FOAMS.

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ABSTRACT: One of the possible alternatives to eliminate the future use of CFC-11 as an essential blowing agent in rigid polyurethane foams, consists of a partial or complete replacement by carbon dioxide. The two major problems associated with the use of high levels of  $CO_2$  are first, a significant deterioration of thermal insulation capacity with time, and second, a potential loss of dimensional stability due to the rapid effusion of  $CO_2$ . The paper gives an overview of the scope and limitations of this approach based on the results of extensive computational analysis and simulation.

KEYWORDS: Polyurethane foams, thermal conductivity, thermal insulation, aging, dimensional stability, CFC-replacement

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# Introduction.

The use of ozone depleting, so-called hard CFC's as blowing agents for rigid polyurethane (PU) foams is presently experiencing considerable political and legislative pressure, to such an extent that a complete ban of these substances is expected within the forseeable future. Consequently, there exists an urgent need for new thermal insulation technologies based on alternative, non-ozone depleting products. Two of the options which are available for these technologies are:

- a. A partial or total replacement of CFC-11 by  $CO_2$ . Problems with this approach include severe dimensional stability problems for unfaced foams at high  $CO_2$  levels, as well as a significant deterioration of thermal insulation capacity (k-factor).
- b. The use of less ozone depleting HCFC's to replace CFC-11. Quite a number of potential alternatives show promising performance, especially with regard to k-factor and dimensional stability. However, availability of these substances on the long term remains as yet uncertain.

A typical profile of foam thermal conductivity versus foam density is shown in Fig.1. The corresponding contributions (%) of the radiation, the polymer conduction and the gas conduction to the total foam k-factor are shown in Fig.2. Fig.3 gives a schematic representation of the relationship of these contributions to the foam material distribution, the cell gas composition and the polymer composition and structure [1]. The coupling between these respective components is a relatively weak one, which means that, under the condition that the foam material distribution remains relatively constant for any modifications in the initial cell gas composition, the changes in the gas k-factor will be reflected directly in the total k-factor [1,2].

In this paper we will investigate what influence the initial cell gas composition and cell pressure have on the foam thermal conductivity and dimensional stability both initial and aged.





Fig. 2 Contributions to foam k-factor



Fig. 3 k-factor problem structure

#### 1.1 Thermal conductivity of gas mixtures.

From the numerous methods to calculate low pressure gas mixture thermal conductivities which have been described in the literature [3-10], the most succesful ones are based on the Wassiljewa equation [3]. This equation states that the thermal conductivity of the gas mixture,  $k_{mix}$  is a weighted sum of the thermal conductivities  $k_i$  of the pure components i:

$$k_{mix} = \sum_{i=1}^{n} w_i \times k_i \tag{1}$$

where the weight factors are equal to:

$$w_i = \frac{x_i}{\sum_{j=1}^{n} x_j \times A_{ij}}$$
(2)

 $x_i$  and  $x_j$  are the mole fractions of components i and j respectively. The  $A_{ij}$  parameters are interpreted [5] as the ratio of the efficiencies by which molecules j and molecules i impede the transport of heat by molecules i. The larger and/or heavier the molecule, the more efficient it is in impeding the transfer of heat by other molecules.

The ratio of the weight factor  $w_i$  over the mole fraction  $x_i$  can be used to define how "active" the various components in the gas mixture are in contributing to the final thermal conductivity of the mixture.

$$a_i = \frac{w_i}{x_i} \tag{3}$$

An activity coefficient  $a_i$  smaller than one denotes a gas which has an "effective" k-factor in the mixture which is lower than the pure gas k-factor. This is generally the case for low molecular weight gases in the presence of heavier, more bulky molecules. Components in the mixture with a higher molecular weight usually have a higher effective k-factor in the mixture i.e. an activity coefficient which is larger than one.

In Fig.4 the activities, the effective k-factor and the pure gas k-factor are shown for each component from a hypothetical equimolar mixture of ten different gases with different molecular weights.

Notice how the activity decreases with decreasing molecular weight of the molecules considered. Also notice that for nitrogen and oxygen the respective activity coefficients are even below 0.5 which means that for this particular gas mixture the effective k-factor of  $N_2$  and  $O_2$  is comparable and sometimes even below the effective k-factor of some of the molecules with higher molecular weights (e.g. HCFC-123). It will turn out that also in the description of k-factor aging as a function of initial cell gas composition it is essential to consider how the effective k-factor of the different components is influenced.



Fig. 4 Effective k-factor in a mixture of gases

In part the applicability of the Wassiljewa equation is due to its mathematical flexibility. For binary mixtures, minima, maxima, inflection points and S-shaped curves can all be represented by a suitable choice of the  $A_{ij}$  parameters. To estimate these interaction parameters, the most accurate method was found to be the Lindsay-Bromley correlation [4]. In this empirical equation the  $A_{ij}$  are expressed in terms of the absolute temperature and the viscosity n, the molecular weight M and the corresponding Sutherland coefficient S of the pure gas components.

$$A_{ij} = 0.25 \times (1 + \sqrt{a})^2 \times b \tag{4}$$

where

$$a = \left(\frac{n_i(T)}{n_j(T)}\right) \times \left(\frac{M_j}{M_i}\right)^{\frac{3}{4}} \times \frac{(T+S_i(T))}{(T+S_j(T))}$$
(5)

and

$$b = \frac{(T+S_{ij}(T))}{(T+S_{i}(T))}$$
(6)

 $S_{ij}$  represents the Sutherland coefficient for the mixture. A good estimate of  $S_{ij}$  is the geometric mean of  $S_i$  and  $S_j$ 

$$S_{ij} = \sqrt{(S_i \times S_j)} \tag{7}$$

The thermal conductivity as well as the viscosity increase with temperature for a pure gas. It was suggested [11] that for thermal conductivity this relation could be described as:

$$k_i(T_2) = k_i(T_1) \times (\frac{T_2}{T_1})^{n_i}$$
(8)

where  $n_i$  is a constant and  $k_i(T_1)$  is a known thermal conductivity of the gas at temperature  $T_1$ . For large temperature ranges, however, it was found that  $n_i$  is not a constant but a linear function of the temperature. So by using an equation of the form:

$$k_{i}(T_{2}) = k_{i}(T_{1}) \times (\frac{T_{2}}{T_{1}})^{(A_{i} \times (\frac{T_{2}}{T_{1}}) + B_{i})}$$
(9)

pure gas thermal conductivities can be represented within experimental accuracy over relatively large temperature ranges (for our purposes typically 250-350 deg.K).

It was found that , using a similar equation, also gas viscosity could be represented over large temperature ranges within experimental accuracy.

For binary mixtures a consideration of the product of the interaction coefficients  $A_{12}$ .  $A_{21}$  (see Eq.4) with respect to the ratio's of the pure gas k-factors  $k_1/k_2$  and  $k_2/k_1$  gives information on the variation of  $k_{mix}$  with composition (see Fig.5).



Fig. 5 Thermal conductivity of low-pressure gas-mixtures. k-mixture = function (mole-fractions, temperature, pressure,  $A_{ij}$ )  $A_{ij}$  = function (temperature, MW, gas-viscosity, boiling point)



Binary mixture Argon/n-Pentane, Temperatures 50, 100 deg.C

o o o Experimental values





Binary mixture Argon/Diethyl ether, Temperatures 50, 100 deg.C

o o o Experimental values



- a. If  $(k_1/k_2) < A_{12}A_{21} < (k_2/k_1)$  for  $k_1 < k_2$ , the  $k_{mix}$  varies monotonically with composition. This situation occurs in most of the cases where the thermal conductivities of the pure components are markedly different.
- b. If  $A_{12}A_{21} < (k_1/k_2)$  the  $k_{mix}$  versus composition curve has a maximum above  $k_2$ . This happens in many cases where one or both components are polar.
- c. If  $A_{12}$ ,  $A_{21} > (k_2/k_1)$  the  $k_{mix}$  versus composition curve has a minimum below  $k_1$ . Unfortunately this type of behaviour has only been observed in a limited number of cases. It appears that the conditions for a minimum to occur are that one of the components is a bulky, heavy molecule while the other one is a more compact and lighter molecule [6].

In figures [6,7] experimental k-factors for the case of the mixture of two non polar gases i.e. argon and n-pentane and for the case of a mixture of a non polar and a polar gas i.e. argon and diethylether, are compared with the thermal conductivities. Calculations were done at two temperatures (50 and 100 deg.C). In both cases a good agreement is found between the theoretical and the experimental results.

## 1.2 Effect of temperature and cell pressure on cell gas composition.

If one or more components of the gas mixture have their boiling point somewhere within, or above, the temperature range considered for the application, condensation can take place. This gives rise to a decrease of the mole fraction of these particular components with a concomittant change in the thermal conductivity of the gas mixture. To know whether or not a gas condenses at a specific temperature, one has to consider the vapour pressure VP(T) of the gas. This data is readily available in the form of the Antoine equation which represents the behaviour of most substances over large temperature ranges rather well.

$$VP_i(T) = e^{A_i - \frac{B_i}{(T+C_i)}}$$
(10)

Now, assuming the mole fractions  $x_i$  as well as the total cellpressure  $P_{tot}$  at ambient temperature  $T_{amb}$  are known, the partial pressure  $pp_i$  for each component in the mixture can be found according to the ideal gas law:

$$pp_i(P_{tot}(T_{amb}), T_{amb}) = x_i(T_{amb}) \times P_{tot}(T_{amb})$$
(11)

the partial pressures at a different temperature, T, can be found from:

$$pp_{i}(P_{tot}(T_{amb}),T) = pp_{i}(P_{tot}(T_{amb}),T_{amb}) \times \frac{T}{T_{amb}}$$
(12)

Condensation takes place when the vapour pressure at temperature T,  $VP_i(T)$  is lower than this partial pressure  $pp_i(P_{tox}(T_{amb}),T)$ . In this case  $VP_i(T)$  should be taken as the effective partial pressure.

$$pp_i^{eff}(P_{tot}(T_{amb}), T) = MIN(VP_i(T), pp_i(P_{tot}(T_{amb}), T))$$
(13)

The total cell pressure at temperature T can be found from:

$$P_{tot}(T) = \sum_{i=1}^{n} p p_i^{eff}(P_{tot}(T_{amb}), T)$$
(14)

Finally the mole fractions  $x_i$  at temperature T are:

$$x_{i}(P_{tot}(T_{amb}),T) = \frac{pp_{i}^{eff}(P_{tot}(T_{amb}),T)}{P_{tot}(T)}$$
(15)

In the next section it will be shown how condensation effects affect the thermal conductivity of gas mixtures.

# 1.3 Condensation in binary gas mixtures. Effect of thermal conductivity and cell pressure.

When used for thermal insulation purposes rigid foam is always exposed to a range of use temperatures. The range might vary from application to application. If no condensation occurs within this range, the insulation performance of the foam will improve with decreasing temperature. If, however, like in most practical cases where CFC-11 (or any of the alternative HCFC's being considered) is used, condensation does occur, the determination of the overall insulation performance of the foam is not straightforward. To illustrate this the gas mixture thermal conductivity as well as the cell pressure are shown as a function of composition and temperature (-10 to 40 deg.C) for the binary systems of CFC-11 with CFC-12, CFC-114 and n-pentane respectively (Figs. 8,9,10).

In all three systems there is a nearly linear change of thermal conductivity with composition in the upper temperature range. At ~20 deg.C CFC-11 starts to condense for the 100 mole% CFC-11 containing system. At -10 deg. C the cell pressure for this system has dropped from 0.84 to 0.25 bar, so foam shrinkage can be expected, especially with low density foams. The lower the CFC-11 mole fraction in the mixture, the lower the partial pressure is of CFC-11 and, consequently, the lower the temperature at which condensation will start (see e.g. Fig. 8b). For the case of the CFC-11 / CFC-12 mixture, CFC-12 does not condense within the temperature range considered (-10 to 40 deg.C) so the dimensional stability of this system is expected to improve the more CFC-12 is present. From ~65 mole% CFC-12 no further improvement is to be expected, because the amount of CFC-11 present is now low enough to prevent condensation, even at -10 deg.C.

Also observe from Figure 8a, the relative increase in the mixture thermal conductivity at low temperature when condensation occurs (the relative mole% of CFC-12 in the gas phase increases and CFC-12 has the higher pure gas k-factor).


Figs. 8,9,10 Condensation and k-factor for binary gas mixtures

The second example shows the behaviour of CFC-11 / CFC-114 mixtures (see Figs 9a,b). This mixture is similar to the CFC-11 / CFC-12 mixture with the additional feature that also CFC-114 condenses in the temperature range considered. It is now clear (see Fig. 9b) that the optimum with respect to cell pressure and expected good dimensional stability at low temperatures is now limited to the range 20-40 mole% of CFC-11.

This effect is even more extreme in the third example which corresponds to a mixture of CFC-11 / n-pentane. In this case n-pentane condenses even before CFC-11 does, so the optimum composition, as far as dimensional stability at low temperatures is concerned, is now displaced to higher levels of CFC-11. The best performance is expected at a level of -65 mole% of CFC-11. Notice, however, that for this mixture it is no longer possible to completely prevent condensation of either one of the components below 5 deg.C.

As far as the thermal conductivity of the mixture is concerned, the compositions with a high level of CFC-11 will again show an increase of the k-factor since the condensation of CFC-11 is predominant in this range (see Fig. 10a). The compositions below  $\sim 60$  mole%, however, will show a relative decrease in k-factor due to an enrichment of the gas mixture with CFC-11. In fact, for the case of the CFC-11 / n-pentane mixture, there exists a range from  $\sim 30$  to  $\sim 80$  mole% where the k-factor is essentially constant.

In practice, considering these systems, this means it should be perfectly possible to use blowing agents with a relatively high boiling point, provided they can be practically used, in combination with some other blowing agent. By selection of the optimal gas composition it is possible to improve the dimensional stability of the foam as a function of temperature, relative to foam made with a single blowing agent. In addition it is possible to benefit from the low k-factor which this high boiling blowing agent might have. These considerations can obviously be applied equally well to ternary systems, or higher.

### 2. Initial cell pressure in the foam.

The initial cell pressure in a PU foam is a function of both the formulation and the processing of the foam. Given the parts by weight of blowing agent per kilogram formulation and the average molecular weight of the blowing agents used, one can calculate the total number of moles gas generated during the foam formation.

$$m_g = 1000 \times \frac{w}{\langle MW \rangle} \tag{16}$$

The effective moles of blowing agent used for foam expansion is related to m<sub>g</sub> by:

$$m=f \times m_g$$
 (17)

where (1-f) represents the fraction of blowing agent lost during foaming. For closed cell foams the fraction f turns out to be very close to one. In the following it will be assumed that no blowing agent is lost, so f=1. The initial cell pressure in the foam can be related to m by using the ideal gas law.

$$P_{ini} = \frac{m \times R \times T_r}{V_g}$$
(18)

with R=8.31441 J/mole.K and  $T_r$ =293.15 deg.K (20 deg.C). The volume of gas,  $V_g$ , is related to the foam and polymer density as follows:

The total volume of the foam consists of polymer and gas,

$$V_f = V_g + V_p \tag{19}$$

conservation of mass gives,

$$D_{f} = D_{g} \times \left(\frac{V_{g}}{V_{f}}\right) + D_{p} \times \left(\frac{V_{p}}{V_{f}}\right)$$
(20)

combining the two previous equations and rearranging gives the well known expression for the volume fraction of the gas,

$$VF_{g} = \frac{V_{g}}{V_{f}} = \frac{D_{p} - D_{f}}{D_{p} - D_{g}}$$
(21)

Assuming the polymer density is known one can calculate the volume of polymer from:

$$V_p = \frac{(1-w)}{D_p} \tag{22}$$

The density of the gas is related to the gas volume by:

$$D_g = \frac{w}{V_g} \tag{23}$$

Inserting these relations in Eq. 21 gives the relation between foam density and gas volume we require:

$$V_{g} = \frac{(1-w) \times (D_{p} - D_{f}) + w \times D_{p}}{D_{p} \times D_{f}}$$
(24)

which leads to:

$$p_{ini} = \frac{1000 \times w \times R \times T_r \times D_p \times D_f}{\langle MW \rangle \times ((1-w) \times (D_p - D_f) + w \times D_p)}$$
(25)

The experimental foam density  $D_{fa}$ , includes the buoyancy effect of air,

$$D_f = D_{fa} + D_a \tag{26}$$

where  $D_a = 1.197 \text{ kg/m}^3$  at 20 deg.C.

Since Eq. 25 applies to any density of the foam (free rise and molded), it can be used to relate the initial cell pressure of a molded foam to the initial cell pressure of the corresponding free rise foam.

$$p_{ini}^{md} = p_{ini}^{fr} \times (\frac{D_{md}}{D_{fr}}) \times (\frac{(1-w) \times (D_p - D_{fr}) + w \times D_p}{(1-w) \times (D_p - D_{md}) + w \times D_p})$$
(27)

Using this equation the initial cell pressure in a foam can be calculated from the experimental foam density and the moles of blowing agent per 100 g of formulation (see Fig. 11). A free rise foam typically has an initial cell pressure in the core of  $\sim 0.7$  bar. Molded foams have higher initial cell pressures, the value of which depends on the difference between molded and free rise density (see Fig.12). The polymer density  $D_p$  ( $\sim 1150 \text{ kg/m}^3$ ) and the average molecular weight of the blowing agents used have only a minor influence on the resulting initial cell pressure.

The effects of foam processing on the initial cell pressure (and foam density), can be more easily quantified using a different expression for  $p_{ini}$ .

$$p_{ini} = P_{\max} \times \frac{T_r}{T_{\max}}$$
(28)

where  $T_{max}$  is the maximum temperature reached in the foam during its formation and  $P_{max}$  is the corresponding pressure. The assumption here is that the foam volume does not change anymore when the foam subsequently cools down to room temperature. A free rise foam usually has a  $P_{max}$  which is only slightly above atmospheric pressure. This is in agreement



Fig. 11 Initial cellpressure [bar] in a foam as a function of the foam density [Kg/m<sup>3</sup>] and the amount of blowing agent in the formulation.



Initial cell pressure at room temperature as a function of density Moles BA/M3 =28.17 ,Free rise density = 19 to 31 by 2





with the experimental fact that polymer modulus buildup only takes place during the curing, i.e. after the final rise height has been reached. The higher the maximum temperature reached in the foam and the lower the associated

pressure, the lower the initial cell pressure (at roomtemperature) will be. Or, in other words, the more efficiently the blowing agent has been used. Related phenomena are:

- a. The decreased blowing efficiency observed when panels with a lower thickness are produced (lower  $T_{max}$ ).
- b. The known dependence of blowing efficiency on changes in atmospheric pressure (changes in  $P_{max}$ ).
- c. The lower free rise densities obtained for systems with an increased  $T_{max}$  (about 1 Kg/m<sup>3</sup> decrease in density for an increase of 10 deg.C in  $T_{max}$ ).
- d. Because the conditions which determine the initial cell pressure in a foam are not uniform throughout the foam, the initial cell pressure is not uniform also.

The initial pressure from Eqs. 25,27 is calculated with the assumption that all of the blowing agent is present in the gas phase. Depending on the temperature to which the foam is exposed, part of the initial cell pressure can be lost due to condensation effects. Another possibility is that, for some blowing agents, part of the cell pressure is lost because of their solubility in the polymer. There is quite some controversy on this last point regarding the actual amounts of blowing agent dissolved [13]. Whereas the effect on the initial cell pressure are probably very small, one cannot exclude the contribution of polymer solubility effects on the long term aging of foams.

### 3. Effect of initial cellgas composition and pressure on thermal conductivity aging.

Both the thermal conductivity of the foam as well as the pressure in the foam cells do not remain constant in time [12-17]. Generally speaking, there is a gradual outward diffusion of the blowing agents with a concomittant inward diffusion of air. The rate at which the concentration of the different gases changes is heavily dependent on the foam material distribution, the polymer composition and structure, the initial conditions and of course the actual aging conditions themselves [13]. There are a number of models available which describe these phenomena in more or less detail [1,2,12-13]. In this paper we will look at the aging from the perspective of the initial conditions (blowing agent composition and cell pressure) and how these affect the long term thermal conductivity and dimensional stability assuming all other factors remain constant.

One of the possible short term alternatives to reduce the level of CFC-11 in rigid PU foams is a partial or a complete replacement by carbon dioxide. In Figures [13-15] the effects of such a replacement on the short term aging of the thermal conductivity and the cellpressure are shown.

For the case of a 100% CFC-11 blown foam, the cell pressure increases with time since the inward diffusion of nitrogen and oxygen is much more rapid than the outward diffusion of CFC-11. So in terms of dimensional stability one would expect foam expansion rather than foam shrinkage. The thermal conductivity increases, but only to a small extend although the CFC-11 is diluted with quite a large amount of air. The reason that the concomittant increase in k-factor is still moderate is due to the fact that there is more than enough





Initial 50% Freon-11, 50% CO2





Fig. 14 Foam aging - 50% CFC-11, 50% CO2 initial



Fig. 15 Foam aging - 100% CO2 initial

CFC-11 available in the gas mixture to "deactivate" the incoming nitrogen and oxygen. It turns out that for this particular case the effective k-factor of nitrogen and oxygen are reduced by nearly 65%, which means a reduction from a k-factor of  $\sim 26$  mW/M.K. for the pure gas to  $\sim 11-12$  mW/M.K. in the presence of CFC-11.

When 50 mole% CFC-11 is replaced by carbon dioxide (see Figs. 14a,b) the cell pressure initially decreases with time since carbon dioxide diffuses out of the foam faster then air diffuses in. In this case there is consequently some potential for foam shrinkage within a few days or weeks after foam production. The initial k-factor of a 50 mole%  $CO_2$  containing foam is increased by a few mW/M.K. but the aging rate is still moderate. For some systems even an initial decrease in k-factor can be observed due to the rapid loss of  $CO_2$  with a concomittant enrichment of the cell gas with CFC-11.

For the case of a fully carbon dioxide blown foam (see Figs. 15a,b) one can expect a very severe dimensional stability problem in view of the significant minimum in cell pressure the system has to go through. This already occurs during the first few weeks because of the rapid loss of carbon dioxide. The k-factor starts at a much higher level (the gas k-factor of carbon dioxide is twice as high as the one for CFC-11) and deteriorates very quickly to a k-factor of an air filled system.

Fig. 16 summarizes the expected short term k-factor aging profiles for systems ranging from 100% CFC-11 initially to 100%  $CO_2$  in steps of 10 mole%. Notice the increasingly larger effect the replacement of another 10 mole% of CFC-11 by carbon dioxide has on both the initial and the aged k-factor despite the fact that each system contains exactly the same





Fig. 17

k-factor aging vs. initial cellgas composition

Gas-activities vs. initial cellgas composition



Fig. 18 k-factor increase vs. initial cellgas composition

amount of air at a given aging time. The reason why the deterioration in k-factor aging progresses more and more rapidly with increasing levels of carbon dioxide is largely controlled by the previously mentioned deactivation effect a high molecular weight compound (CFC-11) has on the low molecular weight components ( $CO_2$ ,  $N_2$ ,  $O_2$ ).

Looking at Fig. 17, which represents a cross section through the graph represented in Fig.16 at 12 days aging, one can observe the relatively constant, low activity of carbon dioxide, nitrogen and oxygen up to a level of about 40 mole% CFC-11. Beyond that point there is a rapid increase in the activities of all components, including CFC-11. This means that not only the effective k-factors of air and carbon dioxide are no longer reduced compared to their pure gas K-f actors, but also the effective k-factor of CFC-11 is increasing by 30 to 80 %. the results is a very rapid increase in the mixture k-factor once the level of CFC-11 drops below  $\sim 40$  mole%. Fig. 18 represents the same data from Fig.16 but now looking at the change in k-factor from the initial value with each curve representing a different aging time. The message is that it is essential to preserve a minimal amount of high molecular weight blowing agents in the system in order to keep the effective k-factors of nitrogen and oxygen as low as possible since, in most applications, these gases will enter the system no matter what.

In Figs. 19-22 the long term aging at ambient conditions as a function of initial cell gas composition and panel thickness are depicted for a number of different systems which contain various levels of CFC-11 (\_\_\_\_), HCFC-123 (---) and HCFC-141b (...). Also included for each case is the effect of an 100% effective (diffusion tight) facer material, the edges of the panel are left uncovered. These long term predictions are based on a combination of



Panel 4 ft x 8 ft, 3D simulation

### Fig. 19 Long term aging - 100% CFC-11 initial



Panel 4 ft x 8 ft, 3D simulation

Fig. 20 Long term aging - 100%/50%/15%/0% CFC-11 initial



Panel 4 ft x 8 ft, 3D simulation





Panel 4 ft x 8 ft, 3D simulation

Fig. 22 Long term aging - 50% CFC-11, HCFC-123 (--) or HCFC-141b (..) initial.

experimental data and a 3D aging model where some of the parameters have been scaled using these data. Again the intention here is to provide an assessment of the relative performance of a number of systems with different initial cell gas compositions. Based on these extrapolations a number of conclusions can be drawn:

- a. Looking at the long term performance, there is little difference between CFC-11, HCFC-123 or HCFC-141b based systems (despite the differences in initial k-factor).
- b. The importance of having a good facer material is quite clear. The simulations assume the presence of a 100% effective facer material with no adhesion problems. In practice, however most facer materials have a much lower effectiveness than this.
- c. Again looking at long term performance, 50% reduced systems show very acceptable aging profiles. The difference with respect to the corresponding 100% CFC-11 blown systems, can be attributed mainly to the increases in initial k-factor.
- d. Aging of unfaced, 100%  $CO_2$  blown systems in unacceptable, irrespective of panel thickness. Notice, however, the significant impact on aging of a good facer in combination with large panel dimensions. Also notice the significant improvements which can be obtained by the addition of a relatively small amount of a high molecular weight blowing agent to a largely  $CO_2$  blown system.

### 4. Foam dimensional stability.

As already hinted at in the previous section, aging of thermal conductivity is not the only factor which is affected when the initial cell gas composition is changed. In many cases also a consideration of the dimensional stability of the foam plays an important role in the optimization process.

Foam dimensional stability is essentially the result of a balance between some external forces and a number of internal forces. The external forces are the ambient pressure and eventually the applied load during a compressive strength measurement. The internal forces are the collapse strength of the foam [18], the cell pressure and eventually an increment to the cell pressure as a result of the compression of the sample (see Fig. 23). In case the sum of the internal components is larger, the foam will expand, otherwise the foam will shrink. The actual driving force for any changes in the foam volume with time is either an increase or a decrease in the total cell pressure due to the different diffusion rates of the gases involved.

In Fig. 24 the profiles of cell pressure versus time are depicted for a range of systems where the initial cell gas composition varies from 100% CFC-11 to 100%  $CO_2$  in steps of 10 mole%. Any volume changes will essentially follow this cell pressure profile (see Fig. 26), provided the foam structure is not destroyed in the process. To determine what density is required to widthstand the maximum pressure difference which occurs between the outside and the inside of the foam, it is important to know:

a. What is the minimum (or maximum) cell pressure the foam is going through during its life time. This is not a function of the absolute differences in diffusion coefficients of the various gases, but only of the ratios between them. In most cases it is not



Fig. 23 Foam dimensional stability



# Fig. 24 Foam cellpressure vs. time

Fig. 25 Volume shrinkage vs. time

really essential to know when this minimum or maximum will occur. As depicted in Fig. 26, the exact value of this minimum or maximum cell pressure depends on the initial cell gas composition, the initial cell pressure, the molded versus the free rise density, the temperature etc. This means that dimensional stability, like thermal conductivity aging, is inherently also very dependent on the actual application the foam is used in.

b. What is the collapse strength of the foam matrix. This information can be obtained from a consideration of the plateau value in the stress-strain profile for foam compression. The collapse strength is proportional to the polymer modulus as well as to the relative density of the foam over the polymer squared [18].

In Fig. 27 the balance of the different components (see Fig. 23) is plotted against density for three different systems with varying levels of carbon dioxide in the initial cell gas composition [30, 60, 100 mole%]. Notice how these lines are shifted to higher foam density values with increasing level of  $CO_2$ . The theoretical non-shrink density is represented by the value where these lines cross the zero line. One of the assumptions for the generation of these lines is that the foam is uniform in density (no density distribution) and that it is isotropic (no cell orientation). For any practical system however this is not generally the case so one has to build in a safety margin which usually amounts to ~50 kPa. This means that the actual density for the 30% CO2 blown system should be around 32 Kg/m<sup>3</sup>. The system with initially 60%  $CO_2$  should be foamed at densities of 35-37 Kg/m<sup>3</sup> or higher to be dimensionally stable whereas the fully  $CO_2$  blown system should be foamed over 40 Kg/m<sup>3</sup> to be stable. Any variation in polymer modulus, density distribution, free rise density etc. will of course also influence the exact value of this non-shrink density but to a minor extent.



Fig. 26 Effects on minimal and maximal cellpressure in a foam





Finite element modelling



Fig. 28 Finite element modelling of foam shrinkage

By means of the finite element method the evaluation of both the aging and the dimensional stability evaluation can be extended to handle more complicated applications where both the initial and boundary conditions can be place and/or time dependent. Fig. 28a,b show two examples where the spatial dimensions have been included into the analysis. This particular example shows that even in case a fully  $CO_2$  blown system is completely covered with diffusion tight facings, it is still necessary to foam at or above the non shrink density. If not, even small imperfections, like pin holes, could lead to a local deformation of the panel within a relatively short period of time.

## Conclusions.

Both thermal conductivity aging and foam dimensional stability are heavily dependent on the initial cell gas composition and cell pressure. To a large extent these effects can be quantified in a relative manner without an explicit consideration of the foam structure.

To preserve favorable long term aging characteristics it is essential to preserve a minimal amount of a high molecular weight blowing agent in the blowing agent compositon. The main function of such a compound would be the "deactivating" effect on the low molecular gases like carbon dioxide, nitrogen and oxygen. Another, very effective, way to preserve good long term aging properties is the use of diffusion tight facer materials in combination with large panel dimensions.

Foam dimensional stability is mainly controlled by the changes in cell pressure during foam aging. Depending on the minimal cell pressure a certain foam density is required to avoid foam shrinkage. This non-shrink density exceeds 40 Kg/m<sup>3</sup> for fully CO<sub>2</sub> blown systems. Unless systems are completely diffusion tight, and no defects are present, one can not avoid the use of increased densities without significant increases in polymer modulus.

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# **Assessments and Properties of Foams**

### J. Richard Booth

# SOME FACTORS AFFECTING THE LONG-TERM THERMAL INSULATING PERFORMANCE OF EXTRUDED POLYSTYRENE FOAMS

REFERENCE: Booth, J. R., "Some Factors Affecting the Long-Term Thermal Insulating Performance of Extruded Polystyrene Foams," <u>Insulation Materials: Testing and Applications, 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: The long-term thermal performance of insulation products is important to ensure their suitability in applications where product life can be fifty years or more. Actual measurements of foams aged under conditions of use are helpful but can only be gathered in real time. While necessary to assure the validity of model predictions, such measurements do not allow early determination of the future performance of new systems. With the need for all users of CFCs to find alternatives, the question of how to ensure longterm thermal performance takes on immediate significance. In this paper we describe some work done in our laboratory to assure the long-term thermal performance of STYROFOAM\* brand insulation foam.

The use of bundles of thin-sliced foam samples provides a technique for accelerating foam aging. Aging curves relating bundle k-factor to a scaled age consolidates the data for measurement of the slope of k-factor with scaled age. This slope can be employed to evaluate the effective diffusion coefficients of candidate CFC replacements in polystyrene foams. The ratio of effective diffusion coefficient to film diffusion coefficient yields insight concerning the cell structure of the foam.

**KEYWORDS:** XPS foams, long-term aging, CFC, HCFC polystyrene foam, cellular plastics, thermal insulation

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### THEORETICAL

Heat is transferred through foam by conduction and radiation in parallel mechanisms; convection is absent in insulating foams. Furthermore, conduction occurs through both solid and gaseous phases. Radiation and solid conduction may be evaluated from initial foam properties, and do not change with aging.

The composition of the cell gas phase changes (in air) and causes the thermal conductivity of the foam to change or "age". The aging curves for foam having different thicknesses can be related by using the **scaled age** ( $\sqrt{time/thickness}$ ). A typical aging response for foam is presented in Fig. 1, which shows differences between the initial k-factor and later kfactor values. Three regions are identified. The **thermal drift region** of the aging curve, represented by the influx of oxygen and nitrogen and the efflux of fugitive blowing agents, has a rapidly changing k-factor. The so-called **blowing agent plateau region** of the aging curve has a (relatively) slowly changing k-factor. This is where the foam is saturated with air, but the captive blowing agent escapes slowly. Finally, k-factor becomes constant when the airsaturated foam is depleted of captive blowing agent.





The aging of a foam then can be described as the sum of the constant contributions (solid conduction and radiation kfactor), and the average gas phase conduction which changes with time [1]. The fraction of heat transfer area for solid conduction is introduced, to partition the conductive heat flux into solid and gaseous contributions. In an ideal isotropic foam, for example, one having a nearly cubic cell structure and all the polymer in the windows, two-thirds of the polymer contributes to solid conduction. The ratio of foam to polymer densities provides a relative measure of the solid fraction in the foam, in the heat flow direction.

$$K_{f,t} = \left[\frac{2}{3}\frac{\rho_f}{\rho_p}\right]K_s + K_r + \left(1 - \left[\frac{2}{3}\frac{\rho_f}{\rho_p}\right]\right)K_{g,t}$$
(1)

where:

K,	= aged foam k-factor, W/m°K
к''`	= solid conduction contribution, W/m°K
ĸŗ	= radiation k-factor contribution, W/m°K
Ќ,	= gas conduction contribution, W/m°K
ρ,	= polymer density, $g/cm^3$
ρŗ	= foam bulk density, g/cm <sup>3</sup>

The average gas phase thermal conductivity of a gas mixture may be computed by a mixing rule such as the Riblett equation [2]. The average gas mixture thermal conductivity and fraction of gas (that is, non-solid) heat transfer area are used to compute the gas contribution to foam k-factor.

$$K_{g} = \frac{k_{O_{2}}M_{O_{2}}^{\prime b}P_{O_{2}} + k_{N_{2}}M_{N_{2}}^{\prime b}P_{N_{2}} + k_{BA}M_{BA}^{\prime b}P_{BA}}{M_{O_{2}}^{\prime b}P_{O_{2}} + M_{N_{2}}^{\prime b}P_{N_{2}} + M_{BA}^{\prime b}P_{BA}}$$
(2)

where:

= individual pure gas conductivity, W/m°K k, = component molecular weight, g/mol Μ,

= component partial pressure, atm pi

There are regions of the aging curve where the partial pressure of one gas component is changing at a much greater rate than the other components. When one examines the aging curve, Fig. 1, it appears that there are regions where the slope of k-factor with scaled age in the thermal drift and blowing agent plateau regions might be determined by piecewise linear regression. Because the solid and radiation conductivity contributions are constant, the slope of the aging curve is equivalent to the slope (or derivative) of gas conductivity with respect to scaled age.

The slope of the gas conductivity-partial pressure relationship may be obtained by differentiating Eg 2 with respect to the partial pressure of a gas component. Note that in Eq 2 the partial pressure of each gas component appears in both the numerator and denominator. The derivative of such an expression must be computed by parts. If only one gaseous partial pressure is changing, one can develop an expression for the derivative of the gas conductivity with respect to partial pressure of that gas component.

$$\left(\frac{1}{C}\right)dK_{g,t} - dp_{BA,t} \tag{3}$$

where:

$$C - M_{BA}^{\prime 4} \left( \frac{k_{BA}}{M_{O_2}^{\prime 4} P_{O_2} + M_{N_2}^{\prime 4} P_{N_2} + M_{BA}^{\prime 4} P_{BA}} - \frac{k_{O_2} M_{O_2}^{\prime 4} P_{O_2} + k_{N_2} M_{N_2}^{\prime 4} P_{N_2} + k_{BA} M_{BA}^{\prime 4} P_{BA}}{(M_{O_2}^{\prime 4} P_{O_2} + M_{N_2}^{\prime 4} P_{N_2} + M_{BA}^{\prime 4} P_{BA})^2} \right)$$

The proportionality parameter in Eq 3 is evaluated from the molecular weight, pure gas thermal conductivity and instantaneous partial pressure of each the components. Multiplying both sides of Eq 3, by the effective thickness L and dividing by  $d(\sqrt{time})$ , one obtains the desired expression, Eq 4, in which the slope of the scaled aging time is related to the derivative of partial pressure with scaled aging time.

$$L\left(\frac{dp_{\mathbf{BA},t}}{d\sqrt{t}}\right) - \frac{L}{C}\left(\frac{dK_{g,t}}{d\sqrt{t}}\right)$$
(4)

where:

K <sub>at</sub>	=	aged gas k-factor, W/mºK	
$p_{RA}^{3,1}$	=	instantaneous blowing agent partial pressure,	atm
с",	=	delta conductivity parameter, W/m°K/atm	
t	=	foam aging time, s	
L	=	effective foam thickness, cm	

The early thermal drift region of the aging curve where several gas components are changing simultaneously, oxygen and nitrogen partial pressures increasing, the partial pressures of the fugitive blowing agents decreasing, is extremely complex and difficult to analyze.

An opportunity **may** exist, late in the thermal drift region, as the foam becomes oxygen-saturated and nearly depleted of fugitive blowing agents. Provided the fugitive blowing agents have much higher permeabilities than nitrogen, only the partial pressure of nitrogen would be changing at a significant rate. Analysis in the thermal drift region thus **requires an identification of all the diffusing species**, an estimate of the average concentration in the foam and permeability of each species. Otherwise, the analysis would be confused and the effective diffusion coefficient estimates would be inaccurate!

However, in the blowing agent plateau region of the aging curve, with air-saturated foam, the analysis is simpler because only the partial pressure of the captive blowing agent is changing.

The interchange of atmospheric components and blowing agents in closed-cell foams by diffusion has been modelled as a semi-infinite continuum [3,4]. Numerical solutions for gas diffusion have been reported for foam structures in an ideal cubic lattice [5,6], with specified anisotropy [7,8] and a variable diffusional resistance (natural skins) [7,9]. A general solution of the diffusion equation with the same initial and boundary conditions, employing the error-function [10], provides an approximation for short diffusion times (that is, when less than 50 percent change in concentration has occurred), for the effective diffusion coefficient, Eq 5.

$$\frac{m_{BA,t}}{m_{BA,\infty}} - \frac{4}{\sqrt{\pi}} \left( \frac{D_{off}t}{L^2} \right)^{\frac{1}{2}}$$
(5)

where:

 $m_{BA,t}$  = the aged mass of component, g  $m_{BA,o}$  = initial or equilibrium mass of component, g  $D_{eff}$  = component effective diffusion coefficient, cm<sup>2</sup>/s

The mass of a component in a foam is the sum of its mass in the gas and polymer phases. The solubility of a component in the polymer phase, when proportional to the external partial pressure of the component, may be computed from Henry's law [10]. Then using the ideal gas law, the partial pressures can be computed by material balance within the foam. The instantaneous and equilibrium (or initial) values of component partial pressures may be substituted into Eq 5 in place of the component mass. The effective solubility of the component is the proportionality between the total material in the foam (in both gas and polymer phases) and the external partial pressure.

$$m_{BA,t} - (H_{BA} + \left[\frac{1}{\rho_f} - \frac{1}{\rho_p}\right]) \left(\frac{M_{BA}}{RT}\right) p_{BA,t}$$
(6a)

$$S_{eff} - (H_{BA} + [\frac{1}{\rho_f} - \frac{1}{\rho_p}])(\frac{T_{STP}}{T})$$
 (6b)

where:

H <sub>RA</sub>	=	Henry's law constant, $cm_{stp}^3/cm_s^3$ atm
Seff	=	effective solubility, cm <sup>3</sup> sup/cm <sup>3</sup> atm
R	=	ideal gas law constant, 82.06 cm <sup>3</sup> atm/gmol <sup>o</sup> K
т	=	gas absolute temperature of foam, °K
TSTP	=	standard temperature, 273.2 °K

After the substitution of partial pressure for mass terms in Eq 5, the derivative of partial pressure with respect to  $\sqrt{t/L}$  provides the means for determining the effective diffusion

$$\left(\frac{L}{P_{BA,\infty}}\right)\frac{dP_{BA,t}}{d\sqrt{t}} - \frac{4}{\sqrt{\pi}}\sqrt{D_{eff}}$$
(7)

coefficient, based on the change in component partial pressure with scaled age. However, direct measurement of the internal component partial pressures is not convenient. One may substitute, the slope of k-factor for slope of partial pressure into Eq 7 to obtain an expression which can be rearranged to provide an estimate of effective diffusion coefficient from aged thermal performance measurements of foam samples.

$$D_{eff} = \frac{\pi}{16} \left( \frac{L}{p_{BA,\infty} C} \frac{dK_{f,t}}{d\sqrt{t}} \right)^2$$
 (8)

### EXPERIMENTAL

### Samples and Method

Samples of STYROFOAM brand insulation foam manufactured with CFC-12 and methyl chloride as a blowing agents were collected during plant audits in 1980. The STYROFOAM samples produced with HCFC-142b and ethyl chloride as blowing agents were collected during experimental trials between 1981 and 1990. The initial concentrations of the blowing agent components in these samples were determined by mass-flow measurements at the production plants during manufacture.

The foam samples were stored in the Dow Product Evaluation Laboratory and were aged at  $24\pm5^{\circ}$ C ( $75\pm9^{\circ}$ F) in air. Individual full-thickness foam samples (with integral skins) were measured, on fresh foam (less than two days for most samples) and later remeasured at various intervals which based on time and thickness, ultimately exceeded a scaled age of 5000 s<sup>#</sup>/cm (about 7 years for a one inch sample). Foam kfactor measurements were made on a HOLOMETRIX, Inc., Model R-MATIC using the procedure described in ASTM C518 [11]. The R-Matic was operated at an average foam temperature of  $24\pm1^{\circ}$ C ( $75\pm1^{\circ}$ F) having an impressed  $27\pm0.8^{\circ}$ C ( $50\pm1^{\circ}$ F) temperature difference.

The samples were thus aged and tested at essentially the same thermal conditions and although slight temperature variations were experienced, the concentrations of the blowing agent(s), oxygen and nitrogen were either at equilibrium or changing in a monotonic manner.

The aged thermal performance data, of a polystyrene foam produced by UC Industries having HCFC-22 as a blowing agent, was reported by Graves, et.al. [12]. The thermal conductivity of the single sample was measured on three different types of apparatus, using two procedures ASTM C518 [11] and ASTM C1114 [13].

The slope of foam k-factor with scaled age was determined by linear least-squares regression. The standard deviation from regression was computed to estimate the precision of the effective diffusion coefficient. This was done in the thermal drift region for the foam samples prepared with CFC-12, HCFC-22 and HCFC-142b with scaled ages less than 800 s<sup>k</sup>/cm (about 50 days for a 1 inch sample). Slopes were also determined in the blowing agent plateau region for the same samples at scaled ages greater than a value of 1200 s<sup>k</sup>/cm (about 80 days for a 1 inch sample).

The initial partial pressures of the blowing agent in the foam were computed using the initial blowing agent concentrations obtained from production measurements, by material balance using literature values for solubility [14] and the measured foam density, at the aging temperature. The delta conductivity parameter was evaluated using pure gas thermal conductivity, from the literature for both the oxygen-saturated and air-saturated portions of the foam kfactor aging curve.

### Results and Discussion

The molecular and transport properties of CFC-12, HCFC-22 and HCFC-142b which are required for the calculation of delta conductivity and permeability in a polystyrene film, obtained from the literature [14], have been presented in Table 1 for the reader's convenience.<sup>1</sup>

TABLE 1 -- Molecular and Transport Properties of Selected Blowing Agents at 25°C

Blowing Agent	Molecular Weight g/mol	Solubility in Polystyrene cm <sup>3</sup> (stp)/cm <sup>3</sup> atm	Permeability in Polystyrene cm <sup>3</sup> (stp) cm/cm <sup>2</sup> s cm Hg	Pure Gas Conductivity W/m K
Nitrogen <sup>a</sup>	28.0	0,0095	2.82E-11	0.0226
CFC-12 HCFC-22 HCFC-142b	120.91 86.5 100.5	5.67 7.49 11.7	2.64E-13 1.67E-11 6.31E-13	0.00997 0.01060 0.01208

<sup>a</sup> Ref [15]



FIG. 2 -- Delta Conductivity Parameter Values

The delta conductivity parameter was computed for a foam density of  $32 \text{ kg/m}^3$  (actual sample foam densities varied from 28 to 34 kg/m<sup>3</sup>). This was done for several initial blowing

<sup>&</sup>lt;sup>1</sup> the units which have been reported for solubility coefficient, diffusion coefficient and permeability coefficient are the SI units which have been employed historically rather than those which have been proposed by the International Organization for Standardization.

agent partial pressures for the oxygen-saturated and airsaturated regions for each of the blowing agent components. The parameter values are presented in Fig. 2 as a function of initial blowing agent partial pressure. The values of delta conductivity parameter for the oxygen-saturated region are positive because nitrogen was considered to be the diffusing in the thermal drift region. The thermal qas late conductivity of nitrogen is greater than the mixed gas conductivity, and k-factor increases as nitrogen partial pressure increases. Later, in the blowing agent plateau region, the values of delta conductivity parameter are negative because the blowing agents have conductivities which are lower than the mixed gas conductivity and the blowing agent partial pressure is decreasing. Hence k-factor increases as blowing agent partial pressure decreases.

<u>CFC-12</u> aging: An aging curve for polystyrene foam containing CFC-12 was prepared from k-factor measurements on samples of thicknesses from 2.5 to 15 cm and ages up to seven years. A scale factor was used to describe aging in Fig. 3.



FIG. 3 -- STYROFOAM manufactured with CFC-12

A rapid increase in k-factor, characteristic of thermal drift was found for scaled age less than about 800 s<sup>#</sup>/cm. In the late thermal drift region, between 200 and 800 s<sup>#</sup>/cm, the slopes  $(4.2 \times 10^{-7}$  to  $1.1 \times 10^{-6}$  W/mK/s<sup>#</sup>) of samples having thicknesses from 10 to 15 cm were determined and recorded in Table 5 (see APPENDIX). A transition occurs between the thermal drift and plateau region at scaled time between about 800 and 1200 s<sup>#</sup>/cm. The blowing agent plateau region began at scaled ages greater than 1200 s<sup>#</sup>/cm, as seen in Fig. 3. The slopes  $(4.7 \times 10^{-8}$  to  $2.7 \times 10^{-7}$  W/mK/s<sup>#</sup>) of individual samples of STYROFOAM manufactured with CFC-12, for thickness from 2.54 to 7.64 cm were determined for the plateau region and recorded in Table 6 (see APPENDIX). The k-factor slope of the late thermal drift region was about 7 times higher than the k-factor slope in the plateau region. The relative standard deviation of the k-factor slopes in the plateau region were about 1.5-3.2 percent. Because the slope must be squared, the effective diffusion coefficient estimates have a precision not better than 5 percent.

Initial partial pressures (0.27 to 0.41 atm) were computed from the CFC-12 concentration, foam density and solubility. The equilibrium partial pressures of oxygen, nitrogen in the atmosphere and CFC-12 in the foam were used with their molecular weights and pure gas conductivities to compute the delta conductivity parameter.

The thermal aging data displayed in Fig. 3 represents an established process where adjustment of the captive blowing agent concentration and optimization of manufacturing process conditions have been used to produce a specified long-term thermal performance for a range of foam thicknesses.

<u>HCFC-22 aging</u>: The thermal performance data of an experimental polystyrene foam, manufactured with a HCFC-22 blowing agent, reported by Graves, et.al. [12] were plotted in an aging curve, Fig. 4.



FIG. 4 -- Experimental Polystyrene Foam Produced with HCFC-22

The thermal drift region of the experimental foam had a rapid increase in k-factor  $(3.17 \times 10^{-6} \text{ W/mK/s}^{\text{H}})$  up to a scaled age of about 800 s<sup>H</sup>/cm. The slope value for the thermal drift was recorded in Table 5. In this sample the plateau region, evaluated for scaled ages greater than 1200 s<sup>H</sup>/cm, maintained a relatively rapid increase in k-factor  $(1.69 \times 10^{-6} \text{ W/mK/s}^{\text{H}})$ with scaled age. The slope for the plateau region was recorded in Table 6. A slight break occurred in the aging curve at about 800 s<sup>H</sup>/cm. A coupled diffusion/k-factor model [9] was used to estimate the initial blowing agent concentration of the HCFC-22 foam sample by matching the predicted k-factors to the aging curve developed from data reported by Graves, et.al. [12]. The initial partial pressure was estimated to be about 0.3 atm for an assumed foam density of 32 kg/m<sup>3</sup>. The slope of k-factor in the late thermal drift region was only about 1.5 times the k-factor slope in the so-called blowing agent plateau region for this foam.

<u>HCFC-142b aging</u>: Aged k-factor data for samples of STYROFOAM brand insulation foam having HCFC-142b as a blowing agent, from experimental plant trials were used to prepare an



FIG. 5 -- STYROFOAM produced with HCFC-142b

aging curve, Fig. 5. The thermal drift region was demonstrated primarily by samples from 4.8 to 20.2 cm thick having scaled ages less than 800 s<sup>#</sup>/cm. The thermal drift region slopes, k-factor with scaled age  $(8.9 \times 10^{-7} \text{ to } 1.7 \times 10^{-6} \text{ W/mK/s}^{\text{h}})$ , were determined and recorded in Table 5. At scaled ages greater than 1200 s<sup>#</sup>/cm, the plateau region slopes of k-factor with scaled age  $(1.6 \times 10^{-7} \text{ to } 3.5 \times 10^{-7} \text{ W/mK/s}^{\text{h}})$  were determined in Table 6. The k-factor slope of the late thermal drift region was about 5 times higher than the k-factor slope in the plateau region.

Initial partial pressures (0.39 to 0.46 atm) were computed from HCFC-142b concentration based on process flow rates, foam density and solubility. Partial pressures of oxygen and nitrogen and the initial HCFC-142b partial pressure were used with their molecular weights and pure gas conductivities to compute the delta conductivity parameter.

Each set of data (in a limited thickness range) seems to follow a consistent aging trend. However, the collection of HCFC-142b containing samples shown in Fig. 5 is not as well behaved as the CFC-12 containing samples shown in Fig. 3. These HCFC-142b containing samples were prepared during short trials without the benefit of optimization or operational experience. Additionally, the initial k-factors of this group of samples were highly variable because the initial composition of HCFC-142b and foam cell size were the primary variables investigated during the trials. <u>Thermal drift region effective diffusion estimates</u>: The thermal drift region of a foam aging curve such as that presented in Fig. 1 **looks** deceptively simple and is a little misleading.

By way of an example, consider two extruded polystyrene foams, having the same bulk density  $32 \text{ kg/m}^3$  and cell size 0.30 mm, and the same 4.9 wt percent initial concentration of permanent blowing agent CFC-12. The first foam initially contains 4.9 wt percent **methyl** chloride and 1.3 wt percent CO<sub>2</sub> as fugitive blowing agents. The second foam initially contains 4.9 wt percent **ethyl** chloride and 1.3 wt percent CO<sub>2</sub> as fugitive blowing agents.

A coupled diffusion/thermal performance model [9], was used to estimate the vapor (gas) component partial pressures and thermal performance. These results were used to prepare component partial pressure aging curves, Fig. 6 and the thermal performance aging curves, Fig. 7 for these extruded polystyrene foams.



FIG. 6 -- Component Diffusion in Polystyrene Foam

Refering to Fig. 6, when one observes the component partial pressure curves with scaled age, the thermal drift region, which was previously described as having values of scaled age less than 800 s<sup> $\frac{1}{2}$ </sup>/cm is quite busy "diffusionly" speaking.

Observe that CO<sub>2</sub> and methyl chloride are quite rapidly depleted from polystyrene foam and oxygen is rapidly saturated, leaving a "window of opportunity" for evaluating the effective diffusion coefficient of nitrogen in the region between 600-1000 s<sup>%</sup>/cm scaled age. The effective diffusion coefficient of nitrogen, evaluated by rearranging Eq 7, using the nitrogen partial pressure curve was  $1.92 \times 10^{-7}$  cm<sup>2</sup>/s.

In a foam which contains ethyl chloride no such "window of opportunity" exists because the partial pressure curves of the fugitive blowing agents ( $CO_2$  and ethyl chloride) and oxygen overlap the nitrogen partial pressure curve and there is no clear region to evaluate the effective diffusion coefficient of nitrogen.





One might persist in using the thermal drift region of the thermal aging curves, such as those presented in Fig. 7, using Eq 8, and the delta conductivity constant for oxygensaturated foam containing about 0.3 atm CFC-12.

The foam containing the methyl chloride fugitive blowing agent provides an interference-free "window" region from 600-1000 s<sup>\*</sup>/cm in which the effective diffusion coefficient of nitrogen is estimated to be  $2.17 \times 10^{-7}$  cm<sub>2</sub>/s.

The foam containing the ethyl chloride fugitive blowing agent provides no such interference-free region. The scaled aging curves are curved in the region 100-1300 s<sup>\*</sup>/cm. The instantaneous slope in the 600-1000 s<sup>\*</sup>/cm region provides an estimate effective diffusion coefficient of  $3.16 \times 10^{-7}$  cm<sup>2</sup>/s, while that in the 600-1000 s<sup>\*</sup>/cm region is  $1.19 \times 10^{-10}$  cm<sup>2</sup>/s.

The effective diffusion coefficient estimates for both foams are different from the value determined by application of Eq 7 and the nitrogen partial pressure curve. Accurate estimates of effective diffusion coefficients determined from the slope of a thermal aging curve are dependent on the calculation of the delta conductivity parameter, which requires a knowledge of the composition and thermal conductivities of all the volatile components in the foam at that instant in the foam and may not be determinate when more than one component composition is changing.

Effective foam and film diffusion: Fig. 8 compares the effective diffusion coefficients of nitrogen and the blowing agents CFC-12, HCFC-22 and HCFC-142b evaluated by Eq 8 to the adjusted diffusion coefficient for polystyrene film. The adjustment to the film diffusion coefficient accounts for the effective solubility of the foam, adding the cell gas volume to the soluble blowing agent for the total blowing agent solubility. The values for the adjusted film diffusion coefficient are recorded in Table 4.

Component	Gas Fraction	Effective Solubility S <sub>eff</sub>	Film Diffusion Coefficient cm <sup>2</sup> /s
Nitrogen <sup>a</sup>	0.969 <sup>b</sup>	1.0	2.14x10 <sup>-9</sup>
CFC-12	0.969	1.195	1.68x10 <sup>-11</sup>
HCFC-22	0.969	1.258	$1.01 \times 10^{-9}$
HCFC-142b	0.969	1.403	$3.42 \times 10^{-11}$

TABLE 4 -- Adjusted Polystyrene Film Diffusion Coefficients

<sup>a</sup> Ref [15]

<sup>b</sup> based on foam density of 33.6 kg/m<sup>3</sup>





When a diffusion model is developed for a heterogeneous structure like foam, one important task is to provide physical significance to the structural characteristics of the foam. Shankland [7] presented an analysis which suggested that the ratio of the effective diffusion coefficient of a gas in foam to the adjusted diffusion coefficient in a polystyrene film was a geometric factor. He employed a (steady-state) transmission cell containing a foam sample as an **extended** membrane to determine the ratio. The gases tested by Shankland, He, CO<sub>2</sub> and O<sub>2</sub> have larger permeability values than those investigated here. The CO<sub>2</sub> data he reported have been plotted for comparison with the present results. The solid line drawn in Fig. 8 has the value of the geometric factor (115) determined by Shankland for a  $33.6 \text{ kg/m}^3$  (2.1 pcf) closed-cell polystyrene foam using high permeability gases. Shankland suggested that because the foam structure determined the value of the ratio between the measured effective (foam) diffusion coefficient and the adjusted (film) diffusion coefficient for a standard gas, this value could be used to **calibrate** a diffusion measurement for another gas.

By comparing the effective diffusion coefficients for nitrogen, CFC-12, HCFC-22 and HCFC-142b to the adjusted film diffusion coefficients in Fig. 8, the geometric factor has been determined for several gases. The value of effective diffusion coefficient determined in the previous section from a model prediction of the nitrogen partial pressure curve compares favorably with the geometric factor reported by Shankland [8]. The effective diffusion coefficients for nitrogen and HCFC-22, evaluated from the slope of the thermal aging curve, were found above the regression line, as though there was some interference from other components.

If the effective diffusion coefficients had been measured using the same foam sample and different gases, the conditions for Shankland's analysis would be satisfied and one could test his suggestion. The process of manufacturing foam with a given blowing agent but different thickness may be expected to cause a change in the foam morphology and hence change the geometric factor. Likewise, foam of a given thickness prepared from a different blowing agent may also have a different morphology and geometric factor.

Shankland remarked that the regression value for the ratio he observed was surprisingly similar to the value of the calculated geometric parameter value based on cell size and cell wall thickness.

The range of geometric factor, the **ratio** of effective (foam) diffusion coefficient to adjusted (film) diffusion coefficient has been computed. Values of effective diffusion coefficient from Table 5 and film diffusion coefficient from Table 4 for nitrogen yield a range of 125 to 1158. Geometric factors may be computed for the blowing agents investigated using values of effective diffusion coefficient from Table 6 and film diffusion coefficients from Table 4. Comparing the range of geometric factor for the blowing agents, CFC-12 (8 to 525), HCFC-22 (1495) and HCFC-142b (55-228) suggests that structural differences exist between these foams and that the specific process conditions for a polymer/blowing agent system may have contributed to the observed differences.

### <u>Conclusions</u>

The values for effective diffusion coefficients for gases and blowing agents estimated from the slope of k-factor with scaled age roughly agree with those which have been

determined by gas transmission through similar polystyrene foam samples.

Effective diffusion coefficients, determined from the plateau regions of aging curves for polystyrene foams having CFC-12 and HCFC142b blowing agents indicate that these foams have a geometric parameter of comparable value to the foams utilized by others.

The large value of the effective diffusion coefficient of HCFC-22 in polystyrene foam makes this material unsuitable as a CFC replacement captive blowing agent for extruded polystyrene foamed products.

HCFC-142b was found to be a suitable CFC replacement captive blowing agent. The slope of k-factor with scaled age of HCFC-142b was found to be the same as that of CFC-12 for the foam samples evaluated in this investigation.

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### APPENDIX

Blowing Agent	Initial Pressure atm	Avg Foam Thick. cm	k-factor slope W/m K/s <sup>1/2</sup>	Delta Conductivity W/m K/atm	D <sub>eff</sub> cm <sup>1/2</sup> /s
CFC-12	0.35	10.44	6.79E-7	0.0142	3.97E-7
	0.28	10.13	9.56E-7	0.0155	9.79E-7
	0.36	10.11	1.06E-6	0.0140	8.80E-7
	0.31	10.33	7.61E-7	0.0149	6.38E-7
	0.26	11.48	9.44E-7	0.0158	1.35E-6
	0.27	11.91	4.21E-7	0.0157	2.68E-7
	0.32	12.93	5.29E-7	0.0147	4.11E-7
	0.27	14.58	5.31E-7	0.0157	6.55E-7
HCFC-22	0.3 <sup>a</sup>	5.08	3.17E-6	0.0151	2.48E-6
HCFC-142b	0.33	7.75	1.16E-6	0.0129	8.75E-7
	0.21	7.75	1.44E-6	0.0146	2.58E-6
	0.42	10.03	1.16E-6	0.0116	1.12E-6
	0.42	9.88	1.16E-6	0.0116	1.09E-6
	0.42	10.21	1.15E-6	0.0116	1.94E-6
	0.44	12.16	1.04E-6	0.0113	1.26E-6
	0.44	20.19	7.80E-7	0.0113	3.11E-6
	0.44	4.88	1.73E-6	0.0113	5.62E-7
	0.44	5.10	1.45E-6	0.0113	4.31E-7
	0.44	6.02	1.63E-6	0.0113	7.59E-7
	0.44	5.99	1.47E-6	0.0113	6.11E-7
	0.56	6.15	1.28E-6	0.00992	3.94E-7
	0.56	5.94	1.34E-6	0.00992	4.03E-7
	0.51	9.91	8.83E-7	0.0104	5.28E-7
	0.44	13.87	8.96E-7	0.0113	1.22E-6
	0.44	14.55	8.72E-7	0.0113	1.27E-6

TABLE 5 -- Evaluation of Diffusion Coefficients in Thermal Drift Region

<sup>a</sup> estimated initial partial pressure

Blowing Agent	Initial Pressure atm	Avg Foam Thick. cm	k-factor slope W/mK/s <sup>1/2</sup>	Delta Conductivity W/mK/atm	D <sub>eff</sub> cm <sup>1/2</sup> /s
CFC-12	0.31	2.54	2.69E-7	-0.0114	7.11E-9
	0.35	2.56	1.63E-7	-0.0107	2.38E-10
	0.27	2.51	1.83E-7	-0.0121	3.79E-9
	0.42	3.76	1.26E-7	-0.0096	2.76E-9
	0.31	5.16	1.53E-7	-0.0115	9.74E-9
	0.34	5.23	1.46E-7	-0.0109	8.26E-9
	0.29	5.05	6.24E-8	-0.0117	1.64E-10
	0.26	5.08	4.70E-8	-0.0123	1.09E-10
	0.41	7.67	1.05E-7	-0.00976	7.91E-9
	0.35	7.64	1.02E-7	-0.0108	8.39E-9
	0.38	7.64	1.79E-7	-0.0102	2.42E-9
HCFC-22	0.3 <sup>a</sup>	5.08	1.69E-6	-0.0103	1.51E-6
HCFC-142b	0.39	1.45	2.77E-7	-0.00868	2.76E-9
	0.39	1.47	3.50E-7	-0.00868	4.53E-9
	0.44	1.47	3.25E-7	-0.00774	3.86E-9
	0.44	1.40	2.65E-7	-0.00774	2.33E-9
	0.46	1.40	2.38F-7	-0.00737	1.90E-9
	0.46	1.40	2.49F-7	-0.00737	2.08E-9
	0.44	1.49	3.14E-7	-0.00774	3.70E-9
	0.44	1.42	2.68E-7	-0.00774	2.45E-9
	0.42	2.54	2.68E-7	-0.00812	7.82E-9
	0.42	2.39	1.62E-7	-0.00812	2.53E-9

TABLE 6 -- Evaluation of Diffusion Coefficients in Blowing Agent Plateau Region

<sup>a</sup> estimated initial partial pressure

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THERMAL PERFORMANCE OF HCFC-22 BLOWN EXTRUDED POLYSTYRENE INSULATION

**REFERENCE:** Yarbrough, D. W., Graves, R. S., and Christian J. E., "Thermal Performance of HCFC-22 Blown Extruded Polystyrene Insulation," <u>Insulation Materials: Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A closed-cell foamboard insulation containing chlorodifluoromethane (HCFC-22) has been monitored in the laboratory and in the field for about three years. Experimental data for this developmental extruded polystyrene insulation has been obtained in two laboratories using three apparatuses. A correlation of apparent thermal conductivity with time gives the thermal resistivity of the two-inch-thick unfaced foamboards to within 3% with 95% confidence. A thermal resistivity of 30.2 m•K/W (4.36 ft<sup>2</sup>•h•°F/Btu•in) was calculated from the correlation for time equal 180 days after manufacture. Based on three years of data, the predicted thermal resistivity is 27.6 m•K/W (3.98 ft<sup>2</sup>•h•°F/Btu•in) as time since manufacture becomes large. Field measurements confirm the decrease with time of the thermal resistivity that was observed in the laboratory.

A three-dimensional model for heat transfer through foamboard was used to predict thermal resistivity as a function of time. A comparison of the experimental results with the model shows a radiative contribution to the total heat flux of about 30%. A one-dimensional model was used to show the effect of initial foamgas pressure and thickness on thermal performance.

**KEYWORDS:** foamboard insulation, thermal resistivity, CFC replacements, closed-cell insulation, apparent thermal conductivity, polystyrene

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```
NOMENCLATURE
     thickness of series resistance
а
       constants in Equation (3)
A,B
F
    percentile of the F distribution
     F_{0.95}(2,24)
k
    thermal conductivity, W/m·K (Btu·in/ft<sup>2</sup>·h·°F)
    k(T), function of temperature
    \boldsymbol{k}_a, apparent thermal conductivity
     k_a(t), function of time
     k<sub>a</sub>(t<sup>*</sup>), average value for t<sup>*</sup> years
     k<sub>a</sub>(180), at 180 days
    k<sub>ap</sub>, predicted from correlation
    k_g, gas phase
    k<sub>r</sub>, radiative component
    \mathbf{k}_{\mathrm{s}}, solid phase
    k_a^{\infty}, at t=0
k_a^{\infty}, as t \rightarrow \infty
l
     insulation thickness
    number of data points
n
    weekly average heat flux
Q
R^*, R^*(t) thermal resistivity (1/k_a), m \cdot K/W (ft<sup>2</sup>·h·°F/Btu·in)
s_y standard deviation in y, \Sigma (y_i - y)^2/(n-2)
s_{xx} \Sigma(x^2) - (\Sigma x)^2/n
t
     time, days (unless stated otherwise)
t' time related function defined by Equation (3)
t*
     lifetime
т
     temperature
∆T weekly average temperature difference
W
    half-width of prediction interval
```

 $\beta$  constant in Equation (1), days<sup>-1</sup> or yr<sup>-1</sup>

### INTRODUCTION

The Montreal agreement [1] to phase-out the use of chlorofluorocarbons has resulted in a need for replacements for CFC-11 and CFC-12 used in closed-cell foamboard insulations. The relatively high thermal resistivities (R\*) achieved by foamboard insulations are due to the low thermal conductivity of gaseous CFCs present in the cells within the foam. Candidate CFC replacements contain less chlorine since this element has been identified as a primary participant in the destruction of the earth's ozone layer [2]. A developmental extruded polystyrene product containing chlorodifluoromethane (HCFC-22) is the subject of the experimental part of this paper. HCFC-22 has an ozone destruction factor of 0.05 compared to CFC-12 of 1.0 [3].

At the time of manufacture foamboard insulation contain a gas that is rich in foaming agent. After manufacture  $O_2$  and  $N_2$  diffuse into cells and the foaming agent diffuses out, thus changing the cell-gas composition and thermal conductivity. The decreasing resistivity with time of foamboard insulations produced with a CFC as the foaming agent is referred to as "aging".

The aging of the nominally two-inch-thick closed-cell foamboard under discussion has been monitored for approximately three years by periodic measurements of the apparent thermal conductivity  $(k_a)$  of four specimens of the product. The foamboard has also been installed and monitored for two years as insulation for a monolithic slab-ongrade foundation. Three experimental apparatuses were used for the measurement of  $k_a$ . A single value for  $k_a$  82 days after production was measured by the manufacturer.

The k<sub>a</sub> data as a function of time and the Method of Least Squares were used to determine the constants k<sub>a</sub><sup> $\infty$ </sup>, k<sub>a</sub><sup> $\circ$ </sup>, and  $\beta$  in Equation (1). The constant  $\beta$  is related to the time required for 50% of the product aging to take place (t<sub>k</sub>) by the expression t<sub>k</sub> =  $ln2/\beta$ .

$$k_{a}(t) = k_{a}^{\infty} + (k_{a}^{\circ} - k_{a}^{\infty}) e^{-\beta t}$$
(1)

Equation (1) for  $k_a(t)$  of a product permits discussion of techniques for labeling the product. The time-average for  $k_a(t)$  over an interval [o,t\*],  $k_a(t^*)$ , is a measure of the thermal performance over time. The parameters in Equation (1) that describe  $k_a(t)$  can be used to calculate  $k_a(t^*)$ .

$$k_{a}(t^{*}) = \int_{0}^{t^{*}} k_{a}(t) dt / t^{*} = k_{a}^{\infty} - \frac{k_{a}^{\circ} - k_{a}^{\infty}}{\beta t^{*}} [e^{-\beta t^{*}} - 1]$$
(2)

Equation (2) clearly shows that  $k_a(t^*)$  approaches  $k_a^{\infty}$  for large values of  $\beta t^*$ . Because the second term of the right-hand-side of Equation (2) is positive,  $k_a(t^*) < k_a^{\infty}$  for all values of  $\beta t^*$ .

The current practice is to evaluate the thermal performance of foamboard insulation blown with CFCs 180 days after manufacture. Equations (1) and (2) can be used to evaluate the ratio  $k_a(t^*)/k_a(180)$ , which is a measure of the suitability of the 180 day criteria. The above ratio has been evaluated using constants obtained for the

unfaced foamboard that was studied in this research. The results presented in Table 1 show that for this product, the 180 day criteria overstates the product by about 7% at five years and perhaps as much as 10% for large  $t^*$ .

t\* (years)  $k_a(t^*)/k_a(180)$ 5 1.069 10 1.082 20 1.089  $\infty$  1.096

TABLE 1 --  $k_a(t^*)/k_a(180)$  for an unfaced foamboard insulation initially containing HCFC-22\*

\*Ratios were computed using  $k_a^{\circ} = 0.03622 \text{ W/m} \cdot \text{K} (0.2511 \text{ Btu} \cdot \text{in./ft}^2 \cdot \text{h} \cdot ^\circ \text{F}), k_a^{\circ} = 0.0238 \text{ W/m} \cdot \text{K} (0.1651 \text{ Btu} \cdot \text{in./ft}^2 \cdot \text{h} \cdot ^\circ \text{F}), \beta = 0.0076 \text{ (days} ^{-1}), \text{ and } k_a(180) = 0.03303 \text{ W/m} \cdot \text{K} (0.2292 \text{ Btu} \cdot \text{in./ft}^2 \cdot \text{h} \cdot ^\circ \text{F}).$ 

Table 2 contains percent differences between apparent thermal conductivities at 180 days after manufacture and average apparent thermal conductivities for t<sup>\*</sup> years. Equation (2) and selected values for the constants were used to calculate the percentages. The largest percentage shown, 21.9%, occurs for a large difference between k° and k°, a small rate constant ( $\beta$ ), and a large t<sup>\*</sup>. Relatively small percentages, 0 to 5%, occur when there is a small difference between k° and k°. A small  $\beta$  can exist for foamboard with impermeable facing and exposed edges.

Equation (2) can also be used to estimate the sensitivity of k(t<sup>\*</sup>) to  $k_a^{\circ\circ}$ ,  $k_a^{\circ}$ , and  $\beta$ . If  $k_a^{\circ\circ} = 0.2$ ,  $k_a^{\circ} = 0.10$ ,  $\beta = 2$ , and t<sup>\*</sup> = 20 then  $\partial k_a / \partial k_a^{\circ\circ} = 0.975$ ,  $\partial k_a / \partial k_a^{\circ\circ} = 0.025$ , and  $\partial k_a / \partial \beta = 0.0012$ . The relatively large  $\partial k_a / \partial k_a^{\circ\circ}$  indicates the importance of an accurate value for  $k_a^{\circ\circ}$ .

#### THERMAL MEASUREMENTS

Apparent thermal conductivity measurements have been made on specimens of HCFC-22 filled closed-cell foamboard using three apparatuses employing test methods ASTM C 518 and ASTM C 1114 [4, 5]. Data for  $k_a$  were obtained for a period extending to three years after manufacture using heat flow meter apparatuses at the Oak Ridge National Laboratory (ORNL) and Tennessee Technological University (TTU). The unguarded thin heater apparatus at ORNL was also used.

The C 518 equipment used to measure  $k_a$  was designed for 0.61x0.61 m (24x24 inch) specimens. The test specimens, however, were slightly smaller than this so a strip of fiberglass batt insulation was used to fill the small void space along the edges. The metering section in both C 518 apparatuses is 0.254x0.254 m (10x10 inches) and is located at the center of the test specimen so no correction for the fiberglass strip was deemed necessary. Thermal measurements were made at a mean specimen temperature of 23.88  $\pm$  0.5°C (75  $\pm$  1°F) and a temperature

difference across the specimen of 27.8°C ( $50^{\circ}F$ ). Heat flow meters were calibrated using SRM 1450b or SRM 1451 [6]. The uncertainty in the measured  $k_a$  has been estimated to be  $\pm$  3%.

TABLE 2 -- Calculated values for the percent difference between apparent thermal conductivity based on a 180 day test and a time-average apparent thermal conductivity

k <sub>a</sub> <sup>oo(a)</sup>	k <sub>a</sub> °	$\beta(yr^{-1})$	t*=5	10	20	50 (yrs)
0.0216	0.0144					
(0.15)	(0.10)	2.0	10.2%	12.0	12.7	13.0
		3.5	4.1	5.1	5.6	6.0
		5.0	1.4	2.1	2.5	2.7
0.0216	0.0173					
(0.15)	(0.12)	2.0	5.8	6.9	7.4	7.7
		3.5	2.4	3.0	3.3	3.5
		5.0	0.8	1.3	1.5	1.6
0.0252	0.0144					
(0.175)	(0.10)	2.0	13.6	16.2	17.5	18.2
		3.5	5.4	6.7	7.4	7.8
		5.0	1.9	2.8	3.2	3.5
0 0050	0 0172					
0.0252	0.0173	0.0	0.5	11 0	10.0	10.7
(0.1/5)	(0.12)	2.0	9.5	11.3	12.2	12.7
		3.5	3.9	4.8	5.3	5.6
		5.0	1.4	2.0	2.3	2.5
0 0288	0 0144					
(0.20)	(0.10)	2.0	16.4	19.5	21.0	21.9
		3.5	6.4	8.0	8.7	9.2
		5.0	2.2	3.2	3.8	4,1
0.0288	0.0173					
(0.20)	(0.12)	2.0	12.6	14.9	16.1	16.8
		3.5	5.0	6.2	6.9	7.2
		5.0	1.7	2.6	3.0	3.2
0.0288	0.0202					
(0.20)	(0.14)	2.0	9.0	10.1	11.6	12.1
		3,5	3.7	4.6	5.1	5.3
	_	5.0	1.3	1.9	2.2	2.4

(a) Units for  $k_a$  are W/m·K (Btu·in/ft·h·°F)

The unguarded thin-heater apparatus has been described previously in the literature [7, 8]. This apparatus was designed for 0.91x1.52 m (3x5 ft) specimens so it was necessary to border the 0.61x0.61 m (2x2 ft) test specimens with extruded polystyrene foamboard to produce test specimens of the required size. The temperatures used to calculate  $k_a$ were obtained in a 0.305x0.305 m (12x12 inch) region centered on the foamboard specimen. The thin-heater data were obtained as a function of temperature in the range 25.6 to 35°C (78 to 95°F) and a linear extrapolation was used to obtain  $k_a$  at 23.88°C.

The test specimens were maintained in a laboratory environment at about  $24^{\circ}C$  (75°F) - 50% RH except for specimen two at TTU. Specimen two was exposed to a  $24^{\circ}C$  (75°F) - 95% RH environment for 80 days to check for water absorption. The specimen absorbed less than 0.1 wt% water during the 80-day exposure [9]. Specimen two was stored in the laboratory after the high-humidity exposure.

Table 3 contains the  $k_a$  for the foamboard specimens at 23.88°C (75°F). The data are described to better than ±4% by Equation (1) with  $k_a^{\circ} = 0.25113$ ,  $k_a^{\circ} = 0.16515$ , and  $\beta = 0.0076$ . The average absolute deviation between the data and the equation is 1.8% and the standard deviation in  $k_a$  is  $0.57 \times 10^{-2}$ . Figure 1 shows the experimental  $k_a$  as a function of specimen age with a solid curve representing Equation (1). The solid curve is bounded by dashed curves that enclose a 95% confidence band for the curve as a whole [10]. The confidence interval shown in Figure 1 was constructed from a series of points calculated from a linear form of Equation (1).

$$k_{an}(t') = A + Bt' \tag{3}$$

 $t' = \exp(\beta t)$ 

$$k_{a}(t') = k_{an}(t') \pm W \tag{4}$$

$$W = \sqrt{2} F_{0.95}(2, 24) \cdot S_{y} \left(\frac{1}{n} + \frac{(t' - t')^{2}}{S_{xx}}\right)^{\frac{1}{2}}$$

The least-square fit to the  $k_a$  data gives the asymptotic value for the apparent thermal conductivity,  $k_a^{\circ\circ}$ , from which the asymptotic value for the resistivity can be calculated. The aged resistivity,  $k_a(t)^{-1}$ , for this foamboard product is 27.5 m•K/W (3.97 ft<sup>2</sup>•h•°F/Btu•m) the initial resistivity was 40.9 m•K/W (5.90 ft<sup>2</sup>•h•°F/Btu•in) and the resistivity at 180 days was 30.3 m•K/W (4.37 ft<sup>2</sup>•h•°F/Btu•in).





Age (days)	Method	k <sub>a</sub> (W/m∙K)	k <sub>a</sub> (Btu•in/ft <sup>2</sup> •h°F)	R* (ft²•h•°F/Btu•in)
82	Manufacturer	0.02914	0.2020	4.950
93	C 518-TTU <sup>(a)</sup>	0.03026	0.2098	4.766
98		0.03085	0.2139	4.675
131		0.03248	0.2252	4.440
204		0.03404	0.2360	4.237
459		0.03473	0.2408	4.153
556		0.03478	0.2411	4.148
732		0.03509	0.2433	4.110
739		0.03553	0.2463 <sup>(b)</sup>	4.060
1058		0.03515	0.2437	4.103
558	C 518-ORNL <sup>(c)</sup>	0.03691	0.2559	3.908
558		0.03717	0.2577	3.880
586		0.03687	0.2556	3.912
741		0.03755	0.2603	3.842
741		0.03734	0.2589	3.863
747		0.03701	0.2566 <sup>(d)</sup>	3.897
747		0.03571	0.2476 <sup>(e)</sup>	4.039
1067		0.03630	0.2517	3.973
1068		0.03629	0.2516	3.975
117	C 1114-ORNL	0.03022	0.2095	4.773
229		0.03345	0.2319	4.312
544		0.03574	0.2478	4.036
732		0.03589	0.2488	4.019
1088		0.03612	0.2504	3.994

TABLE 3 --  $k_a(t)$  and  $R^*(t)$  for an unfaced closed-cell thermal insulation product blown with HCFC-22

(a) Tennessee Technological University - Heat Flow Meter Apparatus

(b) TTU Specimen Two

(c) Oak Ridge National Laboratory - Heat Flow Meter Apparatus

(d) TTU Specimen One

(e) TTU Specimen Two

# A MODEL FOR CALCULATING R\*(t)

 $R^*(t)$  for the foamboard insulation containing HCFC-22 can be calculated using a version of the model proposed by Batty et al [11]. Batty's model includes heat transport by the gas phase and heat transport by solid elements either in series or in parallel with the gas. Radiative transport was taken to be an additive constant and treated as an adjustable parameter. Equation (5) was used to calculate  $k_a(t)$ . The gas phase thermal conductivity,  $k_g(t)$ , was calculated using the method of Lindsay and Bromley [12] for the thermal conductivity of gaseous mixtures. The parameter "a" was found to be 0.00470 using a porosity of 0.974 and an equation given by Batty. The thermal conductivity of polystyrene,  $k_s$ , was taken to be 0.1569 W/meK (1.088 Btuein/ft<sup>2</sup>•h•°F) [13], and the initial pressure of HCFC-22 was taken to be 60.8 kPa (0.6 atms). The conductivities in Eq. (5a) have units Btu•in•/ft<sup>2</sup>•h•°F while the conductivities in Eq. (5b) have units W/m·K.

$$(k_a(t) - k_r)^{-1} = 0.00470/k_s + [0.00865k_s + 1.01112k_a(t)]^{-1}$$
 (5a)

$$= 6.779 \times 10^{-4} / k_s + [0.05997 k_s + 7.0101 k_a(t)]^{-1}$$
 (5b)

Values for  $k_g(t)$  were calculated for cell-gas compositions obtained from an analytical solution of the diffusion equation in three dimensions. The three dimensional solution was represented as a product of the infinite series solutions for the one-dimensional boundary value problem [14] for each of the three spacial coordinates. The result is a function for  $k_g(x,y,z,t)$  that can be integrated to give an average value  $k_g(t)$  for the foamboard [10].

Figure 2 shows  $R^*(t)$  calculated for three values of  $k_r$ . The experimental data have been included in the figure for comparison. The curve for  $k_r = 0.011 \text{ W/m} \cdot \text{K} (0.0763 \text{ Btu} \cdot \text{in/ft}^2 \cdot \text{h} \cdot \text{°F})$  goes through the data. This  $k_r$  results in a predicted radiative flux for the aged foamboard equal to 30% of the total. The curves in Figure 2 do not compare favorably with the data for t<150d. This result suggests that the diffusion coefficients for N<sub>2</sub> and O<sub>2</sub> in polystyrene used in the calculation may be too large. The calculated  $R^*(t)$  for t>200d are insensitive to the N<sub>2</sub> and O<sub>2</sub> diffusion coefficients depending rather on the diffusion coefficient for HCFC-22 in polystyrene.

A one-dimensional model developed by Ostrogorsky [15] was modified and used to calculate apparent thermal conductivities for foamboard insulation produced with CFC-11. Table 4 contains  $k_a$  calculated for t=180 days and for t approaching  $\infty$  (20,000 days). These calculations have been performed for several thicknesses and three values for the initial partial pressure of the foam gas, P°. The entries for thickness of 5.28 cm (2.0 in.) and P°=60.8 kPa (0.6 atm) are close to the values found experimentally for foamboard produced with HCFC-22. These calculated results show the effect of the initial pressure of the foaming gas on both the 180-day value for apparent thermal conductivity and the very long time (20,000-day) value.



Thickness P°		k(180)ª	k(20000) <sup>b</sup>	k(20000)/k(180)
(cm)	(kPa)	(W/m•K)	(W/m•K)	
2.54	60.80 <sup>(c)</sup>	0.02617	0.03525	1.35
5.08		0.02351	0.03192	1.36
7.62		0.02171	0.02953	1.36
10.16		0.02047	0.02846	1.39
30.48		0.01774	0.02575	1.45
2.54	70.93	0.02487	0.03522	1.42
5.08		0.02289	0.03147	1.37
7,62		0.02120	0.02892	1.36
10.16		0.02009	0.02781	1.38
30.48		0.01764	0.02505	1.42
2.54	81.06	0.02487	0.03522	1.42
5.08		0.02239	0.03105	1.39
7.62		0.02081	0.02837	1.36
10.16		0.01978	0.02722	1.38
30.48		0.01757	0.02445	1.39

Table 4. Apparent thermal conductivity as a function of thickness and initial partial pressure of the HCFC blowing agent

(a) 180 days after manufacture

(b) 20,000 days after manufacture

<sup>(c)</sup> 60.8 kPa = 0.60 atms, 70.93 kPa = 0.70 atms, 81.06 kPa = 0.80 atms

# IN-SITU PERFORMANCE OF THE FOAMBOARD CONTAINING HCFC-22

The slab-on-grade of a building at ORNL has been insulated with the developmental foamboard insulation and monitored since February 1989. The in-situ thermal performance has been continuously monitored by heat flux transducers and thermocouples placed in a plane that is perpendicular to the face of the insulation. The sensors were read every minute, averaged and stored hourly. The in-situ apparent thermal conductivity can be calculated during the heating season by Eq. (6).

$$k_a = \overline{Q} \cdot \ell / \overline{\Delta T} \tag{6}$$

Figure 3 shows  $k_a$  from data for 37 weekly heating periods adjusted to a mean insulation temperature of 12.8°C (55°F) [16]. The specimen ages for these field measurements ranged from 430 to 1100 days.



(M\*m/W) eulev-A

FIG. 3 -- Average weekly in-situ  $k_a$  adjusted to 12.7°C (55°F) as a function of specimen age.

Specimen temperatures for heating seasons ranged from 8.3 to 16.7°C (47 to 62°F). The average in-situ  $k_a$  for the heating season from day 430 to day 500 was 0.034 W/m•K (0.24 Btu•in./ft<sup>2</sup>•h•°F) at an average temperature of 12.7°C (55.0°F). An insulation specimen was taken from the field installation on day 586 (July 1989) for a laboratory measurement of  $k_a$ . The result of this measurement,  $k_a = 0.037$  W/m•K (0.26 Btu•in./ft<sup>2</sup>•h•F), is shown in Figure 3. The adjustments to 12.7°C indicated in Figure 3 were based on laboratory measurements of  $k_a$  as a function of mean specimen temperature for the polystyrene insulation being studied.

The field measurements show an aging effect for the insulation. The magnitude of the effect is inconsistent with the last laboratory measurement. The first two laboratory tests appear to agree with the field data. The third laboratory measurement, however, suggests that the thermal performance has remained the same throughout the 1990 summer season. Measuring in-situ  $k_a$  is more difficult than steady-state laboratory measurements and there is, consequently, more scatter in the data. A specimen of the insulation was removed from the installation and tested in the laboratory to give  $k_a$  after 1085 days. The result, however, was 0.036 W/m•K (0.25 Btu•in/ft<sup>2</sup>•h•°F), a number that is very close to the  $k_a$  for the laboratory aged materials. The laboratory measurement is 9% below the average field-measured value for a four-week period, around 1085 days of age.

The field data suggest a slight aging effect (2.8% increase in kvalue) after the second heating season. This magnitude is small, but roughly consistent with the laboratory measurements and model predictions discussed earlier. When the samples have been pulled from the field, 0.3-0.5% moisture by volume has been found on or in the sample. Using the effect of moisture relationships on the thermal performances of XEPS blown with CFC-12, the wet k-value could be estimated to be about 1% more than a dry sample [17].

### DISCUSSION

The data presented in this paper represent an evaluation of a developmental foamboard insulation. Resistivity data obtained over a three year period were described by a relatively simple three parameter equation. The  $k_a(t)$  correlation shows that this unfaced nominally two-inch-thick product dropped from an initial  $R^*$  of 42.0 m•K/W (6.06 ft<sup>2</sup>•h•°F/Btu•in) to an aged  $R^*$  of 27.6 m•K/W (3.98 ft<sup>2</sup>•h•°F/Btu•in). This initial value is 10 to 14% lower than measurements made about one hour after manufacture.

An evaluation was made of the commonly used 180-day aging period. It was shown that the 180-day resistivity value exceeds the time average value by 7 to 10% depending on the length of time used to compute the time average. The equation for time average  $k_a$  was used to calculate the excess thermal resistance that results from using  $k_a$  at 180 days after manufacture to assign R-value. The calculations indicate excess values of over 20% for parameters in the range of the product that was tested.

A one-dimensional model was used to show the effect of foamboard thickness and initial partial pressure of the foaming agent on  $k_a$  [15]. The calculation shows that faced insulation; represented in the table by a thickness of 0.3048 m (12 inches), one-half of the total width; with high initial values have high potential R-value overstatements if the 180-day measurement is made with the facing in place.

The foamboard specimens aged in the field showed a decrease in R-value with time. The R-value decrease for t around 450 days to t around 750 days is greater than the R-value measured in the laboratory. One specimen of the foamboard insulation was removed from the field project and tested at t equals 1085 days. The result  $k_a=0.03655 \text{ W/m} \cdot \text{K} \ (0.2534 \text{ Btu} \cdot \text{in/ft}^2 \cdot \text{h} \cdot ^\circ \text{F})$  is very close to the value 0.03687 W/m \cdot \text{K} \ (0.2556 \text{ Btu} \cdot \text{in/ft}^2 \cdot \text{h} \cdot ^\circ \text{F}) measured at t equals 586 days. The difference between the in-situ  $k_a$  measurements and the laboratory measurements has not been resolved.

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CFC BLOWING AGENTS SUBSTITUTES - A STATUS REPORT

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**ABSTRACT:** Rapidly moving events at the national and global levels mean a virtual elimination of chlorofluorocarbons (CFC) as insulation foam blowing agents by the year 2000. There have been a number of more environmentally acceptable substitutes proposed for CFCs. As replacements for CFC-11, HCFC-123 and 141b have received most attention, while HCFC-22 and HFC-134a have been considered as CFC-12 substitutes.

This paper reports on the progress made to critically evaluate these substitute materials in rigid polyurethane and polyisocyanurate insulating foams. The processing, industrial hygiene, and final foam performance of the substitutes are emphasized. The path to commercialization of HCFC-141b is presented in some detail.

**KEY WORDS:** Thermal conductivity, polyurethane foam, polyisocyanurate foam, chlorofluorocarbon, hydrochlorofluorocarbon, insulating gases.

## INTRODUCTION

Rigid polyurethane and polyisocyanurate foams are commercially important insulating materials. The global prominence of these materials is due, in large measure, to the low thermal conductivity chlorofluorocarbon (CFC) expansion agents used to make these foams. The resulting closed cell matrix of CFC-containing foam provides outstanding insulation in building, appliance and transportation applications.

Unfortunately, CFCs have been implicated in serious global environmental problems. The role CFCs play in the decrease in the earth's protective ozone layer and in predicted global warming has been extensively studied. As a consequence of such scientific studies, CFC regulations have been promulgated.

Leon M. Zwolinski, Technical Supervisor; Gary M. Knopeck, Group Leader; Ian R. Shankland, Research Manager; Allied-Signal Inc., 20 Peabody St., Buffalo, NY 14210 On a world-wide basis, the first action to restrict and decrease CFC production occurred with the 1987 Montreal Protocol agreement, as coordinated by the United Nations Environmental Programme (UNEP). More recently, that agreement was revised to phase out CFCs by the year 2000. In addition, the revision of the Clean Air Act in the United States in late 1990 included Title VI as a section that called for a somewhat more rapid phase down of CFCs. The global and U.S. phase out schedules for CFCs are shown in Table 1.

TABLE 1 Global and U.S. CFC Reduction Schedule					
	Revised Montreal Protocol CFC Reduction June 1990	Clean Air Act CFC Reduction November 1990			
1991	0%	15%			
1992	0	20			
1993	20	25			
1994	20	35			
1995	50	50			
1996	50	60			
1997	85	85			
1998	85	85			
1999	85	85			
2000	100	100			

It should be noted that the Montreal Protocol provisions call for a future assessment to determine if a complete CFC phaseout by January 1, 1997 is achievable. This agreement also included a non-binding declaration of intent to phase out of hydrochlorofluorocarbons (HCFCs), proposed substitutes for CFCs, no later than 2040 and, if possible, no later than 2020. The U.S. Clean Air Act is more specific in regard to HCFCs, putting a production freeze in 2015 on these chemicals and a total production phaseout in 2030 [1].

#### CFC SUBSTITUTES CONSIDERATIONS

In the urethane foam industry, as in other CFC-user industries, finding suitable substitutes for CFCs is of critical importance. CFC producers have had alternates development programs underway for some time. For rigid polyurethane and polyisocyanurate foams, where trichlorofluoromethane (CFC-11) is the principal blowing agent and to a lesser extent, in froth foams, dichlorodifluoromethane (CFC-12), substitutes are needed that have the good properties of these chemicals. Obviously, more acceptable environmental properties are critical in such a selection process along with a number of other factors. Most attention has focused on the shorter-lived HCFCs and hydrofluorocarbons (HFCs). Table 2 cites the ozone depletion potentials (ODP) and greenhouse warming potentials (GWP) of possible alternates compared to regulated CFC blowing agents.

There are a host of other considerations for CFC alternates including toxicology, inherent physical properties, foam performance, commercial availability, and economics factors that enter into the substitutes selection process. The importance of these factors has been

TABLE 2 CFC	and alterna	ates environmental	properties [2]
	Lifetime (years)	Ozone Depletion Potential	Greenhouse Warming Potential
CFCs			
11 (CCl <sub>3</sub> F)	60	1	1
12 ( $CCl_2F_2$ )	120	1	3
113 $(CCl_2F-CClF_2)$	90	0.8	1.4
114 ( $CClF_2-CClF_2$ )	200	0.7	3.9
HCFCs, HFCs			
22 (CHClF <sub>2</sub> )	15	0.05	0.34
123 (CHCl <sub>2</sub> -CF <sub>3</sub> )	2	0.02	0.02
141b ( $CH_3-CCl_2F$ )	12	0.15	0.20
134a (CH <sub>2</sub> F-CF <sub>3</sub> )	16	0	0.26

previously detailed [3-5].

Although a large number of species have been considered as CFC substitutes, the list of practical candidates is short. As a substitute for CFC-11, 1,1 dichloro-1-fluoroethane (HCFC-141b) and 1,1-dichloro-2,2,2-trifluoroethane (HCFC-123) have received most attention (Table 3). As a substitute for CFC-12 in froth foaming, chlorodifluoromethane (HCFC-22) and 1,1,2,-tetrafluoroethane (HFC-134a) are being evaluated (Table 4).

	Jubacitutes	propercies	·
	11	141b	123
Molecular Weight	137.4	116.9	152.9
Boiling Point, •C	23.6	30.0	27.9
Vapor Thermal Conductivity at 25•C, in mW/mK	8.23	10.0	10.4
Flammability,* ASTM E-681 Lower Limit, Vol % Upper Limit, Vol %	None None	7.6 17.7	None None
Flash Point, ASTM D-1310-67 or D-56-82	None	None	None

TABLE 3 -- CFC- and substitutes properties

\* Flammability properties of these materials are not intended to reflect the fire hazards of any resultant cellular or foamed plastics products.

For rigid insulating foams, the thermal conductivity of substitute candidates is of paramount importance. While HCFC-141b and HCFC-123 exhibit slightly higher vapor thermal conductivities than CFC-11 (Figure 1), these values are still substantially better than other gases such as carbon dioxide or air [6].

## CFC ALTERNATES PERFORMANCE IN FOAM

The CFC substitutes candidates are undergoing intensive applications

scrutiny to determine their suitability as rigid polyurethane and polyisocyanurate foam blowing agents. This work is being conducted by CFC producers, polyurethane industry raw material suppliers, foam systems producers, foam insulation manufacturers, appliance producers and trade associations [7]. On several projects, there is government participation and funding.

	12	22	134a
Molecular Weight	120.9	96.5	102.0
Boiling Point, •C	-29.8	-40.8	-26.5
Vapor Thermal Conductivity at 25.C, mW/mK	9.98	10.6	14.0
Flammability, ASTM E-681 Lower Limit, Vol % Upper Limit, Vol %	None None	None None	None None
Flash Point, ASTM D-1310-67 or D-56-82	None	None	None

TABLE 4 -- CFC-12 and Substitutes Properties



for HCFC-123, HCFC-141b and CFC-11 [3].

The body of work done to date can be broadly summarized as follows:

- Neither HCFC-141b nor 123 is a drop-in replacement for CFC-11, but is has been shown that both materials can make acceptable foams.
- Foams made with HCFC-141b and HCFC-123 generally have slightly higher (about 10%) thermal conductivity values than CFC-11-blown foams.
- 3. The greater solvency of HCFC-141b and especially HCFC-123 pose foam/plastics compatibility problems in the home appliance industry.
- The CFC-11 replacement factor of HCFC-141b is more favorable than that of HCFC-123 on both a theoretical and actual basis.
- 5. HCFC-22 has been reported to be a suitable CFC-12 substitute in froth foams [8].
- Foams blown with HCFC-141b and HCFC-123 can be formulated to meet fire resistance tests of building codes and insurance standards.

As can be appreciated, commercial foam-making practices and finished product requirements vary greatly. At present, there is much work in progress to optimize foam formulations to utilize the HCFC alternates. For example, in one plant trial where HCFC-123 was substituted for CFC-11 in a polyisocyanurate boardstock foam, the HCFC-123-blown foam showed lower initial and aged thermal conductivity values than the corresponding CFC-11-blown foams [9]. It is believed that formulation modifications resulted in a more closed-cell, finer foam structure and were responsible for this outcome.

### COMMERCIALIZATION OF ALTERNATES

While HCFC-22 is commercially available from several U.S. producers, the other potential CFC substitutes are not. However, commitments to produce HCFC-123, HCFC-141b and HFC-134a have been announced in North America, Europe and Japan.

The introduction of a major new chemical is today a complex task. For CFC substitutes, in addition to initial properties determination and applications evaluations, a number of other stages must be addressed. A discussion of Allied-Signal's path toward commercialization of HCFC-141b illustrates the magnitude of such an effort.

- May 1988 Allied-Signal announces the start-up of a HCFC-141b pilot plant at the Buffalo, NY research laboratory. The multi-ton facility will make material for customer evaluations [10].
- September 1988 Eight CFC producers, from around the world, announce the formation of the second Program for Alternative Fluorocarbon Toxicity Testing (PAFT II) to study the toxicological properties of HCFC-141b. This longterm testing program, of which Allied-Signal is a member, with an eventual list of ten supporting companies, will pool resources to shorten the time required for testing.
- December 1988 Fourteen chemical companies, including Allied-Signal, reached an agreement to initiate the

Alternative Fluorocarbon Environmental Acceptability Study (AFEAS). The goal of this program is to evaluate the impact, if any, which new fluorocarbons may have on the environment.

- February 1989 The U.S. General Accounting Office (GAO) issued a report on the Environmental Protection Agency's safety assessment for substitutes for ozone depleting chemicals [11]. This document reviewed the assessment approach the EPA will use, under its authority detailed in the Toxic Substances Control Act (TSCA), to identify, assess, and control risks to human health and the environment for CFC substitutes, especially those, like HCFC-141b, that are not on the TSCA inventory list.
- September 1989 The PAFT consortium reported that preliminary toxicity test results of several alternatives to CFCs indicated that the three leading products, one of which is HCFC-141b, do not show significant signs of toxicity [12]. Exposure of test animals to these compounds caused no birth defects and no significant irritation to the skin or eyes. Studies to evaluate the chronic toxicity and carcinogenicity of all three substances are underway with results of twoyear inhalation work expected in 1992-1993.
- October 1989 EPA officials reported, at a Washington conference on CFC and halon alternatives, on the agency's strategy for reviewing the effects of the chemical substitutes on human health and the environment [13]. The agency stated the program has two basic principles: Substitutes must be tested now, and EPA must continually assess the test data as generated.
- October 1989 -The Society of the Plastics Industry, with Allied-Signal as a participant, commented to EPA on a contractor-prepared report on consumer exposure to CFCs and CFC substitutes [14]. The EPA-contracted report by Versar, Inc. (Contract No. 68-02-4254) cited expected consumer exposure levels to CFCs, and by analogy, to CFC substitutes. Blowing agent emissions from rigid polyisocyanurate insulating foam were included in the assessment study. The SPI task force that studied the report commented on instances where flawed calculations and erroneous assumptions may have been used. The SPI agreed to do additional blowing agent-from-foam emissions work to provide more information.
- December 1989 Allied-Signal announced it will build the world's first full-scale commercial plant to manufacture HCFC-141b. The new \$50 million plant will be located in Geismar, LA. with start-up scheduled for late 1991-early 1992.
- June 1990 The PAFT consortium reported at the London meeting of the Montreal Protocol delegates on the status of toxicological work on CFC substitutes [15]. HCFC-141b was reported to be of low acute toxicity by both the inhalation and oral routes. An extensive battery of in-vitro and in-vivo tests for gene mutation and chromosomal damage demonstrated no sign of mutagenic activity with and without metabolic activation. Chronic inhalation studies with the rat species showed no signs of toxicity, at levels used,

and no evidence of carcinogenicity after four months of exposure.

- August 1990 Allied-Signal announced that an agreement had been reached with the EPA which permits the company to commercially manufacture HCFC-141b [16]. Allied-Signal submitted a Premanufacture Notification to the EPA that included information on the manufacture and use of HCFC-141b, together with results of extensive health and safety studies. The agreement allows Allied-Signal to commercially manufacture HCFC-141b while additional toxicological tests are being conducted.
- October 1990 Allied-Signal issued a HCFC-141b storage and handling bulletin [17]. The document pointed out that although HCFC-141b has vapor flame limits, it is similar in this respect to 1,1,1-trichloroethane, a commonly used solvent. It was stated that because HCFC-141b has no flash point, because the rate of burning is slow in its flammable range, and because it's difficult to ignite, the company does not believe classified electrical installations are required. Factory Mutual supports Allied-Signal's position.
- November 1990 -In a letter to the EPA, Allied-Signal reported that studies have been initiated on the stability of HCFC-141b during various use applications, including in foam and solvent systems [18]. The company used the 8 (d) section under TSCA to report on formation of trace quantities of 1-chloro-1-fluoroethane (151a) and vinylidene chloride during rigid polyurethane and polyisocyanurate foaming. Previously, the company had reported to the EPA about breakdown products of HCFC-123 during foaming and in centrifugal chillers. Allied-Signal stated that additional studies are being undertaken and that the company is working with trade association groups to arrive at a formalized safety assessment in this matter.
- December 1990 The EPA was reported to judge chemicals under development as substitutes for CFCs as probably safe for workers, consumers, and the general population, according to preliminary analyses [19]. The EPA stressed, however, that toxicity studies are still incomplete.

## SUMMARY

CFCs are scheduled for a phaseout by 2000 and perhaps sooner if scientific findings of ozone depletion and potential global warming justify an earlier phaseout date. HCFCs and HFCs have been acknowledged to be desirable CFC substitutes for various applications, including rigid polyurethane and polyisocyanurate insulating foams, for a transition period into the next century. However, there are still important toxicological and environmental factors that must be addressed before the substitutes can be used commercially on a large scale. Programs to provide information in these areas are well underway and should be completed in the near future.

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Leon R. Glicksman, Melissa Burke, Arlene Marge, and Mark Mozgowiec

A REVIEW OF TECHNIQUES FOR IMPROVED FOAM CONDUCTIVITY: REDUCING RADIATION HEAT TRANSFER, LIMITING AGING AND INCLUSION OF VACUUM ELEMENTS

REFERENCE: Glicksman, L. R., Burke, M., Marge, A., and Mozgowiec, M., "A Review of Techniques for Improved Foam Conductivity: Reducing Radiation Heat Transfer, Limiting Aging and Inclusion of Vacuum Elements," Insulation Materials: Testing and Applications, 2nd Volume, ASTM STP 1116, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: To comply with international agreement closed cell polyurethane and polyisocyanerate foams must be made with blowing agents which minimize ozone depletion in the stratosphere. Candidate blowing agents have higher vapor conductivities resulting in lower R values for the foams. This paper reviews recent results of research at M.I.T. sponsored by the Department of Energy and the Environmental Protection Agency to find means to recover high R values. Use of small opaque particles or smaller foam cells reduces the radiative transfer in foams which is usually one third of the total heat transfer at standard conditions. The thermal aging of foams made with new blowing agents is a function of the permeability which has been measured for several new materials. The rate of aging can be reduced by proper use of diffusion barriers on the foam surface. The inclusion of vacuum elements in foam should increase the overall R value. A key problem is to find packaging materials for the vacuum elements which preserve the vacuum over the insulation life while

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KEYWORDS: foam, conductivity, heat transfer, foam aging, vacuum insulation

## INTRODUCTION

Closed-cell foam insulation blown with a refrigerant vapor such as CFC-11 has the lowest effective thermal conductivity, or the highest R per unit thickness, of any non-vacuum insulation currently available. With the advent of higher energy standards for buildings and appliances there is a need to improve the R-value even further.

Recently there has been concern that the CFCs used to blow the foams cause depletion of the ozone layer. Under international agreements the use of current day CFCs must be phased out. Recent measurements of ozone depletion in the northern hemisphere may accelerate the rate of CFC phaseouts. If replacement CFCs must be used, they by-and-large have a higher conductivity than the current CFCs. Thus, means must be developed to decrease other forms of heat transfer in foams just to retain present levels of R-values with the new CFCs.

There has been a research program on rigid foam insulation at the author's university for the past seven years. This work has been supported by the SPI, U.S. Department of Energy, Office of Buildings and Community Systems and Development through Oak Ridge National Laboratory, Dow Chemical, Mobay Corporation, U.S. Environmental Protection Agency, and with the active participation of the above along with Jim-Walters Research Corp., Owens-Corning Fiberglas and the Polyurethane Division of SPI.

The research at MIT has concentrated on developing fundamental models to understand and predict the effective thermal conductivity and aging of closed-cell foam insulation. Research recently underway focuses on identification and development of new techniques which will allow foams containing environmentally acceptable blowing agents to have effective conductivities equal to or less than present foams. The aging behavior of foams formed with the new blowing agents is also being measured. This paper will review results of the MIT program to improve foam conductivity. These include reduction of radiation heat transfer in foam, decreasing the rate of foam aging, and the inclusion of vacuum elements in the foam.

# HEAT TRANSFER MECHANISMS

This paper will first review the models for radiative heat transfer through foam. The radiative models will show the influence of cell size reduction on foam performance as replacement blowing agents are used for the foam.

Heat transfer through the foam is due to three mechanisms, conduction along the struts and cell walls of the solid polymer, conduction through the gas within the cell, and thermal radiation. The total heat transfer can be predicted as the sum of the heat transfer by the three mechanisms considered separately.

For a typical foam filled with CFC-11, Schuetz and Glicksman [1] have shown that one half of the total heat transfer takes place by conduction through the gas, one third to one quarter of the heat transfer is by infrared radiation and the balance is by conduction along the cell walls and struts of the solid polymer. The substantial contribution by radiation is because the cell walls are highly transparent to radiation, Schuetz.

Schuetz [1], Glicksman and Torpey [2], and Glicksman, Mozgowiec, and Torpey [3] have reported measurements of the foam transmissivity at several different thicknesses. The transmissivity can be represented as, [4],

$$\tau = e^{-K_s L} \tag{1}$$

where  $K_s$  is the extinction coefficient of the foam material and L is the thickness. The inverse of  $K_s$  is the mean free path of radiation from emission at one location of the foam to absorption at a second location. Thus the term KL represents a ratio of the foam width to the mean free path for radiation. Measurements made by Schuetz, Torpey and Mozgowiec have shown that for foams with cell diameters ranging from 0.5 mm to much smaller sizes, the mean free path is 0.1 mm or less.

In analogy to heat transfer in a gas, when the mean free path is much less than the physical dimensions of the system the heat transfer can be considered a diffusion process. For conduction the Fourier equation applies. For radiation a similar equation, the Rosseland equation, [5] applies,

$$q_{r} = -\frac{4}{3K_{s}} \frac{d(\sigma T^{4})}{dx} = -\frac{16\sigma T^{3}}{3K_{s}} \frac{dT}{dx}$$
(2)

The coefficient of the temperature gradient on the right hand side of equation 2 is the radiative contribution to the effective conductivity of the foam, Glicksman, Schuetz, and Sinofsky [6]. Recent results have shown that the extinction coefficient for conventional rigid foams can be predicted by a simple model of the foam. In the infrared wavelengths, transmission measurements of polyurethane and polyisocyanurate [1] have revealed that cell walls are about 90% transparent to infrared radiation. In the initial model, the cell walls are assumed to be transparent. Measurements of much thicker films, of the order of the strut thickness, have shown that these elements are essentially opaque. Thus the struts serve as the main barrier to thermal radiation through the foam.

The extinction coefficient is proportional to the total surface area of the strut per unit volume of the foam. Taking the cells as pentagonal dodecahedrons, and the struts as inscribed within equilateral triangles, the strut surface area can be shown to be proportional to the square root of the foam density and inversely proportional to the cell diameter [2]. The resulting expression for the extinction coefficient becomes

$$K_{s} = 4.10 \frac{\sqrt{f_{s} \frac{\rho_{foam}}{\rho_{poly}}}}{d}$$
(3)

where d is the cell diameter,  $\rho_{foam}$  is the foam density,  $\rho_{poly}$  is the polymer density, and  $f_s$  is the fraction of the polymer in the struts.

The contribution of the cell walls to the foam extinction coefficient has been recently predicted for two conditions: optically thin and small cell size [3]. The results of both cases agree within 7% for cells of size 1 mm or smaller, foam density of 32 kg/m<sup>3</sup>, and fraction of polymer in the struts of 0.6 or greater. The total foam extinction coefficient becomes the sum of the effects of the struts, given by equation 3 and the effects of the cell walls given by the optically thin limit of

$$K_{TW} = \frac{(1-f_s)\rho_f}{\rho_s}K_W$$
(4)

where  $K_{\mu}$  is the wavelength averaged extinction coefficient of the solid polymer. Earlier measurements [1] indicated the value of  $K_{\mu}$  was approximately 1600 cm<sup>-1</sup>.

Combining this with the equation for thermal radiation for the diffusion limit yields the following expression for the contribution to the effective foam conductivity due to radiation:

$$k_{r} = \frac{16}{3} \sigma T^{3} \left[ \frac{4 \cdot 1 \sqrt{f_{s} \frac{\rho_{foam}}{\rho_{poly}}}}{d} + \frac{(1 - f_{s})\rho_{f}}{\rho_{s}} K_{W} \right]^{-1}$$
(5)

As the cell diameter is decreased at a fixed foam density, the number of struts, which inhibit the radiation, is increased. Similarly, as the foam density is increased, the thickness and surface area of the strut and the cell wall are increased, thus decreasing radiation heat transfer.

Samples of isocyanurate and polyurethane foams were obtained from several different sources. The foams had densities near  $32 \text{ kg/m}^3$ . Table 1 gives the cell diameter density and measured extinction coeeficient of the foams. The extinction coefficient, given by equation 5, was compared to values obtained from transmission measurements of the foams. The predicted and measured values are quite close, as shown in figure 1. Predictions of the extinction coefficient are shown for the struts alone, equation 3, and the struts plus cell walls, equations 5 for a polymer extinction coefficient, K, of 1600 cm<sup>-1</sup>. Recently, Cunningham, Jeffs, Rosbotham, and Sparrow [7] measured the contribution of radiation heat transfer for foams with different cell size but approximately constant density. The radiation contribution was obtained by subtracting predicted solid conductivity contributions from the measured effective conductivity. Figure 2 shows a comparison of their experimental results versus the prediction given by equation 5. Results are shown for three values of the polymer extinction coefficient. The results with an extinction coefficient of 600 cm-1 give the closest agreement with the results. However in figure 1 a value of 1600 cm<sup>-1</sup> gives better agreement. Use of 600 cm<sup>-1</sup> in figure 1 would give a line about one third the distance between the struts only and the case of 1600 cm  $^{-1}$ . There is still a need to confirm these results for the smallest cell sizes, of the order of 0.1 mm, where the experimental determination of k. has the highest uncertainty.

d,mm	$\rho,g/cm^3$	$\sqrt{\rho/d}$ , $\frac{1}{2}/cm^{5/2}$	K <sub>b</sub> , cm <sup>-1</sup>
0.55	0.0269	2.98	17.0
0.47	0.0294	3.65	22.5
0.43	0.0261	3.76	25.6
0.40	0.0268	4.09	25.7
0.39	0.0264	4.17	25.2
0.38	0.0262	4.26	21.9
0.36	0.0293	4.75	30.5
0.31	0.0300	5.59	26.5
0.34	0.0363	5.60	27.9
0.29	0.0300	5.97	30.7
0.26	0.0531	7.21	27.3
0.21	0.0289	8.10	38.0
0.22	0.0497	10.13	51.8

TABLE 1 -- Extinction coefficient, density and celldiameter data for polyurethane foams.



FIG. 1 -- Extinction coefficients of foam theory for struts only and for struts plus cell walls versus measured values for various foams.



FIG. 2 -- Radiation contribution to foam conductivity theory versus data of Cunningham and Sparrow.

INFLUENCE OF REPLACEMENT BLOWING AGENTS ON THE FOAM THERMAL CONDUCTIVITY

theoretical models of the effective thermal The conductivity can be used to predict how much the effective conductivity will increase when replacement blowing agents are used. See for example Richard and Shankland [8], and Kennedy, Lin and Bhattacharjee [9]. The models will also be useful in identifying possible measures to restore the conductivity to its original level. Table 2 compares the conductivity of a fresh foam filled with CFC-ll to fresh foam filled with HCFC-123 and HCFC-141b, two possible alternate blowing agents with the same cell size, foam density, and percentage of polymer in the struts. The alternate blowing agents when available will cause about an 11% increase in the thermal conductivity if all of the other foam properties can be maintained constant. Shorter term solutions, which may involve partial or total substitution of CO, for CFC-11, can cause much larger changes.

TABLE 2 -- Effects of various blowing agents on the thermal conductivity of closed cell polyurethanes or polyisocyanurate foams.

 Cell gas			Foam	effective	cond	ductivity
-				W/m°C	(BTI	U/hrft°F)
 CFC-11				0.0	180	(0.0104)"
HCFC-123				0.0	200	$(0.0116)^{b}$
HCFC-141b				0.0	197	$(0.0114)^{b}$
Air (100%)				0.03	64	(0.021)
CO, (100%)				0.02	266	(0.0154)
Air/CFC-11	(50/50)	(mass	fractior	n) 0.03	17	(0.0183)
CO,/CFC-11	(33/66)			0.02	27	(0.0131)
 CO2/CFC-11	(50/50)		_	0.02	41	(0.0139)

"Base line, assumed

<sup>b</sup>Based on measured vapor thermal conductivity Richard, R.G. and Shankland, I.R., Int. J. Thermophysics <u>10</u>, 673, 1989.

In addition, the aging phenomenon, the increase of thermal conductivity due to air diffusion into the foam, occurs more rapidly since there is less CFC-11 available to mix with the incoming increasing air. When there is less CFC-11 present, a given rate of air diffusion leads to a higher mole fraction of air to CFC within the foam. Total substitution of CFC-11 with CO<sub>2</sub> causes the effective foam conductivity to rise by 44%. Note that over time, the original cell gas will be replaced by air and all of the foams will ultimately reach the effective conductivity of the 100% air filled foam if they have the same density, polymer composition, and cell geometry.

## FOAM ENHANCEMENT: CELL SIZE REDUCTION

There are several modifications to the foam which will counteract the increased conductivity caused by the substitution of  $CO_2$  for CFC-11. The first modifications are directed at reducing the radiation heat transfer. When the average cell size is reduced while maintaining a constant density, the total length and surface area of the struts are increased. The influence of cell-size is given by equation 3.

Since the foam density has an influence on both the radiation heat transfer and the solid conductivity, it is useful to look at the effect of foam density and cell size at the same time. Figure 3 shows the effective conductivity of a foam filled with pure CFC-11 for two different cell diameters and a range of foam densities. These results were obtained from equations 3 and 4. For a fixed cell diameter the conductivity initially decreases as the density is increased due to the reduction in the radiation heat transfer. As the density is increased further, the conductivity begins to rise due to the increased heat transfer by conduction through the solid. The same behavior holds for smaller cell sizes, although the level of the effective conductivity is lower due to the lower radiation and the minimum point occurs at smaller densities.



FIG. 3 -- Foam conductivity, filled with CFC-11.

The decrease in cell size may also change the fraction of material in the cell walls. If this fraction increases the rate of gas diffusion and the rate of foam aging will decrease, Glicksman, Ostrogorsky, and Reitz [10]. The conduction through the solid will increase as the fraction of polymer in the cell wall increases. For these results it is assumed that the fraction in the cell will remain constant.

Figure 4 shows similar results for a foam filled with pure CO<sub>2</sub>. The magnitude of the effective conductivity is higher than that for CFC-11-filled foam but the advantages of using small cell sizes and modest densities remain. If the cell size of the 100% CO2-filled foam can be reduced to 0.1 mm (and the simple radiation model is still valid--this has yet to be experimentally substantiated) the R-value of the foam is predicted to be more than 7 (BTU-in/ft<sup>2</sup>°F). This corresponds to an effective conductivity of 0.020 W/m°C, about the same as a HCFC-123-filled foam.

Figure 5 shows the predicted foam conductivity versus weight fractions of  $CO_2$  for two different cell sizes. The base line case is assumed to be the same as that in table 2 for a foam containing 100% CFC-11 with 0.5 mm cells. Note that by halving the cell diameter, a foam filled with 20% by weight  $CO_2$  has the same predicted conductivity as a foam filled only with CFC-11.



CO<sub>2</sub> Filled Foam

FIG. 4 -- Foam conductivity, filled with CO2.



FIG. 5 -- Foam conductivity with mixtures of CO<sub>2</sub> and CFC-11.

DIFFUSION BARRIERS

There are several important practical barriers to blowing foams with CO, while reducing radiation by using small cells, making fine-cell foam is difficult, especially for applications such as refrigerators. Additionally, an effective barrier must be designed to prevent CO, from diffusing out of the foam and to prevent air from diffusing into the foam. In a typical unprotected foam, CO<sub>2</sub> will diffuse out in a matter of days. Many present foam products made as boards have inadequate adhesion between the facing material and the foam so that even if the facing is impermeable, air and  $CO_2$  can easily move in a lateral direction under the interface from the edge to the center of the board (Ostrogorsky and Glicksman [11]).

Conversely, if a impermeable facing is properly attached to both faces of a foam board the time averaged performance will be substantially improved. This improvement may be more than enough to offset the handicap of using a blowing agent with a higher conductivity than CFC-11.

Figure 6 shows the predicted rate of conductivity increase for a board 122 cm (4 ft) wide initially filled with CO<sub>2</sub>. The time for the conductivity change has increased substantially over an unfaced foam because the time varies with the square of the characteristic thickness. In this simulation it is assumed that facing is impermeable and there is no lateral diffusion under the interface. Certainly in appliances, where it is possible to provide tight interfaces, a pure CO<sub>2</sub> filled foam with small cell sizes provides good performance if such a foam can be made to fully fill the wall cavity. Note in figure 6 that the CO<sub>2</sub> blown foam with no lateral diffusion has much better performance over time that a CFC-11 blown foam with a permeable facing.



FIG. 6 -- Predicted aging behavior, CFC-11 filled foam with no diffusion barrier versus CO<sub>2</sub> filled foam with ideal barrier on top and bottom.

# OPAQUE FLAKES

A second technique to reduce radiation heat transfer is to include opaque flakes in the polymer cell walls, making them less transparent. Figure 7 shows the increase in the radiative extinction coefficient as pure graphite supplied by Consolidated Astronautics and aluminum flakes are added The extinction coefficient is calculated from to the foam. transmission measurements made over the infrared wavelengths where appreciable thermal radiation is emitted at room The radiation heat transfer, which varies temperature. inversely with the extinction coefficient, can be reduced by 15% to 20% by the use of the flakes (Glicksman and Torpey [12]). However, part of this improvement is offset by the increased solid conductivity of the polymer containing the metal flakes. The effective conductivity of the foam decreases as flakes are added; a large amount of flakes causes the effective conductivity to increase, figure 8.



FIG. 7 -- Extinction coefficient for foams filled with aluminum and graphite particles.



FIG. 8 -- Measured conductivities foams filled with aluminum and graphite particles.

Present research is underway with the goal to find flakes which will minimize the increase in the solid conductivity. Flakes of low thermal conductivity materials, such as plastics, are coated with a very thin layer of graphite to make them opaque. The coated flakes will be a radiation barrier with a low effective thermal conductivity.

# VACUUM PANELS

As a third technique the effective conductivity of foamfilled walls can be increased by using foams in conjunction with vacuum insulations. Flat evacuated panels containing fine particles encapsulated within flexible envelopes have demonstrated insulation levels three times higher than fresh CFC-11 filled foams of the same thickness. Kollie et al [13] give a review of flexible vacuum filled panels. There are several issues to be addressed before these panels are commercially feasible. First, the flexible envelope must be impermeable enough to maintain the desired vacuum over the lifetime of the insulation. Evacuated panels commercially introduced into a line of Japanese refrigerators several years ago have been discontinued, presumably because of this problem. The envelopes cannot be made of heavy metal foil, which is a good gas barrier, because heat transfer around the circumference of the package will cause substantial thermal "short-circuiting" of the vacuum and severely increase the effective conductivity. If the panel is directly attached to a steel wall of an appliance, the steel wall acts as a short-circuiting element; this effect can be reduced by the use of a layer of foam insulation between the vacuum panel and the steel wall.

The use of powder-filled panels with envelopes made of thin ceramics or glass is being investigated. The envelope will be impermeable to gases over the panel's lifetime. Glass and ceramics have a low thermal conductivity minimizing heat transfer around the circumference. It is planned to make these panels small, similar in size to ceramic tiles which could be assembled into a wall panel. Damage of a few panels would not significantly effect the performance of the wall.

Proposed vacuum insulation panels for wall insulation of a building or an appliance would have a configuration as shown on figure 9.

The interior of the insulation can be filled with a fine powder which is evacuated. The powder is encapsulated in a flexible barrier which prevents air diffusion into the vacuum. Although the barrier does not have to be rigid, it does have to prevent diffusion over the lifetime of the insulation. The most common barrier material which can perform this function is a metallic layer such as aluminum.


FIG. 9 -- Evacuated insulation and envelope, thermal boundary conditions.

First, simple two-dimensional analysis can be made by assuming that the panel length is larger that its width or thickness so that conduction in the third dimension can be neglected. When this is not correct the secondary conduction will be larger than the values presented in this paper. As shown in figure 10, the envelope wall acts like a fin to conduct heat around the periphery of the vacuum panel.



FIG. 10 -- Heat flow paths for evacuated insulation.

Two cases will be considered for the boundary conditions: the panel surface is directly exposed to the exterior or interior air with a heat transfer coefficient to account for this exterior convection, figure 10, or the panel is attached to a thicker metal surface, e.g. the metal wall of a refrigerator, figure 11. The heat transfer through the interior of the panel will be assumed to be one-dimensional, either radiation between parallel low emissivity walls for the empty panel or conduction through the powder with an effective conductivity based on the experimental results obtained at Oak Ridge [13].



FIG. 11 -- Evacuated insulation attached to a conducting wall.

The envelope surrounding the panel, both top and bottom, is divided into equal sized increments as shown in figure 10. An energy balance for an element along the bottom envelope becomes,

$$q(y+\Delta y) - q(y) = h_{effective} \Delta y (T(x) - T(y))_{x=y} - h_2 \Delta y (T-T_2)$$
(6)

where

$$q(y+\Delta y) - q(y) = \frac{kt}{\Delta y} \left[ T(y+\Delta y) + T(y-\Delta y) - 2T(\Delta y) \right]$$
(7)

and

$$h_{effective} = h_r$$
 for rigid wall vacuum insulation (8a)

$$=\frac{k_{effective}}{(w-2t)}$$
 for powder insulation. (8b)

A similar form of expression is used for the elements along the top of the panel to find T(x). When the envelope along the top of the panel is in contact with a metal wall then the product of the conductivity and the thickness is the sum of that for the envelope and the wall. The solution for the temperature distribution along the top of the envelope is found simultaneously with the solution for the bottom. The top and bottom were divided into as many as 20 increments for the solution, it was found that 10 increments gave satisfactory results, within 5 percent of the results for the larger number.

Figure 12 shows results using flexible aluminum film envelopes around evacuated powder. Fine powder with an effective conductivity of  $0.004 \text{ W/m}^\circ\text{C}$  (.0023 BTU/hrft°F) when evacuated was used in the calculations. Note that the powder covered with a .025 mm, .001 inch, aluminum foil has an overall insulating value equivalent to evacuated rigid panels made from stainless steel. As the foil thickness is reduced the influence of the two-dimensional heat transfer is diminished.

Figure 13 shows similar results when an inorganic ceramic or glass envelope is used to encapsulate the powder. The envelope has a conductivity of 0.865 W/m°C (0.5 BTU/hrft°F). For a thickness between 0.13 mm, (.005 in) and 0.26 mm the ceramic panel has insulating performance superior to the stainless or thick aluminum. Also with glass or ceramic films much smaller panel widths can be used without causing substantial degradation of the thermal performance. Thus, multiple vacuum spaces can be employed so that damage to one vacuum space will only cause a modest degradation of the overall panel performance. The envelope surrounding the panel, both top and bottom, is divided into equal sized increments as shown in figure 10. An energy balance for an element along the bottom envelope becomes,

$$q(y+\Delta y) - q(y) = h_{effective} \Delta y (T(x) - T(y))_{x=y} - h_2 \Delta y (T-T_2)$$
(6)

where

$$q(y+\Delta y) - q(y) = \frac{kt}{\Delta y} \left[ T(y+\Delta y) + T(y-\Delta y) - 2T(\Delta y) \right]$$
(7)

and

$$h_{effective} = h_r$$
 for rigid wall vacuum insulation (8a)

$$=\frac{k_{\text{effective}}}{(w-2t)} \text{ for powder insulation.}$$
 (8b)

A similar form of expression is used for the elements along the top of the panel to find T(x). When the envelope along the top of the panel is in contact with a metal wall then the product of the conductivity and the thickness is the sum of that for the envelope and the wall. The solution for the temperature distribution along the top of the envelope is found simultaneously with the solution for the bottom. The top and bottom were divided into as many as 20 increments for the solution, it was found that 10 increments gave satisfactory results, within 5 percent of the results for the larger number.

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FIG. 12 -- Predicted R value of evacuated insulation with aluminum foil envelope.



FIG. 13 -- Predicted R value of evacuated insulation with glass envelope.

If the vacuum spaces or tiles which are used to form the insulation on the wall have approximately equal height and width dimensions, both of which are much larger than the panel thickness, then the two dimensional representation used for the previous results must be modified. To simplify the analysis the individual panels will be assumed to be sections of right circular cylinders which have axisymetric conditions. Thus the top and bottom surfaces of the panel will be circular areas with the temperature across the areas varying with the radius. The powder will be assumed to be a coarser less expensive one with a thermal conductivity of 0.00476 W/m°C (R 30 per inch) when evacuated. The same form of finite difference solution will be employed, see equations 6 and 7, except x and y are given in radial coordinates and the variation of the cross-sectional area of the glass material with radius must be included. The upper and lower surfaces of the panel are assumed to be covered by an 3.2 mm, 1/8 inch, thick foam insulation which as a conductivity of 0.018 w/m°C, 0.125 BTU in/mft<sup>2</sup>°F.

Figure 14 shows the results for the right circular panels when the thickness is held constant at 1.27 cm, one half inch, and the thickness of the glass or ceramic envelope is allowed to vary from .005 mm, 0.002 inches to 0.38 mm, 0.015 inches, the later is about 60 percent of the thickness of the glass in a commercial light bulb. Figure 15 shows the thermal resistance when the glass thickness is held constant at 0.015 inches, and the thickness of the panel is varied between 1.27 cm, one half inch, and 2.54 cm. In all of these



FIG. 14 -- R value of evacuated insulation with glass envelope, cylindrical geometry, thickness 1.27 cm.



FIG. 15 -- R value of evacuated insulation with 0.38mm glass envelope, cylindrical geometry.

cases with modest panel diameters the overall panel gives thermal resistances which are far superior to that of foam insulation of equal thickness. For example, 1.9 cm thick panel of radius 10 cm and wall thickness of 0.38 mm give the same thermal resistance as 5 cm of CFC-11 filled foam.

One of the main advantages of a vacuum element with a glass or ceramic envelope is that if the panel is not physically damaged the glass is impermeable enough so that the pressure within the panel will effectively remain unchanged over the life of the panel. The use of a thin layer of foam over the surface of the panel will serve to protect the glass surface from damage.

#### CONCLUSIONS

The overall heat transfer through a closed cell foam is due to three contributions: conduction through the cell gas, conduction along the polymer forming the cell walls and struts, and radiation. As substitute blowing agents such as HCFC's and  $CO_2$  are used which are environmentally acceptable, the overall conductivity of the foam will rise because the gas conductivity will be higher than for CFC-11.

Radiation heat transfer through the foam is reduced by interaction with struts and cell walls. Smaller cells give larger struts surface areas per unit volume and thus reduce radiation. The effect of the cell walls stays about constant as the cell size changes if the foam density and percent of material in the cell wall is unchanged.

Radiation heat transfer can also be reduced by the addition of opaque flakes or powders to the foam.

The time averaged conductivity of a foam panel can be substantially improved if an impermeable barrier is added to both faces of the panel. The foam must be tightly bonded to the barrier to prevent lateral diffusion from the unprotected edges. The improvement due to the barrier may offset the handicap of using a blowing agent with a higher conductivity than CFC-11.

Vacuum insulation panels filled with powders hold the potential for substantial insulation performance per unit thickness. Use of thin glass or ceramic as the envelope for the powder will guarantee that the vacuum will not be compromised by diffusion of air components into the panel over the lifetime of the panel.

#### ACKNOWLEDGEMENTS

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NOMENCLATURE

- d Cell diameter cm
- Fraction of polymer in struts f,
- k Conductivity W/m°C
- Conduction though solid W/m°C k<sub>cs</sub>
- k<sub>eff</sub> Effective conductivity of foam W/m°C
- Gas conductivity W/m°C
- Polymer conductivity W/m°C
- kg kp kr Radiant contribution for effective conductivity w/m°C
- K Extinction coefficient cm<sup>-1</sup>
- Ks Extinction coefficient due to struts cm<sup>-1</sup>
- Extinction coefficient due to cell K<sub>TM</sub>
- walls, optically thin limit cm<sup>-</sup>
- ĸ Extinction coefficient of the solid polymer cm<sup>-1</sup>

- $P_u$  Percent of solid in cell wall,  $100 \cdot (1-f_s)$
- T Absolute temperature °K
- $\delta$  Fraction of foam volume occupied by gas
- $\rho_f$  Foam density kg/m<sup>3</sup>
- $\rho'_{s}$  Solid density kg/m<sup>3</sup>
- $\sigma$  Stefan-Boltzmann Constant W/m<sup>2</sup>(°K)<sup>4</sup>

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# **Convection in Fibrous Insulation**

#### William B. Rose and David J. McCaa

# THE EFFECT OF NATURAL CONVECTIVE AIR FLOWS IN RESIDENTIAL ATTICS ON CEILING INSULATING MATERIALS

REFERENCE: Rose, W. B., and McCaa, D. J., "The Effect of Natural Convective Air Flows in Residential Attics on Ceiling Insulating Materials", *Insulation Materials: Testing and Applications, 2nd Volume, ASTM STP 1116*, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Four test cells of a field laboratory have been used to determine the effect of natural convection on the thermal performance of insulating materials in various typical flat-ceiling residential attic constructions. Both blanket and loose fill mineral insulating materials are used in both vented and unvented assemblies. Temperature, air movement, heat flux and air humidity are continuously measured in each cell, and outdoor weather conditions are monitored. For the purpose of this study, data from the month of January 1991 are used; during that period the attic temperature remained colder than the "indoor" temperature. Field measurements of temperature at the top and bottom of the insulation, and heat flux measurements, are used to determine temperature difference across the insulation and instantaneous R-value. The results show that attic ventilation has negligible impact on the thermal performance of blanket insulation. The results also show an effect of elevated wintertime temperature differences upon loose-fill insulating materials.

KEYWORDS: attic insulation, thermal conductivity, natural convection, loose-fill insulation, blanket insulation

#### PURPOSE

The purpose of this research is to investigate the performance of mineral fiber insulating materials in typical residential attic assemblies under natural conditions. Convection, or air movement, is expected in most residential attics. Literature on attic convection includes early significant work from environmental chamber studies[1], from smoke decay studies in existing structures[2], and from studies involving tracer gas measurements[3]. There is a growing body of literature which investigates convection within insulating materials in theoretical and experimental studies [4,5]. However, the research need which this work attempts to address is the description and quantification of the effect upon insulating materials of air flows under natural conditions in typical residential attic constructions.

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Figure 1. Photo of Building Research Laboratory, taken from the north. Laboratory contains 10 compartments, of which 8 are study bays. Bay 0 is at right. Photo taken in early morning during cold weather.

In this study, surface temperature and heat flux in attic insulation are measured in test cells designed for attic performance research. These values permit calculation of temperature difference across the insulation ( $\Delta T$ ) and thermal resistance (R-value). These resistance values are compared among test cases with respect to variations in construction detailing.

### TEST BAYS

In 1989, a research facility was constructed which contains eight adjacent test bays for the study of residential attic performance. The bays measure  $8ft. \times 20ft. (2.44m \times 6.1m)$  and are constructed using methods and materials common in residential construction. A photo of the facility is shown in Figure 1. Four of the test bays (bay 2- bay 5) are pertinent to this study. These four bays are framed with wood trusses and contain insulation material laid directly on the gypsum sheetrock ceiling. Other studies, including thermal performance of cathedral ceilings, and wind flow patterns in flat ceiling bays, are being conducted concurrent with this study but the results are not included here for reasons of brevity.

The characteristics of each individual bay are shown in Table 1. A plan of the test bays is shown in Figure 2, and a section showing eave and ridge details is given in Figure 3. Briefly, bays 2 and 3 have no venting accessories while bays 4 and 5 have soffit and ridge ventilation. Bays 2 and 4 have loose-fill fiberglass insulation and bays 3 and 5 have fiberglass batt insulation.

Bay #	2	3	4	5	
ventilation	no	no	yes	yes	
insulation	loose-fill	batt	loose-fill	batt	

#### TABLE 1 -- CONFIGURATION OF STUDY BAYS

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Figure 2. Plan of bays 2 through 5, Building Research laboratory.



Figure 3. Half-section of Building Research Laboratory, showing location of heat flux transducer and thermocouples.

#### INSULATION PROPERTIES

All the batts (nominal R-30) were produced in the insulation plants during May and June of 1990. At the same time that bags of insulation were sent to the test site, two bags were sent to Blue Bell, PA for testing in the CertainTeed thermal test laboratory. Specimens of each type of insulation were tested in a ASTM C-518 apparatus. Further, the actual batt insulation pieces located above the transducers in the attics, were individually measured in the laboratory. Based on the weight of the insulation installed and the geometry of the installation, the values given in Table 2 for the installed batt insulation were derived.

**TABLE 2 -- INSULATION PROPERTIES AT TIME OF INSTALLATION** 

Bay#	2	3	4	5
Thickness: in. (cm)	12.01 (30.5)	9.74 (24.7)	13.97 (35.5)	9.74 (24.7)
Density: lb/cf (kg/m <sup>3</sup> )	0.523 (8.38)	0.524 (8.39)	0.491 (7.87)	0.524 (8.39)
R-value: ft <sup>2</sup> -hr- <sup>•</sup> F/Btu (m <sup>2</sup> - <sup>•</sup> K/W)	30.11 (5.30)	29.88 (5.26)	34.34 (6.05)	29.88 (5.26)

For the loose-fill insulation a direct comparison is more difficult. At the time of manufacture bags were sent to both the test site and the testing facility at Blue Bell, PA. At Blue Bell four test specimens were prepared using a commercial pneumatic blowing machine and tested following ASTM C-687. The resulting curve of apparent thermal conductivity versus density is given in Figure 4.



Figure 4. Thermal conductivity v. density plot for loose-fill insulating material.

At the test site the attic bays were blown using the same type of pneumatic blowing machine. All bags of insulation were weighed before being blown and, from the weight of insulation installed and the measured installed thickness, an average installed density was calculated. From this density and Figure 4 the installed apparent thermal conductivity was found and finally an R-value calculated. These R-values are given in Table 2.

#### INSTALLATION

Unfaced 10" (25.4cm) R-30 batts were installed in bays 3 and 5. Over the heat flux transducer, a fiberglass screen apparatus was installed to maintain the insulation batt at a uniform thickness above the ceiling drywall.

The insulation in the two bays with blown in loose-fill insulation was installed by a commercial insulation contractor. In Bay 4 the installer was asked to blow as light a density as he could and in Bay 2 a standard R-30 installation was asked for.

In bay 4, a problem arose that required the heat flux transducer to be removed, repaired and replaced. This necessitated the removal and hand replacement of the blown insulation in Bay 4. It was replaced to the thickness of the original installation, but the insulation was no longer installed to the lightest possible density, as originally intended.

#### INSTRUMENTATION

There is a weather station at the site which records outdoor temperature, relative humidity, wind speed and direction, and solar insolation. The test bays are designed for the measurement of heat flux, air temperature, surface temperature, air velocity, relative humidity and sheathing moisture content in each of the cells. The sensors most pertinent to this study are the temperature measuring devices (RTD's and thermocouples) and heat flux transducers.

#### Temperature sensors

Platinum resistance temperature detectors (RTD) are used for measurements of air temperature in the test cells and as reference junctions for thermocouple measurements. Air temperature sensors have radiant protection. Type T thermocouples were used for surface temperature measurements, with lead lengths identical for each sensor. The calculated error of the temperature measurements is  $0.2^{\circ}$ F ( $0.1^{\circ}$ C) relative and  $1.0^{\circ}$ F ( $0.5^{\circ}$ C) absolute. The critical temperature measurements for this study are  $\Delta$ T which are relative temperature measurements. Thus, the expected error for the  $\Delta$ T measurements is less than  $0.2^{\circ}$ F ( $0.1^{\circ}$ C).

### Heat flux sensors

In the flat ceiling bays, large  $10" \times 10" (25 \text{ cm} \times 25 \text{ cm})$  transducers are used. The transducers were calibrated in an ASTM C-518 apparatus using a NBS SRM 1450 board as a known insulation material. This SRM has been characterized as a function of temperature and density over wide ranges. By mounting the transducers such that they were in contact with the SRM, and using the hot and cold plate temperatures in the C-518 apparatus, it was possible to measure the transducer output at constant temperature over a range of heat flux values.

This was done by keeping the mean temperature constant (i.e. the temperature of the transducer) while varying the  $\Delta T$  between the hot and cold plates. This was done at mean temperatures of 65 and 75°F (18.3 and 23.9°C) which should cover the expected range of temperatures on top of the gypsum ceiling boards.



Figure 5. Calibration plots from four heat flux transducers.

The resulting calibration plots of transducer voltage output versus heat flux are shown in Figure 5. From these data linear regression analysis gave the functional form of the calibration constants for each transducer. The calibration correlation was performed using a computed (not a zero) intercept.

#### Air Flow Sensors

The air flow sensors (Shibaura Electronics F6801B) are heated thermistor anemometers. They were calibrated at the Illinois facility, and found to have a 4% interchangeability error. A single second-order polynomial was used for calibration. The sensors were placed at the air chutes in vented bays 4 and 5. The opening area in the vented bays at each soffit end is roughly estimated to be 1 sq. ft. $(0.09 \text{ m}^2)$  Thus, a 1 fpm (18.35 m/sec) velocity at the chute area corresponds roughly to 1 cfm (1.64 m<sup>3</sup>/sec) flow through that area. The fact that there are three openings, with flow varying in and out, prevents these flow calculations from being used in an air change rate calculation.

#### Data Logging

There are three data loggers (Campbell Scientific CR10) used to record over three hundred channels of data. Data are sampled every ten minutes, averaged, and stored at hourly intervals. One exception to this interval is a much shorter sampling interval for the heat flux sensors in this study. A shorter interval was selected to fall within the typical on-off cycle of the thermostat-controlled electric resistance heater. A digital microvoltmeter was used to verify the millivolt output from the heat flux sensors. The error arising from the comparison of the two voltmeters is 2%.



Figure 6. Weather conditions. Temperature is shown as indoor temperature - outdoor temperature in degrees F. Scale of solar insolation is reduced; units are  $0.1 * Btu/ft^2$ -hr. Wind speed at the soffit is shown in feet per minute.



Figure 7. Temperature difference across ceiling insulation over time, bays 3 and 5. Temperature difference is strongly dependent on outdoor air temperature and solar insolation. Bay 3 (not vented) has a lower temperature difference particularly during sunny periods.

#### FINDINGS

The study bays are continuously monitored. The study period—January 1991—was selected because the outdoor temperature for this period remained below the "indoor temperature" (maintained at 70°F, 21.1°C). During this period, all the heat flow was upward, from the warmer indoor space to the colder attic space. This uniformity of direction of flow simplifies the calculation of R-values. Findings from this period have been compared to findings over more extended time periods. The one-month period selected is representative of overall performance during the winter season. Thermal performance during summer conditions will be the subject of a future paper.

Figure 6 shows the weather conditions for the period. The outdoor temperatures remained below the indoor temperature of 70°F. There were both cloudy and sunny days. The wind is shown as the wind speed measured at the air chutes (see Figure 3 for anemometer locations).

The findings are presented in order of their derivation from natural conditions; i.e., weather conditions are used to explain the measured temperature difference, heat flux are shown as a function of temperature difference, R-value are shown as a function of heat flux and temperature difference.

#### Temperature difference as a function of outdoor weather conditions

A single chart (Figure 6) is used to represent the weather conditions during January 1991. Figure 7 shows temperature difference across the insulation in bays 3 and 5 during the same time period. Table 3 shows the correlation coefficient ( $\mathbb{R}^2$ ) for linear regression between the temperature difference measurements in the four study bays, and the outdoor conditions.

bay #	2	3	4	5	
soffit wind speed (WS)	.001	.0001	.001	.004	_
solar insolation (SO)	.156	.098	.028	.025	
outdoor air temp. (ATO)	.607	.671	.838	.871	
SO + ATO	.800	.801	.886	.917	
WS + SO + ATO	.805	.810	.889	.918	

TABLE 3 - CORRELATION COEFFICIENT ( $\mathbb{R}^2$ ) BETWEEN  $\Delta T$  and weather

Clearly the temperature difference is primarily a function of outdoor air temperature and solar insolation. During sunny periods in winter, the attic air temperature in a vented bay (bay 5) will remain lower than the attic air temperature in a bay without venting. Thus, during sunny periods, the temperature difference across the insulation in a vented bay will remain greater than the temperature difference across insulation in an unvented bay. It is important to note that the differing performance in  $\Delta T$  between the vented and unvented bays should be attributed to air exchange within the attic and cannot be attributed to differences in insulation thermal performance. However, the  $\Delta T$  difference between vented and unvented bays introduces a variable into this study, for it is apparent that the conditions to which the insulation products are subjected in the four bays are not identical.

The "indoor" conditions are maintained at 70°F. The underside of the insulation product is separated from the indoor by a sheet of 1/2" gypsum sheetrock. Thus the underside of the insulation is not maintained at strictly isothermal conditions. The temperature measuring locations are shown in Figure 4. Table 4 gives the average  $\Delta T$  measurements for the four bays.



Figures 8 and 9. Heat flux  $(Btu/ft^2-hr)$  compared to temperature difference, bays 2 and 3 (not vented).



Figures 10 and 11. Heat flux  $(Btu/ft^2-hr)$  compared to temperature difference, bays 4 and 5. (vented).

Heat Flux as a function of temperature difference

Heat flux measurements are compared to the temperature difference measurements for the four bays in Figures 8 through 11. Several effects are readily apparent. The performance of bays 3 and 5 (both with batt insulation) appear similar, with the exception that bay 3 (unvented) has data points in the 0 to20°F (-18 to -6.6°C) range, due to the venting effect upon  $\Delta T$  discussed above. The data from bays 2 and 4, containing loose-fill insulation, show an increase in slope with increasing  $\Delta T$ , and these two slopes are not similar. There is a noticeable scatter in the data. Much of this scatter can be attributed to hysteresis. The heat capacity of the insulation is not negligible, so as the temperature on the top surface varies, the variation is felt by the heat flux transducer only after a period of time. A function was introduced into the comparison of  $\Delta T$  and heat flux data sets, which sought to correlate heat flux with a value derived from the current and the previous  $\Delta T$ . The value is taken as that lag time which brings the two data sets into the tightest correlation. The data sets for bay 3, for example, shown in Figure 9, correlated most closely with a 35 minute lag time. Figures 8 through 11 show simultaneous data, not modified by the lag function. Table 4 gives the heat flux averages over the month of January for the four study bays.



Figures 12 and 13. R-values, bays 2 and 3 (not vented). Values along Y-axis in  $ft^2$ -hr- F/Btu.



Figures 12 and 13. R-values, bays 2 and 3 (not vented). Values along Y-axis in  $ft^2$ -hr-F/Btu.

#### R-value as a function of temperature difference and heat flux

Thermal resistance is calculated as the ratio of the temperature difference across the insulation to the rate of heat flux through the insulation. In this study this ratio is taken as the instantaneous ratio between simultaneous  $\Delta T$  and heat flux, with no allowance for lag time, and no averaging of values. Figures 12 through 15 show the R-value as a function of  $\Delta T$ . For this study period, the heat flux remained >0, so the common difficulty of defining R-value under conditions of changing direction of heat flux was not encountered.

Again certain effects are apparent. The performance of bays 3 and 5, both with batt insulation, appear similar (again data points appear in the 0 - 20°F [-18 to -6.6°C] range in bay 3). There is an increase in R-value with increasing  $\Delta T$ . This is due to the fact that the underside of the insulation is somewhat isothermal, and increased  $\Delta T$  indicates that the insulation is colder—it has a lower mean temperature. At lower mean temperature the radiant component of heat transfer is reduced, so the overall thermal conductivity is reduced, thus the R-value is increased. Loose fill insulation appears to perform differently. In bay 2, the R-value appears to decrease with increasing  $\Delta T$  in the range > 25°F (14°C). In bay 4, the R-value appears to increase up to  $\Delta T$ =40°F, (22°C) then decrease. Currently, the most plausible explanation of this change in performance is the possible onset of convection within the insulation[6]. This convection would be due, not to the presence of attic ventilation (indeed the vented cavity has a higher threshold of convection onset), but may be due to the initiation of convective air loops within the insulation.

#### Average values

Bay #	2	3	4	5
ΔT: 'F ('C)	37.95 (3.30)	37.81 (3.23)	38.70 (3.72)	40.75 (4.86)
Heat Flux: Btu/ft <sup>2</sup> -hr (W/m <sup>2</sup> )	2.210 (6.96)	1.340 (4.22)	1.469 (4.63)	1.373 (4.32)
R-val.: hr-ft <sup>2</sup> -*F/Btu (m <sup>2</sup> -*K/W)	17.6 (3.10)	27.9 (4.91)	27.5 (4.84)	29.5 (5.19)

#### TABLE 4 - AVERAGE VALUES FOR JANUARY

Table 4 shows the averaged values for temperature difference, heat flux and R-values for the entire study period. The field R-values for batt insulation are very close to the laboratory R-values (compare Table 2 and Table 4). A comparison between bays 3 and 5 indicates that ventilation more directly affects  $\Delta T$  than heat flux. The R-value difference between the vented and non-vented conditions, therefore, reflects variables in test conditions, and thus poorly represents differing actual thermal performance.

The average field R-values for loose-fill insulation are lower than anticipated. Among the possible causes are: convection, settling of insulation, and actual density being less than desired density. Some settling appears to have occurred, especially in Bay 2. The actual density of the material above the heat flux transducers has not been measured.

### Moisture effects

The interior spaces are maintained at 50% relative humidity (varying up to 55% rh). The ceiling is 1/2" gypsum sheetrock and there is no vapor retarder. There are no holes for convective air movement between the humidified indoor space and the attic air volume. The attic air relative humidity is measured hourly. During the study period, the relative humidity of the attic air varied greatly, between 20 and 98%. During this period, two condensation events were noted, in the early morning during particularly cold weather (outdoor air temperature °F). During those events, frost was detected on the nail points protruding through the underside of the sheathing on bays 2 and 3 (unvented) but not on bays 4 and 5 (vented). Beading water was visible near the top surface of the batt insulation in bay 3, but there was no apparent beading in the loose fill insulation in bay 2, nor in any of the insulation in bays 4 and 5. The events were transient, with all traces of moisture gone within two hours after sunrise.

A review of the data appears to indicate that neither the heat flux nor the thermal performance in bay 3 (batt insulation, no ventilation) were greatly affected by the transient presence of moisture. It appears implausible that moisture would have played a significant role in the thermal performance of loose-fill insulations, given that the noted decreases in R-value occur at low  $\Delta T$  in bay 2, and also occur in bay 4, where no condensation was noted.

#### Energy implications

The units which most directly indicate the energy performance of a study bay are the heat flux units, not the R-value performance units. Table 3 shows that the heat flux performance between bays 3 and 5 (non-vented and vented) is slight.

#### CONCLUSIONS

The presence of attic ventilation in a flat ceiling bay with proper detailing does not seem to detract from the thermal performance of the insulation material. The thermal performance of insulation materials in flat ceiling bays with this construction, does not seem to vary with air velocity, measured at the soffit air chute. The thermal performance of loose fill insulation is affected by higher  $\Delta T$  across the insulation.

#### ACKNOWLEDGEMENTS

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THERMAL PERFORMANCE OF ONE LOOSE-FILL FIBERGLASS ATTIC INSULATION

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ABSTRACT: A series of experiments on one commonly available loose-fill fiberglass attic insulation has been performed using the guarded hot box capability of the Large Scale Climate Simulator at the Oak Ridge National Laboratory. These experiments were conducted using an attic test module that was built to simulate typical residential construction. Under simulated winter conditions, the surface-to-surface thermal resistance between the bottom of the gypsum board and the top of the insulation was found to decrease substantially as the surface-tosurface temperature difference increased from 10 to 40 K. At the largest temperature differences, the resistances were as much as a factor of two lower than the estimated nominal thermal resistance of the insulation. This change in thermal resistance has been attributed to natural convection within the insulation. It should be noted that the present experiments were performed with one type of loose-fill fiberglass insulation, and the results may not be applicable to other loose-fill insulations with different air flow and thermal properties. A series of experiments was also performed with various materials laid on top of the insulation. The most effective of these covering systems produced increases in thermal resistance of more than 100 percent at the largest temperature differences, and appeared to nearly eliminate heat transfer by natural convection in the temperature range investigated.

KEYWORDS: loose-fill, fiberglass, insulation, attic, experiment, natural convection

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#### INTRODUCTION

Heat flow through fiberglass insulation is usually considered to be due to a combination of conduction through still air, conduction through the glass fibers, and radiation through a scattering, absorbing, and emitting medium. Heat transfer by natural convection has usually been considered to be negligible. Experiments performed by Wilkes and Rucker [1], using an attic test module, showed that convection is negligible for fiberglass batt attic insulation under normal operating conditions. However, their data for one type of loose-fill fiberglass attic insulation available at that time showed that under certain normal winter conditions, heat transfer by natural convection can also be an important mechanism. From tests on a box of another type of loose-fill fiberglass insulation, Langlais et. al. [2] also found that natural convection can be an important heat transfer mechanism under winter conditions, but concluded that a significant deterioration in performance would only occur under very severe climatic conditions. Using several experimental setups, Bullock [3,4] studied the benefits of membranes in improving the efficiency of fiberglass insulation, including the reduction of heat transfer by air movement.

The objective of this study was to determine whether the results of Wilkes and Rucker could be corroborated using another attic test facility insulated with a type of loose-fill fiberglass attic insulation that is commonly available from current production. If natural convection were observed, another objective was to determine the effectiveness of various types of covering materials (developed by the cosponsor) in lessening the deterioration of thermal performance due to heat transfer by convection. In addition to tests performed under winter conditions, a limited number of tests were performed under summer conditions; however, these summer results are not reported in this paper. For a complete summary of attic test results, see the report by Wilkes et. al. [5].

#### EXPERIMENTAL FACILITY

#### Large Scale Climate Simulator

The experiments were performed using an attic test module built in the Large Scale Climate Simulator (LSCS) at the DOE Roof Research Center.[6] A schematic of the LSCS is shown in Figure 1. In this facility, the test panel separates the upper climate chamber from the lower metering and guard chambers. Temperatures in the climate chamber can be controlled between -40 and 66°C (-40 to 150°F) under either steady-state or dynamic conditions. In addition, the upper surface of the test panel can be heated up to 93°C (200°F) using infrared lamps.

Temperatures in the metering and guard chambers can be independently controlled between 7 and 66°C (45 and 150°F). Normally, both these chambers are maintained at the same temperature to minimize heat flows across the metering chamber walls. The metering chamber walls are insulated with 102 mm (4 in.) thick polyisocyanurate foam. The automatic control systems for the guard and metering chambers are capable of maintaining temperatures to within  $\pm 0.06$  K ( $\pm 0.1^{\circ}$ F). The control system for the climate chamber is also capable of this same level of temperature control, except when the infrared lamps are operating.

The metering chamber contains both heating and cooling equipment. Circulation fans and heaters are supplied with direct current, and heat inputs are obtained from measurements

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#### LARGE SCALE CLIMATE SIMULATOR

CROSS SECTION

Figure 1. Cross-section of the Large Scale Climate Simulator. The climate chamber at the top simulates outdoor conditions, and the metering chamber and guard chambers at the bottom simulate indoor conditions.

of currents and voltages. Cooling is supplied by a chilled water coil, and the energy removed is obtained from a flow rate measured with a turbine flowmeter and from a temperature rise measured with a differential platinum resistance temperature device. Heat flows through the metering chamber walls are estimated using 32 differential thermocouples across the walls.

The metering chamber chilled water coil must be operated when heat is flowing from the climate chamber to the metering chamber. It must also be operated when heat is flowing from the metering chamber to the climate chamber and the heat input from the fans is larger than the heat flow through the roof panel. For all the experiments reported in this paper, the chilled water coil was turned off.

#### Attic Test Module

The attic test module was built to simulate typical residential construction, within the size constraints imposed by the LSCS. A schematic of the test attic is shown in Figure 2. It is a gabled attic, approximately 4.3 by 4.9 m (14 by 16 feet), and is constructed with 2X4 wood joists and rafters, 0.6 m (24 in.) on centers. The 5 in 12 slope roof is made of 12 mm (0.5 in.) plywood, and is covered with roofing felt and medium gray asphalt shingles. The ceiling consists of 12 mm (0.5 in.) gypsum board, and the gables consist of 12 mm (0.5 in.) plywood.

The attic is ventilated using a blower which is connected to ducts along both eaves. Cardboard baffles are attached to the rafters near the eaves to prevent insulation from covering the cave vents, and to prevent ventilation air from blowing directly through the



Figure 2. Schematic of the Attic Test Module, which was designed and built to simulate common characteristics of residential roofs.

insulation. Ventilation air exits through a ridge vent. Air flow rates are measured in the ducts using hot-wire anemometers.

The attic test module is instrumented with more than 125 thermocouples. These include arrays of 21 thermocouples that measure temperatures in the metering chamber air, the bottom surface of the gypsum board, the top surface of the insulation, and the attic air 75 mm (3 in.) above the insulation surface. The thermocouples on these arrays are arranged midway between the joists. Arrays of four additional thermocouples are placed in line with the joists on the top of the insulation and on the bottom of the gypsum board. Thermocouples for the top of the insulation and the attic air are carried on a moveable frame that is lowered so that the bottom array of thermocouples is in contact with the insulation surface. Other thermocouples are mounted on the outside of the roof (under the shingles), the underside of the roof, the inside and outside of the gables, in the attic air space, at the vent inlets and exit, and on top of the gypsum board. All thermocouples are made from Type T special limits of error wire, and all thermocouples for the bottom of the gypsum board and top of insulation are taken from the same spool of wire.

#### Data Reduction

Data are collected by a computer with all signals being recorded every four minutes. Usually, individual tests are run for a period of approximately 24 hours, with the most stable 8 hour period near the end of the test being used for data analysis. A heat balance on the metering chamber is performed for each four minute interval, and these values are averaged over the (usually) 8 hour period to obtain an average heat flow through the ceiling. Likewise, temperatures are averaged over the (usually) 8 hour period, and are also averaged over the array for a particular surface. Using these averaged quantities, a surface-to-surface thermal resistance, R, is calculated from the relationship:

$$R = \frac{A \Delta T}{Q}$$
(1)

where A is the effective area of the ceiling exposed to the metering chamber (taken to correspond to the centerlines of the 100 mm (4 in.) thick metering chamber walls, or  $6.45 \text{ m}^2$  (69.4 ft<sup>2</sup>)),  $\Delta T$  is the absolute difference between the average temperatures on the bottom of the gypsum board and the top of the insulation, and Q is the net heat loss or gain through the ceiling of the metering chamber.

#### Accuracy of Results

The accuracy of the thermal resistance measurements has been assessed by measurements on a 102 mm (4 in.) thick panel made of expanded polystyrene foam for which the thermal conductivity is well known, and also by measurements with the attic insulated with 127 mm (5 in.) of expanded polystyrene foam from the same lot of material. Results of these measurements are reported in Wilkes, et. al.[5] Under simulated winter conditions, with the metering chamber chilled water turned off, these experiments showed that the accuracy of measured thermal resistances is better than 5 percent, and the precision is about 1.5 percent.

#### MATERIALS TESTED

The materials tested included two specimens of the same brand of commonly available bonded loose-fill fiberglass insulation, and the loose-fill insulation with various covering materials.

The loose-fill insulation specimens were both installed by a local certified insulation contractor, with the same blowing machine and operators. All the material used to produce both specimens was from the same lot. Specimen 1 was installed two days after the material was produced, while Specimen 2 was installed about 6 months later. The target nominal R-value for both specimens was  $3.35 \text{ m}^2 \text{-} \text{K/W}$  (19 hr•ft<sup>2</sup>•°F/Btu). For this R-value, the label minimum thickness is 0.21 m (8.25 in.) and the label density is 8 kg/m<sup>3</sup> (0.5 lb/ft<sup>3</sup>). Rulers were placed in the attic in several locations to allow a determination of approximate blown thickness, and the total amount of insulation installed was also determined to allow estimates of average blown density.

The installed thickness of Specimen 1 was 0.23 to 0.25 m (9 to 10 in.). After testing was completed, the total weight of insulation in the metered area was determined to allow an estimate of the installed density of the metered section, which was 7.2 to 8.0 kg/m<sup>3</sup> (0.45 to 0.5 lb/ft<sup>3</sup>). Assuming a simple proportionality between thickness and R-value gives estimated nominal R-values between 3.7 and 4.0 m<sup>2</sup>•K/W (21 and 23 hr•ft<sup>2</sup>•°F/Btu). An additional adjustment is needed to account for the variation of apparent thermal conductivity with density. Assuming that the apparent thermal conductivity is the sum of the thermal conductivity of air and a radiation term that varies inversely with the density gives estimated nominal installed R-values between 3.7 and 3.8 m<sup>2</sup>•K/W (21 and 22 hr•ft<sup>2</sup>•°F/Btu). (Note, for this calculation, the thermal conductivity of air was taken to be 0.0260 W/m•K (0.18

Btu•in./hr•ft<sup>2</sup>•°F) and the apparent thermal conductivity was taken to be 0.0626 W/m•K (0.434 Btu•in./hr•ft<sup>2</sup>•°F) at a density of 8.0 kg/m<sup>3</sup> (0.50 lb/ft<sup>3</sup>)).

The installed thickness of Specimen 2 was about 0.24 m (9.5 in.). Using the total weight of insulation removed from the metered area at the conclusion of testing gave an installed density of about 6.4 kg/m<sup>3</sup> (0.40 lb/ft<sup>3</sup>). This estimate of density gives an estimated nominal R-value of 3.4 m<sup>2</sup>•K/W (19 hr•ft<sup>2</sup>•°F/Btu).

Covering materials tested in combination with Specimen 1 were: a white perforated polyethylene film, similar to those used for disposable diapers (emittance is 0.80 to 0.86); a radiant barrier with a spunbonded polyolefin film substrate (emittance is 0.08 on reflective side and 0.4 on the nonreflective side); a "pillow" consisting of a 25 mm (1 in.) thick fiberglass blanket having a density of about 13 kg/m<sup>3</sup> (0.8 lb/ft<sup>3</sup>) sandwiched between two sheets of the perforated polyethylene film; and a 25 mm (1 in.) thick fiberglass blanket having a density of about 16 kg/m<sup>3</sup> (1 lb/ft<sup>3</sup>), here called a "blanket".

The covering material tested in combination with Specimen 2 was an 0.05 mm (0.002 in.) thick continuous (nonperforated) white polyethylene film covered with the same fiberglass "blanket" as was used for Specimen 1. This exact configuration is not recommended for general use because of the potential for moisture problems. Some degree of perforations would be necessary to avoid moisture condensation.

#### **RESULTS AND DISCUSSION**

#### Loose-Fill Fiberglass Insulation Only

Data obtained under winter conditions with the loose-fill fiberglass specimens with no covering have been reported by Wilkes et. al.[7], and are repeated here for completeness. Results of measurements on these two specimens under winter conditions are given in Tables 1 and 2. Measured thermal resistances are plotted as a function of temperature difference in Figure 3. Also plotted in Figure 3 are the results of Wilkes and Rucker [1] obtained at another experimental facility using a test attic and an earlier loose-fill fiberglass insulation product.

The measured thermal resistances for Specimen 1 are as low as 1.6  $m^2$ ·K/W (9.2 hr•ft<sup>2</sup>•°F/Btu), with the resistance decreasing as the temperature difference increases. Thus the thermal resistances at the largest temperature differences are as much as a factor of two lower than the estimated nominal value of 3.7 to 3.8  $m^2$ ·K/W (21 to 22 hr•ft<sup>2</sup>•°F/Btu).

The measured thermal resistances for Specimen 2 range from a high of  $3.1 \text{ m}^2$ -K/W to a low of 2.0 (17.7 to 11.1 hr•ft<sup>2</sup>•°F/Btu), again with the resistance decreasing as the temperature difference across the insulation increases. At small temperature differences, the measured thermal resistances for Specimen 2 are about 7 to 11 percent lower than the estimated nominal R-value of  $3.4 \text{ m}^2$ -K/W (19 hr•ft<sup>2</sup>•°F/Btu). Also, it should be noted that there is very little difference between the resistances measured at temperature differences of 12 and 18 K (21 and 32°F). At large temperature differences, the resistance of Specimen 2 is about 20 percent higher than that of Specimen 1. The reason for this difference has not as yet been identified. However, based on the precision and accuracy of tests with expanded polystyrene foam insulation, this appears to be a real difference between the two specimens of loose-fill insulation.

Apj Tempera	plied itures, °C						
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F)*	Ventilation Rate, m/s**	Cciling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> •K/W (hr•ft <sup>2</sup> •°F/Btu)		
21	-13	28.3 (51.0)	0.0005	13.5	2.11 (12.0)		
21	-20	33.8 (60.8)	0.0005	18.1	1.87 (10.6)		
21	-28	38.9 (70.0)	0.0006	24.1	1.62 (9.2)		
	After running tests with radiant barrier covering:						
21	-20	34.2 (61.5)	0.0006	18.7	1.83 (10.4)		
After ru	inning tests w	vith perforated po	olyethylene film	/fiberglass	blanket "pillow":		
21	7	11.7 (21.1)	0.0004	3.36	3.49 (19.8)		
21	-20	34.8 (62.6)	0.0005	15.0	2.32 (13.2)		
	After n	unning tests with	fiberglass "blan	ket" coveri	ng:		
21	7	11.5 (20.7)	0.0004	3.69	3.12 (17.7)		
21	-20	35.3 (63.6)	0.0005	13.4	2.64 (15.0)		
After run	After running tests with fiberglass blanket covering, and after intervening tests with polystyrene foam panel:						
21	-20	36.7 (66.0)	0.0006	14.6	2.50 (14.2)		

# TABLE 1 – LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 1

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.

As noted in Table 2, one test was performed with no ventilation in the attic space. The resistance measured in this test was in very good agreement with the test under similar conditions with the attic ventilated, indicating that the imposed ventilation of the attic space did not alter the thermal resistance of the insulation.

As seen in Figure 3, the trends for thermal resistance as a function of temperature difference for both Specimens 1 and 2 are very similar to that obtained by Wilkes and Rucker. This trend is the characteristic that distinguishes the natural convection mode of heat transfer from conduction and radiation. If conduction and radiation were the only heat transfer mechanisms present, the thermal resistance would be expected to increase slightly as the mean temperature decreases, and hence as the temperature difference increases. For a porous material heated from below, it is expected that conduction and radiation alone will be

Apj Tempera	plied tures, °C				
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F) <sup>•</sup>	Ventilation Rate, m/s	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> •K/W (hr•ft <sup>2</sup> •°F/Btu)
21	7	11.5 (20.7)	0.0003	3.86	2.99 (17.0)
21	0	18.1 (32.6)	0.0005	5.81	3.12 (17.7)
21	-7	23.6 (42.5)	0.0005	8.2	2.87 (16.3)
21	-13	28.8 (51.9)	0.0006	11.4	2.54 (14.4)
21	-13	29.3 (52.7)	0	11.6	2.52 (14.3)
21	-20	34.4 (62.0)	0.0006	16.0	2.14 (12.2)
21	-28	40.2 (72.4)	0.0007	20.6	1.95 (11.1)

# TABLE 2 – LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 2

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.

present below a certain temperature difference, and heat transfer by natural convection will start at this critical temperature difference, and will become larger as the temperature difference increases.

Further evidence for heat transfer by natural convection was obtained from infrared scans of the top surface of the insulation, as reported by Wilkes et. al.[7]. These scans resembled the classical hexagonal Benard cell patterns that are observed for natural convection in fluid layers heated from below.[8] The scans suggested that cold, dense air from the attic space flowed down into the insulation at the cores of the hexagons, was warmed by heat from below and the warmer, less dense air flowed up out of the insulation at the perimeters of the hexagons.

#### Effect of Covering Materials

After a series of tests was performed with loose-fill Specimen 1 in the as-installed condition, the spunbonded polyolefin radiant barrier material was laid on top of the insulation, with the low emittance side facing the roof. Two tests were performed under winter conditions, with the results shown in Table 3 and in Figure 4. The effect of the radiant barrier is to increase the thermal resistance by 24 to 29 percent, with the percentage increase being larger at the larger temperature difference. Likewise, the radiant barrier produces reductions in heat flux of 19 to 22 percent, again with the larger percentage reduction occurring at the higher temperature difference. The effect of the radiant barrier under winter conditions appears to be that it blocks direct flow of air from the attic space into



Figure 3. Thermal resistance of loose-fill fiberglass insulation.

## TABLE 3 – LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 1 Covered with a Radiant Barrier

Ap Tempera	plied tures, °C				_
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F) <sup>•</sup>	Ventilation Rate, m/s**	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> •K/W (hr•ft <sup>2</sup> •°F/Btu)
21	-20	33.9 (61.0)	0.0006	14.6	2.32 (13.2)
21	-28	39.1 (70.4)	0.0006	18.7	2.10 (11.9)

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.

the insulation, but does not eliminate convection within the loose-fill insulation itself, as evidenced by the trend of the resistance to decrease with increasing temperature difference.

The radiant barrier was then removed, and a test with the loose-fill only was repeated, with the results shown in Table 1. These results are in good agreement with the results of the first sequence of tests with loose-fill only, indicating that the installation and removal of the radiant barrier did not alter the thermal performance of the loose-fill insulation.

Next, the loose-fill insulation was covered with the perforated polyethylene film. The results of three tests performed under winter conditions are shown in Table 4 and in Figure 4. The effect of this film is to increase the thermal resistance by 5 to 10 percent, and to lower the heat flow by 4 to 9 percent. It appears that this film is much less effective in blocking air exchange between the attic space and the insulation than was the radiant barrier, as might be expected because of the perforations.

Following these tests, the "pillows" were laid on top of the loose-fill insulation. As described in the section on materials tested, the pillows consist of a 25 mm (1 in.) 13 kg/m<sup>3</sup> (0.8 lb/ft<sup>3</sup>) fiberglass blanket sandwiched between two sheets of perforated polyethylene film. The results of three tests under winter conditions are shown in Table 5 and in Figure 4. These results show that the effect of the "pillow" is to increase the resistance by as much as 104 percent, and to decrease the heat flow by as much as 48 percent. The resistance decreases slightly with increasing temperature difference, indicating that some natural convection is still occurring.



Figure 4. Thermal resistance of loose-fill fiberglass Specimen 1 with and without covering materials.

# TABLE 4 -- LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 1 Covered with a Perforated Polyethylene Film

Apj Tempera	plied itures, °C				
Metering Chamber Air	Climatc Chamber Air	Surface-to- Surface Temperature Difference, K(°F) <sup>•</sup>	Ventilation Rate, m/s	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> •K/W (hr•ft <sup>2</sup> •°F/Btu)
21	-13	28.8 (51.8)	0.0005	12.3	2.32 (13.2)
21	-20	34.0 (61.3)	0.0005	17.4	1.95 (11.1)
21	-28	39.4 (71.0)	0.0005	22.9	1.72 (9.8)

\*Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.

TABLE 5 -	· LSCS Results	for Winter C	Conditions wit	h Loose-Fill Fib	erglass Insulation
Specimen :	1 Covered with	a Perforated	Polycthylene	Film/Fiberglass	Blanket "Pillow"

Ap Tempera	plicd itures, °C				
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F) <sup>•</sup>	Ventilation Rate, m/s**	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> • K/W (hr • ft <sup>2</sup> • ° F/Btu)
21	0	18.3 (33.0)	0.0001	4.44	4.14 (23.5)
21	-20	35.9 (64.6)	0.0005	9.80	3.66 (20.8)
21	-28	41.4 (74.6)	0.0006	12.6	3.31 (18.8)

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.

After the "pillows" were removed, it was apparent that the pillows had caused compaction of the loose-fill insulation by about 25 mm (1 in.). A repeat test performed with the loose-fill only showed that the resistance had <u>increased</u> by about 25 percent at a temperature difference of 35 K ( $63^{\circ}$ F), as shown in Table 1. When the thickness of fiberglass insulation is decreased, it is usually believed that the resistance also decreases. The increase observed here is thought to be due to the compaction leading to a decrease in the air flow permeability of the insulation, and a consequent reduction in the heat transferred by natural convection.

The effect of the pillows is thought to have been fourfold. First, the weight of the pillows appears to have compacted the loose-fill insulation, causing a decrease in convection heat transfer, and an increase in resistance at a given temperature difference across the loose-fill insulation. Second, the thermal resistance of the pillows should cause a reduction in the temperature difference across the loose-fill insulation itself, since a part of the total temperature difference will be across the pillow. A smaller temperature difference across the loose-fill insulation would decrease the amount of convection heat transfer, and would result in a resistance for the loose-fill part that would be larger than it would have been if the full temperature difference had occurred across the loose-fill. Third, the 25 mm (1 in.) thick 13 kg/m<sup>3</sup> (0.8 lb/ft<sup>3</sup>) fiberglass blanket inside the pillow is estimated to have a thermal resistance of 0.5 to 0.7 m<sup>2</sup> K/W (3 to 4 hr + ft<sup>2</sup> • °F/Btu) which would be added to the thermal resistance of the loose-fill insulation. Finally, the relatively high air flow resistance of the pillow would have provided a barrier to interchanges of air between the attic air space and the loose-fill insulation.

The next set of tests involved uncovered 25 mm (1 in.) thick fiberglass "blankets" having a density of about 16 kg/m<sup>3</sup> (1 lb/ft<sup>3</sup>) that were laid on top of the loose-fill insulation. The results of three tests under winter conditions are shown in Table 6 and in Figure 4. These resistances are 12 to 16 percent larger and the heat flows are 10 to 13 percent lower than the results with the pillows in place. A large part of this increase appears to be due to a further increase in the resistance of the loose-fill insulation. After removal of the blankets, the resistance of the loose-fill insulation at a temperature difference of 35 K (64°F) was found to be 14 percent higher than the value measured under the same conditions after the pillows had been removed (see Table 1). Thus, the blanket is only slightly more effective than the pillow, possibly due to the blanket having a slightly larger thermal resistance and/or air flow resistance due to its higher density.

After these series of tests, the attic module was removed from the LSCS, and tests were performed with the expanded polystyrene foam panel. Then the attic was reinstalled, without disturbing the loose-fill insulation. After reinstallation, the resistance measured under one winter condition was found to be within about 5 percent of the value measured just before the attic was removed, as shown in Table 1.

After the experiments had been performed with loose-fill Specimen 2 with no covering, experiments were performed with the loose-fill insulation covered with a nonperforated white polyethylene film and the 25 mm (1 in.) 16 kg/m<sup>3</sup> (1 lb/ft<sup>3</sup>) fiberglass blankets. Results under winter conditions are given in Table 7 and in Figure 5. These results show that the effect of the covering is to increase the thermal resistance by 33 to 120 percent, and to decrease the heat flows by 24 to 51 percent, with the larger percentage changes occurring at the largest temperature differences.

Apj Tempera	plied itures, °C				
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F) <sup>•</sup>	Ventilation Rate, m/s**	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> • K/W (hr • ft <sup>2</sup> • ° F/Btu)
21	-13	30.3 (54.5)	0.0005	6.95	4.35 (24.7)
21	-20	36.4 (65.5)	0.0005	8.84	4.12 (23.4)
21	-28	42.5 (76.5)	0.0005	11.0	3.86 (21.9)

### TABLE 6 -- LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 1 Covered with a Fiberglass Blanket

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

\*\*Ventilation rate is cubic meters per second per square meter of ceiling area.

With this covering in place, the resistance starts at 3.98  $\text{m}^2$ •K/W (22.6 hr•ft<sup>2</sup>•°F/Btu) at the smallest temperature difference, increases to a high of 4.33  $\text{m}^2$ •K/W (24.6 hr•ft<sup>2</sup>•°F/Btu) at temperature differences of 30 to 37 K (54 to 66°F), and then decreases slightly to 4.30  $\text{m}^2$ •K/W (24.4 hr•ft<sup>2</sup>•°F/Btu) at the highest temperature difference.

This trend with the covering in place suggests that heat transfer by natural convection is almost completely eliminated except at the highest temperature differences. The slight trend for the resistance to increase with increasing temperature difference is attributed to the dependence of the apparent thermal conductivity (i.e., conduction plus radiation) on mean temperature. Using the four data points at the smallest temperature differences, a linear regression of resistance versus <u>mean</u> temperature gives a normalized slope of about 0.8 percent per °C, which is in reasonable agreement with the normalized slope of 0.5 percent per °C that was estimated by Wilkes and Rucker [1] for loose-fill fiberglass insulation.

Extrapolating this regression equation to a mean temperature of 24°C (75°F) gives a resistance of 3.70 m<sup>2</sup>•K/W (21.0 hr•ft<sup>2</sup>•°F/Btu). Since the resistance of the fiberglass blanket was estimated to be 0.5 to 0.7 m<sup>2</sup>•K/W (3 to 4 hr•ft<sup>2</sup>•°F/Btu), the resistance of the loose-fill only would be estimated to be 3.0 to  $3.2 \text{ m}^2$ •K/W (17 to 18 hr•ft<sup>2</sup>•°F/Btu). Considering that the blanket has probably compacted the loose-fill insulation by about 25 mm (1 in.), these last values are in reasonable agreement with the values of 2.99 and 3.12 m<sup>2</sup>•K/W (17.0 and 17.7 hr•ft<sup>2</sup>•°F/Btu) that were obtained for the loose-fill before the covering was applied, and also are in reasonable agreement with the estimated nominal R-value of 3.4 m<sup>2</sup>•K/W (19 hr•ft<sup>2</sup>•°F/Btu) that was obtained from the estimate of installed density.

### TABLE 7 -- LSCS Results for Winter Conditions with Loose-Fill Fiberglass Insulation Specimen 2 Covered with Polyethylene Film and Fiberglass Blanket

Apj Tempera	plied tures, °C				
Metering Chamber Air	Climate Chamber Air	Surface-to- Surface Temperature Difference, K(°F)*	Ventilation Rate, m/s	Ceiling Heat Flux, W/m <sup>2</sup>	Thermal Resistance m <sup>2</sup> •K/W (hr•ft <sup>2</sup> •°F/Btu)
21	7	11.7 (21.0)	0.0007	2.93	3.98 (22.6)
21	0	18.4 (33.1)	0.0005	4.36	4.21 (23.9)
21	-7	24.3 (43.7)	0.0006	5.72	4.24 (24.1)
21	-13	30.1 (54.2)	0.0006	6.97	4.33 (24.6)
21	-20	36.7 (66.0)	0.0006	8.45	4.33 (24.6)
21	-28	43.2 (77.6)	0.0006	10.1	4.30 (24.4)

<sup>\*</sup>Temperature difference is measured between bottom of gypsum wallboard and top of insulation. Thermal resistance is calculated using this temperature difference and measured heat flux.

"Ventilation rate is cubic meters per second per square meter of ceiling area.



Figure 5. Thermal resistance of loose-fill fiberglass Specimen 2 with and without a covering material.
### Comparison with Theory for Onset of Convection

Theories for the onset of natural convection in a porous medium heated from below are well known. These theories are usually cast in terms of the dimensionless Rayleigh number, Ra, which is defined as

$$Ra = \frac{g\beta\rho C_{p}LK\Delta T}{vk}$$
(2)

g is the acceleration of gravity;  $\beta$ ,  $\rho$ ,  $C_p$ , and  $\nu$  are the volume expansion coefficient, density, specific heat, and kinematic viscosity of air; L, K, and k are the thickness, air flow permeability, and apparent thermal conductivity of the porous medium; and  $\Delta T$  is the temperature difference across the porous medium.

The theories predict that natural convection will not occur until the Rayleigh number exceeds a certain critical value. Nield shows that the critical Rayleigh number depends upon both the thermal and mechanical boundary conditions on both sides of the porous medium.[9] For a porous medium confined between two isothermal impermeable boundaries, the critical value of Ra is  $4\pi^2$ , or 39.5. As any of the boundary conditions is relaxed, the critical value of Ra decreases. For example, for an isothermal impermeable bottom boundary, and an isothermal permeable top boundary, the critical value of Ra is 27.1. Further, if the top boundary condition is changed to permeable with a uniform heat flux, the critical value of Ra becomes  $\pi^2$ , or 9.9. Experiments performed by Silberstein et. al.[10] with a box of loose-fill insulation with an unvented air space over it gave a critical Rayleigh number of about 12, while the attic experiments of Wilkes and Rucker gave critical Ra values of about 10 to 30.

For the type of insulation used in this experiment, measured air flow permeability values range from 6.2 X  $10^8$  to 9.3 X  $10^8$  m<sup>2</sup>.[11,12] It is difficult to determine accurate critical temperature differences at which convection is initiated from the data shown in Figures 3 through 5. For Specimen 2, the critical temperature difference may be about 22 K. Using a thermal resistance of 3.12 m<sup>2</sup>·K/W (17.7 hr·ft<sup>2</sup>·°F/Btu) for Specimen 2 gives estimated critical Rayleigh numbers of 13 to 20.

Although the critical Rayleigh numbers derived from the present experiments vary greatly, they are in qualitative agreement with the results of Silberstein et. al. and Wilkes and Rucker in that they are much less than the classical value of 39.5 that applies for isothermal impermeable boundaries. This factor is significant since these are just the boundary conditions that are usually imposed for routine small scale tests of loose-fill insulation.[13] Routine tests are usually performed at a mean temperature of 24°C (75°F) and a temperature difference of 22 K (40°F). For a thermal resistance of  $3.35 \text{ m}^2 \cdot \text{K/W}$  (19 hr  $\cdot \text{ft}^2 \cdot ^\circ \text{F/Btu}$ ) and an air flow permeability of 9.3 X  $10^8 \text{ m}^2$ , the corresponding Rayleigh number is 18. Since this is so much less than 39.5, it is not surprising that natural convection effects are not observed with the routine tests.

# SUMMARY AND CONCLUSIONS

A series of experiments has been completed on the thermal performance of one loosefill fiberglass attic insulation under winter conditions using an attic test module in a guarded hot box facility. The results showed that as the temperature difference between the bottom of the gypsum board and the top of the insulation increased from 10 to 40 K, the surface-tosurface thermal resistance decreased substantially. At the largest temperature differences, the thermal resistances were as much as a factor of two lower than the estimated nominal R-value of the insulation. This decrease in resistance has been attributed to natural convection within the insulation. It should be noted that the present experiments were performed with one type of loose-fill fiberglass insulation, and the results may not be applicable to other loose-fill insulations with different air flow and thermal properties.

Various covering materials showed varying degrees of success in lessening the deterioration in resistance due to convection. It was found that a single perforated polyethylene film or a radiant barrier was only moderately effective in blocking convection. However, coverings that included a 25 mm (1 in.) layer of fiberglass blanket were very effective in blocking the effects of convection. The most effective of these coverings, consisting of a nonperforated polyethylene film covered with a fiberglass blanket, produced as much as a 120 percent increase in resistance at the highest temperature differences.

With no covering on the loose-fill insulation, the lowest resistances were measured at the largest temperature differences, at which the heating loads on a house would be the greatest. Thus, under winter design conditions, the effects of convection could be significant. For example, for Minneapolis, the 99 percent winter design temperature is  $-27^{\circ}$ C ( $-16^{\circ}$ F).[14] This corresponds closely with the most extreme conditions used in this study, for which the thermal resistance was as low as one-half of its value at mild conditions. Hence, the contribution of the ceiling heat loss would be about twice as large as would have been expected in the absence of convection. However, very cold conditions will only occur for a limited amount of time, depending upon the climate, and further analyses are required to determine the net impact of convection on the seasonal heating load on a house.

Analysis of the measured data shows that in the attic module, the onset of convection occurs at a critical Rayleigh number of 13 to 20. This is in qualitative agreement with critical Rayleigh numbers of 10 to 30 reported in the literature for loose-fill insulation. These critical Rayleigh numbers are much smaller than the value of 39.5 which would apply to routine small-scale tests of loose-fill insulation. As a consequence, the small-scale tests do not properly simulate the thermal performance of attic insulations that may be affected by natural convection.

# RECOMMENDATION

Because of these findings, it is recommended that efforts be made to modify the routine small-scale testing procedures to simulate more closely the performance of loose-fill fiberglass insulation in an attic environment.

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FORCED CONVECTION EFFECTS IN FIBROUS THERMAL INSULATION

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ABSTRACT: We relate experiments carried out both in the lab, in a guarded hot box, and in experimental test houses. It is shown that forced convection does not significantly affect the thermal properties of the insulation under the air velocities usually found in the cavities, unless the construction mode and/or workmanship allow multiple air entry zones, or discontinuities in the insulation or in the internal air barrier.

A computer model was also developed to help us understand the convective flow patterns inside the porous insulation, as induced by the forced convection above it. A fairly good agreement with experimental results could be obtained.

KEYWORDS: insulation, convection, thermal conductivity, model of convection, roof, ventilated façade

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# INTRODUCTION

The thermal properties claimed by insulating products manufacturers are obtained in ideal laboratory conditions, the insulant being confined between two isothermal plates. In reality, current construction practice, or bad workmanship will often place the insulation in building components in contact with air spaces, ventilated or not.

Natural convection takes place for instance in the large air volume of an attic space, or in cavity walls whenever workmanship allowed the presence of air gaps in the cavity. Both of these cases have been dealt with in previous studies [1], [2], and the impact of natural convection on the effective thermal properties of various types of insulating materials has been discussed. Depending on workmanship, it was found that these air gaps, in combination with joints between panels, may significantly reduce the thermal resistance of the insulated structure.

Forced convection is also often present in buildings, due to the necessity of ventilating away moisture that may have diffused or convected from the inside of the structure. This is the case for example of insulated flat or tilted roofs (cathedral ceilings), and of ventilated façades (Figure 1). These air flows are usually parallel to the surface of the insulation, and should be distinguished from flow patterns through the insulation, a consequence of bad workmanship which can be considered as covered by J. Lecompte's study [2]. Many parameters may have an eventual influence on the increased heat losses due to air flow along the insulation in building structures. Among them, one can cite: the air velocity in the air gap (which is linked to the pressure difference over the air gap, and to the air gap's geometry), the air permeability of the insulation (linked to the product's density and homogeneity), the insulation thickness, the pressure difference across the insulation, etc ...

To the authors' knowledge, very little work has been published that relates the influence of forced convective air movements on the thermal performance of building elements. An extensive experimental (laboratory experiments) and modeling study was carried out by C. Bankvall [3] for the specific case of a ventilated external house wall. It was shown that even under rather severe experimental conditions (16 kg/m<sup>3</sup> insulation board, no wind protection against the insulation layer, and air velocities along the insulation of 2.5 m/s) a ventilated external wall lost at most 10% in thermal resistance. If on the other hand the insulation was installed in the wall with workmanship defects such as vertical cracks, a significant decrease of up to 40% in the overall thermal performance was observed.



Figure 1 -- Schematics of a ventilated pitched roof structure (left), and a ventilated façade (right). (1) inside lining; (2) ventilated air gap; (3) insulation; (4) vapor/air barrier; (5) roof covering; (6) external wall

But despite apparent good workmanship, recent measurements carried out on a typical European ventilated pitched roof (Figure 1a) showed a significant increase of the heat losses, dependent on the external wind speed, whenever the top surface of the insulation wasn't protected from air penetration by an impermeable sheet [4].

Similarly, hot box experiments performed at the Agrément Board [5], in England, indicate a 50% increase in heat losses when an air flow of at most 1 m/s was blown above the surface of a 10 to 12 kg/m<sup>3</sup> glass fiber blanket.

These results, in apparent contradiction with Bankvall's study, triggered the following analysis. In a first part, we report on experimental results obtained in a guarded hot box under forced convection conditions. These results have been used in a second part to validate a computer model developed as a support for understanding the flow patterns within the insulant. In a third part, field data are analyzed with the double purpose of determining realistic velocities in the cavity, and of characterizing the heat transfers in vented roof and wall structures; these experiments are still going on, so the results that we present should not be considered as exhaustive. Finally, in the light of these results, we conclude with some advice for the construction of the best possible thermally safe vented structure.

### EXPERIMENTAL AND NUMERICAL RESULTS

Many research laboratories both in Europe and in the United States are currently working on the subject of air infiltration in walls and roofs, and their consequences on the simultaneous hygric and thermal behavior of building elements. Because very little data are available today regarding air velocities in these vented cavities, a majority of these labs have chosen to run full scale experiments, in specially built test houses. So they can be more representative of the reality, and at the same time acquire some experience on the relationship between the external wind velocity/orientation and the velocity of air in the cavity. Among these laboratories, we have closely followed the studies undertaken at the Small Homes Council (University of Illinois at Urbana Champaign, USA), and the Fraunhofer Institut (Holzkirchen and Stuttgart, West Germany), and we report on a study carried out in the National Building Research Institute test house (Denmark).

On the other hand, interpretation of field data is seldom straightforward, and it often helps to use numerical modeling to understand the various phenomena involved. Having in mind that model validation requires extremely well defined laboratory conditions, and that data concerning the behavior of low density products (less than 10 kg/m<sup>3</sup>) were missing, despite their common use for cathedral ceiling insulation in southern Europe and in the United States, we chose to run hot box experiments on these very air permeable products.



Figure 2 -- Schematic of Certain Teed's experimental equipment.

Test	#	Test thickness (m)	Test density (kg/m³)	Air permeability (m²)	Nominal k-value (W/m.K)	Open	Sealed
		0.254	8.25	4.16 x 10-9	0.0708		x
В		0.254	8.25	4.16 x 10-9	0.0708	х	
C		0.254	7.73	3.78 x 10-8	0.0690		x
D		0.254	7.73	3.78 x 10-8	0.0690	x	

TABLE 1 -- Hot Box test configurations

# <u>Guarded hot box experiments</u>

In order to obtain data under more controlled laboratory conditions, guarded hot box experiments were carried out in CertainTeed's Thermal Laboratory, in the Levecque Technical Center (Blue Bell, Pennsylvania).

The specimen box, 1.15 x 1.15 x 0.254 m, was placed in a support frame 0.165 m thick which is filled with polystyrene foam that fills the entire 3 x 3 m area of the box. On the upper side of the specimens, cooling was achieved by allowing liquid nitrogen to evaporate into the cold box with the flow of liquid N<sub>2</sub> controlled with a temperature controller and a solenoid in the liquid N<sub>2</sub> line. In addition, on the cold box side, an air diffuser was constructed which allowed the cold air (N<sub>2</sub>) to be blown across the specimen (Figure 2). A fixed polyethylene sheet was placed 0.032 m above the specimen's surface to delimit a well defined ventilation cavity, and ensure that the air was really flowing above the whole surface.

Test	#	Tc	Th	Т	Tmean	Average	Heat flux	k-value	Nusselt
						air flow			number
		(°C)	(°C)	(K)	(°C)	(m/s)	(W/m²)	(W/m.K)	
A		-10.4	27.2	37.6	8.4	0	11.02	0.0744	1.00
		-11.7	27.6	39.3	7.9	0.8	10.92	0.0705	1.00
		-12.1	28.1	40.2	8.0	2.2	11.21	0.0709	1.00
в		-10.8	28.4	39.2	8.8	0	10.41	0.0674	1.00
		-12.2	28.3	40.5	8.1	0.8	11.35	0.0712	1.00
		-13.6	27.5	41.1	6.9	2.2	19.73	0.1220	1.72
С		-10.6	26.8	37.4	8.1	0	9.93	0.0675	1.00
		-10.6	27.7	38.3	8.6	0.8	11.01	0.0731	1.00
		-11.0	28.3	39.3	8.6	2.2	10.47	0.0677	1.00
D		-11.0	27.1	38.1	8.0	0	10.12	0.0674	1.00
		-11.0	26.6	37.6	7.8	0.8	12.47	0.0842	1.22
		-13.0	26.4	39.4	7.7	2.2	20.08	0.1294	1,88

TABLE 2 -- Hot Box test data



Figure 3 -- Experimental variation of the Nusselt number (i.e. ratio of heat flux with convection to heat flux without convection) with air velocity above the insulation. Also shown is the result of a numerical simulation of forced convection above the open batt insulation.

The entire box is instrumented in accordance with ASTM C-236 Test Method [6], using Type T thermocouples and DC voltages to the fans and heaters. In operation, a HP 9816 computer interfaces with a Fluke 2240 data logger to compute average temperatures and power levels which are then used to control the metering box and guard box heaters.

Two types of glass fiber products carefully placed on a gypsum board drywall were tested, one being a very low density, rather air-opened loose-fill material (tests C-D), and the other a glass fiber blanket, specially manufactured to reach the same thermal conductivity level as the loose-fill product, despite its much lower air permeability (tests A-B). For each of these products, tests were run with and without a polyethylene cover above the top cold surface. The four configurations are described in Table 1. Air velocities between 0 and 2.2 m/s were used.

Test data are given in Table 2, and experimental results are summarized in Figure 3. As expected, no heat transfer increase appears when either one of the test specimens is covered by an impermeable sheet. On the other hand, if the interface with the ventilated cavity is open, we start to see an effect of air penetration on the measured heat flux almost as soon as some air flow is forced above the insulation. This interpretation of the



Figure 4 -- Temperature profiles obtained for the loosefill product (a) for varying interface conditions, and (b) at two different distances away from the air inlet for an air speed of 2.2 m/s.

heat transfer results is confirmed by an analysis of the temperature profiles in the height of the specimens, which show an marked bow (implying convective movements in the bulk of the specimen) when the top facing is removed (Figure 4a).

Although the loose-fill product is more sensible to air penetration than the blanket is, no significant effect is to be found for any of the two products with air velocities under about 0.5 m/s. The glass fiber blanket is little affected even for air speeds of around 1 m/s.

## Description of the numerical model

We propose here a modeling of the thermal behavior of a porous material submitted to a tangential flow on one of its permeable faces. Two different approaches are presented: the first one takes fully into account the thermal and dynamical coupling between the fluid inside and outside the porous medium (Complete Modeling A). However, since for physical and numerical reasons we cannot reach the real outflow conditions, we developed another model limited to the porous layer, the external flow appearing only as a simplified boundary condition (Reduced Modeling B). Both models A and B are summarized in Figure 5.

<u>Model A</u>: The geometrical configuration (Figure 5a) consists of a channel (total length L, total height H) bounded on its top and bottom by two impermeable planes. The (cold) fluid flow enters the channel from the left hand side with a mean velocity  $V_f$ , licks the porous layer



Figure 5 -- Geometrical configurations used in the two numerical models: (a) complete modeling; (b) reduced modeling.

(height  $h_p$ , conductivity  $\lambda_p$ , permeability K) located in the lower part of the channel, and eventually penetrates it. The lower face is kept to a uniform hot temperature  $T_h$ . The outflow is assumed to be free (i.e. longitudinal gradients of temperature and velocities equal to zero).

The set of equations is identical to the one used for classical problems of mixed convection in fluid/porous media. We refer the reader to our previous paper [1] where the equations and the attached numerical procedure are described in detail. This problem is entirely defined by a set of dimensionless parameters:

the Reynolds number: $REY = V_f H/v$ the Prandtl number: $PR = v/a_f$ the Darcy number: $DA = K/H^2$ the aspect ratios:A = L/Hmand finally, the conductivity ratio: $\Lambda = \lambda_p/\lambda_f$ 

(v designates the kinematic viscosity of the fluid and  $a_f$  its thermal diffusivity).

For our purpose, and because the aim of the present work is not to present a complete parametric study, we consider that most of these parameters are fixed. Computations are carried out for an air flow (PR=1), an ideal porous material ( $\Lambda = 1$ ), while the geometrical configuration is defined by A=5, B=0.5.

Considering a fluid flow of 1 m/s in a channel of a total height of 0.1 m, the Reynolds number reaches  $10^4$ , while for low density insulating materials, the reduced permeability, or Darcy number, is of the order of  $10^{-6}$ .

### STREAM LINES

**ISOTHERMS** 



lso-Ψvalue [-0.05,0.] Ψ=0.005 [0.,+0.0025] ∀=0.0005 Iso-T value [-0.5,+0,5] T+0.1

Figure 6 -- Calculated flow and thermal patterns for two values of the Reynolds number.

The high value of the Reynolds number leads one to predict that the fluid flow may have a turbulent behavior. Such flows are not easy to compute, so we chose to reduce the Reynolds to  $10^3$  to be able to produce numerical results, while at the same time increasing the Darcy number to  $10^{-4}$ , or even  $10^{-3}$ , to remain in similitude conditions (constant REY.DA<sup>0.5</sup> product). Further, the experimental values of both Rayleigh and Reynolds numbers allow one to neglect free convection in the fluid with regard to forced convection (RAY/REY<sup>2</sup> < 1).

Typical calculated flow and thermal patterns are presented in Figure 6 (here with an aspect ratio A equal to 2 instead of 5 for graphic presentation convenience) for a highly permeable porous layer ( $DA = 10^{-3}$ ): the streamlines are deviated into the porous part of the channel, showing some penetration (for a rigorously impermeable porous material, the streamlines would remain straight and only located in the fluid part of the channel). This deviation corresponds in the temperature field to a densification of the isotherms on the left hand part of the porous layer, and therefore to a local increase of the heat transfer. When the flow is fully established, the temperature field is quasi-conductive, so that there is in fact no local increase of the heat transfer. We notice that for a higher value of the Reynolds number (Rey =1000 instead 100), penetration occurs farther away from the inlet, and consequently a recirculating flow (but with a very low velocity) may appear in the left corner of the porous layer.

Interestingly, this rather localized convective behavior in the insulation was also measured in the hot box experiment: Figure 4b shows that for a given air velocity, a temperature profile taken a little away from the air inlet is hardly deformed by convective flows.

An analysis of the vertical velocity profile on the top surface of the porous layer would show that the zone affected by the penetration increases with the Reynolds number, but the flow rate seems to remain constant as soon as the established regime is reached.

<u>Model B</u>: We now consider only the porous layer. In this case, and after observation of the velocity profiles obtained by a complete modelling of the system, we idealize the vertical velocity distribution on the insulation top surface by a linear profile (Figure 5b), negative when air penetration occurs, and positive when it leaves the porous material. Rigorously, we would have to consider a realistic temperature distribution at the top, depending on the Reynolds number of the external flow. As a first assumption, we consider here that this temperature is uniform and equal to the one of the incoming flow (true if the Reynolds number is high enough: see the isotherm patterns in Figure 6). The penetrating flow rate is estimated from the established velocity profile in the porous layer:

$$\int_{0}^{L} V_{z}(x,z=h) dx = \int_{0}^{n_{p}} V_{x}(x=L,z) dz = V_{p} h_{p}$$
(1)

One can easily show that a relation between the flow rates in the porous and fluid parts of a channel (of respective heights  $h_p$  and  $h_f$ ) may be written in the following way:

$$V_p = 12 (K/h_f^2) V_f$$
 (2)

(rigorously this relation is an approximation which is only valid if the ratio  $K/h_f^2$  is lower than  $10^{-3}$ ). One can note in relation 2 the identical role played by the fluid flow rate  $V_f$  and the permeability K.

The fluid flow and heat transfer inside the porous material have been analyzed for different flow rates in conditions identical to CertainTeed's experiments  $(h_p=0.254 \text{ m}, h_f=0.0317 \text{ m}, L=1.15 \text{ m}, K=5.10^{-8} \text{ m}^2)$ . We observe an increase in the heat transfer (estimated by a Nusselt number NU, ratio of the effective heat transfer to the purely conductive one) with the mean velocity of the external fluid flow. The predicted tendency is in qualitative agreement with the experimental data, even though the model overestimates the heat flux increase (Figure 3). This discrepancy may be partly explained by the assumption made in the model of a vertical velocity profile, but maybe more important is the fact that we assumed a uniform top temperature which obviously penalizes the calculated heat transfer.



Figure 7 -- Air velocity in ventilated air space vs external wind velocity, for an inclined roof - pitch is 22°, and overall length is 3 meters (from ref. [7]).

# Air velocities in the cavity

Until about a year ago, extremely scarce data could be found giving actual air velocities in ventilated cavities of building elements, and existing laboratory experiments were using "probable" values, usually of the order of a few meters per second. Two recent studies, carried out respectively on a ventilated cathedral ceiling at the Small Homes Council [7] (University of Illinois at Urbana Champaign, USA) and on a ventilated façade at the National Building Research Institute [8] (Denmark), give us new insight of the actual air velocities in vented cavities.

The test house of the University of Illinois, oriented east-west with roof slopes facing north-south, is designed to study simultaneously, side by side, heat and moisture transfers in flat ceiling attics and cathedral ceilings. The roof has a pitch of about 22° and a length of about 3 meters.

The Danish experiment was done on a 3 meter high ventilated façade. Care has been taken to install correctly the insulation material, and to make sure that the internal wall sheet was absolutely air-tight.

Figures 7 and 8 show the results obtained with these two facilities: in both cases, wind speeds remained below 10 m/s. In the pitched roof case, the air velocity in the cavity peaked to 1.5 m/s, but remained on average around 0.4 m/s. In the case of the ventilated façade, air



Figure 8 -- Air velocity in the ventilated space of a wall, as a function of wind speed (from ref. [8]).

velocities reached at most 0.2 m/s in the cavity. The magnitude of these values are in good agreement with those given by Liersch [9], whose measurements never showed velocities above 0.2 m/s.

If we assume that the pressure difference over the height of the cavity is expressed by [3]:

$$\Delta P / \Delta x = \beta \rho V^2 S / 8 A \tag{3}$$

where:  $\beta$  is a friction factor (defined in ref. [10]),  $\rho$  is the density of air,

- V is the air velocity,
- S is the cavity perimeter, and
- A is the cavity cross-sectional area,

a velocity of 0.2 m/s in the studied cathedral ceiling would correspond to a pressure difference between the top and the bottom of the roof of approximately 0.01 Pa, and the pressure difference would reach a maximum of 0.22 Pa for the maximum observed velocity.

# Analysis of the results

Further to the heat transfer measurements performed in the previously mentioned test facilities, the Fraunhofer Institut in Stuttgart (Germany) carried out full scale measurements on a 4 x 2 m (width x height) ventilated test roof placed in a very large climatic simulator [11]. Two types of products, representative of the roof insulation used in Germany were used, with densities of 14 and 18 kg/m<sup>3</sup>. Test conditions in all three full scale studies are summarized in Table 3.



Figure 9 -- Measured heat flux as a function of external wind speed: influence of the presence of an air-tight wind protection on a 23 kg/m<sup>3</sup> rock wool insulation (from ref. [8]).

Of all three experiments, the University of Illinois test configuration is the only one that uses very low density products, and that is representative of realistic conditions of workmanship, but for various reasons, there wasn't enough data for these cathedral ceilings available for analysis at the time of writing of this paper. These data will be published in a later paper. In the remaining experiments, extreme care has been taken to ensure a perfect fit and air-tightness on the inner side of the building element, although the wind protection used in the Fraunhofer Institut's experiment was discontinuous at the joists.

As expected from the modeling predictions at equivalent air velocities, existing data (Figures 9, 10, and 11) show very little, if any, increased heat losses due to air infiltration into the insulation. There is further no measurable difference between a low density glass wool product, and a rock wool product of higher density, since unfiberized particles, which add weight to the material, do not in fact play any role in the air flow resistance of the product. However, the effective thermal resistance of the materials (faced or not) tested in the German experiment decreases of about 3% with increasing air velocity, probably due to air infiltrations around the installed product close to the joists. Such a result indicates that unless the insulation is completely confined between two impermeable facings, a decrease of the effective thermal resistance will take place.

The question thus remains to understand the abovementioned German and British results, which show a drastic decrease of the thermal performance of the building

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		TABLE	3 Test	t condi	tions	of the f	ield stu	ıdies.	
Location of study	Mater test	ial ed	Average density (kg/m3)	Wall	Roof	Max. me gap ve] (m/	asured .ocity s)	Conditions of fit of the insulation	Max. observed heat losses increase
Denmark	glass mineral	wool wool	13 23	××		0 0	0 0	perfect, with air-tight internal lining	< 18 < 18
Germany	glass glass	wool wool	14 18		××	. 0 . 0	2 2	perfect, with air-tight internal lining	< 7% < 3%
USA	glass	wool	appr. 8		×	л.	5 K	raft paper on the inside, with usual workmanship	•

element. As shown in Figure 6, increased heat losses occur only very locally, close to the air entry and exit zones. If the air flow pattern remains close to one-dimensional in the cavity (or in other words if the velocity component perpendicular to the air flow remains small, which is a fair assumption even in the presence of turbulent flows considering the small dimensions of the cavity), the only disturbed regions will be the top and the bottom of the



Figure 10 -- Measured heat flux as a function of external wind speed: comparison of a 23 kg/m<sup>3</sup> rock wool product with a 13 kg/m<sup>3</sup> glass wool product (from ref. [8]).



Figure 11 -- Measured thermal resistance as a function of the air velocity above the insulation surface: influence of an air-tight wind protection on a 18 kg/m<sup>3</sup> glass wool product (from ref. [10]).

building element (wall or roof). Related to the total height of the structure, these regions will have a negligible influence. On the contrary, in a tiled roof such as the one used in the German study, the extreme air openness of the tiles allows multiple air entries, and consequently multiplies the number of regions affected by air infiltration. The same reasoning is true for the British hot box roof measurement: there, the roof element was facing a large air volume. It is therefore quite legitimate to believe that despite the almost constant measured parallel component of the velocity, there may have been some convective loops formed above the insulation layer, hence here again creating multiple air penetration zones and much more noticeable heat losses.

#### CONCLUSIONS

This paper was aimed at collecting as much information as possible from various ongoing studies regarding the effect of forced convection on the thermal properties of fibrous insulants. These studies are still being pursued, and further results shall be obtained in the coming year, concerning in particular cathedral ceilings.

The use of a numerical model allowed us to understand the flow patterns in a system where the permeable surface of a porous insulant is in contact with a forced tangential air flow. A simplified model was then derived, which showed a fairly good agreement with our hot box experiments, and confirmed that air penetration into the insulating material is not sensitive for commonly encountered flow velocities of about 0.5 m/s and less. Higher flow velocities will obviously be found in the ventilated cavities, when there is a very strong outside wind. But these situations do not occur very frequently, and their influence on the thermal resistance of the building element should be integrated over the year to reflect the actual loss of energy.

From the exposed experimental and numerical results, we can summarize important features of ventilated pitched roofs or external walls:

1) velocities in the cavities remain very low, and have little dependence on the external wind speed. On average, one can consider that velocities remain around 0.5 m/s, with peaks around 1.5 m/s for high wind speeds.

2) for these velocities, and if the outer wall or roof covering is fairly air-tight, even extremely air permeable products will not show any marked decrease of their thermal properties if the <u>inside air-tightness</u> (continuity of an external air and/or vapor barrier) was carefully ensured, and the workmanship good.

3) on the other hand, if we are in a situation of poor workmanship, and/or the outer wall or roof covering is not air-tight, multiple air entry zones exist that give rise to a marked increase of the heat losses.

In consequence, and to ensure the best possible thermal performances of the building, we propose the following construction tips:

1) in all cases, inside air-tightness is MANDATORY.

2) <u>ventilated façades</u>: outer skins of external walls are usually fairly air-tight, and wall insulating materials of higher density than roof's. Hence, provided that the façade is not notably high (which would give rise to an unusually high air velocity in the cavity), and that good workmanship can be ensured, air infiltration will be negligible for common conditions of wind and wall heights. As a result, is not necessary to place an additional wind barrier on top of the insulation.

3) ventilated pitched roofs: no problems of air infiltration will appear in ventilated pitched roofs if the roof covering is air-tight. This is the case in particular for the majority of American roofs, where a wood product panel sheathing is used to support the shingles. It is also true of slate roofs, which are quite impermeable to air penetration. If we are dealing with tiled roofs, it is recommended to place an air-tight lining directly above the insulation. But care should be taken that this sheet be extremely vapor open in order to allow the evacuation of the moisture diffusing and/or convecting from the inside of the building.

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**Tests and Models** 

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EXPERIMENTAL TEST RESULTS OF INTERIOR VS. EXTERIOR INSULATION IN EXTREMELY HOT CLIMATES

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ABSTRACT: Three room-sized building modules were constructed in Elhasa, Saudi Arabia during the summer of 1988. These test modules were identical except for the manner in which they were insulated. One had no insulation, another had insulation on the exterior surface of the exterior walls, and a third had insulation on the inside surface of the exterior walls. Hypotheses were developed and tested based upon an equation expressing the anticipated relationship of lifecycle costs among these three test modules. The equation is: Lifecycle cost (E) < Lifecycle cost (I) < Lifecycle cost (U); where (E) is exterior insulated condition, (I) is interior insulated condition, and (U) is uninsulated condition.

To estimate present worth and equipment size, MRT field measurements were used to fine tune an hour-by-hour energy simulation program called ENERCALC. The thermal characteristics of the buildings were entered along with the annual weather data for the site of the experiment. The buildings' annual heat gains and losses, peak heat gains and losses, summer hourly air-conditioning demand profile, and the present worth of each building were then compared. The comparison indicated that the life-cycle cost of the externally insulated building was the lowest among the three buildings.

KEYWORDS: external insulation, hot climate, mean radiant temperature, peak loads, life-cycle cost.

Ibraheem A. Al-Mofeez is an assistant professor at King Faisal University, Dammam 31451 Saudi Arabia. Paul K. Woods is an assistant professor of Construction Science at Texas A&M University, College Station, Texas, USA, 77843-3137 Since the 1973 energy crisis, the notion of covering external surfaces of a building envelope with thermal insulation has gained the attention of scientists and engineers. Combining a large thermal mass with external insulation improves the thermal performance of buildings in some climates. Mass materials have the ability to store energy, and insulation has the ability to slow its transfer. A building envelope with mass inside and insulation outside reduces the amplitude of the inside-air-temperature oscillation during the diurnal temperature cycle. This reduces the maximum temperature and the peak energy load for cooling or heating. It also reduces energy consumption, reduces the size of cooling or heating equipment, creates uniform room air and surface temperatures, and brings the mean radiant temperature close to the room air temperature (1).

External insulation acts to isolate the internal thermal mass from outside temperature variations. This mass is affected more by internal heat sources than by outside heat sources. External insulation allows for a more uniform temperature inside buildings. An experiment in Victoria, Australia showed that applying external insulation resulted in small daily oscillations of the interior wall surface temperature with minimum variations from dry bulb internal air temperature (2). The result is lower energy consumption for cooling and heating. The mass has a slow response time relative to ambient temperature variations; therefore the inside dry bulb temperature dominates the thermal condition of the mass. In the case of winter, the concept also applies because the mass is protected from outside weather variations, assuming the cooling and heating equipment are utilized in both cases. Applying external insulation greatly reduces the possibility of a low internal surface temperature (2). The mass thermal storage becomes a thermal sink for winter heat loss and summer heat gain. Consequently, the internal temperature is raised in winter and lowered in summer.

Rigid insulation boards have high thermal resistivity and low thermal capacity, but thermal mass has low thermal resistivity and high thermal capacity. External insulation, when combined with mass in the building envelope, reduces heat transmission from the outside and stores the internal heat generated from heating and cooling systems. The thermal lag of the building envelope is increased, allowing the diffusion of heat from outside surfaces of the mass to the outside environment before heat is transmitted to the inside surface and internal space. The result is a lower rate of heat transfer in summer and lower cooling loads. At the U.S. National Bureau of Standards (NBS), an experiment indicated that a room with external insulation consumed the lowest energy for cooling among several buildings: 1) with internal insulation, 2) without insulation, and 3) wood frame with insulation (3). In the NBS experiments, thermal mass is not utilized in the roof. In hot-arid climates the heavy roof mass has a significant effect upon the thermal behavior of a room because of higher sun angles in equatorial regions.

Applying thermal insulation materials to the outside of the building envelope increases the thermal lag and energy diffusion of the mass, thus reducing the energy transferred into the system during peak hours of cooling demand. For example, a building envelope consisting of 20 cm thickness of concrete blocks and 20 cm thickness of reinforced concrete slab insulated externally with 5 cm thickness of insulation has a time lag of 13 hours (4). In hot-arid climates where there is a large daily temperature swing, the temperature oscillation inside a building is reduced and heat is diffused before getting into the living space. Thermal diffusion reduces the internal temperature oscillation and reduces the operating time and the required size of cooling equipment. In the summer, when the ambient temperature at night is cooler than or in the comfort zone, ventilation could be used to cool the mass. During the day, mass could be used as a heat sink, thus saving energy used for cooling and reducing the size of cooling or heating equipment (4). For example, with external insulation, when ambient maximum temperature reaches 33.9 C the internal air temperature reaches 26.1 C; without insulation the internal air temperature could reach 36.7 C, a difference of 10.6 C. This difference could result in extra operating time for air conditioning equipment (5).

The thermal lag of the building envelope is dependent upon the thickness of mass, insulation materials, and the placement of these materials. For instance, the time lag of a wall consisting of 20 cm of concrete block will be 5 hours. Applying 5 cm of insulation internally will increase the time lag to 8 hours. External application will increase the time lag to 13 hours. Time lag can be determined for any building envelope configuration. Buildings may be designed to have different time lags for each wall and space orientation to suit household activities (4). The desired time lag for each room and wall can shift the cooling load to times of off-peak demand. This concept could be used to mitigate the financial consequences of peak-demand surcharges for electrical consumption.

It has been recognized that applying insulation materials to the exterior surface of buildings greatly increases interior thermal comfort by shielding the building skin from weather variations (4). It also greatly reduces the energy consumption for heating and cooling. In hot-arid climates, the daily temperature oscillation is very high; therefore, the building envelopes are exposed to large thermal expansions and contractions which may cause cracks at weaker joints. Applying insulation to the outside surfaces of the building skin will reduce cracks in hot-arid climates. It will also reduce condensation in hot-humid climates. Applying insulation to the external walls and roof could also enhance building appearance and increase flexibility for future renovations.

#### EXPERIMENTAL FACILTITES

Three test modules (small, one-room buildings) have been constructed in a hot-arid climate, Elhasa, Saudi Arabia. Before insulation, these modules are identical in size, color, shape, orientation, and surface areas (see Fig. 1). Each uninsulated building measures 3.24 m long by 3.04 m wide by 3 m high (see table 1). The materials consist of a 15 cm reinforced concrete slab and concrete blocks which measure 20 cm thick. These particular materials are selected because they are widely used in hot-arid climates and have high thermal capacitance.

The buildings are of the same size to eliminate the need for thermal scaling. In addition to the basic materials, 5 cm thick rigid polyurethane insulation boards were installed on the external surfaces of one module and on the internal surfaces of another. The third building was left uninsulated as a control. A door of 2.02 m by 80 cm and a window of 70 cm by 60 cm were installed in the north wall of each module. For the externally insulated building, the insulation started 18 cm below the floor level which is.0.4 m above the ground level. For the internally insulated room, the insulation started at the floor level. All of the thermal insulation boards used in this experiment

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have the same thermal resistance value as determined by the manufacturer. Tape was applied at the door and window perimeters to reduce the effect of infiltration and exfiltration.Thermocouple sensors where place at the center of each wall and roof on the inside and outside surfaces.

Building	Outer surface sq m	Inner surface sq m	Volume cu.m	wall	Average (C . sq roof	R-value m / W) window	* door
Uninsulated Internal	47.5	35.9	19.7	0.31	0.33	0.19	1.80
insulation External	47.5	35.2	18.2	2.38	2.40	0.19	1.80
insulation	50.0	35.9	19.7	2.38	2.40	0.19	1.80

TABLE 1 -- Building Physical and thermal characteristics.

\* R-values include air films.



Measurement Locations

- 1. Ambient Temperature
- 2. Mean Radiant Temperature
- 3. Roof Outside Surface Temperature
- 4. Wall Outside Surface Temperature
- 5. Wall Inside Surface Temperature
- 6. Roof Inside Surface Temperature

FIG. 1 -- The basic test module showing the location of sensors.

Each building has a 3-speed ventilation fan in the south wall 15 cm below the ceiling. Each fan has three speeds. The rated air displacement volumes for the three speeds are: 1.1 cubic meters/minute, 1.4 cubic meters/minute and 1.9 cubic meters/minute. For this experiment the fan apertures in each building were completely sealed with tape and the rate of air change was assumed to be 0.0 cubic meters/minute.

#### HYPOTHESES

 The building with externally applied insulation will experience the lowest life-cycle cost of the three buildings.
The building with internally applied insulation will experience a life-cycle cost greater than the externally insulated building and less than the uninsulated building.
The uninsulated building will experience the highest life-cycle cost of the three buildings.

These three hypothesis may be stated symbolically in an equation: Life-cycle cost(E) < Life-cycle cost(I) < Life-cycle cost(U). "Lifecycle cost" is defined as the net present value of first and operating cost over the expected life-span of the building. The letters "E", "I", and "U" represent the three experimental building types: insulated on the exterior, insulated on the interior, and uninsulated.

#### DATA COLLECTION

Data for this experiment were collected three times per hour from one outside temperature sensor and one mean radiant temperature sensor for each of the three modules. The eighty-six-hour experiment consisted of 1032 observations. The experiment was conducted on a clear sky. The effect of solar radiation was refelected on the MRT readings. It should be noted that windows were not protected from direct solar radiation and they were exposed to it 4.5 hours each day.

#### ANALYSIS

Hourly averages were computed for each sensor and plotted as a function of time (see figures 2, 3, and 4). These graphs make it easier to visually compare the temperature regimes among the three buildings. The plots may also be used to graphically determine the time lag, decrement factor, daily temperature range, time of maximum, and time of minimum temperature.

The overall average mean radiant temperature was computed for each building. The differences among these averages were tested for significance using Scheffe's S method for multiple comparisons at a probability of 0.05. These values were required as input to the thermal analysis program, ENERCALC.

ENERCALC (7) was utilized to extrapolate the results from the four-day experiment to include cooling and heating energy as well as

the effect on sizing HVAC equipment for these one-room buildings. ENERCALC has a floating temperature routine and calculates a MRT value based on weighted surface temperatures.



FIG. 2 -- Temperature profile of the uninsulated building: sealed.



TIME IN HOURS

FIG. 3 -- Temperature profile of the internally insulated building; sealed.



FIG. 4 -- Temperature profile of the externally insulated building: sealed

According to Al-Mofeez (8), measured black-globe temperatures are a reliable estimate in range and average values for MRTs in unairconditioned space. Observed MRT values were used as a bench mark for matching corresponding values generated by ENERCALC. The process included changing the input file for building thermal characteristics such as U-values of walls and roofs, solar absorptivity and decrement factor until ENERCALC-calculated, weighted surface temperatures equaled observed MRT temperature measurements (see table 2). The 1989 weather data for the site was used in ENERCALC. The data were analyzed to test the effect of insulation on total cooling loads, 24-hour cooling load distribution, inside temperature oscillation, and inside surface temperature variation. The analysis required some calculation of raw data such as the 24-hour cooling load and the monthly cooling load.

TABLE	2	~-	Building	Physical	and	thermal	characteristics.
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Building	Internal mass kg/m <sup>2</sup>	Average decrement factor	Average solar absorption	Average (C . sq roof	R-value m / W) wall
Uninsulated Internal	171	0.73	0.38	0.33	0.31
insulation External	122	0.62	0.38	2.40	2.38
insulation	220	0.20 _	0.38	2.40	2.38

A. Initial value	S
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Building	Internal mass kg/m <sup>2</sup>	Average decrement factor	Average solar absorption	Average (C . sq roof	R-value m / W) wall
Uninsulated Internal	293	0.72	0.38	0.26	0.3
insulation External	146	0.56	0.58	2.20	2.5
<u>insulation</u>	439	0.20	0.38	2.50	2.5

#### B. Adjusted values

Time lag and decrement factor were measured by using the time differences for maximum temperature occurrences of ambient temperature and inside surface temperatures, ambient temperature and MRTs, or outside surface temperatures and inside surface temperatures [Degelman, 1981].

#### RESULTS

During the course of the experiment the ambient temperature range was 15 C with maximum of 44.4 C and minimum of 29.4. The maximum temperature occurred at 1:00 p.m. and the minimum occurred at 4:00 a.m. The magnitude and time of occurrence are different for each building.

The uninsulated building exhibited the most extreme temperature range and time lag. This building was very hot both day and night.

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Its maximum and minimum MRTs were approximately 6 C higher than the maximum and minimum outside air temperatures. These results demonstrate a requirement for more cooling due to the higher average cooling load on the building envelope and larger peak cooling load than either of the insulated buildings. The time lag of this module was 6 hours as indicated by the time difference in the maxima of ambient and mean radiant temperature.

The maximum and minimum MRTs of the internally insulated building were respectively 7 C lower than the maximum and 4 C greater than the minimum ambient temperatures. The extremes were more moderate. The daily minimum, was 33.3 C and the maximum was 39.4 C, a range of 6.1 C. The time lag between the two peaks was 1 hour. The anticipated time lag was greater than 6 hours, as predicted by the Mackey and Wright (6) method. The difference is attributed to solar heat penetration during early morning through the north-facing window.

The externally insulated building exhibited the lowest fluctuation in MRT. The daily range was only 1.2 C. The maximum was 35.6, and the minimum was 34.4. This indicates an exceptionally stable inside air temperature. The daily range could have been even lower if the window had been shaded during early morning from sunrise to 9:00 a.m. and from 4:00 p.m. until sunset. This is indicated from the graph of the temperature profile where the occurrence of the high temperature is observed at the mentioned times.

The statistical analysis found significant differences among some of the overall building temperature means at a probability of 0.05 (see table 3). The uninsulated building experienced the highest average MRT, about 41.1 C. Both insulated buildings experienced an average MRT of about 36.1 C. As can be seen from Table 3, the differences in average MRT between each insulated building and the uninsulated building were statistically significant. Therefore, hypothesis three is accepted. Hypotheses one and two are rejected since the 0.05 C average temperature difference between them is not significant. The measured mean radient temperature values were used in hypotheses testing and not calculated cost values.

Building	Scheffe Grouping	Mean C Degrees	N
Uninsulated	A*	41.16	258
Insulated Interior	В	36.05	258
Insulated Exterior	_B	36.00	258

TABLE 3. -- Scheffe's S Comparison of mean radiant temperatures by Building

\* Buildings with the different letters are significantly different.

### Summer Hourly Demand Profile

The analysis showed similar airconditioning demand profiles for the insulated buildings. The externally-insulated building indicated an almost constant demand during the 24-hour period of 0.3 kW while the internally-insulated building showed a higher demand from 1:00 p.m. to 8:00 p.m. of 0.4 kW. The uninsulated building exhibited more demand fluctuation during the 24-hour period. Its minimum demand occurred at 6:00 a.m. and was 0.6 kW, and its maximum demand occurred at 4:00 p.m. and was 1.4 kW (Fig. 5).



FIG. 5 -- Summer average hourly air-conditioning demand profile.

### Peak Heat Gains and Losses

The peak heat gains and losses were different. The externallyinsulated building gained the least among the three buildings. The uninsulated building gained and lost the most. Peak values for the externally-insulated building were a gain of 1105 W and a loss of 601 W. The internally-insulated building gained 1175 W and lost 618 W for respective summer and winter maxima (Fig. 6). Summer and winter peak demands reflect required HVAC system sizes.

#### Estimated Annual Heat Gains and Losses

The annual heat gains for the buildings were different. These results were expected and in line with MRTs results. The externally insulated building gained the least annual energy, 3299 kW. The uninsulated building gained the highest annual energy, 11949 kW. The internally insulated building gained 3595 kW in the same period (Fig. 7). Differences between the two insulated buildings can only be attributed to the position of the insulation.

### Average Seasonal Hourly Air-Conditioning Demand

The three buildings exibited different hourly air-conditioning demand. As expected the uninsulated building was the most sensitive among the three. It required air-conditiong in all seasons with less magnitude in winter and fall. The externally insualted building required the least air-condinioning demand of the three buildings. It demanded 0.3 kWh on average while the internally insulated building required 0.33 kWh (Fig. 8).



HEAT GAINS AND LOSSES

FIG. 6 -- Peak heat gains and losses which affects the HVAC's designed size.



FIG. 7 -- Annual heat gains and losses in kw for three builings.



SEASONAL COOLING

FIG. 8 -- Average seasonal hourly demand for air-conditioning in kw.

#### Present Worth Analysis

The assumed building serviceability was 30 years. The first cost and future cost of maintaining and operating the buildings were included in the analyses (see table 4). Based in these assumptions, the externally-insulated building cost the least, US\$ 9677.00. The internaly-insulated building present worth was US\$ 9928.00, about US\$ 300.00 higher than that of the externally-insulated building (Fig. 9). The uninsulated building present worth was the highest among the three buildings, US\$ 17240.00, although its first cost was the least of all three buildings (see table 5). There are some factors which were not considered that may result in financial benefits when using external insulation. One of these is the reduction of thermal contraction and expansion that may reduce cracks in walls and roofs.

TABLE	4		Financial	assumptions.
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			Bldg & maint	Elect	ric	Water	-
Project life (yrs)	HVAC life (yrs)	Discount rate	cost escalation	\$ per e kwh	escalation	\$ per 1K Gal	escala- tion
30	15	0.08	0.10	0.065	0.08	1.0	0.05

## CONCLUSIONS

The sealed building experiment suggests that when a building is insulated from the outside under these experimental conditions, fluctuation in the mean radiant temperature will be minimized, and thermal comfort may improve. Engineering estimates show that smaller cooling equipment and lower life-cycle costs will result from the application of external insulation on massive buildings in very hot climates.



PRESENT WORTH

FIG. 9 -- Net present worth of constructing and operating the buildings.

Cost in US \$	Uninsulated	Int. Insul.	Ext. Insul
First cost	5378.00	5927.00	5913.00
Present worth cost main.& operation	of 11763.00	4001.00	3764.00
Total present worth	17140.00	9928.00	9677.00 _

TABLE 5 -- Present Worth of the three Buildings.

For the building with insulation on the inside, the diurnal range of MRT is moderated. While this range is less than either that of an uninsulated building or the ambient temperature, it is greater than that of the externally insulated building. ENERCALC-generated estimates show that the size of cooling equipment would be greater for the internally insulated building than the externally insulated building, initial equipment costs would be greater, and life-cycle costs would also be greater. The potential operational cost savings due to smaller equipment size would be less for an internally insulated building than an externally insulated one. Although estimates for the uninsulated building show that its initial cost is less than either of the insulated buildings, it is the most expensive when viewed over its life span.

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# R. Gerry Miller and William P. Goss<sup>1</sup>

HOT BOX INSTRUMENTATION, CALIBRATION AND ERROR ESTIMATION - A SURVEY

REFERENCE: Miller, R. G. and Goss, W. P. "Hot Box Instrumentation, Calibration and Error Estimation - A Survey", <u>Insulation Materials</u>: <u>Testing and Applications. 2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A survey questionnaire dealing with instrumentation, calibration and error estimation of guarded and calibrated hot boxes was mailed to United States and Canada hot box laboratories capable of measuring the R-value of building assemblies. Questions pertaining to types of instrumentation, estimated accuracy of the specific instruments and error analysis calculations were asked. Responses from those laboratories responding to the survey are summarized in the paper. This information will be beneficial to future revisions of the ASTM guarded and calibrated hot box test methods.

KEYWORDS: Guarded hot box, calibrated hot box, calibration, error estimation, instrumentation.

# **INTRODUCTION**

In 1987 one of the authors published a paper on the results of a survey dealing with hot box operating techniques and procedures [1]. Some of the information from that survey was used in the latest revisions of the ASTM C236 Guarded Hot Box Test Method [2] and the ASTM C976 Calibrated Hot Box Test Method [3]. This paper summarizes a second survey mailed to 39 possible hot box operators in the United States and Canada. The survey questionnaire, a copy of which is reproduced as the Appendix to this paper, consisted of 30 questions dealing with testing capabilities, instrumentation, calibration specimens, measurement error estimation and error analysis calculations.

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Of the 39 survey questionnaires mailed out to thermal testing laboratories who were identified as possible laboratory hot box operators, 4 laboratories were identified as solar calorimeters. These were not included in the present results, along with 1 laboratory that had a guarded hot plate rather than a hot box. In addition, 6 laboratories had hot boxes which had been mothballed (more will be said about this situation in the conclusions section of this paper) and did not participate in the survey because the personnel who could provide the survey information were no longer employed by the laboratory. Also, 2 laboratories who participated in the survey recently had their main hot boxes mothballed and did not know when these boxes might, if ever, be made operational again. Therefore, the original 39 potential hot box laboratories were reduced to 28 of which 24 laboratories responded to the survey (an 86% return rate!). Table 1 is an alphabetical listing of 23 of the 24 laboratories participating in the survey. (Note: one laboratory provided information over the telephone, but preferred to remain anonymous as per company policy.) The list of participants includes insulation manufacturers, insulation users, window manufacturers, commercial testing laboratories, government research laboratories and academic research laboratories.

#### **GENERAL INFORMATION**

Questions 1 to 5 on the survey asked for general information about each box, such as box type, sample size, maximum sample thickness, test procedure(s) followed, and National Voluntary Laboratory Accreditation Program (NVLAP) participation. Table 2 summarizes this information. Of the 24 respondents, 17 had guarded hot boxes (13 used the ASTM C236 test procedure [2], 3 the AAMA 1503.1 test procedure [4], and 1 the Institute for Research in Construction/National Research Council of Canada test procedure), 6 had calibrated hot boxes using the ASTM C976 [3] procedure and 2 (one using the appropriate ASTM test procedure and the other using the AAMA test procedure) could be run in either the guarded or calibrated mode. This totals 25 boxes, as laboratory 9 reported 2 boxes, a guarded and a calibrated hot box. The test specimen metered areas ranged from  $0.81 \text{ m}^2$  (9 ft<sup>2</sup>) to 13.8 m<sup>2</sup> (150 ft<sup>2</sup>) with the larger sample areas being the calibrated hot boxes. All but 2 laboratories reported they had the capability of testing a sample up to 15.2 cm (6 in.) thick, the majority being able to test samples greater than 30.5 cm (12 in.). Of the 19 boxes capable of being run in the guarded mode, 8 are accredited through NVLAP. This breaks down to 7 of the 12 guarded hot boxes in the United States using the ASTM C236 test procedure [2], 1 of the 4 guarded hot boxes using the AAMA 1503.1 test procedure [4], and none of the 3 guarded hot boxes in Canada where NVLAP accreditation is not available. All the 7 hot box laboratories in the United States which use the ASTM C976 test procedure [3] would like to see ASTM C976 added to the list of tests that can be accredited by NVLAP.

	4	Th	•			1	
TABLE		- Participants	ın	the	hot	h∩x	questionnaire
	-	T WI HOID WILLO			1100	004	queomonium

Architectural Testing, Inc. York, PA Attn: D. J. Wise

Butler Mfg. Co. Grandview, MO Attn: W. Sanders

Certainteed Corporation Blue Bell, PA Attn: D. J. McCaa

Dynatherm Engineering Lino Lakes, MN Attn: J. B. Funkhouser

Geoscience, Ltd. Solano Beach, CA Attn: H. F. Poppendiek

Holometrix, Inc. Cambridge, MA Attn: A. Desjarlais

Institute for Research in Construction National Research Council of Canada Ottawa, Ontario, Canada Attn: W. C. Brown

Institute for Research in Construction National Research Council of Canada Ottawa, Ontario, Canada Attn: H. Elmahdy

Jim Walter Research Corp. St. Petersburg, FL Attn: R. G. Miller

Kawneer Co. Norcross, GA Attn: J. S. Vermeersch and C. Touton

National Certified Testing Laboratories York, PA Attn: J. Hill

Manville Corp. Denver, CO Attn: D. G. Ober National Institute of Standards and Technology Gaithersburg, MD Attn: D. M. Burch

Oak Ridge National Laboratory Oak Ridge, TN Attn: K. E. Wilkes

Ortech International Mississauga, Ontario, Canada Attn: G. Hildebrand

Owens Corning Fiberglas Granville, OH Attn: J. R. Mumaw

PPG Industries Pittsburgh, PA Attn: A. E. Thompson, Jr.

Sparrell Engineering Research Corp. Damariscotta, ME Attn: J. K. Sparrell

Twin City Testing Corp. St. Paul, MN Attn: J. McFee

University of Illinois Department of Agricultural Engineering Urbana, IL Attn: L. Christianson

University of Massachusetts Department of Mechanical Engineering Amherst, MA Attn: W. P. Goss

U. S. Army CRREL Hanover, NH Attn: A. R. Greatorex

Warnock Hersey International, Inc. Middletown, WI Attn: R. Curkeet

	C976 Included	in NVLAP?		Yes	NAd	NA <sup>d</sup>	NA <sup>d</sup>	NAd	Yes	NA <sup>d</sup>	Yes	NAd	Yes	NAd	Yes	NAd	Yes	Yes	NR <sup>t</sup>	NAd	NAd	NAd	NA <sup>d</sup>	NA <sup>d</sup>	Yes	NAd	NAd	NA <sup>d</sup>					cedure.
	NVLAP	Participant		NAd	Yes	Yes	Yes	No	Yes	No	No	No	NAd	Yes	۵NAd	No	NAd	NAd	No	No	Yes	No	No	No	PA₫	Yes	No	Yes			cring area.		AAMA ICSI PIU
	m Sample	kness	. <u></u>	24	80	t Limited	12	12	24	t Limited	12	eral Feet	eral Feet	10	9	10	90	24	4%	9	12	14	7	S	24	9	12+	œ			refers to met		rateo dox anu
	Maximu	Thic	5	61.0	20.3	No	30.5	30.5	61.0	No	30.5	Sev	Sev	25.4	15.2	25.4	76.2	63.5	10.8	15.2	30.5	35.6	17.8	12.7	61.0	15.2	30.5 +	20.3			set of numbers	Not applicable	: guarded/canu
E 2 General Information	ple Size	h x height)	ft. x ft.	15 × 10	9 x 9/4 x 4 <sup>c</sup>	12 x 12/8 x 8 <sup>c</sup>	8 x 8/4 x 4 <sup>c</sup>	6.5 x 7.2/6.5 x 7.2 <sup>c</sup>	8 x 8/6 x 6 <sup>c</sup>	12 x 12/8 x 8 <sup>c</sup>	10 x 10/8 x 8 <sup>c</sup>	6 x 7.2/4 x S <sup>c</sup>	6.8 x 7	5.6 x 5.6/3 x 3 <sup>c</sup>	4 x 8	8 x 8/8 x 8 <sup>c</sup>	8 x 8	10 x 8	9 x 8.8/6.3 x 8 <sup>c</sup>	6 x 6/4 x 4 <sup>c</sup>	5.5 x 5/2.7 x 3 <sup>c</sup>	8 x 8/5 x 5 <sup>c</sup>	4 x 6/4 x 6 <sup>c</sup>	10 x 10/8 x 8 & 5 x 8 <sup>c</sup>	12 x 8	8 x 8/4 x 4 & 2.7 x 2.7 <sup>c</sup>	$10 \times 10/8 \times 8^{c}$	10.4 x 10.4	4.2 x 10.4	4 x 8	units. <sup>c</sup> Second	MA test procedure $d NA = 1$	$f$ /NKUU test procedure $\sqrt{1}$ UR = N
TABLI	Sair	(lengt	mxm	4.6 x 3.0	2.7 x 2.7/1.2 x 1.2 <sup>c</sup>	3.7 x 3.7/2.4 x 2.4 <sup>c</sup>	2.4 x 2.4/1.2 x 1.2 <sup>c</sup>	2.0 x 2.2/2.0 x 2.2 <sup>c</sup>	2.4 x 2.4/1.8 x 1.8 <sup>c</sup>	3.7 x 3.7/2.4 x 2.4 <sup>c</sup>	3.0 x 3.0/2.4 x 2.4 <sup>c</sup>	1.8 x 2.2/1.2 x 1.5 <sup>c</sup>	2.1 x 2.1	1.7 x 1.7/0.9 x 0.9 <sup>c</sup>	1.2 x 2.4	2.4 x 2.4/2.4 x 2.4 <sup>c</sup>	2.4 x 2.4	3.0 x 2.4	2.7 x 2.7/1.9 x 2.4 <sup>c</sup>	1.8 x 1.8/1.2 x 1.2 <sup>c</sup>	1.7 x 1.5/0.8 x 0.9 <sup>c</sup>	2.4 x 2.4/1.5 x 1.5 <sup>c</sup>	1.2 x 1.8/1.2 x 1.8 <sup>c</sup>	3.0 x 3.0/2.4 x 2.4&1.5 x 2.4°	3.7 x 2.4	2.4 x 2.4/1.2 x 1.2 <b>&amp;</b> 0.8 x 0.8 <sup>c</sup>	3.0 x 3.0/2.4 x 2.4 <sup>c</sup>	3.2 x 3.2	1.3 x 3.2	1.2 x 2.4	ersions were made to obtain SI	G/A: guarded box and AA	U/I: guarded DOX and LIKL sing ASTM C736 or C976
	Test	Procedure		C/976 <sup>b</sup>	G/236 <sup>b</sup>	$G/236^{b}$	$G/236^{b}$	G/A <sup>b</sup>	G/C <sup>b</sup>	$G/236^{b}$	$G/I_{p}$	$G/236^{b}$	C/976b	$G/236^{b}$	C/976b	$G/236^{b}$	C/976b	C/976 <sup>b</sup>	G/C/A <sup>€</sup>	G/236 <sup>b</sup>	G/236 <sup>b</sup>	G/236 <sup>b</sup>	G/236 <sup>b</sup>	G/A <sup>b</sup> 3	C/976 <sup>b</sup>	G/236 <sup>b</sup> 2	G/236 <sup>b</sup>	G/A <sup>b</sup>			or English units, conve	ated box/ASTM C976	ed box/ASTM U230 ed or calibrated hov u
	1 -1	Laboratory		1	2	3	4	5	6	7	8	ğ	9C	10	11	12	13	14	15	16	17	18	19	50	21	22	23	24			<sup>a</sup> Survey asked f	<sup>b</sup> C/976: calibr	G/2:0: guard

#### **TEMPERATURE INSTRUMENTATION**

In the previous survey [1], questions were asked about thermocouple types, sizes, numbers and locations. In this survey, the authors were more concerned with the calibration and error estimate of the temperature instrumentation used in the hot boxes. Table 3 summarizes this information from questions 6 to 8. In response to question 6, all but one laboratory indicated thermocouples (TC) were being used in the hot boxes. Two laboratories use TC for data collection and resistance temperature detectors (RTD) for control; another also uses thermistors in the box. One respondent reported using TC, RTD and infrared (IR) thermography for their temperature measurements. One laboratory uses the generally more accurate, but more expensive RTD for all of their temperature measurements.

In retrospect, survey questions 7 and 8 on calibration and temperature measurement error estimation should have been more explicit by asking if the answers were for just the temperature sensor or for the entire temperature measurement system (eg. thermocouple/datalogger/computer). Eighteen (18) out of the 24 laboratories reported that they "calibrate" their temperature sensors. One of the laboratories that does not "calibrate" their TC reported using special error limits TC wire (as most laboratories do) on both sides of the hot box sample (or test specimen). Moreover, the term "calibrate" appeared to be interpreted in many ways. Usually the term "calibrate" implies that a known standard is used. The National Institute for Standards and Technology (NIST) provides calibration services for all commonly used thermocouples. In addition, there are a number of NIST and ASTM publications related to temperature sensors and thermocouple calibration. It is recommended that, based on the above information, future versions of ASTM C236 [2] and C976 [3] should provide hot box laboratories with more detailed information on how to calibrate their temperature measurement systems.

Of the laboratories responding to the survey, the "calibration" procedures used could be roughly grouped into two methods: temperature sensors only or the entire temperature measurement system. Many of the respondents reported checking the temperature sensors only or the entire temperature measurement system with an ice bath. It is strongly recommended that the entire temperature measurement system (each temperature sensor and the associated data acquisition equipment) be truly calibrated at least at one temperature. Ideally, the entire measurement system should be calibrated over the entire range of temperatures that the sensors will operate. In response to question 8, a wide range of temperature measurement error estimates were provided. It appears that many of the laboratories estimate their temperature measurement errors to be beyond the capability of the sensor alone. For most special limit TC sensors, the estimated uncertainty is approximately  $\pm 0.5 \,^{\circ}\text{C}$  ( $\pm 0.9 \,^{\circ}\text{F}$ ). This again implies the need to specifically spell out the temperature measurement system calibration procedures in ASTM C236 and ASTM C976.

	TABLE 3 Ten	iperature instrumen	tation <sup>a</sup>			
	Temperature		Estimate of T	emperature	Measurement	Error <sup>e</sup>
Laboratory	Instrumentation Used	<b>Calibration</b>	Indivi	dual	Diffe	rence
			ŝ	۰F	ပို	٩
1	Thermopile - Primary Individual TC - Secondary	Yes	±0.06	±0.1	±0.1	±0.2
2	TC	No	±0.3	±0.5	±0.1	±0.2
3	TC	Yes	±0.3	±0.5	±0.1	±0.2
4	TC	Yes	1.1	2	0.3	0.5
5	TC	Yes	±0.1	±0.2	±0.1	±0.2
6	TC	Yes	±0.1	±0.2	±0.2	±0.3
7	TC	No	±0.6	±1	<0.6	<1
œ	TC for data; RTD for control	Yes	±0.1	±0.2	±0.2	±0.4
6	TC	Yes	<0.2	<0.3	<0.3	< 0.6
10	TC	No	0.3	0.5	0.3	0.5
11	TC	Yes	0.1	0.2	0.1	0.2
12	TC, RTD and IR thermography	Yes		<1% of	Reading	
13	TC for data; RTD for control	Yes	0.1	0.2	0.06	0.1
14	TC, Thermistors	Yes	±0.3	±0.5	±0.01	±0.25
15	TC	Yes	±0.1	±0.2	±0.2	±0.4
16	TC	Yes	±0.2	±0.3	±0.03	±0.05
17	TC	Yes	±0.3	±0.5	±0.6	±1.0
18	TC	No	:	NR <sup>b</sup>	÷	NR <sup>b</sup>
19	TC	No	0.3	0.5	0.3	0.5
20	RTD	Yes	±0.03	±0.05	±0.03	±0.05
21	TC	No	±0.6	±1.0	±0.6	±1.0
22	TC	Yes	±0.01	±0.25	±0.01	±0.25
23	TC for data, RTD for control	Yes	<0.1	< 0.1	< 0.1	<0.2
24	TC	Yes	±0.1	±0.2	±0.2	±0.4
<sup>a</sup> Survey contained English unit <sup>b</sup> Not reported.	ts. <sup>c</sup> Not known if result	s are for sensor only or	entire temperatu	re measureme	nt system	

### **HEIGHT/WIDTH MEASUREMENTS**

Questions 9 and 10 dealt with the laboratories' measurement of the heat transfer area. For those operating guarded hot boxes, this area is considered to be from the center-to-center of the metering box contact edges. Most of the laboratories use an accurate metal tape measure to determine height and width of the heat transfer area. Two laboratories use a metal rule, see Table 4. The majority of respondents reported capability of measuring these linear dimensions to 1.6 mm (1/16 in.) or better.

As the question was not asked, it is unknown the procedure laboratories follow to measure height and width. However, it is recommended that a minimum of 3 measurements be made in each direction and the average of the three measurements be used in calculating the heat transfer area. It should be noted that in measuring these height and width dimensions, it is very important to ensure that the three measurements be made as parallel as possible since the nonparallel measurement errors could turn out to be greater than the root mean square (RMS) error of the height and width measurements.

#### POWER MEASUREMENTS

Table 5 summarizes questions 11 to 13 on instrumentation used to measure the heat flow through the test specimen. The use of alternating current (AC) or direct current (DC) power was nearly evenly divided among the 24 laboratories; 13 use AC power and 11 use DC power. For those laboratories reporting DC power, the most common method was measuring the voltage drop across the metering box heater and determining the current flow through an in-series known precision shunt resistor. The product of the voltage and the current gives the DC power to the metering box. Of those laboratories listing voltage instrumentation, digital voltmeters (HP 3852 and HP 3456A) and dataloggers (Fluke) were mentioned. For those laboratories using AC power, most use watt-hour meters with 5 laboratories listing various Ohio Semitronics models and 3 laboratories listing various Scientific Columbus models.

The results to question 14 on the use of heat flux transducers (HFT) indicated that HFT are not used extensively in hot boxes. Only 7 laboratories responded that they used HFT, and only 3 used them on a regular basis. However, the use of HFT to provide quantitative information on the various hot box heat flow rates may be worth considering. Question 15 asked the respondents to estimate the accuracy of their heat flow measurements. The results, as shown in the last column of Table 5, are quite variable, ranging from a low of 0.001 watts to  $\pm 2$  watts and from  $\pm 0.01\%$  to  $\pm 5\%$ . There did not appear to be any clear power type (AC or DC) that was more accurate than the other. Similar to the situation for temperature measurements, there is a need to specify methods of calibration of the heat flow

	Length/Width	Error	Estimate
Laboratory	Measurement Device	mm	inch
1	Tape Measure	1.6	1/16
2	4 ft. Steel Rule	±1.3	±0.05
3	Tape Measure	±1.6	±0.0625
4	Steel Rule	0.8	1/32
5	Tape Measure	±1.6	±1/16
6	Tape Measure	±1.6	±1/16
7	Tape Measure	< 3.2	<1/8
8	Tape Measure	±2	$\pm 0.08^{b}$
9	Tape Measure	3.2	1/8
10	Tape Measure	2.5	0.1
11	Tape Measure	0.8	1/32
12	Tape Measure	$\mathbf{NR}^{c}$	$\mathbf{NR}^{c}$
13	Tape Measure	0.8	1/32
14	Tape Measure	±1.6	±1/16
15	Tape Measure	1.6	1/16
16	Tape Measure	NR <sup>c</sup>	$NR^{c}$
17	Tape Measure	3.2	1/8
18	Tape Measure	0.0	01%
19	Tape Measure	±0.8	±1/32
20	Cole Parmer Laboratory Grade Rule	+0.8	+1/32
21	Tape Measure	±1	±0.04
22	Tape Measure	0.8	1/32
23	Tape Measure	< 3.2	<1/8
24	Tape Measure	0.8	1/32
Survey asked units. Survey conta	d for English units, conversio	ns were made	e to obtain S

		TABLE 5 Power measurements	
Laboratory	Type of Power	Instrumentation	Estimated Accuracy of Heat Flow Measurements
1	DC	Current: measure voltage drop across a 0.1 $\Omega$ precision shunt resistor and Ohm's law, voltage: digital voltmeter	steady-state: ±2% dynamics: ±4%
2	DC	Fluke 2240B datalogger and 0.1 $\Omega$ precision resistors	±0.8 watts
3	AC	Ohio Semitronics watt transducer (Hall Effect)	±0.5 watts @ 1000 watt input
4	DC	Voltage drop across heater measured with HP 3852 voltmeter. Voltage drop across calibrated resistor in series with heater measured with HP 3852 voltmeter.	0.001 watts
5	AC	An Ohio Semitronics PC5-103D watt meter measures meter box fans energy input. An Ohio Semitronics W10DT/10K watt-hour meter measures meter box heater.	"still calibrating box"
6	AC	Ohio Semitronics WM374 watt-hour meter WM374 (Hall effect transducers)	±2 watts
7	DC	Model 3581 Kinetic Systems A/D converter	"still learning"
8	DC	Precision resistor used as current shunt and precision voltage divider.	±0.01%, ±0.3 watts
9	AC	G.E. IB-10 watt-hour meter. Westinghouse type CA watt-hour meter.	GE: calibrated to 0.1% Westinghouse: calibrated to ±0.3%
10	DC	Current shunt and voltage divider with Biddle potentiometer	1%
11	AC	Scientific Columbus	±3 Btu/h
12	DC	Kepco JQF-100-10M power supply. Direct measurement of voltage. Current converted to voltage using a current shunt.	<1%
13	AC	Scientific Columbus DL5C5A2-6070-8 Digilogic watt and watt-hour transducer with 6152R single 7 digit register.	0.25 watt-hours
14	DC	Current is measured by measuring voltage drop across a precision resistor. Voltage measured by HP 3456A 6½ digit voltmeter with 0.1 $\mu$ V resolution.	not listed
15	AC	Rochester Instruments PCE 15 watt transducer and SC- 1356 Linear Integrator with current transducer	±3 to 4%
16	AC	Ohio Semitronics EN5-1O-2W watt-hour meter	±1%
17	AC	GE single stator I-50-A watt-hour meter	NR
18	AC	Scientific Columbus WTSC-5 Hall Effect watt-hour transducer.	±5% for transmittal and conductance, ±12% for surface coefficients
19	AC	F. W. Bell PX-2203B (50W), PX-2205B (200W) PX- 2202B (1000B) Hall effect transducers calibrated with Valhalla Z101 (6 to 6000 W) digital power analyzer. Computer calculates average power.	From 0.25 to 5W depending on transducer used.
20	DC	HP DC power supply, measure current and voltage using standard precision resistors, HP computer calculates average power.	"very accurate"
21	AC	Ohio Semitronics WH-3 Hall effect watt-hour meter	"have not used in a long time"
22	AC	Watavi watt meter that is calibrated by an Electrical Standard Laboratory.	±5%
23	DC	Current shunt and voltage divider using precision resistors.	less than 1W
24	DC	HP 6112A DC power supply, current shunt, with HP 3497A data acquisition system and HP 86 computer.	±1% of watts measured for heaters in metering room over full scale.

measurement systems in ASTM C236 and C976.

#### HOT BOX CAPABILITIES

### Temperature Controller, Air Infiltration and Humidity Instrumentation

Questions 16, 18 and 19 in the survey inquired about the types of temperature controllers used in the hot boxes, as well as air infiltration measurement and humidity control capabilities. As compared to the previous survey [1], more laboratories have added computer control in their boxes. Table 6 shows that of those laboratories reporting specific temperature controllers, Eurotherm models appear to be the most popular.

Since the last survey, additional boxes have added capabilities for performing air infiltration measurements, as well as humidity control. Table 6 summarizes the pressure differential ( $\Delta P$ ) range and flow rate in cubic feet per minute (cfm) capability of the boxes with air infiltration instrumentation. Of the 6 laboratories reporting humidity control capabilities, ranges went from 12% relative humidity (RH) to 40-90% RH and from 9 to 35°C (-15 to 95°F) dew point.

#### Velocity, Temperature Ranges and Heat Flow Direction

Table 7 summarizes the responses to question 17 concerning air velocity ranges and directions in the hot boxes. All of the boxes use some type of fan arrangement to circulate the air in the hot and cold chambers (and also in the meter box of ASTM C236 guarded hot boxes). Approximately half of the laboratories have the capacity to run the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) exterior wind conditions of 3.35 m/s (7.5 mph) for summer and 6.7 m/s (15 mph) for winter [5]. For the cold chamber wind direction, 16 laboratories (15 from the United States) reported that theirs is parallel to the test specimen, 5 laboratories (3 from the United States) reported a perpendicular wind direction and 3 laboratories (all from the United States) can conduct tests with a either a parallel or perpendicular wind direction.

Table 8 gives the responses to questions 20 and 21 on heat flow directions and temperature ranges in the hot and cold chambers. The results indicate that many of the hot chambers can also be cooled below ambient or room conditions. A typical hot chamber temperature range is from about  $16 \circ C$  ( $60 \circ F$ ) up to 49 to  $66 \circ C$  (120 to  $150 \circ F$ ). About half the cold chambers can also be heated with a typical temperature range from 16 to  $21 \circ C$  ( $60 \circ F$ ) down to  $-29 \circ C$  ( $-20 \circ F$ ). One laboratory reported a capability of reaching  $-62 \circ C$  ( $-80 \circ F$ ) with liquid nitrogen evaporation.

Also included in Table 8 is a summary of heat flow directions. Fourteen

		Humidity Capability	metering chamber: 40-90% RH (Dew point measured with chilled mirror dew point hygrometer)	No	°N N	No	Q	No	No	0-50% RH Vaisala HMI-33 and related humidity sensors	No	No	No
y instrumentation.		Air Infiltration Capability	0-0.5" H2O (Validyne AP, model DP103-18) and Magnehelic AP gauge.	No	0-1" H <sub>2</sub> O (Dwyer inclined manometer ΔP; air flow-orifice plates	No	±2.0" H <sub>2</sub> O (pressure-modus pressure transducer), Flow 0-50 cfm Hastings mass flow meter	No	$S^{*}$ H <sub>2</sub> O and 15 cfm (not presently being used).	air leakage by tracer gas method and gas chromatograph	No	No	No
s, air infiltration and humidity		<u>Mcter</u>	:	same as hot chamber	1 to 5 12 watt fans used in any combination, resistance heater is computer controlled by adjusting variable transformer	Eurotherm 984 used in early test stage to set up gradient. Direct DC during steady-state	ECS 6103-1-20 zero voltage SCR controlled by a 4-20 Ma signal from computer	L & N Electromax III null balance	Same as hot	Same as hot	Same as hot 	Eurotherm differential (null) controller	Ŧ
E 6 Temperature controller	Controllers	Cold	Microcon Model 82330	chilled water circulator, Lauda RC20	variable onifice hot gas bypass valve controlled by temperature feedback from TC, custom electronics	Eurotherm 984	Cooling is provided by an R-502 refrigeration system with a hot gas bypass valve that opens and closes intermittently and is computer controlled.	same with Thermotron refrigeration system	Same as hot	Same as hot	Same as hot YSI	Neslab temperature controlled glycol/water circulator	West RTD
TABLI		Hot	PID software function running on computerized data acquisition system. Metering chamber air temperature controlled.	Heater voltages controlled by a HP 9836 computer.	3,000 Btu/h AC runs constantly - variable electric resistance heater is adjusted by computer to control temperature	Eurotherm 984 null balance	Environmental room heated or cooled. Heat provided by an ECS 7201-01-320 SCR controlled by a 4-20 Ma signal from the computer.	L & N Electromax IV	Micristar Model 828D PID controller	Honeywell Dialatrol Model 7355 Q	guarded: homemade calibrated: YSI	adjusted & regulated power input	West RTD
		Laboratory	1	7	ς,	4	Ś	9	٢	80	6	10	11

		Humidity Capability	Yes - not reported	No	N.	Ŷ	Ŷ	No	No	Ŷ	-15 to 95°F dew point for both cold and hot box, chilled mirror system	ambient to 80% RH on hot side, 65-75% on cold side (set by dew point of coll), Humidifier humidistat and Vaisala humidity transmitter HMP 1137	No	°Z.	1 2 % a n d u p. Hydrodynamics model L15- 7012 humidity transducer	
imidity instrumentation		Air Infiltration Capability	No	No	0-50 Pa, tracer gas system	± 10 psf, Dwyer Instruments "Hook Gage"	No	No	No	0-0.22" H <sub>2</sub> O, 2 Dwyer manometers	$\pm 0.7$ " H <sub>2</sub> O, pressure transducer	0-4" H <sub>2</sub> O, Dwyer manometer	No	No	0-0.25" H <sup>2</sup> O, Hastings flowmeters and computer controlled transducers	-
rollers, air infiltration and hu		<u>Meter</u>	Eurotherm 983	÷	÷		Honeywell thermostat	NR <sup>a</sup>	Research Inc. 3 mode controller	Barber Coleman 560	lgorithms	:	laboratory temperatures and run	Honcywell with RTD temperature sensor with remote set point	7A	
ontinued Temperature cont	Controliers	Cold	Enercorp RCK model PB6 Air-M controller (hot gas bypass system)	NR <sup>a</sup>	Eurotherm 984, Eurotherm 931 SCR output	Harrel Model TCG54P	simple proportional valve to operate air conditioning unit	NR <sup>a</sup>	Research Inc. 3 mode controller	Barber Coleman 560	re control is through HP computer a	Omega Model CN2012(T)-03	the use of variacs, closely controlled	Honeywell with RTD temperature sensor with remote set point	6 computer and data acquisition 349	
TABLE 6 C		Hot	Eurotherm 984	Eurotherm 984 with SCR assembly 931	Eurotherm 984 and Lauda DC power supply	Honeywell Model R7355D	Honeywell thermostat	NR <sup>a</sup>	Research Inc. 3 mode controller	NR <sup>a</sup>	Temperatu	YSI proportional Model 72	Manually vary power to heaters by tests in early morning.	Honeywell null-voltage controller with thermopile AT sensor	HP 8	
		Laboratory	12	13	14	15	16	17	18	19	30	21	23	23	24	<sup>a</sup> Not reported.

		<u>ųd</u> m	:	0-1.1	NC <sup>c</sup> -1.4	0-1		0.5			NC <sup>6</sup> -1.1	0.3-1	;		:	:	NC	NC	NC	8	0.13	0.2-2	i	NC	NR <sup>b</sup>	NR <sup>b</sup>			
	Meter	Direction	1	parallel	parallel	parallel	mall fans with an stable speed contro	parallel		itural convection	parallel	parallel	÷		i	÷	parallel	parallel	parallel	parallel	parallel	paraliei	÷	parallel	parallel	parallel			
		<u>s/m</u>	:	0-0.5	NC <sup>6-0.6</sup>	0-0.4	4 s adjus	0.2		ä	NC <sup>2-0.5</sup>	0.1-0.4	:		÷	÷	NC	NC	NC	0-3.6	0.05	0.08-0.9	:	NC	NR <sup>b</sup>	NR <sup>b</sup>			
xes <sup>a</sup>		<u>uph</u>	0.7-6	0-1.1	NC <sup>c</sup> -15	0-15	15	0.5-15	ds at present		NC <sup>e-1</sup>	0.8	0-1	wind machine	NC <sup>c-15</sup>	0-15	5-18	2	15	80	4-15	0-20	0-15	NÇ	NR <sup>b</sup>	0-15			
abilities of hot bo	Cold	Direction	parallel	parallel	perpendic. & parallel	parallel	perpendic.	parallel	measuring wind speed	NRCC wind machine, perpendicular	parallel	parallel	parallel	<b>NRCC</b> perpendicular	perpendic. & parallel	parailei	perpendic.	parallel	parallel	parallel	parallel	perpendic. & parallel	parallel	parallel	parallel	perpendic.			
Velocities capa		s/m	0.3-2.7	0-0.5	NC <sup>e</sup> -6.7	0-6.7	6.7	0.2-6.7	est facility - not 1	-	NC <sup>6</sup> -6.7	0.4	0-0.4	laboratory 12 - 1	NC <sup>-6.7</sup>	0-6.7	2.2-8	0.9	6.7	3.6	1.8-6.7	0-8.9	0-6.7	с N	NR <sup>b</sup>	0-6.7			
TABLE 7 1		<u>uph</u>	0.7-1.2	0-1.1	air	0-1	NR <sup>b</sup>	0.5-15	Roof t		NC <sup>e</sup> -1.1	NR <sup>b</sup>	0-1	Same as	NC <sup>c</sup> -15	0-5	NC	NC	NC	8-0	NR <sup>b</sup>	NR <sup>b</sup>	NC	NC	NR <sup>b</sup>	NR <sup>b</sup>			
	Hot	Direction	parallel	parallel	ill fans to circulate	parallel	NR <sup>b</sup>	parallel		atural convection	parallel	NR <sup>b</sup>	parallel		parallei	parallel	parallel	parallel	parailel	parallel	parallei	parallei	parallel	parallel	parailei	parallel			
		<u>s/m</u>	0.3-0.5	0-0.5	4 sma	0-0.4	NR <sup>b</sup>	0.2-6.7		c	NC <sup>2-0.5</sup>	NR <sup>b</sup>	0-0.4		NC°-6.7	0-2.2	NC	NC	NC	0-3.6	NR <sup>b</sup>	NR <sup>b</sup>	NC	NC	NR <sup>b</sup>	NR <sup>b</sup>	ned English units.		ction
		Laboratory	1	2	3	4	S	6	7	8	6	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	<sup>a</sup> Survey contai	<sup>b</sup> Not reported.	' Natural conve

	F	ABLE 8 Temperature cap	abilities of heat flow	directions in hot boxes. <sup>a</sup>		
		Temperature R	anges			
		Hot	U	old		
ttory	10 to 66	<u>°F</u> \$0 to 1\$0	40 to 66	40 to 150	<u>Heat Flow Direction<sup>b</sup></u> H	
	24 to 60	75 to 140	-18 to 24	0 to 75	H, D, U	
		9-)	52 with LN2 evaporation)	(-80 with LN <sub>2</sub> evaporation)		
	16 to 49	60 to 120	-29 to 21	-20 to 70	Н	
	16 to 57	60 to 135	-29 to 38	-20 to 100	H, D, U	
	Room to 49	Room to 120	-18 to Room	0 to Room	Н	
	Room to 66	Room to 150	-29 to Room	-20 to Room	H, D, U	
	7 to 66	45 to 150	-40 to 66	-40 to 150	D, U	
			[IR heating sys roof surface tempera	tem can increase ture to 93°C (200°F)]		
	18 to 29	65 to 85	-40 to -1	-40 to 30	Н	
(7)	-1 to 71 (G <sup>o</sup> )	30 to 160 (G <sup>o</sup> )	-18 to 43 (G <sup>o</sup> )	0 to 110 (G <sup>*</sup> )	Н	
()	16 to 27 (C <sup>*</sup> )	60 to 80 (C)	-32 & up (C)	-25 & up (C)		
	27 to 49	80 to 120	4 to 16	40 to 60	H, D, U, T	
1	16 to 66	60 to 150	-29 to 24	-20 to 75	Н	
0	0 to 30	32 to 86	-40 to 30	-40 to 86	Н	
-	4 to 66	40 to 150	-34 to 66	-30 to 150	H, D, U, T	
	18 to 66	65 to 150	-34 to 66	-30 to 150	H, D, U	
	18 to 38	65 to 100	~18 to 10	0 to 50	Н	
	16 to 49	60 to 120	-9 to 21	15 to 70	Н	
	10 to 38	50 to 100	-18 to 10	0 to 50	H, D, U	
	13 to 52	55 to 125	-29 to 10	-20 to 50	н, р, U	
-	17 to 29	63 to 85	-32 to -18	-25 to 0	Н	
_	-40 to 49	-40 to 120	-40 to 49	-40 to 120	Н	
	Room to 49	Room to 120	-34 to 10	-30 to 50	Н	
	21 to 149	70 to 300	-10 to 0	0 to 32	Н, D, U	
	16 to 27	60 to 80	-46 to 10	-50 to 50	Н	
	18 to 43	65 to 110	-37 to 10	~35 to 50	Н	
rvey contained I Horizontal	English units. D: Down	U: Up T: Can tilt to any ang	<u>ə</u>			
Guarded	C: Calibrated					_

boxes are operated in the horizontal heat flow direction only, 7 laboratories reported that their hot box could be operated in both horizontal and vertical (up and down) heat flow directions and 2 boxes are able to operate at any angle (horizontal, vertical up and down, and tilted). The remaining hot box, a roof test facility, can only be used in a vertical mode.

## "CALIBRATION" AND TEST SPECIMENS

Tables 9 and 10 summarize information from questions 22 and 27 on "calibration" specimens and the range of specimen R-values tested. The results from question 28 on the type of products/systems tested are not included in the table since the whole range of building envelope components (opaque building assemblies like walls, roofs, ceilings and floors, and fenestration products like windows, doors and curtain walls) were listed by many laboratories. One of the concerns noted in the previous survey [1] was that the range of tested R-values was usually much greater than the R-value range of the "calibration" specimens. It appears from Table 10, that this R-value range discrepancy has improved. Many of the laboratories reported more than one "calibration" specimen and these now tend to cover the R-value range of specimens tested. One of the more commonly reported "calibration" specimens was the nominal 10 cm (4 in.) homogeneous expanded polystyrene boards used in the hot box interlaboratory comparison program conducted by ASTM C16.30 [6]. Additional interlaboratory comparisons at other R-values in addition to nonhomogeneous systems are needed.

In addition, those laboratories accredited through NVLAP should retain their proficiency materials which can serve as additional "calibration" specimens. It is recommended that those laboratories that did not participate in the ASTM C16.30 interlaboratory comparison program, contact the ASTM C16.30 Subcommittee Chairman for participation in future ASTM sponsored hot box interlaboratory comparison programs.

### DATA COLLECTION TECHNIQUES

Questions 23 to 26 asked for information on how the laboratory collected its temperature and power data. In response to question 23, 17 laboratories reported that they used some type of computer/data logger automatic data acquisition system (computer, data logger/computer and scanning digital voltmeter/computer were the most often mentioned examples), 3 laboratories said they used a data logger only, 1 laboratory indicated the use of a combined data logger and manual system and 3 laboratories performed the temperature and power data collection manually. With automatic data acquisition systems, the requirements in ASTM C236 [2] and ASTM C976 [3] for collecting data at least every 60 minutes could easily be met. Only 2

	TABLE 9 "Calibration" specimens used in hot boxes <sup>a</sup>
Laboratory	Specimen(s) Description
1	· 10 cm (4 in.) and 20 cm (8 in.) molded bead expanded polystyrene boards.
2	· 10 cm (4 in.) expanded polystyrene boards (Hot Box round-robin material)
3	<ul> <li>15 cm (6 in.) EPS boards</li> <li>2.5 cm (1 in.) foil-faced P.I.R. boards</li> <li>2 cm (% in.) plywood</li> </ul>
4	<ul> <li>2.5, 5, 8, 10 cm (1, 2, 3 and 4 in.) "Styrofoam"</li> <li>10 cm (4 in.) expanded polystyrene boards (Hot Box round-robin material)</li> </ul>
5	· AAMA 1503 and polystyrene panels (still in process of calibrating box)
6	• 10 cm (4 in.) expanded polystyrene boards (Hot Box round-robin material)
7	• 10 cm (4 in.) expanded polystyrene boards
8 <sup>6</sup>	<ul> <li>1.3 cm (0.5 in.) expanded polystyrene boards sandwiched between 2 sheets of 0.3 cm (0.1 in.) glass; 3 different sizes:</li> <li>0.8 by 1 m (2.6 by 3.3 ft.), 1.02 by 1.02 m (3.4 by 3.4 ft.) and 1 by 1.6 m (3.3 by 5.2 ft.)</li> </ul>
9	<ul> <li>1.3 cm - 13 cm (0.5 - 5 in.) usually foam or fiberglass</li> <li>10 cm (4 in.) EPS</li> </ul>
10	• NVLAP samples
11	· 2.5, 5, 8, 10 cm (1, 2, 3, 4 in.) polystyrene foam
12	· 10 cm (4 in.) expanded polystyrene boards (Hot Box round-robin material)
13	<ul> <li>10 cm (4 in.) expanded polystyrene boards (Hot Box round-robin material)</li> <li>5 and 10 cm (2 and 4 in.) high density fiberglass boards</li> </ul>
14	• 13 and 30 cm (5 and 12 in.) 2 pcf fiberglass board
15	<ul> <li>3.7 cm (1 7/16 in.) glass sheathed fibrous glass board and dual glazed test panel for surface coefficient calibration</li> </ul>
16	Different insulation samples
17	• 10 cm (4 in.) polystyrene boards
18	• 10 cm (4 in.) sample from 1984 NVLAP round-robin
19	• 8 unframed windows with the edges buried in urethane walls
20	· Various insulation samples
21	<ul> <li>2.5, 5, 10 and 20 cm (1, 2, 4 and 8 in.) Dow styrofoam boards, 10 cm (4 in.) round-robin material</li> </ul>
22	<ul> <li>2.5 to 6.4 cm (1 to 2.5 in.) cork boards, fiberglass boards and foam boards, NVLAP samples</li> </ul>
23	<ul> <li>10 and 20 cm (4 and 8 in.) foam plastic boards, 10 cm (4 in.) foam plastic/wood sandwich</li> </ul>
24	<ul> <li>5 cm (2 in.) AAMA calibration panel; 1.3, 2.5, 3.8, 5, 7.6 and 10 cm (0.5, 1, 1.5, 2, 3 and 4 in.) extruded polystyrene; 0.5 cm (3/16 in.) single sheet of glass</li> </ul>
<sup>a</sup> Survey con <sup>b</sup> Survey con	tained English units. stained SI units

Table	10 Range o	of R-values test	ted versus "calibration"	' specimens <sup>a</sup>
	-	R	ange of R-values	
	Tested St	pecimens	"Calibration" S	Specimens
<u>Laboratory</u>	$\frac{m^2K}{W}$	<u>hft²R</u> <u>Btu</u>	<u>m²K</u> W	<u>hft²R</u> _Btu_
1	1.2-6.9	7-39	3,6	17,34
2	0.4-8.5	2-48	2.8	16
3	0.1-4.4	0.5-25	0.1,1.2,4.2	0.7,7,24
4	0.2-5.3	1-30	0.9,1.8,2.6,2.8,3.5	5,10,15,16,20
5	0.2-1.8	1-10	Values not r	reported
6	0.2-6.2	1-35	2.8	16
7	0.4	2-24	2.8	16
8	0.5-5.8	3-33	Values not r	reported
9	0-7.0	0 <sup>6</sup> -40	up to 3.5	up to 20
10	0.9-6.2	5-35	Values not r	reported
11	0.2-3.9	1-22	0.9,1.8,2.6,3.5	5,10,15,20
12	0.9-7.9	5-45	2.8	16
13	0.4-3.5	2-20	1.1,2.1,2.8	6,12,16
14	0.4-7.0	2-40	3.5,8.8	20,50
15	0.2-5.3	1-30	0.9	5
16	0.01-2.1	0.5-12	0.2-1.4	1-8
17	0.2-4.2	1-24	2.8	16
18	0.7-5.3	4-30		NR°
19	0.2-1.0	0.9-5.9	0.4-1.0	2-5.6
20	0.02-6.7	0.1-38		NR°
21	2.8	16	0.9,1.8,3.5,7.0	5,10,20,40
22	0.2-4.4	1-25		NR°
23	0.4-7.0	2-40		NR°
24	0.1-7.0	0.75-40	0.4	2. other NR <sup>c</sup>

TAE	BLE 11 Data collection periods
Laboratory	Data Collection Frequency <sup>a</sup>
1	20 s (hot side heater voltage) 1 min. (all other data)
2	20 min.
3	at least 1 reading every 10 min.
4	1 min., 1 hr. averages
5	10 min.
6	10 min.
7	4 min., continuously plot results
8	0.2 sec.
9	60 min.
10	4 hr.
11	continuously, 60 min. averages
12	30 sec., 5 min. averages
13	2 min., 10 min. averages
14	5 min., continuously plot results
15	30 min.
16	1 min., 15 min averages
17	1 hr.
18	30 min.
19	1 min., 1.5 hr. averages
20	1 min., continuously plot results
21	continuously, 4 hr. averages
22	continuously, 1 hr. averages
23	continuously, 15 min. averages
24	5 min., continuously plot results
<sup>a</sup> How f	requently data is collected and analyzed.

laboratories (both using manual data collection) collected data at this rate or less frequently; most laboratories collect data every 10 minutes or less as indicated in Table 11. The responses to questions 24 and 26 on what criteria is used to determine if steady-state conditions have been attained and what criteria are used to terminate a test after obtaining sufficient test data were quite mixed. In general, most laboratories run their tests according to the criteria specified in the procedures sections of ASTM C236 (Section 8) [2] and ASTM C976 (Section 9) [3]. The steady-state criteria for ASTM C236 [2] is to attain a 4 hour period where the surface temperatures do not vary by more than  $\pm 0.06 \,^{\circ}C(\pm 0.1 \,^{\circ}F)$ , the average power does not vary more  $\pm 1\%$  and the temperature and power data does not change For ASTM C976 [3], the steady-state criteria is left to the unidirectionally. judgement of the hot box operator, although it is recommended that a combined hot box and test specimen "time constant" be "calculated from the dimensions and estimated physical properties" and used in making this judgement as to when steadystate conditions might be attained. The required test times after achieving steadystate conditions is at least 8 hours for both ASTM C236 [2]] and ASTM C976 [3] where the average of the results for two or more successive 4 hour time periods As more operational experience using automated data agree to within 1%. acquisition equipment is obtained it may be possible to develop criteria that will allow the data acquisition system to determine when steady-state conditions have been attained and when the tests may be terminated after obtaining the steady-state test data.

# DATA ANALYSIS AND ERROR ESTIMATION

Questions 29 and 30 were more open-type of questions which asked how the laboratory reduced (or analyzed) the basic experimental data input (eg. voltage. temperature, power measurements) to a final output R-value or U-factor (often called U-value) characterizing the thermal performance of the test specimen being studied, and if an estimate of the error of this final output result is made. Since the questions were open-ended, there were a wide range of answers made to these two questions. Essentially, most hot box operators analyzed the test data in a manner similar to that prescribed in ASTM C236 [2] or ASTM C976 [3]. That is, the appropriate temperatures across and heat flux through the test specimen are monitored to see if the ASTM (or similar) steady state criteria were being met. Once the steady state criteria had been satisfied, time averaged values of the appropriate temperature difference and test specimen heat flux were used to calculate the test specimen R-value or U-factor. Personal computers that communicate with a data logger were used most often to perform the data analysis. Several hot boxes also carry out a finite difference or finite element analysis of the entire hot box, test frame and test specimen system heat transfer to compare with the experimental results. With regards to error estimation (or analysis), there was a wide range of responses, including several no responses. As has been discussed previously, most hot box operators performed some type of sensor/instrument/data acquisition system calibration and also used calibration specimens for estimating the

overall accuracy of their R-value or U-factor results. Several hot box operators have performed a detailed uncertainty analysis on their data acquisition system and a number of other operators indicated that they intend to perform such an analysis in the near future. This topic of error estimation should definitely be addressed in future versions of the ASTM hot box documents.

### CONCLUSIONS AND RECOMMENDATIONS

Many specific conclusions and recommendations are presented with the prior discussion of the responses to the specific questions in the survey questionnaire. Several general conclusions can be made here. One is the number of hot boxes which have been mothballed in the last several years. Telephone conversations with personnel at the laboratories where this occurred indicated that most of the reasons for mothballing were economic in nature, usually due to one or more of the following: unfriendly takeovers, the recent downturn in the economy, early retirement or transfer to other jobs of key hot box personnel with no action taken to replace them. Unless the previous hot box operator had left a fairly detailed operations manual behind, many hot boxes become practically obsolete after several years of non-use. This common trend of deep cuts in needed research and development activities when a company is in trouble will probably reduce the viability of North American building material companies to compete with new and improved products in the world economy of the future. Significant outlays for sophisticated research and development instrumentation and equipment and training new personnel will be needed to start up the mothballed hot boxes and to start up new hot box laboratories. Recently, there has been several new hot boxes built in the United States, primarily due to anticipated thermal (U-value) testing of fenestration products (windows, glass doors and glass curtain walls) for energy labels as proposed by the National Fenestration Rating Council (NFRC).

In general, it appears that most of the hot box laboratories that responded to the survey are conducting tests in accordance with the procedures outlined in ASTM C236 [2] and ASTM C976 [3]. Some concerns, however, still exist as to steady-state stabilization time, and the time laboratories are allotting to collect the steady-state data used to determine the final results. This area of concern needs to be better specified in the procedure sections of the two ASTM test methods.

Another concern is the calibration of the temperature measurement and the power measurement systems. Most laboratories are "calibrating" their temperature sensors, but only a few "calibrate" the entire temperature measurement system. It is recommended that periodic true calibration be carried out over the range of temperatures encountered in the hot boxes. Similarly, there needs to be regular periodic calibration of the power measurement systems. There is a need to specify methods and frequency of calibration of the temperature measurement and power (heat flow) measurement systems in future revisions of the ASTM test methods. It was found that hot box instrumentation is becoming more automated with direct links to personal computers. This is understandable with the emphasis laboratories are putting on minimizing test personnel. It is much more economically efficient to utilize this automation than to "tie-up" a technical person collecting and analyzing data. However, it should be recognized that there still needs to be a technically skilled hot box operator to interpret the results that the automated data acquisition system produces and to recognize that all those significant figures on a computer printout may not be very significant if the various measurement systems have not be adequately calibrated.

There has been a trend towards more hot boxes having the capability of air infiltration measurement and humidity control. This should be addressed in future versions of the ASTM hot box test methods.

In summarizing the "calibration" specimens used in the various laboratories, the trend is to have a range of R-value materials available. This is a step in the right direction relative to the results found in the last hot box survey [1]. However, some laboratories still only have one specimen and others still do not cover the full R-value range encountered in their day-to-day testing. One partial solution is to proceed with additional interlaboratory comparisons for nonhomogeneous test specimens under the leadership of ASTM subcommittee C16.30. Another partial solution is for more guarded hot boxes to obtain NVLAP accreditation and for NVLAP to start accrediting calibrated hot boxes. Work is progressing slowly in these areas, and needs to be accelerated.

The final questions on data analysis and error estimation indicate the need to have methods of error analysis (an uncertainty analysis of the data analysis procedure and calibration procedures to provide the basic measurement errors) specified and examples provided in future updates of ASTM C236 [2] and ASTM C976 [3].

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### APPENDIX

The Hot Box Instrumentation and Error Analysis Survey that was sent out to 39 potential hot box laboratories in the United States and Canada is included as an Appendix on the following pages.

# Hot Box Instrumentation and Error Analysis Survey By R. G. Miller and W. P. Goss

(If you wish to give a more detailed answer than space allows, then please use the back of this form or a separate sheet of paper....Thank you.)

Name	)	
Comp	bany Na	me
Addre	ess	
Telep	hone #	Fax #
1.	What 1	type(s) of hot box(es) do you have? guarded hot box calibrated hot box guarded/calibrated hot box (same box)
2.	What please	is the sample size and meter area of your hot box? (English units, )
	guarde	ed sample size meter area
	calibra	ted sample size
3.	What	is the maximum sample thickness that you can test in your hot box?

- 4. What test procedure(s) do you follow?
  - □ ASTM C236, guarded hot box
  - □ ASTM C976, calibrated hot box
  - AAMA 1503.1, thermal transmittance of windows
- 5a. If you have a guarded hot box, is it accredited by NVLAP?

□ Yes □ No

- 5b. If you have a calibrated hot box, would you like NVLAP to include ASTM C976 as one of the accredited test methods under the Thermal Insulation Program?
  - □ Yes □ No
- 6. What type of instrumentation do you use to measure temperature?
  - □ thermocouples
  - thermistors
  - □ RTD's
  - □ other, please explain on reverse side
- 7. Do you calibrate your temperature sensors?
  - □ Yes, briefly □ No describe on reverse side
- 8. What do you estimate is the error of your temperature measurements?

individual temperatures: \_\_\_\_\_\_E\_\_\_\_

- 9. What instrumentation is used to measure test specimen length and width to determine heat flow area?
  - tape measure
  - □ other, please explain on reverse side

- 10. What do you estimate is the error of your length/width measurements? units?
- 11. What type of power is supplied to the meter area (guarded hot box) or test specimen area (calibrated hot box)?
  - AC, go to question 12
  - DC, go to question 13
- 12. What instrumentation is used to determine the heat flow through the test specimen?

□ Watt and time measurement, type and model

\_\_\_\_\_

□ Watt-hour meter, type and model

other, please explain on reverse side

Go to question 14.

13. How is the power determination made?

measure current and voltage, types and models

- □ other, please explain on reverse side
- 14. Do you use any heat flux transducers in your hot box facility?
  - Yes, briefly
     No explain on reverse side
- 15. Estimate the accuracy of your heat flow measurements, please give units.

16.	What type and model temperature controllers are used in your hot box?
	hot chamber
	cold chamber
	meter box (guarded mode)
17.	What wind velocity ranges and direction are available in your hot box? (mph, please). What instrumentation is used for measurements? hot chamber
	cold chamber
	meter box (guarded mode)
18.	Is you hot box capable of providing air infiltration measurements?
	If yes, what is the pressure differential range? What type and model instrumentation is used?
	∡p range
	instrumentation

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f yes, what humidity ranges are available and what type and nstrumentation is used? humidity range		Yes, contir	nue		No, go te	o que	stion 2	0		
humidity range         Instrumentation         What heat flow directions are possible with your hot box?         Image: Instrumentation         What heat flow directions are possible with your hot box?         Image: Instrumentation         Image: Instrumentation	If yes, what humidity ranges are available and what type and instrumentation is used?									
What heat flow directions are possible with your hot box?	humidity range									
What heat flow directions are possible with your hot box?   horizontal  vertical-down vertical-up other, please explain What are the temperature ranges in the hot and cold chambers?	nstru	umentation			_					
<ul> <li>horizontal</li> <li>vertical-down</li> <li>vertical-up</li> <li>other, please explain</li> <li></li></ul>	What heat flow directions are possible with your hot box?									
<ul> <li>vertical-up</li> <li>other, please explain</li> <li></li> <li>What are the temperature ranges in the hot and cold chambers?</li> </ul>			horizontal vertical-do	wn						
What are the temperature ranges in the hot and cold chambers?			vertical-up other, plea	se expl	ain _		-			
What are the temperature ranges in the hot and cold chambers?				. <u> </u>						
	Wha	t are the terr	nperature ran	ges in t	he hot and	d colo	l cham	bers?		

23. How do you collect temperature and power data?

manuai data logger computer/data logger

Briefly describe with a schematic on reverse side.

24. Once a test is started, what criteria are used to determine if the hot box has reached steady-state?

- 25. Both C236 and C976 require 4-hour test periods. How often within the 4-hour periods do you collect data?
- 26. What criteria are used to terminate a test?
- 27. What range of R-values have you <u>tested</u> in your hot box? (English units, please)
- 28. What type of products/systems do you test in your hot box?

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- 29. Do you use any data reduction techniques? If so, please describe. Continue on reverse if necessary.
- 30. Do you perform any error analysis to estimate the accuracy of your final test specimen R-value or U-factor?

□ Yes, continue □ No

If yes, please provide a sample error analysis calculation and an estimate of the above accuracy with units. Please complete on a separate sheet.

Thank you for taking the time to complete this survey. Could you kindly return this form by FAX or by mail to the address below:

> R. Gerry Miller Jim Walter Research Corp. 10301 9th Street, North St. Petersburg, FL 33716 (813) 576-4171

FAX (813) 576-0318

Ravi Gorthala, Jeffrey A. Roux, William P. Levins, and Kenneth E. Wilkes

COMPARISON OF HEAT TRANSFER MODELING WITH EXPERIMENTAL RESULTS FOR RESIDENTIAL ATTIC INSULATIONS

**REFERENCE:** Gorthala, R., Roux, J. A., Levins, W. P., and Wilkes, K. E., "Comparison of Heat Transfer Modeling with Experimental Results for Residential Attic Insulations," <u>Insulation Materials: Testing and Applications. 2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A heat transfer model was developed for fibrous insulations typically used in residential attic applications. The model predictions were compared with the experimental results obtained at the Oak Ridge National Laboratory for the three Karns Research Houses located near Knoxville, Tennessee. The model was comprised of onedimensional coupled conduction and radiation heat transport; the unsteady energy equation and radiative transport equation were solved simultaneously. A control-volume-based finite-difference scheme was used to solve the energy equation. The radiative transport equation was solved by the method of discrete ordinates which was inherently required for solving the energy equation. Standard R-30 fiberglass insulation batts without a foil radiant barrier were studied for both summertime and wintertime conditions. The predicted heat fluxes were in good agreement with the experimental heat flux data for the entire one week period for both summer and winter. A key point of this paper is the comparison of the theoretical results with the experimental data for a week's period which has not been attempted previously.

KEYWORDS: heat transfer model, Karns Research House, standard R-30 fiberglass, summertime, wintertime, heat flux

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#### INTRODUCTION

Knowledge of the heat transfer behavior of fibrous insulations in residential attic environments is important for conserving energy. Study of heat transfer in various types of fibrous insulations can help select the optimum insulation according to the thermal environment. Therefore, it is essential to develop a numerical model to study heat transfer and compare the results with the existing experimental data for verification of the model. Some work has been done in this field [1,2,3,4,5,6,7]. Rish [8] has developed a numerical heat transfer model for fibrous insulations used in residential attic environments. His model includes the basic heat transfer mechanisms (conduction and radiation). According to his study, volumetric radiation within low density fiberglass insulation (10 kg/m  $^3$ ) accounts for approximately 45 percent of the total heat flux. Rish [8] used his model for a span of 24 hours only, since data over a longer span were not available. The non-availability of radiative properties was also a major problem in the heat transfer modeling of attic insulations. Yeh [9] has determined the radiative properties of fiberglass insulation material from reflectance data in the 4-20 µm wavelength region. Yajnik [10] has predicted the radiative properties for fiberglass insulations for the 4-40 µm wavelength region.

Typical insulations used in residential attics are fiberglass and rockwool, ranging in thicknesses from 0.15 to 0.28 m (or higher). The basic heat transfer mechanisms that can exist inside these insulations are gas conduction, intrafiber conduction, free convection and volumetric radiation. Verschoor and Greebler [1] in their experimental investigation found that gas conduction played the most significant role for overall heat transfer through the insulating media. Radiation was also found to be a significant contributor to the insulating media; intrafiber conduction and convection effects were found to be negligible.

In the present work, heat transfer through a standard R-30 (Rvalue of 5.3  $m^{4}K/W$  and 0.24 m thickness) fiberglass insulation batt for both summertime and wintertime was studied. Here, the same numerical approach as in [8] was employed for obtaining heat transfer results for a wide range of temperature variations. The important point to note is that this study focused on the increased time duration (one week time period) as well as different seasons (winter and summer); reference [8] only considered summer conditions and compared with data for a time period of only one day. The radiative properties developed in [10] were used as opposed to properties of [9] (which were used in [8]). Solution to the radiative transport equation was based on the method of discrete ordinates as presented in Refs. [11], [12] and [13]. The methodology used in solving the problem was control-volume-based finite-difference technique; а а detailed description of the approach can be found in Ref. [14]. The temperature data required for the boundary conditions were provided by the Oak Ridge National Laboratory (ORNL). The instrumentation for obtaining the temperature data as well as heat flux data were located in a test house situated near Oak Ridge, TN. A data logger (Accurex Autodata Ten/10) was used in conjunction with a microcomputer (Tandy Model 2000) for this purpose. The theoretical heat fluxes predicted by the numerical model were compared with the experimental heat flux data provided by ORNL.

#### STATEMENT OF THE PROBLEM

A significant part of the total energy consumption in the United States is used for space heating and cooling. A decrease in the energy consumption of buildings can result in a significant decrease in the national energy consumption and also yield environmental benefits. Once the sources of heat losses are identified, remedies can be found and applied. Therefore, a better understanding of the heat transfer mechanisms is essential. The present area of focus is low density residential attic fiberglass batt insulation (10.92 kg/m<sup>3</sup>). The primary objective of this study was to accurately model the transient thermal behavior of a fibrous insulating medium. The attic geometry for a residential home with attic insulation is shown in Fig. 1. The boundaries for the fiberglass insulation are air at the top and gypsum wallboard (ceiling) at the bottom. As shown in Fig. 1, T<sub>r</sub> is the temperature at the top surface of the insulation, T<sub>s</sub> is the temperature at the bottom surface of the insulation, is the temperature of the ceiling, and T<sub>r</sub> is the inside room ambient temperature.

The fiberglass constitutes a matrix of glass fibers in a gas phase (air). Therefore, the true thermal conductivity of the fiberglass is nearly that of air. The heat transfer behavior inside the fiberglass insulation was modeled as coupled conduction and radiation. In the fiberglass insulation, the radiant heat can be absorbed, scattered (isotropically or anisotropically) and emitted. The boundary temperatures required for solving the one-dimensional energy equation and the radiative transport equation are the inside roof temperature



FIG. 1--Attic geometry and identification of variables.

 $({\rm T}_{\rm r})$ , the substrate (bottom of the insulation) temperature  $({\rm T}_{\rm s})$ , and the temperature of the top surface of the insulation  $({\rm T}_{\rm s})$  as functions of time. Either emissivities or reflectivities must be known at the roof and substrate surfaces. It was assumed that the fiberglass is homogeneous and that the fiberglass radiative properties are independent of temperature or thickness; the specific heat of the fiberglass is considered constant. Also, the fiberglass is considered dry (no moisture transport). The properties of fiberglass insulation determined by Yajnik [10] are used in the present problem. Both summertime and wintertime situations are studied; previously only summertime results were presented for 24 hours by Rish [8]. The temperature data as well as the experimental heat flux data were provided by ORNL.

#### EXPERIMENTAL SETUP

The experimental heat flow data used for comparisons in this work were obtained from a research facility operated by the Oak Ridge National Laboratory. This facility consists of three very similar unoccupied ranch-style houses (Fig. 2) located in Karns, Tennessee, a suburb of Knoxville. The houses were located on relatively flat lots, which were devoid of trees or shrubbery, and there were no tall buildings nearby. Consequently, there was very little shielding of the houses from the wind, and very little shading from the sun. Each house



FIG. 2--Ranch style houses with test cells (attics) located in Karns, Tennessee.

had lll  $m^2$  of conditioned space, located over a crawl space. The houses faced north and were heated and cooled with heat pumps. Figure 3 shows a typical floor plan of the houses.

The roof was constructed with 2X4 wood trusses, 0.61 m on center, with a pitch of 5 in 12. The roof was covered with 12 mm plywood sheathing, roofing felt, and brown asphalt shingles. The attic space was vented with soffit vents having a net free area of about 0.293 m<sup>2</sup> and gable vents having a net free area of 0.465 m<sup>2</sup> The ceiling consisted of 12 mm gypsum wallboard, with two layers of fiberglass batt insulation. The bottom layer of insulation had a nominal R-value of 1.94 m<sup>2</sup>K/W, was approximately 89 mm thick, had a kraft paper vapor retarder, and was placed between the 2X4 lower truss chords with the vapor retarder next to the gypsum wallboard. The top layer had a nominal R-value of 3.35 m<sup>2</sup>K/W, was approximately 152 mm thick, and was laid perpendicular to the bottom layer.

The houses were highly instrumented with a data acquisition system consisting of an Accurex Autodata Ten/10 and a microcomputer. Approximately 50 data channels were scanned on a 30-second interval, with 15-minute averages stored by the microcomputer. Data channels included sensors for temperatures throughout the house and attic, temperatures in the ground in both the crawl space and 30 feet from the house, ceiling heat flux, indoor humidity, outdoor temperature, outdoor humidity, wind speed, wind direction, and solar insolation on a horizontal surface. It should be noted that all the experimental variables mentioned above were not used in this work.

The ceiling heat flux data were collected with 57 mm X 57 mm X 1.6 mm heat flux transducers manufactured by Thermonetics. The heat flux sensors were located between the kraft paper vapor retarder and the gypsum wallboard, midway between the bottom chords of the roof trusses. One heat flux transducer (HFT) was placed over the hallway and the other was placed over Bedroom No. 1 (see Fig. 3).

#### ANALYSIS

#### Solution of the Energy Equation

For the single-layer insulation model, the energy equation for transient, coupled conduction and radiation heat transfer is given by

$$\frac{\partial}{\partial y} \frac{\partial T}{\partial y} - \frac{\partial q_r}{\partial y} = \rho f C_{pf} \frac{\partial T}{\partial t}$$
(1)

where  $k_f$  is the true conductivity,  $\rho_f$  is the bulk density,  $C_{pf}$  is the specific heat, T is the temperature of the insulation and  $q_r$  is the radiative heat flux. As shown in Fig. 1, the coordinate system origin was taken at the insulation substrate and the boundary conditions are:

at the top surface of the insulation

$$T = T_0(t)$$
 at  $y=y_0 > 0$  (2a)





and at the substrate of the insulation

$$T = T_{S}(t) \qquad \text{at } y=0 \tag{2b}$$

The system of discretization equations used to solve the energy equation numerically, was obtained by the control volume method of Patankar [14].

#### Radiative Transport Equation

The net radiative heat flux,  $q_r$ , is required for solving Eq. (1);  $q_r$  is obtained by solving the radiative transport equation. The onedimensional radiative transport equation for an absorbing, emitting, and anisotropically scattering medium from Ref. [13] can be written as

$$\mu \frac{dI(\tau,\mu)}{d\tau} = -I(\tau,\mu) + \frac{\omega}{2} \int I(\tau,\mu')\Phi(\mu,\mu')d\mu' + n^2(1-\omega)I_b(T(\tau))$$
(3)

Here,  $\tau$  ( $\tau$ =By and  $\beta$  is the extinction coefficient) is the optical depth,  $\mu$  is the cosine of the polar angle  $\theta$ ,  $\omega$  is the single scatter albedo, n is the refractive index of the medium,  $I(\tau,\mu)$  is the radiative intensity at depth  $\tau$  in the specified direction of  $\mu$ ,  $I_b$  is the emitted radiative intensity at depth  $\tau$  as specified by Planck's blackbody intensity function,  $\Phi(\mu,\mu')$  is the scattering function or phase function. Equation (3) can be written in discrete ordinates form as [13]

$$\frac{dI(\tau, \mu_{j})}{d\tau} = -I(\tau, \mu_{j}) + \frac{\omega}{2} M \qquad L \\ \sum_{m=1}^{L} a_{m}I(\tau, \mu_{m}) \sum_{\ell=0}^{L} A_{\ell} P_{\ell}(\mu_{j}) P_{\ell}(\mu_{m}) \\ + n^{2}(1-\omega)I_{b}(T(\tau)) \qquad j = 1, \dots, M \qquad (4)$$

where the scattering function expanded in terms of Legendre polynomials  $(P_{g}(\mu))$  as

$$\Phi(\mu, \mu') = \sum_{\ell=0}^{L} A_{\ell} P_{\ell}(\mu) P_{\ell}(\mu')$$

The discrete ordinates form was obtained by using Gaussian quadrature to approximate the integral term in Eq. (3) to obtain Eq.(4) (a system of linear non-homogeneous first-order differential equations); here  $\mu_{\rm IR}$  are quadrature points and  $a_{\rm IR}$  are quadrature weights. The boundary conditions for Eq.(4) become

$$I(0,\mu_{\rm m}) = \rho_{\rm s}I(0,-\mu_{\rm m}) + (1-\rho_{\rm s})n^2 I_{\rm b}(T_{\rm s}), \quad \mu_{\rm m} > 0, \ {\rm m} = 1,2,..{\rm M}/2$$
(5)

$$I(\tau_{0},-\mu_{m}) = \rho_{n}I(\tau_{0},\mu_{m}) + (1-\rho_{n})n^{2}I_{b}(T_{r}), \quad \mu_{m} > 0, m = 1,2,...M/2 \quad (6)$$

where  $I_b(T_r)=e_b(T_r)/\Pi$  and for a gray body  $e_b(T_r)=\sigma T_r^4$ . Here,  $\rho_s$  is the reflectivity of the substrate,  $\rho_{\Pi}$  is the reflectivity of the fiberglass-air interface at the insulation top boundary. It should be noted that  $\rho_s=0.07, \rho_{\Pi}=0$ , and n=1 for this problem. The inhomogeneous

nature of these equations require that the entire solution be formed as a set of linear sums of homogeneous solutions and particular solutions for the system (Eq.(4)) of differential equations. The solution procedure has been discussed in Refs. [11, 12, 13]. The radiative heat flux is given by definition as:

$$q_{r}(\tau) = \int \int I(\tau, \mu) \mu d\mu d\phi$$

$$0 -1$$
(7)

Since the radiative heat flux is a highly non-linear function of temperature, coupling the radiative and conduction heat transfer, both the energy equation and the radiative transport equation were solved simultaneously by an iterative scheme. A non-uniform grid with a finer grid near the boundaries and a coarser grid for the interior nodes was used in the insulation. The number of total grid points used was 13. A 10-minute time step was used in the computation. The temperature data provided by ORNL had average hourly temperatures, so a linear interpolation was used to obtain temperatures at 10-minute intervals. The program used only 100 seconds of CPU time on a mainframe computer (Amdahl 470) for running two weeks of data.

#### RESULTS

Heat transfer results are presented for low density fiberglass batt insulation (R30) in an attic environment for both summertime and wintertime conditions. The radiative properties (extinction coefficient and albedo) and other thermal properties (density, specific heat and thermal conductivity) for fiberglass used in the computations are given in TABLE 1 (Ref. [10]). Figure 4 shows the typical type of boundary condition temperature data used for solving the heat transfer problem. The roof temperature and the temperatures of the top and bottom of the insulation are the boundary condition

TABLE 1-- Radiative and thermal properties used in numerical computations

٩f	10.92 kg/m <sup>3</sup>
Cpf	844.4 J/kg-K
kf	$a + bT + 8.5537 \times 10^{-5} \rho_f$ W/m-K $a = 4.97576 \times 10^{-3}$ , $b = 7.00025 \times 10^{-5}$
ω	0.201 (Isotropic)
в	3.70 (Isotropic) cm <sup>-1</sup>


FIG. 4--Temperature-time histories data provided by ORNL for R-30 (Standard fiberglass insulation without foil radiant barrier) for September 3, 1987.



FIG. 5--Comparison of measured substrate heat fluxes with predicted heat fluxes for September 3, 1987.

information needed for solving the energy equation and radiative transport equation.

(ceiling-fiberglass interface) fluxes Substrate heat were calculated using the present model. The predicted heat fluxes were compared with the experimental data obtained by ORNL. All the are based on the assumption that the fiberglass calculations insulation is dry (no moisture transport). Figure 5 shows a comparison between the predicted heat fluxes and the experimental heat fluxes for R30 fiberglass insulation during summertime (September 3, 1987). A positive sign in heat flux indicates that heat transfer takes place from the attic into the room and a negative sign in heat flux indicates that heat transfer takes place from the room into the attic. It can be seen from the Fig. 5 that the predicted heat flux agrees well with the experimental heat flux data. Heat fluxes are negative until about 10:00 a.m. indicating heat transfer from the room into the attic. This is in agreement with the temperature-time histories shown in Fig. 4. The temperature of the substrate is higher than that of the top of the insulation until about 10:00 a.m. Then the insulation temperature rises and reaches a peak temperature in the afternoon. The substrate temperature is essentially constant. This temperature gradient between the top of the insulation and the substrate causes the heat transfer to occur from the attic into the room. Therefore, the heat flux attains a maximum positive value when the temperature gradient reaches a maximum value. The magnitudes of the theoretical fluxes conform well with the experimental values. But, there is a slight shift in the occurrence of the peak heat flux indicating a predicted peak earlier in time. Also, the predicted heat fluxes are somewhat higher in magnitude until 10.00 a.m. This may be because



FIG. 6--Comparison of predicted substrate heat fluxes with the ORNL data for a six-day period in summertime (August 30, 1987 - September 4, 1987) for R-30.

this model considers the fiberglass to be dry and neglects the effects of moisture adsorption and desorption. It should be noted that the hourly averaged temperature data for the boundary conditions was used in the model; the interpolated data for a 10 minute time interval was used in the computations. This may also induce a shift between the predicted values and measured values. The predicted integrated flux value over 24 hours for September 3, 1987 was higher by 6 percent compared to the experimental integrated flux value. However, the present model may be considered very good for its agreement with the experimental results, considering that the heat flux meters used are accurate to + 5 percent.

The predicted and experimental heat fluxes for a six day period of summertime from August 30, 1987 to September 4, 1987 are shown in Fig. The dashed line indicates the experimental data whereas the 6. continuous line denotes the model. It can be observed that the predicted results are consistently in good trend-wise agreement with the experimental results for all six days. The model reasonably predicts the magnitudes as well as the trend. Only in the mornings until about 10:00 a.m. is there some deviation between the theoretical results and the experimental fluxes which might be attributable to the presence of a small amount of adsorbed moisture. The maximum positive heat flux (heat transfer from the attic into the room) is about 7.5 watts/m<sup>2</sup> and the maximum negative heat flux (heat transfer from the room into the attic) is about 2.0 watts/ $m^2$ . The data used are a good test of the model since these data correspond to a wide variety of moderate to hot summer conditions. The peak roof temperatures ranged from 38°C to 71°C. Also, a comparison of the predicted substrate integrated heat fluxes for the theory and data is shown in TABLE 2. The positive sign in the percentage error denotes that the experimental value is higher, whereas the negative value indicates that the predicted value is higher. The percentage errors for the first three days are relatively large compared to the latter three days, but it should be noted that the first 3 days have smaller heat fluxes than the last three days and hence the percentage errors will be magnified. Also, the net heat fluxes are probably small, since there are negative heat fluxes for these 3 days. This deviation might be attributable to the influence and presence of moisture. The present model considers that the fiberglass is dry and the radiative properties of fiberglass used in this paper are for a dry insulation only [10]. The relative humidity data for these days revealed that the relative humidities for the first three days were much higher between late afternoon (about 2:00 p.m.) and early morning (8:00 a.m.) compared to the latter three days. High relative humidity at the top of the insulation batt could cause adsorption of  $H_2O$  present in the attic air into the phenolic fiberglass binder. When the  ${\rm H_{2}O}$  is adsorbed, the heat of adsorption released will be given off to the insulation. This would contribute to reducing the magnitude of the morning substrate heat flux. Hence, the presence of moisture which is neglected in the analysis of this model, might be considered responsible for some of the deviation in the integrated flux values in TABLE 2. Also, the effect of moisture on the radiative properties is an unknown factor. The daily integrated heat fluxes from the model were in the range of -3.5% to 43.0% for a six day period with an average difference of 12%. However, for the first three days (see Fig. 6) the net heat fluxes were lower in magnitude and hence the percentage errors were magnified. It should also be kept in mind that

the heat flux meters were accurate to + 5 percent; only the hourly averaged temperature data for the boundary conditions was used in the model. The thermal properties used in the model were nominal. Any deviation in the properties used in the model from the true values may impact the predicted values. It should be noted from Fig. 6 that no two days are identical. Moreover, the amount of moisture may be quite different from one day to the other. This may be one of the reasons for the underpredicted as well overpredicted values of the calculated integrated heat flux (dry case) in addition to the inaccuracies mentioned above. It should be noted from TABLE 2 that the underpredcited values are only 2.5% and 3.5% which are very close to the data and within the accuracy of the heat flux meters. The dry model predicts very close to the data for these two days because the relative humidity and hence the amount of moisture may be very small. However, the overpredicted values range from 6.2% to 43.0%; this may be due the presence of higher amounts of moisture whose effects were not included in the dry model; the large percentage differences for the summertime overpredictions correspond to days of higher relative humidity and hence higher amount of moisture.

Also, the substrate heat fluxes were calculated for the wintertime period of January 7, 1987 through January 12, 1987. Figure 7 shows the comparison between the predicted heat fluxes and experimental heat fluxes for this wintertime period. From Fig. 7, it can be seen that the theoretical results match well trend wise with the experimental data for all six days. During this period, heat fluxes are mostly negative indicating heat transfer from the room into the attic. This is because of higher substrate temperatures in the winter and rarely is the temperature of the top of the insulation higher than the substrate temperature. The maximum negative flux is about 4.5 watts/m<sup>2</sup> and the maximum positive flux is about one watt/m<sup>2</sup> during this period. Even for the wintertime, the data provided a good testing tool with a wide range of variations in winter conditions from sunny to cloudy days. TABLE 3 shows a comparison between the 24-hour integrated



FIG. 7--Comparison of predicted substrate heat fluxes with the ORNL data for a six-day period in witertime (January 7, 1987 - January 12, 1987) for R-30.

TABLE 2-- Percentage difference for 24-hour integrated fluxes between the predicted and experimental data during summertime.

Day	Percentage Error#
August 30, 1987	23.0
August 31, 1987	21.0
September 1, 1987	43.0
September 2, 1987	-2.5
September 3, 1987	6.2
September 4, 1987	-3.5

# A negative sign in the percentage indicates that the predicted integrated flux value is higher and the positive sign indicates that the experimental integrated flux value is higher.

TABLE 3--Percentage difference for 24-hour integrated fluxes between the predicted and the experimental data during winter.

Day	Percentage Error <sup>#</sup>
January 7, 1987 January 8, 1987 January 9, 1987 January 10, 1987 January 11, 1987 January 12, 1987	$ \begin{array}{r} -7.0 \\ -17.0 \\ -4.0 \\ -2.0 \\ -17.0 \\ -13.0 \\ \end{array} $

# A negative sign in the percentage indicates that the predicted integrated flux value is higher. fluxes for the model and the measured data. The predicted values were higher than the experimental values. This may be somewhat attributableto the presence of moisture. During wintertime, usually the substrate temperature is higher than the insulation top temperature. If the moisture is present in the insulation, heat transfer occurring from the substrate will heat up the insulation and the moisture as well, thereby decreasing the temperature gradient between the insulation top and the attic air. For the dry case, the opposite is true. Therefore, the heat flux for the dry case may be higher than the measured heat flux. But, the overall predicted integrated heat flux for the six day period differed by only an average of 10 percent. Thus, agreement of the theoretical results with the experimental data for both summertime and wintertime subjected to a wide range of conditions may be considered good.

Here, only R-30 fiberglass insulation has been studied. The study can be extended to various types of fiberglass insulations with foil radiant barriers to verify the effect of foil radiant barriers on heat transfer. In the future, the effect of moisture adsorption and desorption will be included in the model to make the model more comprehensive and complete and also to evaluate the importance of moisture transport as compared to the coupled conductive and radiative transfer.

## CONCLUSIONS

A one-dimensional transient heat transfer model with coupled conduction and radiation was used to model the heat transfer behavior for R-30 fiberglass insulation for a six-day period during summer and winter conditions. The boundary condition temperature data were measured at ORNL. The substrate heat fluxes determined by the model agree reasonably well with the experimental data provided by ORNL for both summertime and wintertime. The model is consistent and reasonably accurate in predicting the magnitudes as well as the trend of the heat fluxes. Thus, the model is useful for a wide range of temperatures and seasonal changes and also for understanding the heat transfer mechanisms occurring within the attic insulation.

NOMENCLATURE

Α	Coefficient of phase function
Cp	Specific heat, J/kg-K
I <sup>-</sup>	Intensity, W/m <sup>2</sup> -steradian
k	Thermal conductivity, W/m-K
n	Refractive index of medium
P	Legendre polynomial
q	Total heat flux, W/m <sup>2</sup>
q <sub>r</sub>	Radiative component of total heat flux, q
Τ	Temperature, K
T <sub>a</sub>	Attic air temperature
$T_{C}^{-}$	Ceiling temperature

- T<sub>i</sub>..... Room temperature  $T_0$ ..... Temperature at the top of the insulation T<sub>5</sub>..... Substrate temperature Tr..... Roof temperature t ..... Time, sec y ..... Physical thickness, m  $\beta$  ..... Extinction coefficient, cm<sup>-1</sup>  $\theta$  ..... Polar angle, degrees  $\rho_f....$  Density,  $kg/m^3$

# Subscript

- b ..... Blackbody
- f ..... Fiberglass insulation
- j ..... Summation index
- m ..... Summation index

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COMBINED CONDUCTION, RADIATION HEAT TRANSFER AND MASS TRANSFER IN FIBROUS ATTIC INSULATIONS

**REFERENCE:** Gorthala, R., Roux, J. A., and Fairey, P. W., "Combined Conduction, Radiation Heat Transfer and Mass Transfer in Fibrous Attic Insulations," <u>Insulation</u> <u>Materials: Testing and Applications. 2nd Volume. ASTM STP</u> <u>1116.</u> R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A numerical model to study heat transfer in an attic insulation has been developed. This model comprises combined conduction, radiation, and moisture transport. The radiative one-dimensional energy equation, transport equation and species equations for liquid H<sub>2</sub>O and vapor H<sub>2</sub>O were solved simultaneously. A control volume based finite difference method was used to solve the energy equation and the species equations. A guasi-analytical technique (method of discrete ordinates) was used in solving the radiative transport equation. Standard fiberglass insulations with and without a foil radiant barrier were modeled. Heat transfer results were compared with the experimental data measured at the Florida Solar Energy Center. Predicted heat fluxes agree with measured fluxes quite well. All the cases studied with moisture present were also studied for the dry cases (no mass transfer) and the results were compared. Results with moisture being included indicate that heat transfer from the attic into the house decreases for fiberglass insulations during summer as compared to dry insulation.

KEYWORDS: moisture, attic insulations, adsorption, desorption, conduction, radiation

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## INTRODUCTION

Study of simultaneous heat and mass transfer in fibrous insulations in a residential attic environment is important for reducing heat losses or gains and thereby conserving energy. Rish [1] developed a numerical heat transfer model for fibrous insulations subjected to typical residential attic thermal environments. His model included heat transfer mechanisms of conduction and radiation. According to his study, volumetric radiation within the fiberglass batt accounted for approximately 45 percent of the total heat flux. His model did not include the moisture phenomenon which is also often present in an attic insulation when subjected to a humid environment. The non-availability of radiative properties has always been a problem in the heat transfer predictions for fibrous insulations; Yeh [2] has developed a technique for determining the radiative properties for fiberglass residential insulation.

Much work has been done on simultaneous heat and mass transfer in porous insulations [3, 4, 5, 6]. But until now, no one has studied the coupled conduction, radiation heat transfer and mass transfer situation. So, the present study focuses on development of a total numerical model for the prediction of heat transfer in an attic insulation which combines mass (moisture) transfer to the coupled radiation and conduction problem. Thicknesses of residential attic insulations vary typically from about 8 cm to 0.28 m. The basic heat transfer mechanisms that can exist inside these insulations are gas conduction, intrafiber conduction, free convection and volumetric radiation. Verschoor and Greebler [7] found that gas conduction played a significant role for the overall heat transfer through the insulating media. Radiation was also found to be a significant convection effects were found to be negligible.

In studying simultaneous heat and mass transfer, it is important to understand the interaction of moisture transport with heat transfer. Moisture transport is generally a complex problem; it includes an air-vapor mixture flow due to convection, free convection, and infiltration through openings. It also includes vapor transport by diffusion and convection, flow of liquid due to gravity, capillarity and diffusion and condensation or evaporation accompanied by release or consumption of latent heat. At moderate temperatures moisture adsorption/ desorption may be equally important depending on the insulation binder properties. There is a large body of literature available dealing with the general concepts of heat transfer and moisture in porous media from formulation of the phenomenon [8, 9] to specific applications.

The objective here was to develop a total model for heat transfer coupled with moisture transport in fibrous insulations. The heat transfer portion comprises transient conduction and radiation and is the same model as developed by [1] and [2]. Solution to the radiative transport is based on the method of discrete ordinates as presented in Refs. [10], [11] and [12]. For the mass transfer analysis, the equations are derived from control volume fundamentals. The model comprises the diffusion of both vapor and liquid. Because of the very low velocities, convection is neglected. The phenomenon of moisture adsorption/desorption, analogous to the phenomenon of condensation and evaporation, has been incorporated in the model. Two standard fiberglass insulations R30STD (R-value of 5.3 m<sup>2</sup>K/W and 0.24 m thickness) and R19STD (R-value of 3.3 m<sup>2</sup>K/W and 0.15 m thickness), are studied. Since the properties for moisture adsorption/desorption for the fiberglass binder (phenolic) were not known, it is assumed that the binder behaves similarly to the mineral zeolite.

The wet bulb temperature and dry bulb temperature data required for the boundary conditions were provided by Florida Solar Energy Center (FSEC) for a summertime situation. The methodology used in solving the problem is a control-volume based finite difference scheme. A detailed description of the approach can be found in Ref. [13]. Heat transfer results were compared with the experimental results as well as dry case (no mass transfer) results. Comparison of the predicted heat transfer results with that for the dry case results depict the influence of moisture.

# STATEMENT OF THE PROBLEM

The primary objective of this study was to develop an analytical/ numerical model which can simulate the transient thermal behavior of a fibrous insulating medium in the presence of heat (conduction and radiation) and moisture (mass) transport. The typical environment to which the insulating medium is subjected is a residential attic. The attic geometry for a residential home with attic insulation is shown in Fig. 1. The boundaries for the fiberglass insulation are air at the top and gypsum board (ceiling) at the bottom. As shown in Fig. 1, T<sub>r</sub> is the inside roof temperature, T<sub>a</sub> is the attic air temperature, T<sub>b</sub> is the temperature at the top surface of the insulation, T<sub>b</sub> is the temperature at the bottom surface of the insulation (interface between gypsum board and insulation), T<sub>c</sub> is the temperature of the ceiling,



FIG. 1--Attic geometry.

and T, is the inside room ambient temperature.

The presence of moisture in the insulation is expected to influence the overall heat transfer. Hence, diffusion of vapor and liquid along with mass adsorption/desorption were considered in the energy transfer. Because of the mass concentration gradients, vapor and liquid diffusion will occur. As these species diffuse through the insulation, energy (enthalpy) is carried with the associated species. Also, when the vapor leaves the insulation (desorption), it carries latent heat of desorption with it. Likewise moisture heats up the insulation, when it is adsorbed by the insulation binder. The influence of vapor diffusion, liquid diffusion and adsorption/ desorption are included in the energy equation. With the specified boundary conditions (relative humidities and hence concentrations), moisture concentration profiles are determined which will be required to predict the temperature field and heat transfer. The species conservation equations for liquid and vapor are required for obtaining their respective spatial concentrations.

In the fiberglass insulation heat can be conducted and the radiant heat can be absorbed, scattered and emitted. The inside roof temperature, the substrate temperature and the temperature of the top surface of the insulation are considered to be known (measured) functions of time (the data provided by FSEC). Either emissivities or reflectivities must be specified at the roof and substrate surfaces. In this analysis, only the fiberglass is considered and the sheetrock is not considered. Radiative properties of fiberglass determined by Yeh [2] were used for the present problem with the assumption that the presence of moisture will not strongly effect the radiative properties of the insulation.

## ANALYSIS

The governing equations related to this problem are the energy equation, the species equations of the liquid water and vapor, and the radiative transport equation. A finite difference model was used to solve the energy equation and the species equations simultaneously. A quasi-analytical method (method of discrete ordinates) was utilized in solving the radiative transport equation.

The basic assumptions made in the development of this model are given as below: 1) one-dimensional combined conduction, radiation heat transfer and moisture transfer; 2) fiberglass insulation is an absorbing, emitting and scattering (isotropic or anisotropic) medium; 3) gray radiative properties; 4) top insulation interface is transparent to radiation; 5) constant bulk density and homogeneous chemical composition; 6) convection of liquid and vapor is neglected; 7) relative humidities at the boundaries are known functions of time; and 8) boundary temperatures are known functions of time.

The one-dimensional energy equation for transient, coupled conduction, radiation heat transport and moisture transport is given by

$$\frac{\partial}{\partial y} \frac{\partial T}{\partial y} \frac{\partial q_{r}}{\partial y} \frac{\partial}{\partial y} \frac{\partial q_{r}}{\partial y} \frac{\partial}{\partial y} \frac{\partial m_{v}}{\partial y} \frac{\partial}{\partial y} \frac{\partial m_{l}}{\partial y} \frac{\partial}{\partial y} \frac{\partial m_{l}}{\partial y} \frac{\partial}{\partial y} \frac{\partial m_{l}}{\partial y} \frac{\partial}{\partial t} \frac{\partial}{\partial t} (\rho_{f} C_{f} T) \quad (1)$$

with boundary conditions:

$$T = T_{S}(t) \text{ at } y = 0$$
$$T = T_{O}(t) \text{ at } y = y_{O}$$

The first term on the left hand side of the Eq. (1) represents heat conduction; the second term corresponds to radiative heat transfer; the third and the fourth terms denote heat transferred by diffusion of vapor and liquid H<sub>2</sub>O; and the last term on the left hand side,  $\dot{q}$ , denotes volumetric heat adsorption /desorption (heat source or sink) due to moisture adsorption /desorption within the insulation. In Eq. (1),  $k_f$  is the true thermal conductivity (i.e.  $k_f$  refers to conduction heat transfer only and is not a type of effective or apparent conductivity) of the insulation,  $\rho_f$  is the bulk density of the insulation,  $c_f$  is the specific heat of the insulation; these properties are functions of temperature within the insulating medium. The symbols  $\gamma_1$  and  $\gamma_V$  are the diffusion coefficients and  $i_1$  and  $i_V$  are the specific enthalpies of the H<sub>2</sub>O liquid and H<sub>2</sub>O vapor respectively.

The net radiative heat flux,  $q_r$ , is required for solving Eq. (1);  $q_r$  is obtained by solving the radiative transport equation. The onedimensional radiative transport equation for an absorbing, emitting, and anisotropically scattering medium from Ref. [12] can be written as

 $\frac{dI(\tau,\mu)}{\mu_{-----}} = -I(\tau,\mu) + \frac{\omega}{2} \frac{1}{-1} I(\tau,\mu') \Phi(\mu,\mu') d\mu' + n^2(1-\omega) I_b(T(\tau)) (2)$ 

Here,  $\tau$  ( $\tau$ =By) is the optical depth,  $\mu$  is the cosine of the polar angle  $\theta$ ,  $\omega$  is the single scatter albedo, n is the refractive index of the medium, I( $\tau$ , $\mu$ ) is the radiative intensity at depth  $\tau$  in the specified direction of  $\mu$ , I<sub>b</sub> is the emitted radiative intensity at depth  $\tau$  as specified by Planck's blackbody intensity function,  $\Phi(\mu,\mu')$ is the scattering function or phase function.

The boundary conditions for the Eq. (2) are given by:

$$\begin{split} \mathbf{I}(0,\mu) &= \rho_{\mathrm{S}}\mathbf{I}(0,-\mu) + (1-\rho_{\mathrm{S}})n^{2}\mathbf{I}_{\mathrm{b}}(\mathbf{T}_{\mathrm{S}}), \qquad \mu > 0 \\ \mathbf{I}(\tau_{\mathrm{O}},-\mu) &= \rho_{\mathrm{D}}\mathbf{I}(\tau_{\mathrm{O}},-\mu) + (1-\rho_{\mathrm{D}})n^{2}\mathbf{I}_{\mathrm{b}}(\mathbf{T}_{\mathrm{O}}), \quad \mu > 0 \end{split}$$

where  $I_b(T_R)=e_b(T_R)/\pi$  and for a gray body  $e_b(T_R)=\sigma T_R^4$ . Here,  $\rho_s$  is the reflectivity of the substrate,  $\rho_\pi$  is the reflectivity of the fiberglass-air interface at the insulation top boundary. It should be noted that  $\rho_{\Pi}=0$  and n=1 for this problem. The radiative heat flux is given by definition as

$$\begin{array}{rcl} 2\Pi & 1 \\ q_{r}(\tau) \approx \int \int I(\tau, \mu) \mu d\mu d\phi \\ 0 & -1 \end{array}$$

For one-dimensional unsteady moisture transport, the species conservation equation of liquid (convection being neglected) can be expressed as

$$\frac{\partial m_{1}}{\partial t} - \frac{\partial m_{1}}{\partial y} = m_{1}$$
(3)

Here,  $m_1$  is the concentration (kg H<sub>2</sub>O/kg of dry fiberglass) of liquid,  $\gamma_1$  is the liquid diffusion coefficient,  $\dot{m}_1$  is the rate of adsorption of liquid (source-term). Similarly, the species conservation equation for vapor without convection can be written as

$$\frac{\partial \mathbf{m}_{\mathbf{v}}}{\partial \mathbf{t}} - \frac{\partial \mathbf{m}_{\mathbf{v}}}{\partial \mathbf{y}} = \mathbf{m}_{\mathbf{v}}$$
(4)

In the above equation, all the terms denote the same as in Eq. (3) excepting that the subscript "v" represents vapor. The source terms in Eqs. (3) and (4) account for adsorption and desorption. With the assumption that the binder is 5 percent by weight in the fiberglass, it is formulated as below

$$\dot{m}_{1} = 0.05 \rho_{f} \frac{\partial x}{\partial t} = -\dot{m}_{v}$$
(5)

# TABLE 1--Radiative and thermal properties used in numerical computations.

٩f	10.92	kg∕m <sup>3</sup>
c <sub>f</sub>	844.4	J/kg-K
k <sub>f</sub>	$a + bT + 8.5537 \times 10^{-5} \rho_f$ $a = 4.97576 \times 10^{-3}, b = 7.00025 \times 10^{-5}$	₩/m-K
ω	0.201	
β	3.70	cm <sup>-1</sup>
Yv	2.6x10 <sup>-5</sup>	m <sup>2</sup> /s
¥1	1.185 x 10 <sup>-9</sup>	m <sup>2</sup> /s
ΔX	- ΔT(K)/667 kg H <sub>2</sub>	D/kg Binder

where X is the mass of adsorbed water per unit mass of the binder. This is an inherent function of time, since it is a function of the insulation temperature. Since properties for the fiberglass binder are not available, it is assumed that the binder acts as zeolite whose adsorption properties of moisture are available. Hence, data available for zeolite were used for computational purposes. Discretization of Eq. (5) yields  $\Delta X$  which can be found using the formula shown in Table 1. Details of the computation of  $\Delta X$  can be found in Ref. [14]. The boundary conditions for Eqs. (3) and (4) are given by:

and  $m_{1} = (m_{1})_{S}(t) \quad \text{at } y = 0$   $m_{1} = (m_{1})_{O}(t) \quad \text{at } y = y_{O}$   $m_{V} = (m_{V})_{S}(t) \quad \text{at } y = 0$   $m_{V} = (m_{V})_{O}(t) \quad \text{at } y = y_{O}$ 

Because of the complicated nature of the source terms in the energy equation (its coupling with moisture transport terms and radiative transport terms), an iterative (numerical) technique was employed. It has been demonstrated in Refs. [1], [2] that a control volume based finite difference technique outlined in Ref. [13], is an effective tool for solving the energy equation. This same technique was utilized here in solving the energy equation and species equations simultaneously.

The system of discretized equations was obtained by the control volume method of Patankar [13] for the energy equation and the species equations for liquid and vapor. These discretized equations were solved simultaneously using an iterative scheme. The radiative heat flux  $q_r$ , present in the energy equation, was obtained by solving the radiative transport equation; the solution procedure can be found in Refs. [1, 2, 12]. The transient solution is obtained by marching through time with an appropriate time step (10 min per time step). The largest time step that yielded no change in the converged temperatures was considered the appropriate time step.

Experimental (FSEC) data for boundary temperatures were used to obtain mass transfer and heat transfer results. The radiative properties (extinction coefficient and albedo) and other thermal properties (density, specific heat, and conductivity) used for calculating the heat transfer results are shown in Table 1. The detailed analysis for the present problem can be found in Ref. [14].

#### RESULTS

Results for four different cases and their comparison are presented. These include summertime cases with various types (thicknesses) of fiberglass batt insulations. Heat flux results for all cases are compared with the experimental measurements made at FSEC [15] for validation of the present model. Also, results are compared with that of the dry model (no mass transfer) to demonstrate the influence of moisture. Heat transfer through standard fiberglass insulations as well as fiberglass insulations with radiant barriers are presented.

Data for boundary temperatures and boundary relative humidities (wetbulb temperature) were provided by FSEC for summertime conditions. Wetbulb temperatures at the substrate (i.e. bottom of insulation) were not known for the summertime conditions. Hence, the wetbulb temperature at the substrate was computed based on the wetbulb temperature at the top of the insulation so that the relative humidities of the top and the bottom have a known trend. It was shown in Ref. [16] that the relative humidities of the top and bottom were out of phase and the minimum relative humidity of the top surface shifts forward in time in relation to the peak relative humidity of the bottom boundary. The standard fiberglass insulations that are studied here are R30STD (9.5-inch thick batt) and R19STD (6-inch thick batt). The fiberglass insulations Rl9RBS (6-inch thick batt with foil radiant barrier) and RllRBS (3.5-inch thick batt with foil radiant barrier) were also studied. A detailed discussion of the results is presented only for R30STD; however, the comparative results (see Table 2) of heat fluxes for all the cases are presented.

Data obtained from FSEC [15] for R30STD (a standard 9.5-inch fiberglass insulation without radiant barrier) on September 30, 1985 were analyzed by the present model. Data were obtained at 15-minute intervals for a 24 hour period. The data included boundary temperatures (drybulb) (shown in Fig. 2), wetbulb temperatures at the top of the insulation, and substrate heat flux recorded as functions of time.



FIG. 2--Temperature-time histories data provided by FSEC [15] for R30STD (Standard fiberglass insulation without radiant barrier) for September 30, 1985.



FIG. 3--Relative humidities computed from wetbulb temperatures as functions of time for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.

Figure 2 shows the drybulb temperature data for September 30, 1985 as a function of time. The roof temperature and the temperatures of the top and bottom of the insulation are the inputs for solving the energy equation and radiative transport equation. As shown in Fig. 2, heating of the roof and the insulation top boundary begins just before 8:00 a.m. and the temperatures of the roof, the top and the bottom of the insulation approach being steady at about 6:00 p.m. The substrate temperature is essentially constant during the day.

Boundary conditions for solving the moisture transport equations require that either the relative humidities or the wetbulb temperatures must be known at the insulation top surface and the substrate as well. The FSEC data included only the wetbulb temperature at the top surface. Therefore, the wetbulb temperature at the substrate is parameterized. The relative humidity at the substrate is computed from the wetbulb temperature. Non-availability of this data at the time of this work forced parameterization of the wetbulb temperature/ relative humidity. One encouraging factor is that the relative humidities of the top surface and the substrate follow a known trend [16]. It should be noted that the experimental data in Ref. [16] was used only to check the trend and it was not used in the computations of the present model. As shown in Fig. 3, the relative humidities are out of phase and there exists a shift between the peaks. The relative humidity peak at the substrate is advanced in time relative to the top surface minimum.

Using this data for boundary conditions, total heat fluxes were predicted by employing the present numerical model. Also, the fluxes

were predicted for the dry insulation case (no mass transfer). A comparison of the predicted substrate heat fluxes (both dry and with moisture) with the data supplied by FSEC is shown in Fig. 4. It is evident from Fig. 4 that the predicted heat fluxes are in very good agreement with the experimental data. The inherent oscillations present in the data are somewhat predicted by the model. As shown in Fig. 4, the predicted heat fluxes for the dry insulation case (no mass transfer) do not agree well with the experimental fluxes. The behavior of the dry insulation case variation is very smooth and does not predict the heat flux oscillations shown in Fig. 4. It is important to note that there is a dip in the experimental heat flux data at about 8:00 a.m.. This is well predicted by the present moisture model, but the dry insulation model does not predict this dip. In this analysis, the diffusion coefficient of vapor was parameterized. For a vapor diffusion coefficient of  $y_v=2.6x10^{-5}$  m<sup>2</sup>/s which is the value for diffusion of water vapor through air, the magnitude of the dip in the flux agrees well with the experimental value. An attempt will be made later to explain this phenomenon. The heat flux is positive between 10:00 a.m. and 6:00 p.m.(i.e. heat being transferred from the attic into the room). From 6:00 p.m. onwards, the heat transfer tends to be near zero.

Figure 5 shows the influence of the vapor diffusion coefficient. It can be observed that with an increase in the value of the diffusion coefficient, there is an increase in the magnitude of the dip (at about 8:00 a.m.). An overall shift of the curve towards the right is observed as the diffusion coefficient increases. Figure 6 depicts the spatial concentration variation of liquid  $H_2O$  within the batt. A steep concentration gradient is seen at about 4 mm thickness from the bottom and the top; otherwise, the concentration seems to be almost constant. Since the value of the liquid diffusion coefficient is very low, the liquid diffusion is not very important. As shown in Fig. 6, during the heating period (at t=14.00 HR), the liquid concentration was decreasing towards the top of the insulation indicating a probable desorption of moisture. It was found that liquid diffusion has a negligible influence on the heat flux, because of the low specific enthalpy of liquid and the low diffusion coefficient of the liquid. It was also found that the heat of adsorption/desorption did not have a strong influence on the heat flux. Moisture adsorption/desorption mainly causes generation/ consumption of the vapor which in turn impacts the diffusion of vapor.

Spatial vapor concentration profiles are shown in Fig. 7. The vapor concentration near midnight (t=0.25 HR) seems to be almost constant in the insulation. But at about t=8.25 HR, there is a significant concentration gradient with a maximum concentration at the bottom of the insulation and minimum concentration at the top. This seems to cause the dip at about 8:00 a.m. in the heat flux as shown in Fig. 4. Though the temperature of the top the insulation is greater than the substrate temperature at that time (t=8.25 HR), the flux assumes a negative value. This denotes that heat transfer occurs from the substrate into the attic. This heat transfer from the substrate into the diffusing vapor which is now (t=8.25) diffusing from the bottom to the top of the insulation batt.



FIG. 4--Measured substrate heat flux-time histories and predicted flux-time histories for both moisture and dry cases for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.



FIG. 5--Effect of the vapor diffusion coefficient on heat fluxes for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.



FIG. 6--Spatial liquid  $H_2O$  concentration (kg  $H_2O/kg$  of dry fiberglass) profiles for three different times for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.



FIG. 7--Spatial vapor concentration (kg  $H_2O/kg$  of dry fiberglass) profiles for three different times for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.



FIG. 8--Spatial temperature profiles for the moisture case at three different times for R30STD (fiberglass batt without radiant barrier) for September 30, 1985.

Spatial temperature profiles for the moisture case are shown in Fig. 8. At time t=0.25 HR, the temperature is decreasing towards the top of the insulation which in accordance with the trend of temperatures in Fig. 2. For the cases t=8.25 HR and t=14.0 HR, the temperatures are monotonically increasing towards the top of the insulation. Also, a temperature inversion is observed near the top of the insulation, which according to Rish [1], can be attributed to radiative absorption and re-emission by the fiberglass. It was found that moisture had no significant effect on the temperature profiles as predicted by the model for both the moisture and no-moisture (dry insulation) cases.

The predicted substrate heat fluxes were integrated over time for a 24 hour period for all the cases to determine the heat transfer for a full day. By comparing the integrated heat flux for the moisture case with that of the dry insulation case (no mass transfer), the effect of moisture, whether it is detrimental or helpful can be determined. The total heat fluxes for various fiberglass insulations for the summertime are shown in TABLE 2. The negative sign in the percentage denotes savings with moisture included. For R30STD and R19STD (standard fiberglass insulations without radiant barrier) on a summer day, the heat transfer (heat gain) into the room is found to be lower with moisture included by about 12.3 percent and 13.2 percent respectively; this is for the data provided by FSEC. There was no pertinent data available for the wintertime situation so as to assess the influence of moisture on heat transfer during winter. At least. for summertime, it may be concluded that the presence of some moisture appears to be helpful in reducing heat gains for standard fiberglass insulations. For insulations with a roof mounted foil radiant barriers) analyzed for a summer day, it is found that heat transfer into the room (heat gain) is reduced by 25.8 and 38.0 percent respectively for R19RBS and R11RBS when moisture was considered. These magnitudes are much higher than that of R19STD and R30STD. A valid comparison may be between R19STD and R19RBS since these insulation batts have the same thickness and the data used corresponds to the same day. Moreover, attention has to be focused on the absolute reduction in the total heat flux rather than the percentage reduction. It can be seen from TABLE 2 that the absolute reductions in the total heat flux due to the presence of moisture for R19STD and R19RBS are 15.0 kJ/m<sup>2</sup> and 14.7 kJ/m<sup>2</sup> respectively. This indicates the consistency in the model's predictions.

Though the cases studied here are few, tailoring an insulation depending on the climatic conditions based on these results may be significant. It could be possible that an insulation with a binder that can adsorb moisture could be used in a hot and moist climate for an advantage. In climates like Arizona, where it is hot and dry in summer, insulation with a binder and forced attic humidification could reduce heat transfer. From these results, it can be seen that using a radiant barrier in the summer, more energy can be saved. Likewise, with the knowledge of the local climatic conditions, a prudent judgment can be made selecting an optimum insulation.

## CONCLUSIONS

Heat transfer results were compared with the experimental data provided by the FSEC. Predicted substrate heat fluxes with moisture present, were in good agreement with the measured heat fluxes. Results indicated that heat transfer from the attic into the house was reduced by about 12.3 percent for R30STD and about 13.2 percent for R19STD during summer when moisture was present. The heat transfer was found to decrease significantly through the fiberglass insulations with a roof mounted foil radiant barrier for summer conditions. Heat transfer from the attic into the house was reduced by 25.8 percent for R19RBS and by 38.0 percent for R11RBS.

Temperature inversions were observed near the top surface of the insulation which can be attributed to radiative absorption and reemission by the fiberglass insulation. These inversions were seen for the standard fiberglass batts (without a roof mounted foil radiant barrier) in summer as well as winter. The presence of moisture had no significant influence on the temperature profiles. The vapor diffusion coefficient which was parameterized was found to strongly influence the heat fluxes. Moreover, this numerical model may be useful in tailoring a residential attic insulation for a known environment.

It should be noted that the present study focused mainly on understanding the moisture phenomenon in the attic insulations. Also, the results presented were only for summer and did not cover a wide range of seasons. Future work will comprise of improving the model with the adsorption/desorption properties of phenolic (fiberglass binder), testing and validation of the model with a wide range of data, detailed tailoring of insulations for a variety of situations, and estimating the economic impact.

TABLE	2Comparison	of	predicted	24-hour	integrated	substrate	heat
			flux	kes.			

Date	Insulation	Moist case (kJ/m <sup>2</sup> )	Dry case (kJ/m <sup>2</sup> )	Difference (percent)	
Sept.30, 1985	R30STD	36.3	41.4	-12.3	
Sept. 2, 1985	R19STD	98.0	113.0	-13.2	
Sept. 2, 1985	R19RBS	42.3	57.0	-25.8	
Sept.30, 1985	R11RBS	27.5	44.4	-38.0	

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**Performance Factors I** 

# V. Freitas, P. Crausse and V. Abrantes

MOISTURE DIFFUSION IN THERMAL INSULATING MATERIALS

REFERENCE: Freitas, V., Crausse, P., and Abrantes, V., "Moisture Diffusion in Thermal Insulating Materials," <u>Insulation Materials:</u> <u>Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

moisture behaviour of building materials, ABSTRACT: The especially the kinetics of wetting and drying, is very important because it can influence the hygrothermal performance of the building envelope and the internal comfort. The mathematical model of heat and mass transfer, in porous media, proposed by LUIKOV - PHILIP and DE VRIES shows that moisture migration, in isothermal conditions, is a function of the humidity gradient and the moisture diffusivity coefficient. This paper reports values of moisture diffusivity coefficients for some thermal insulating materials and how these are calculated. The results of numerical simulation of drying of these materials and a comparison with the data obtained from measurements using gamma-ray spectroscopy in the determination of the moisture distribution are also presented.

KEYWORDS: moisture, moisture migration, moisture diffusivity coefficients, insulating materials, drying, wetting, gamma-ray spectroscopy.

# INTRODUCTION

The envelope of a building interacts with moisture that originates from either the interior or the exterior air mass.

Moisture can damage the building elements and significantly diminish their thermal performances.

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Knowledge of the mechanisms of moisture transfer, especially the kinetics of wetting and drying and its modeling are important for the evaluation of the behaviour of the materials and components that form the envelope of the building.

The materials used in building construction and, in particular, the thermal insulating materials can have moisture content values varying from the zero to saturation. However, these two situations are rarely reached in natural conditions.

A conventional description of the different moisture values that a material can have, is presented in Figure 1. It is easily possible to distinguish the zones that correspond to the hygroscopic and capillary domain, which form the domain of usage of the materials and of application of the mathematical model.





The hygroscopic moisture - Wh can be zero if the material is in contact with an ambient environment with zero relative humidity. The maximal moisture value, Wmax, corresponds to a value of moisture that, in volume, is the same as porosity. The saturation moisture, Wsat, is obtained when the sample immersed in water and reaches stabilize mass. Finally, the critical moisture, Wcr, is the value of moisture of a material in contact with a saturated ambient environment.

In this paper we present the formulation proposed by LUIKOV -PHILIP and DE VRIES for the modulation of mass transfer, adapted to the conditions of the isothermal regimen, that led to the mathematical model. We describe how the moisture diffusivity coefficient. Dw. is determined for building materials.

We also describe the experimental apparatus, available in the Hygrothermal Laboratory of the Faculdade de Engenharia da Universidade do Porto, for the determination of the values of moisture in the interior of the samples, which is based on the attenuation of radiation (gamma-ray spectroscopy).

We give data for the moisture diffusivity coefficients of different thermal insulating materials which represent needed information for the modeling of moisture transfer. It is important to note that a more complex study about the heat and mass transfer for wetting and drying in multilayer walls is being developed in Oporto University. It will also allow the study of the moisture influence on their thermal performances.

The current results permit a comparative analysis of the drying of a cellular concrete sample using the numerical simulation and experimentation, which leads to a good agreement with the results. Results of the drying of mineral wool are also presented, and the influence of moisture diffusivity coefficient in the variation of the water content profiles is analyzed.

#### MATHEMATICAL MODEL

# Numerical solution of the equation for moisture transfer

Several authors have already presented the numerical solution of the equation for heat and mass transfer (1,2,3) and, in particular, the equation for mass transfer under isothermal condition (4).

The equation for transfer that states the phenomena in study is the following:

$$\frac{\partial w}{\partial t} = \frac{\partial}{\partial \chi} \left( D_w(w) \cdot \frac{\partial w}{\partial \chi} \right)$$
[1]

where:

W— Moisture content
$$(kg/kg)$$
 $D_w(w)$ — Moisture diffusivity coefficient $(m^2/s)$ 

The influence of the gravity forces has been negligible, but is important for several building materials.

Since the governing equation is non-linear, it is customarily solved by the method of finite difference or of finite element.

For the solution of this equation, a knowledge of the derivative of W(x) with respect to the interval  $(0, +\infty)$ , and the second derivative with respect to space over the interval (0, L), that satisfies equation (1), is needed.

The method of finite differences that has been used, aims to replace the differential operators in relation to space, by using Taylor development:

$$U_{i\pm 1} = U_i \pm \left(\frac{\partial U}{\partial \chi}\right)_i + \frac{\Delta \chi^2}{2} \left(\frac{\partial^2 U}{\partial \chi^2}\right)_i \pm \dots \qquad [2]$$

To resolve the equation in relation to time, we used Crank-Nicholson's method, that is:

$$\chi^{m+\theta} = \theta \cdot \chi^{m+1} + (1-\theta) \cdot \chi^{m} \quad \text{, with } \theta = \frac{1}{2} \qquad [3]$$

In each interior node there is an equation and three variables. The two equations of the interface nodes, in contact with the ambient, are the necessary conditions to determine the n equations correspondents to the n system nodes. The linearization is obtained by calculating the coefficients Dw in the moment m+1/2  $\Delta t$ .

The system solution is achieved by using the Gauss elimination method.

# Solution of the equations nodes in contact with the ambient

In the exterior nodes, that are in the plane that separates the material from the ambient, a condition of flow during the drying is imposed:

$$\rho_{s} \cdot D_{w} \frac{\partial w}{\partial \chi} = \beta \left( PS_{suf} \cdot HR_{suf} - PS_{ext} \cdot HR_{ext} \right) \qquad [4]$$

where:

ρ	-	Density of dry material	(kg/m³)
ß	—	Surface mass transfer coefficient	(m/s)
PS	_	Saturation vapour pressure	(kg/m <sup>3</sup> )
HR		Relative humidity	(%)
suf	—	Surface	
ext	—	Exterior	

The exterior humidity conditions are given for each time step by the relative humidity values. The relative humidity in the interface is calculated using the limit nodes values of humidity in the material sorption/desorption curves.

## Moisture Diffusivity Coefficient

The described solution of the equation for mass transfer requires knowledge of the moisture diffusivity coefficient, Dw, that expresses the migration humarcy - vapour and liquid phases - under the moisture gradient. The coefficient is variable and has been experimentally determined by several authors (4,5,6,7,8).

The method that we use for the determination of the cellular concrete coefficient, Dw, consists in obtaining the experimental profiles, in wetting, by the experimental apparatus of attenuation of the gamma-ray, that is described later. The different profiles calculated in several moments can be transformed in unique type-profile using the Boltzman's variable  $U = x/\sqrt{t}$ . Figure 2 presents the cellular concrete wetting type-profile.

The coefficient Dw can be calculated from the type-profile (5) based in the following equation:

$$D_{w} = -\frac{1}{2\frac{\partial W}{\partial U}} \int_{W_{1}}^{W_{2}} U \, dw$$
 [5]



FIG. 2 - Experimental wetting of cellular concrete - type-profile

In Figure 3 and 4 some experimental values for the moisture diffusivity cellular concrete and mineral wool coefficients-Dw are presented.



FIG. 3 - Moisture diffusivity coefficient-Dw of cellular concrete



FIG. 4 - Moisture diffusivity coefficient-Dw of mineral wool (7)

A data base on other thermal insulating materials is presented in Figure 5, with values of Dw for expanded polystyrene, polyurethane and phenolic foam.



of insulating materials (7,8)

## EXPERIMENTAL APPARATUS

The experimental measurement apparatus of moisture content values, that we have in the Hygrothermal Laboratory of the Faculdade de Engenharia da Universidade do Porto, consists of:

- Radioactive source;
- Detector;
- Electronic measurement;
- Metallic support apparatus.

The disposable radioactive source is the Americium 241, that has an activity of 100 miC and an energy of 60 KeV. This source has the advantage of having a high resolution.

The detector and its protection, and also the electronic measurement were made and built in Portugal by the LNETI-National Laboratory of Industrial Engineering on request by the GCC-FEUP.

The metallic support structure allows the detector and the radioactive source to be moved horizontally, with a precision of about 0.5 mm (see Figure 6).



FIG. 6 - Partial view of the gamma-ray equipment

Both the detector and the radioactive source have collimators, with a diameter of 5 mm, that leads to a reduction of the measuring errors by reducing particle divergence

# Measuring principle

The method is based on the principle of attenuation of gammaray to determine the values of moisture content in porous media has been used by several authors (5,7,8). The method allows the detection of space and time variations of moisture depending on the relation of intensity of emitted and transmitted radiation. The relation between these two intensities is expressed by the following expression:

$$\mathbf{I} = \mathbf{I}_{0} \cdot \mathbf{e}^{-\mu\rho\chi}$$
 [6]

where:

I - Intensity of transm	itted radiation	(counts/s)
I Intensity of emitted	d radiation	(counts/s)
$\mu$ — Absorption coefficient	ent of the material	(m²/kg)
$\rho$ — Density of the mate	erial	(kg/m <sup>3</sup> )
$\chi$ — The thickness of th	e material sample	(m)

The porous materials have the form of a solid skeleton containing a liquid phase, and surrounding them, when they are being tested by several impermeable layers. Under these circumstances the relation between the emitted and the transmitted intensity is given by the following expression:

$$I = I_o^{\bullet} e^{-\mu} w^{\rho} w^{\chi, \theta}$$
[7]

where:

Io* — Intensity of radiation transmitted	
by the impermeable sample	(counts/s)
$\mu_w$ — Absorption coefficient of the water	(m²/kg)
$\rho_{w}$ — Water density	(m²/kg)
$\theta$ — Moisture content in the volume	$(m^{3}/m^{3})$

The value of moisture content can easily be calculated from the latter equation, as long as we keep in mind the coefficient of water absorption :

$$\theta = -l_n \frac{I}{I_o^*} \cdot \frac{1}{\mu_w \rho_w \chi} \quad (m^3/m^3); \quad w = \frac{\theta}{\rho_s} (kg/kg) \qquad [8]$$

Nielson (9) and Kumaran-Bomberg (10) experimentally measured the absorption coefficient for water to the Americium 241 source. They found identical values  $\mu = 0,0192 \text{ m}2/\text{kg}$ .

## RESULTS

Since our experimental device obtains water content profiles, we proceeded to the compare the theorical and experimental results to validate the mathematical model. We will present results that characterize the kinetics of drying and the influence of moisture diffusivity coefficient on the development of the water content profiles.

Figures 7 and 8 show good agreement of experimental water content profiles of drying of a cellular concrete specimen and by using a numerical simulation, which shows a good agreement between the mathematical model and the experimentation, is presented.

The conditions of test and simulation are the following:

- Previously saturated specimen;
- Relative humidity of the ambient at 50%;
- Mass superficial transfer coefficient  $\beta$  = 0.00067 m/s (experimental value).



FIG. 7 - Drying water content profiles of cellular concrete -Experimentals results-

The samples used have a section of  $0.07 \times 0.07$  m2 and were made impermeable at four faces with an aluminium adhesive band in order to assure the one dimensional flow of moisture.

It was extremely difficult to assure a uniform saturation of the samples of cellular concrete in the longitudinal direction, and there was a decreasing moisture gradient from the sample core to the surface. However in the numerical simulation, the initial conditions

W kg/kg



FIG. 8 - Drying water content profiles of cellular concrete -Numerical simulation results-

of the test were reproduced. As the test was done with the sample horizontally, profiles were determined at different sample heights, and it was verified that a significant influence of gravity does not exist for this material.

Analysis of the results shows that during the first phase of the drying, the profiles are almost linear and horizontal. The same results were achieved with micromodels (11). This means that the moisture transported to the surface is equal or greater than the quantity that goes to the ambient. During the second phase, there is a quicker drying near the surface due to the diminishing of moisture values, and therefore of the moisture diffusivity coefficient.

The mineral wool drying leads to different drying profiles as can be seen in Figure 9. This numerical simulation result was with the same external conditions as the cellular concrete.

The difference in behaviour between the cellular concrete and the mineral wool, in drying, is due to the fact that the values of the moisture diffusivity coefficient of the materials is different. If the values of Dw are high, then the probability of uniform drying is also higher. If those are low, then there is a higher probability of a quicker drying of the superficial layers, and only after that, an interior drying.



FIG. 9 - Drying water content profiles of mineral wool -Numerical simulation results-

#### CONCLUSIONS

The modeling of the moisture transfer can be an important tool for prediction behavior wetting and drying, of the thermal insulating materials and to anticipate the reduction in the thermal performances of the material due to moisture absorption.

Moisture diffusivity coefficient values, Dw, are presented for cellular concrete, mineral wool, expanded polystyrene, polyurethane and phenolic foam. However results from different authors show some relevant discrepancies due to the variability of the properties of the material tested and the technique of experimental evaluation.

The moisture diffusivity coefficient allows the prediction of the variations of the water content profiles in wetting and drying.

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George A. Tsongas

THE EFFECT OF EXTERIOR INSULATING SHEATHING ON WALL MOISTURE

**REFERENCE:** Tsongas, G. A., "The Effect of Exterior Insulating Sheathing on Wall Moisture," <u>Insulation Materials: Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: For some time the impact of the use of rigid foam exterior insulating sheathing on moisture levels in residential wood-frame walls has been uncertain. Recently a field study of 86 newly-constructed Washington and Montana homes was completed that provides clear evidence. Measurements were made of the moisture content of wood members within the wall cavities, almost one-third of which had exterior insulating sheathing. Unacceptably high moisture levels were measured in the walls of a large percentage of the test homes. However, the use of exterior insulating sheathing was found to greatly reduce wall moisture levels. Furthermore, the higher the R-value of the sheathing, the lower were the wall moisture levels. Those walls were found to be drier because the insulating sheathing keeps the wall cavity wood members warmer and also because the sheathing is an excellent exterior moisture barrier or break that keeps wet siding from transmitting or wicking moisture into the wall cavity better than other exterior moisture barriers. Moisture entry from the outside, called "splashback," is an extremely important and common wall cavity wetting mechanism that heretofore has not been widely recognized. The results of this study indicate that the use of exterior insulating sheathing is one way of providing additional wall insulation while at the same time reducing the potential for wall moisture problems.

KEYWORDS: moisture, walls, residential, insulating sheathing, field study, insulation

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# INTRODUCTION

The use of exterior insulating sheathing has become more common in new residential construction. But questions have remained regarding its impact on moisture levels of wood members within the cavities of above-grade walls. Some hypothesize that the sheathing might act as an exterior vapor retarder that would trap moisture inside wall cavities during the winter in residences located in cold winter climates, leading to excessively high moisture levels within walls. Others argue that addition of insulating sheathing increases temperatures within wall cavities and hence reduces the incidence of condensation and high moisture levels in wall wood members.

Recently a comprehensive, multifaceted moisture field study known as the "Northwest Wall Moisture Study" [1] was completed. Some of the results of that study provide clear evidence regarding this question of the effect of insulating sheathing on moisture levels in walls. In one part of that study the exterior walls of 86 newlyconstructed houses in the Pacific Northwest were cut open from the outside and examined to determine if building them to energyefficient standards with more insulation (at least R-3.3 m<sup>2</sup>-K/W [R-19 ft<sup>2</sup>-h-°F/Btu]) and an air-vapor retarder causes unacceptably high levels of moisture or moisture damage within walls. The test houses were located in three climate regions: 50 in the metropolitan Seattle-Olympia area (designated hereafter as the "metro" region), 16 on the rainy Washington coast, and 20 in the cold Montana region. Almost one-third of the houses (none of the coastal homes) had exterior insulating sheathing.

Measurements of the moisture content of wood members within the wall cavities were first made between January and March 1987. Unacceptably high moisture levels were measured in the walls of a large percentage of the test homes. The criterion for unacceptably high levels was chosen to be any wood moisture content greater than 20% because any time wood members are over 20% moisture content the wood is considered "wet" and there is a possibility of wood decay occurring, although decay only occurs during warmer weather [2]. Actual decay does not start until about 30% moisture content [2], but when 20% is measured other parts of the structure may be wetter. Hence when moisture levels are greater than 30% the wood is considered "very wet" and there is major cause for concern.

Over half of the test homes had at least one wall wood member with over 20% moisture content. The highest moisture content measured was approximately 55%. The 28 homes with the wettest walls were reopened in the same wall locations during July and August 1988 to check for drying. Some of the homes still had elevated wall moisture levels (above 20%). The 16 homes with the highest summer moisture contents above 20% were reopened once more during March 1989. Elevated moisture contents above 20% again were measured.

A detailed statistical analysis of the field data was undertaken to try to determine what factors influenced or "caused" the elevated wall moisture contents. Some specific causes of the elevated wall moisture levels were discovered, including high indoor relative humidities due to lack of control of moisture generated indoors and certain construction practices--especially those that allowed moisture to enter the wall from the outside. It was this latter cause that is being considered in this paper. It is generally believed that moisture enters a wall cavity from the inside during the winter in cold climates, and that certainly is true. However, the results of the Northwest Wall Moisture Study [1], as well as an earlier field study of older homes [3], indicated that when walls get excessively wet, it is most often caused by water entering the wall cavity from the outside. That is a major new and unexpected finding.

As a further correlated result of that statistical data analysis, it was found that the presence of exterior insulating sheathing had a major effect on the moisture contents of wall cavity wood members. This paper describes that important effect in detail.

#### TEST HOME CHARACTERISTICS

A detailed description of the characteristics of the test houses is presented in reference [1]. Selected house characteristics are presented in Table 1. There was a mix of wall vapor retarder types. Most of the cold region houses had a continuous polyethylene airvapor retarder, whereas most of the coastal houses did not. About 60% of the metro houses used an interior polyethylene wrap; an additional 20% used interior foam sheathing. Exterior foam sheathing was used in almost one-third the test homes, but in none of the homes in the coastal region. Often, it was used in combination with interior polyethylene.

					_
	<u>Cold</u>	<u>Coastal</u>	<u>Metro</u>	<u>A11</u>	
No. of Test Houses	20	16	50	86	
Wall Air-Vapor Retarder Type:					
Interior Noncontinuous	0	8	9	17	
Interior Poly, Walls Only	0	3	1	4	
Interior Poly Wrap	19	2	29	50	
Interior Foam Sheathing	1	1	10	12	
Exterior Foam Sheathing	10	0	15	25	
Plywood Sheathing	8	12	23	43	
Fiberboard Sheathing	3	4	0	7	
No Sheathing	0	0	8	8	
ADA (Airtight Drywall)	0	4	2	6	
Exterior Tyvek	7	8	33	48	
T1-11 Plywood Siding	0	5	14	19	
Sealed Receptacles	19	15	38	72	
AAHX (air-to-air heat exhanger)	20	10	43	73	
Dehumidifier	1	6	7	14	
Wall Insul. Mean R (m <sup>2</sup> K/W)	5.8	3.5	3.9	4.2	
(ft <sup>2</sup> h <sup>0</sup> F/Btu)	33	20	22	24	
Wall Insul. Max. R (m <sup>2</sup> K/W)	7.2	4.4	5.3	7.2	
(ft <sup>2</sup> h <sup>0</sup> F/Btu)	41	25	30	41	
Triple Glazing	13	7	24	44	

#### TABLE 1 -- Selected test house characteristics

All of the coastal houses used 2 by 6 single stud walls with R-3.3 (R-19) fiberglass batt wall insulation. The metro homes tended to use similar wall construction, except that half added external or internal rigid foam sheathing for additional insulation. The cold region houses utilized a wide variety of framing types, including single and double stud construction. Eleven of the 20 homes used external or internal rigid foam sheathing. The cold region houses had the highest average wall insulation R-value (foam sheathing plus batts or blow-in or spray-on insulation); they also had the highest maximum of R-7.2 (R-41).

The insulating sheathing was rigid board type and was either isocyanurate with foil facing, extruded polystyrene, or molded polystyrene (beadboard). The insulating sheathing was used in thicknesses from 13 mm (0.5 in.) to 51 mm (2 in.). The interior foam insulation usually was nailed to the inside of the studs, taped at the seams to form an air/vapor barrier, and covered with sheetrock (gypsum board). The exterior foam insulation was typically used like a wood sheathing, being attached to the outside of the studs and then covered with wood siding. Sometimes plywood sheathing was nailed over the outside of the foam or plywood was covered on the outside with building paper, 15# felt, or Tyvek.

All the test houses were electrically-heated (see [1] for details). Heating season natural infiltration air change per hour (ach) values for 34 of the homes were obtained from previous blower door tests using the LBL (Lawrence Berkeley Laboratory) methodology. The mean ach values for the regions are: coastal, 0.22; cold, 0.30; metro, 0.27; all, 0.28. The range of ach values for all the test homes was very broad: from 0.05 to 0.86; the standard deviation for all the test homes was 0.17. Air change rate results were available for only one coastal house, 16 of the 50 metro houses, and 17 of the 20 cold region homes. Thus the mean values may not be representative of all the houses in the metro and coastal regions. In part, given the lack of polyethylene in half of the coastal houses, they are probably leakier than the one 0.22 ach value would suggest.

Occupant lifestyle characteristics, including relative humidities measured in each living room, are described in reference [1]. The average living room relative humidity in the test homes in the cold, metro, and coastal regions was 40%, 47%, and 53%, respectively.

# FIELD TEST METHODOLOGY

Details of the field test methodology, including training, occupant interviews and house inspection, determination of wall opening locations, wall openings, data collection and recording, and sheathing foam insulation sample moisture content determination are presented in [1], as is a discussion of how the test homes were identified and selected. The exterior wall cavities of each house were opened from the outside in three places; the openings were located where the highest wall moisture contents were expected. In each wall cavity opening the wood moisture content of the sole plate, each stud, sheathing, rim joist (header), subflooring, and the sill (mud) plate was measured using a Delmhorst model RDX-1 moisture meter with a wood temperature correction setting and a penetrating electric resistance probe with two insulated pins. Wood moisture contents were determined to the nearest half percent. Measurements were made at one-quarter the member depth from the surface in order to determine the average moisture content. They were also made as close to the coldest outer surface as possible to measure the highest moisture content. In addition, one sample of foam insulation from each wall opening, when present either on the exterior or the interior of the wall, was removed and later analyzed gravimetrically in a laboratory to determine its moisture content by weight.

## **RESULTS AND FINDINGS**

## Moisture Contents of Wall Cavity Wood Members

As noted in the Introduction and shown in Table 2, unacceptably high moisture levels (above 20%) were measured in the walls of a large percentage of the test homes; wall cavity wood members with moisture contents above 20% were considered "wet," whereas those above 30% were considered "very wet." Extensive statistical analysis of the field data was undertaken to try to discover what factors influenced or "caused" the high moisture levels. Details of the statistical analysis are presented in reference [1].

Region	No. Test Homes	Moisture ≥20	Content 3 ≥30	Range (%) ≥40
Coastal	16	15	3	1
Metro	50	29	8	5
Cold	20	8	3	2
A11	86	52	14	8

# Table 2 -- Number of homes with any one wood member with high moisture content

It should be noted that moisture contents were measured in each of the three wall openings in each house in seven different wall cavity locations, including the sheathing. For some of the statistical analysis, the data was broken down into two major categories for each house: the average sheathing moisture content of the three openings and the overall average wall moisture content of all the readings in the three openings. In those houses that had exterior insulating sheathing there usually was no wood sheathing, so the average moisture content typically did not include that for sheathing. In order to fairly compare the effects of exterior insulating sheathing on average wall cavity moisture contents in the statistical data analysis, the averages were calculated without including the sheathing moisture contents.

<u>Wall wetting from the outside</u>: One of the major causes was moisture entering from the outside by what is referred to as "splashback" [1,3]. Splashback involves siding whose lower edges or back sides are not satisfactorily painted or sealed absorbing rain water or melted snow that splashes up from the ground, and that moisture is transmitted into the wall cavity wood members. Based on the statistical data analysis results [1], as well as observations of the condition and moisture content of the different siding materials, houses with plywood panel siding or vertical grain siding or shingles that were unprotected on their bottom edges were especially prone to moisture entry from the outside.

The absence of an exterior moisture or air barrier such as building paper or Tyvek resulted in substantially higher sheathing moisture levels. Relative to not having a barrier, the presence of any type of exterior barrier resulted in greatly reduced sheathing moisture levels; there were no statistically significant differences between types except one. The use of exterior insulating sheathing as a barrier led to significantly lower average wall moisture contents than with any other type of barrier. The results suggest that an exterior barrier of any type should be installed. Clearly it helps keep moisture from entering the wall cavity from the outside.

Incidently, the presence of a vapor retarder such as polyethylene sheet on the warm side of the walls and ceilings did result in slightly lower wall moisture levels. However, the differences were not statistically significant. This gives further credence to the belief that the major source of wet and very wet walls is moisture entering from the outside rather than from the inside.

<u>Effect of amount of wall cavity insulation</u>: A rather disconcerting finding was that increased wall cavity insulation Rvalues (not including exterior or interior insulating sheathing) correlated with increased wall moisture levels. That is most likely because the more cavity insulation there is the lower the temperatures of the outer layers of the wall cavity. Thus, building houses with increased wall cavity insulation levels to improve energy efficiency may pose an increased risk of eventual wall moisture damage relative to houses with less wall cavity insulation. That is especially true of homes in cold winter climates that require a substantial thickness of wall insulation.

Effect of exterior insulating sheathing: A fortuitous result is that the addition of exterior insulating sheathing led to a statistically significant reduction in wall moisture levels relative to those walls without it. Walls with exterior insulating sheathing were the driest, whereas walls with no sheathing were the wettest on average, and the difference was statistically significant. Moreover, the greater the thickness (and R-value) of the exterior insulating sheathing, the lower the moisture contents of the wall cavity wood members.

Furthermore, as shown in Table 3, no walls with moisture content readings greater than 30% were noted among the test homes with exterior insulating sheathing (EIS). For all homes with exterior insulating sheathing the highest moisture content was 28.5%, whereas for homes without it there were many higher readings, ranging as high as 55%. Of the 30 metro and cold region homes that had any moisture content reading equal to or above 20%, only 5 (or 17%) had exterior insulating sheathing. Thus, only 17% of the homes with exterior insulating sheathing had walls with moisture levels equal to or greater than 20%, while 83% of the houses without exterior insulating sheathing had high moisture levels. Clearly the addition of exterior insulating sheathing measurably reduced wall moisture levels.

Incidently, since no coastal region homes had exterior insulating sheathing, they were not included in this comparison. Such comparisons really only should be made on a region by region basis. While there are not big differences in the average wall cavity moisture contents (not including sheathing) between those with and without external insulating sheathing, that is not surprising since the numbers of houses with EIS in the homes and with readings above 20% is very small. Note that the numbers in Tables 2 and 3 appear to be different. That is because the numbers in Table 2 include sheathing, whereas the comparison values shown in Table 3 do not.

Range of Moisture <u>Readings</u>	Region	No. Homes With EIS	No. Homes Without EIS	Avg Moistur <u>With EIS</u>	re Content (%) <u>Without EIS</u>
<20%	Metro	13	12	14.4	14.7
	Cold	7	8	12.0	11.0
20-30%	Metro	2	18	17.9	16.5
	Cold	3	1	14.2	14.7
>30%	Metro	0	5		18.7
	Cold	0	1		23.4

TABLE	3	 Characteristics	of	homes	with	and	without	exterior
		insulating	g sl	neathi	ng (El	[S)		

Overall, as shown in Table 2, 14 homes without exterior insulating sheathing had very wet walls, and 8 had moisture levels above 40%. While 52 of the 86 homes had at least one wood member with a moisture content above 20% (see Table 2), 43 of the 52 did not have exterior insulating sheathing.

Only two of the 28 worst homes (those with any wall moisture content readings above 25% during the winter of 1987) reopened during the 1988 summer and one of the 16 of those homes that still had moisture levels above 20% during the summer that were reopened during the 1989 winter had exterior insulating sheathing. Conversely, 50% of the 42 driest houses (lowest average moisture contents) had exterior insulating sheathing. Other investigations also have found that walls with exterior foam insulating sheathing added were dryer or no worse than with regular wood sheathing [4,5].

It has been postulated that the incorporation of exterior insulating sheathing warms the wall cavity wood members and keeps

them drier because there is less condensation potential. That is likely true. However, the substantial increase of the temperature of the outer edge of the studs or plates at the sheathing--cavity insulation interface when exterior insulating sheathing is used is not well known or appreciated.

An illustrative example is presented to compare the sheathing-cavity insulation interface temperature with and without insulating sheathing. Both a conventional 2 by 4 wall and a 2 by 6 wall were assumed, as were indoor and outdoor air temperatures of  $21^{\circ}C$  (70°F) and -1.1°C (30°F). The walls had standard gypsum board on the inside, wood lap siding, and either 13 mm (0.5 in.) plywood sheathing or isocyanurate rigid insulating board in various thicknesses as exterior insulating sheathing. The R-values for the wall components and the air films were taken from the ASHRAE Handbook of Fundamentals [6], and the cavity insulation--sheathing interface temperatures were calculated using the method outlined in the Handbook [6]. A PC microcomputer program written to predict the a11 interface temperatures is available upon request. The results for both the 2 by 4 and 2 by 6 walls are described below; those for the 2 by 6 wall are illustrated in Figure 1.

The temperature on the warm side of the plywood sheathing was  $8.0^{\circ}$ C (46.4°F) for an uninsulated 2 by 4 wall and  $7.8^{\circ}$ C (46.1°F) for an uninsulated 2 by 6 wall, respectively. Adding standard fiberglass batt insulation to the 2 by 4 and the 2 by 6 walls reduced the cavity insulation--plywood sheathing interface temperature to  $2.0^{\circ}$ C (35.6°F) and  $1.1^{\circ}$ C (33.9°F), respectively. Replacing the plywood sheathing increased the cavity insulation--sheathing interface temperature back up to  $5.7^{\circ}$ C (42.3°F) and  $3.9^{\circ}$ C (39.0°F), respectively. Doubling the insulating sheathing thickness for the two walls changed the interface temperatures to  $8.6^{\circ}$ C (47.5°F) and  $6.4^{\circ}$ C (43.5°F), respectively.

Adding fiberglass batt cavity insulation to the uninsulated 2 by 4 and 2 by 6 walls with plywood sheathing cooled off the sheathing interface temperature by  $6.0^{\circ}$ C  $(10.8^{\circ}$ F) and  $6.7^{\circ}$ C  $(12.2^{\circ}$ F), respectively. However, replacing the plywood sheathing with the same thickness insulating sheathing increased the interface temperatures back toward what they were with the uninsulated wall cavity. Adding the double or 25.4 mm (1.0 in.) thickness insulating sheathing made the interface temperatures about the same as with the uninsulated walls with plywood sheathing. In both the 2 by 4 and 2 by 6 walls changing from plywood sheathing to 25.4 mm (1.0 in.) of insulating sheathing increased the cavity insulation--sheathing interface temperature by  $6.6^{\circ}$ C  $(11.9^{\circ}$ F) and  $5.3^{\circ}$ C  $(9.6^{\circ}$ F), respectively. That is significant warming. Even greater warming would occur with thicker insulating sheathing.

Thus, as this example illustrates, the use of exterior insulating sheathing rather than conventional wood sheathing (plywood or other similar products) dramatically increases the temperatures of the outer layers of wood members in wall cavities. In fact the use of exterior insulating sheathing makes the outer layers of the wood members (studs and plates) about as warm as or warmer than in an uninsulated wall cavity.

That is important because field studies of wall moisture levels in older conventional homes [3,7] have shown that uninsulated walls stay dry, most likely largely because without wall insulation the



a) Plywood sheathing, uninsulated



b) Plywood sheathing, insulated



FIG. 1 -- Temperature profiles for different 2 by 6 wall configurations showing the warm side sheathing temperatures.

heat flux levels are relatively high and the outer layers of the wall stay relatively warm. With warmer wood there is less opportunity for condensation, and the wood should dry faster.

As shown in the example, adding insulation reduces the heat flux, and that results in considerably cooler outer wall layer temperatures. But adding exterior insulating sheathing in place of the plywood sheathing makes the outer layers of the stud and plate wood members much warmer; the outer wall temperatures then behave like those in an uninsulated wall. This example clearly points out that adding exterior insulating sheathing substantially warms the outer layers of wall cavities, thus reducing the opportunity for condensation and elevated moisture contents in wood members.

It also appears that the insulating sheathing acts as an excellent exterior barrier that prevents moisture from getting into the wall cavity from the outside as easily as it does with other exterior barrier types. While it has generally been thought that moisture gets into the wall cavity primarily from the inside during the winter in mild and cold climates, the results of the Northwest Wall Moisture Study [1] strongly indicate that considerable moisture also can get in from the outside, and when it does the wall cavity can experience elevated moisture levels. Perhaps the reason why most of the walls with exterior rigid sheathing were relatively dry is because the sheathing helped keep moisture from getting into the wall cavity from the outside while the air-vapor barrier on the inside of the wall helped keep moisture from getting into the wall cavity from the inside. Only three of the 44 houses with walls that were wet or very wet had both external insulating sheathing and a polyethylene air-vapor barrier.

Fortunately, the addition of exterior insulating sheathing negated the previously mentioned effect of the increased wall cavity insulation. Thus, the use of exterior insulating sheathing is one major way of reducing the potential for wall moisture problems. Its use would be especially wise with walls in cold climates that require large amounts of cavity insulation and also with Tl-ll (plywood panel) or shingle siding that are especially prone to moisture entry from the outside and subsequent elevated moisture levels, particularly since it provides an excellent moisture break between the siding and the wall cavity wood members.

While there has been considerable concern that external insulating sheathing might act as an external air-vapor retarder and trap moisture migrating from inside the house in the wall cavity, that does not appear to be a problem. It is possible that some moisture is trapped that way, but less moisture gets into the wall cavity from the outside--resulting in a net drying effect with the use of external insulating sheathing.

Some of the houses, especially in the metro region, used interior rigid insulation right behind the gypsum board. As might be expected, its use did not correlate with lower moisture levels in the wall cavity wood members. Only 14% of the 42 driest houses had interior foam. It provides neither the wall cavity warming effect nor the exterior moisture break that exterior insulating sheathing does.

# <u>Rigid Insulation Moisture Content</u>

In addition to measuring wood member moisture contents in each

wall opening, samples were taken of rigid foam insulation used as interior or exterior foam sheathing, sealed in a glass jar, and sent to a laboratory for gravimetric analysis of their moisture content by weight (for details, see [1]). In addition, there were a few samples of fiberboard with asphalt-impregnated surfaces. It was used like exterior wood sheathing.

The results of the laboratory analysis of the moisture contents of the foam samples are presented in Table 4.

Insul. <u>Type<sup>a</sup></u>	Location	Region	No. of <u>Samples</u>	Moisture <u>Avg</u>	e Conte Max	nt by W Min	Veight (%) Std Dev
MPS	Interior	Cold	6	<0.10	<0.10	<0.10	0
EPS	Interior	C-M <sup>b</sup>	10	0.74	2.07	0.10	0.56
ISO	Interior	C-M Cold	5 10	1.99 1.77	2.64 3.33	1.33 0.81	0.60 0.89
MPS	Exterior	Cold	4	0,68	1.49	<0.10	0.69
EPS	Exterior	Metro	33	1.85	5.32	0.82	1.01
ISO	Exterior	Metro Cold	41 24	3.95 2.09	14.6 5.55	1.09 0.50	2.28 1.40
FIB	Exterior	C-M Cold	6 2	18.1 7.41	43.0 7.65	10.3 7.16	12.6 0.35

#### TABLE 4 -- Moisture content in wall opening rigid insulation

<sup>a</sup>ISO: Isocyanurate

EPS: Extruded polystyrene

MPS: Molded polystyrene

- FIB: Fiberboard
- <sup>b</sup>C-M: Coastal-Metro

The results indicate the extent to which the different rigid insulations have or hold moisture during the winter. It is presumed the insulations would be drier in the summer, as was the case with the wall cavity wood members. The results are listed for interior applications first and then for exterior applications. Within each interior or exterior group the different types of insulations are listed in increasing order of their average measured moisture content by weight. Samples were taken from the three wall openings in each test house that had rigid insulation.

Generally speaking, both the isocyanurate and the polystyrenes stayed relatively dry. The polystyrenes were considerably dryer than the isocyanurate in both interior and exterior applications in similar climates. The molded polystyrene was much dryer than the extruded polystyrene. As might be expected, the interior samples

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were dryer than the exterior samples. In the case of exterior isocyanurate, the maximum value of 14.6% was for a case where it was applied over a rim joist; the next highest value was 7.29%. The isocynaurate results have been presented for two separate climates because the results were different for the exterior cases and similar for the interior cases. That is to be expected. Finally, the fiberboard sheathing samples acted much like wood sheathing with a substantially higher average and maximum. The asphalt coatings on each surface did not keep moisture out of them.

## CONCLUSIONS

By comparing moisture levels in walls with different amounts of cavity insulation, it was found that building walls with more cavity insulation definitely leads to increased moisture levels in them. Fortunately that adverse finding is balanced by the positive finding that walls with exterior insulating sheathing are significantly drier than walls without it. Almost none of the wettest walls had such sheathing, and half of the driest had it. These walls were found to be drier because the insulating sheathing keeps the wall cavity wood members warmer and also because the sheathing is an excellent exterior moisture barrier or break that keeps wet siding from transmitting or wicking moisture into the wall cavity better than other exterior moisture barriers. Moisture entry from the outside, called "splashback," is an extremely important and common wall cavity wetting mechanism that heretofore has not been widely recognized.

In addition, the in-field moisture content of different types of rigid foam insulation used as both interior and exterior sheathing was measured. Generally speaking both the molded and extruded polystyrenes as well as the isocyanurates stayed relatively dry, whereas the asphalt-coated fiberboard was much more moist and acted more like wood. The molded polystyrenes were the driest, followed by the extruded polystyrenes, the isocyanurates, and the fiberboard.

## RECOMMENDATIONS

The following recommendations are suggested, especially to builders and contracters, to help reduce the potential for long term moisture damage in walls and any associated homeowner complaints, callbacks, or litigation. The small added cost of following these preventative actions is likely to be well worth it.

1. Never install wood siding without a moisture or air barrier behind it, such as building paper or 15# felt, Tyvek, or insulating sheathing. The barrier provides a capillary break between siding that gets wet and the wall cavity wood members and reduces moisture penetration. Walls without such a moisture or air barrier have a greater probability of being wet. Aside from insulating sheathing, which is the best, there is little difference in wall cavity moisture levels between exterior barrier types. Providing an air space behind the siding may be an acceptable alternative means of providing a capillary break, although that method was not examined in this study.

- 2. Use exterior insulating sheathing in place of or in addition to plywood or fiberboard or other wood sheathing to greatly reduce the chance for high wall cavity moisture levels and the long-term potential for wood decay. Such insulating sheathing reduces wall moisture levels by keeping the wall cavity warmer and by providing an excellent moisture barrier or break to reduce wicking of moisture into the wall cavity from wet siding. The greater the thickness, the better the protection. With rigid insulating sheathing sealed at the seams, it is not necessary to install any other exterior moisture or air barrier. That offsets the extra cost of insulating sheathing over other sheathing.
- 3. With siding materials such as T1-11 plywood panel siding or shingles that are unprotected from water absorption on their lower edges, paint or seal them there. Water splashing up from the ground is easily absorbed into those unprotected edges. This has been clearly shown to lead to very wet wall cavities [1]. Walls with these siding materials were among the wettest and have one of the greatest potentials for moisture-induced wall damage. Their performance with exterior insulating sheathing that acts as a good moisture barrier or break should be satisfactory, but that is unproven.
- 4. To help avoid moisture damage from "splashback," maintain a minimum clearance of at least one foot and preferably about two feet between the ground and the bottom edge of the siding. Definitely avoid the use of splashback-prone siding materials if smaller clearances are used.

## SUGGESTED FURTHER RESEARCH

One question that was not directly answered by the Northwest Wall Moisture Study [1] is whether adding exterior insulating sheathing will eliminate the high wall cavity moisture levels that have occurred with the use of panelized T1-11 plywood siding or shingle siding or other vertical grain siding types that are prone to moisture entry from the outside of the wall cavity. While the study results strongly suggest that the addition of exterior insulating sheathing would help reduce wall moisture levels, it would be worthwhile to verify that recommendation with field test results.

Thus it is recommended that a new wall moisture field test be initiated. It would involve continuously monitoring over a full year or two the variation in moisture content of wall cavity wood members in a coastal region test home that was found to have high moisture contents both in the summer and winter and where summer drying is least effective. The major purpose would be to test different exterior wall constructions side by side in the same room of the same house to compare their moisture performance. For example, one wall section could have the problematic T1-11 siding with plywood sheathing, one could have the siding bottom edges sealed, one could have an air space added behind the siding, and another could have unsealed siding with exterior insulating sheathing added. The wall moisture monitoring could be accomplished inexpensively by reopening

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the walls every month and measuring the moisture content of selected wood members with a hand-held moisture meter. The wall cavities could be inspected at the same time to see if any wood decay has occurred. Pin moisture sensors also could be installed in selected wood members of the wall cavities to enable weekly moisture content measurements to be made using a hand-held moisture meter. True continuous monitoring with in-wall moisture sensors and a datalogger would provide more information, but such testing would be more expensive.

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## T. Ashworth and Eileen Ashworth

THERMAL CONDUCTIVITY OF SEVERAL CONCRETES AS A FUNCTION OF MOISTURE

**REFERENCE:** Ashworth, T., and Ashworth, E., "Thermal Conductivity of Several Concretes as a Function of Moisture," <u>Insulation Materials: Testing and Applications. 2nd Volume,</u> <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A simple laboratory system for measuring thermal conductivity has been developed which utilizes a heat-flux transducer on each side of the specimen. Because previous studies have characterized the thermal contact resistance, the need to embed temperature sensors in the specimen was avoided. The specimen can be sealed in a plastic bag. This enables measurement of the thermal conductivity of specimens with varying amounts of moisture. Four different concretes have been studied: a standard concrete, a micro-concrete used for model structural analysis, a standard shotcrete with metal fibers, and a foamed light-weight (cellular) shotcrete. Thermal conductivity increased linearly with moisture up to high moisture levels; the values at 27°C range from 1.00 to 1.85 W/(m-K) for concrete, 1.46 to 2.03 W/(m-K) for microconcrete, 0.98 to 1.83 W/(m-K) for shotcrete, and 0.14 to 0.36 W/(m-K) for cellular shotcrete. Data have a reproducibility of better than 2%. The conduction processes in the materials are discussed briefly.

KEYWORDS: thermal conductivity, apparatus, moisture, concrete, microconcrete, cellular concrete, shotcrete

## INTRODUCTION

The authors have had a long-standing interest in the thermal properties of geological materials. Previous works summarized in Clark [1] and Robertson [2] contain values for dry and saturated rocks which indicate that the thermal conductivity of many rocks is dependent on moisture content; from our own measurements it has become

Dr. T. Ashworth is Professor and Head of the Physics Department and Prof. E. Ashworth is Associate Professor in the Mining Engineering Department, both at South Dakota School of Mines and Technology, Rapid City, SD 57701. apparent that moisture content has a strong effect upon thermal conductivity in some materials [3,4]. Recently, the authors described a testing system which was developed specifically to measure the thermal conductivity of rock specimens taken from standard cores with minimal specimen preparation [5]. This system allows measurements on specimens which are individually sealed in plastic bags and which contain specified amounts of moisture. Also, it is effective for a wide range of solids whose thermal conductivity is in the range 0.1 to 10.0 W/(m-K) and which can be fabricated into disks about 50 mm diameter by 10 mm thick.

Specimens for this apparatus can easily be fabricated from many construction materials. However, the size of the specimen can sometimes be an important consideration. There are advantages and disadvantages of using large or small specimens. Non-homogeneous materials, such as concrete with large aggregate, and some insulating materials, should be evaluated using methods which utilize large The resulting "average" value is then applicable in specimens. engineering design calculations. When a small specimen is used, the variability of thermal conductivity within the material can be determined by measuring a selection of specimens. In addition, for materials in which differential expansion can cause warping, the problems associated with thermal contact can be more serious than the possible non-homogeneity problems. Absolute precision of thermal conductivity data is always important, but, for a study of its dependence on parameters, repeatibility is of greater importance.

There are a number of previous studies of the thermal conductivity of concretes in the literature. Some of the most pertinent to the current work are: Tye and Spinney [6], who measured thermal conductivity of several concretes by guarded-hot-plate (including the effect of moisture), heat-flow transducer, and guarded-hot-box methods; Sparks [7], and Lentz and Monfore [8] measured the conductivity of concrete mortar and expanded shale concrete, respectively, with moisture content over a wide range of temperatures; Moore et al [9] reported the transport properties of limestone concrete; Campbell-Allen and Thorne [10] measured the conductivity of several concretes as a function of moisture by a radial flow method; Valore [11,12] summarized the properties of a wide range of insulating and cellular concretes, including the effect of moisture.

The study reported here was initiated because data were needed for the engineering applications of heat flow into underground openings. Concretes are often required to give stability in excavations and could be used simultaneously to provide thermal insulation. Since the conditions underground can be dry, saturated or anywhere in between, thermal conductivity values were needed for a wide range of moisture The concretes studied were chosen because of their contents. availability to the authors, their applicability to engineering problems, and, in one case, the availability of previous data using another method [13,14]. These particular materials did not present any serious problem relating to specimen size. One material (the worst case) was stated "to contain 3/8" aggregate", but, as can be seen later in Fig. 2, the aggregate was generally considerably smaller than the stated size and certainly did not penetrate the thickness of the specimen.

#### DESCRIPTION OF THE APPARATUS

Conceptually, the thermal conductivity stack of the system combines the principles of a comparative system [15,16,17] and a heatflow transducer system [18]. Although developed independently, the resulting system is similar to the ASTM designation F433-77 [19], but with two significant differences:

- in the present apparatus, heat flux transducers (HFT's) are placed on BOTH sides of the specimen, and
- 2) contact resistance is considered in a different manner -- in the present work, it is treated as PART of the measured thermal resistance and then subtracted, whereas in F433 it is included in the calibration factor.

Fig. 1 shows a photograph of the stack; details of the fabrication of the components and the operation of the system are given in [4,5]. Two HFT's, between which the specimen is placed, and a heater assembly are the primary components. Each transducer consists of 1.27 mm thick disk of nylon (of known thermal conductivity) sandwiched between two copper disks of thickness 3.81 mm. Extreme care was taken to ensure the faces of the disks were both parallel and flat. Highly stable thermistors are located in the copper disks. Leads from the thermistors are attached to electrically isolated copper posts at the edge of the disks for thermal anchoring and ease of attachment of the external leads. The thermistors are calibrated using a thermometer calibrated by the National Bureau of Standards. Any change in calibration vertically shifts the thermistor calibration curve; thus, temperature differences used in data analysis are still determined



FIG. 1 -- Photograph of the thermal conductivity measuring stack.

Shown, from top to bottom, are the stiff testing machine head, the nylon spacer, the copper heater disk, the upper heat flux transducer, the specimen in a plastic bag, the lower heat flux transducer, and the stiff testing machine hydraulic anvil. accurately. HFT outputs are attached to a chart recorder to monitor when steady state is attained. When the stack temperature has reached steady state, the resistance of each thermistor and the heater input power are measured directly using a Keithley 178 DMM.

Without direct installation of thermocouples in the specimen surfaces, the treatment of the thermal contact resistance becomes a major consideration in the design and operation of the stack. In the past, materials such as greases [20] and heat-sink compounds [18] have been used to minimize contact resistance. The principle used here is not to minimize the contact resistance but to minimize its variability. A known contact resistance can then be subtracted from the measured total thermal resistance to give the thermal resistance of the specimen. This approach has been successful in reducing the uncertainty in the contact resistance and has eliminated problems associated with the viscosity of greases [20] and specimen contamination. Several candidate interfacial materials were tried. The most successful in terms of repeatability is Teflon tape, which is about 0.025 mm thick. Interfacial resistance is dependent upon a number of factors including the applied load and the roughness of the copper and the specimen surfaces. These have been studied in detail [4,5].

Plastic (Ziploc) bags have also been found to give a constant value of interfacial resistance for repeated specimen mounting/ demounting cycles [3], but the results are not reproducible from bag to bag due to variations in the thickness of the plastic. то determine the interfacial resistance for a specific bag, measurements are taken first with the specimen equilibriated at ambient humidity with Teflon tape as interfacial medium. The measurements are repeated with the specimen enclosed in the plastic bag (without the Teflon tape). This characterizes the contact resistance of the specific plastic bag. Subsequently, measurements as a function of moisture content of the specimen are performed using the same bag. Immediately after each conductivity measurement is made, the bagged specimen is taken out of the stack and weighed within and without the bag to obtain accurate values of moisture content within the specimen and any possible water condensation.

The thermal conductivity stack is mounted between the anvil and head of a stiff-testing machine. It is surrounded by fiberglass insulation, which in turn is surrounded by polystyrene cut to fit the system, to reduce heat losses. Most of the measurements were made using a 141 kg load (the weight of the stiff-testing machine head) on the stack for consistency in the contact resistance. A nylon cylinder was placed above the heater to transmit the pressure and to act as a thermal insulator against upward heat flux. A ball-bearing on the top of the nylon cylinder helps to ensure uniform pressure distribution. For the light-weight shotcrete the use of a lighter load was required; a special 43 kg head was built for this purpose. The large steel components of the stiff-testing machine, which are used as heat sinks, provided some measure of thermal stability. However, the recent addition of an insulated, temperature-regulated enclosure around the machine has greatly improved the thermal stability of the heat sinks. Simple off-on control maintains the air temperature in the enclosure to  $\pm$  0.25°C, with heat-sink temperatures stable to  $\pm$  0.01°C over a period of days.

Careful attention has been given to evaluating heat losses. The controlled environment around the stack serves the function of a guard heater at a constant, known temperature. Thus, radial heat-flows can be estimated both analytically and by numerical modeling [21]; both methods indicated that these losses were significantly less than 5% of the total heat flux. Under these circumstances, the use of HFT's both above and below the specimen largely compensates for any remaining radial heat losses. Sweet [22] analyzed in detail the very similar situation in comparative measurement and found that averaging of the heat flux by upper and lower standard reference materials (SRM's) gave thermal resistance values accurate to 0.07%. As an additional test, values of thermal resistance of the stack with specimens of conductivity of about 1.0 W/(m-K) have been taken with widely varying amounts of insulation around the stack. The results are in accordance with Sweet's analysis.

Calibration curves of the thermistors have been entered into a Quattro-Pro spreadsheet. The data is recorded and analyzed within the spreadsheet, including specimen specifications, regression analysis, and the graphs presented in the results section.

## CALIBRATION OF THE APPARATUS

Initially, the calibration used for the HFT's was estimated using the dimensions of the nylon disk and the thermal conductivity of nylon. Subsequent tests using specimens of nylon and sintered cordierite, both of which had been measured in our laboratory with both guarded and unguarded hot-plate systems, indicated that a small correction was needed. The same correction was indicated by the measurements on both of the materials, and it was consistent with the assumption of a very small contribution to the overall thermal resistance of the HFT's by the epoxy used to cement the nylon to the copper thermometer plates. This modified calibration was then checked by measurements on a SRM specimen (Pyrex 7740 obtained from Lafayette Instruments for which a value of thermal conductivity of 1.15 W/(m-K) was given). Over a period of a year, nine measurements were made yielding an average thermal conductivity of  $1.147 \pm 0.015 \text{ W/(m-K)}$  at  $22.5 \pm 0.5^{\circ}C.$ 

Based upon the recommended values for Pyrex 7740 [23] and the values obtained in the 1985 round-robin [24] in which our laboratory participated, the thermal conductivity of this material at 22.5°C is 1.09 W/(m-K). At the present time, we have chosen not to modify the HFT calibrations to align with this lower value for Pyrex 7740 for a number of reasons: The piece of Pyrex used here was carefully measured at TPRL, Purdue, by Powell's group against their SRM's; values obtained at 50°C in the round-robin varied from 0.95 to 1.2 W/(m-K); some experts in the field believe that the NBS recommended values may be too low (Sweet et al [22] suggest 8% too low). Funds for NBS traceable SRM's have not been available. (Purchase of such materials, and subsequent recalibration of the system, is of high priority and it is hoped to be accomplished in the near future.) However, the nature of the moisture dependence and the values reported here for percent increase in conductivity for a 1 percent increase in moisture content are independent of the system calibration.

## SPECIMENS

#### <u>Preparation</u>

Specimens were prepared by coring bulk samples with a 50 mm diameter diamond drill. The cores were sliced into disks about 13 mm thick using a diamond saw. The disks were then finished on a centerless grinder to provide a disk between 6 mm and 12 mm thick with smooth surfaces which were flat and parallel (to less than  $1^{\circ}$ ). Perpendicularity with the sides was attempted though not critical. The foamed light-weight concrete was not structurally strong enough to withstand this method of preparation. It was prepared by hand using a regular hacksaw, a template, and various grades of emery paper placed on a lapping table.

The equipment used for coring, slicing and grinding required water as a cooling and cutting fluid. Thus, after fabrication, the specimens were conditioned before the series of measurements commenced by allowing them to come to the "ambient" relative humidity within the laboratory (between 20% and 40% at present). Following the measurement with Teflon interfaces, measurements as a function of moisture content began from ambient humidity, from soaked, or from a fully dried state. The term moisture content is being used and calculated as the ratio of the weight of water added to the dry weight of the specimen.

Moisture was added to the specimens simply by soaking them in distilled water. For the highest moisture levels used, soaking was continued until there was no further increase in the moisture content as determined by weighing. With these porous specimens, it is assumed that the highest moisture level thus attained is close to saturation and represents the moisture conditions of many engineering applications. Since, the data are presented as an added weight percent, comparison with other work is possible where the quoted "saturated" data specifies the actual moisture content of the specimen.

Specimen drying procedures were not nearly as straightforward. In previous work on microconcrete, specimens had been dried by placing them in an oven at a temperature slightly above 100°C [13]. This follows the procedure used by hydrogeologists for evaluating the water content of aquifer materials [25]. Heating to these temperatures does invoke some risk of removing structural water molecules, and we wished to avoid this possibility. Thus, an attempt was first made to vacuum dry the specimens. A liquid nitrogen trap was used between a rotary vacuum pump and a desiccator to reduce the partial pressure of water vapor in the desiccator and to prevent oil from backstreaming from the Water removal was accomplished, but test samples of material pump. dried with the specimens showed significant additional weight reduction when baked at 105°C for a few hours. This result indicated that it was necessary to use the baking procedure to dry the specimens thoroughly. Since all the concretes used were "mature", there was little concern for removing structural water. The results for weight loss obtained during the vacuum drying will be discussed below.

Thermal conductivity results presented later indicate that there was no hysteresis in cases where the specimens were wetted and

dried several times as measurements were taken. Repeated cycling may affect the values of thermal conductivity as shown in a weathering study [3] but was not considered in the current study. All the concretes considered were older than 3 years.

## <u>Description</u>

Each sample is described below in as much detail as available for publication. Fig. 2 is a photograph of the four prepared specimens used in this study. It should be noted that the dark areas in the three high density materials are pieces of aggregate, whereas the small circular dark areas in the lightweight shotcrete are pores. A reinforcing metal fiber can possibily be seen slightly below the center of the shotcrete specimen. The density of each specimen is included in Table 1 which contains a summary of the results.

<u>Concrete</u>: The concrete sample was supplied by Dr. Jan Wolski of the Department of Mining, Environmental and Geological Engineering at New Mexico Institute of Mining and Technology. The concrete is being used as part of a research program on the effects of heating mine air, particularly relating to mine fires. He indicates that "it is 2,000 psi concrete made from Portland Cement Type I with 3/8" aggregate. Small molds were poured in December 1987 and then moist cured for 30 days. The specimens were then cored from these molds."

<u>Microconcrete</u>: The microconcrete was prepared at South Dakota School of Mines and Technology in 1974 for a previous research project. Its thermal conductivity was previously measured by a drift technique [13,14].



FIG. 2 -- Photograph of the four prepared specimens.

Microconcrete is a type of concrete used in model analysis to augment numerical simulation. It is used to construct a physical model of a structure which is then tested to simulate actual conditions. Physically, microconcrete differs from normal concrete only in that the sizes of the aggregate are scaled down; it is required to satisfy the laws of similitude and to possess reproducible mechanical properties and geometric stability. The mix design used was 63% aggregate, 25% Portland cement Type I, and 25% water by weight. The aggregate was in a 2:2:1 ratio for #8-16, #16-30, and #30-pan sieve analysis.

The aggregate used in the samples was river sand taken from the Cheyenne River near Oral, South Dakota (a silica sand with negligible clay content satisfying ASTMC Standard 33). "Specimens were poured in a standard 3" concrete mold. They were then cured at room temperature under lime water for 28 days" [13].

Shotcrete: The shotcrete was retrieved from a local underground mine and had been in place in one of the main airways for a number of years. It contains standard aggregate and metal fibers, most of which lie perpendicular to the direction of heat flow in the thermal resistance measurement.

Lightweight Shotcrete: This material was obtained from Eugene H. Skinner at the Spokane Reseach Center of the U.S. Bureau of Mines. It is a non structural grade, lightweight shotcrete which is used as a crushable (frangible) liner with "a 28 day compressive strength of about 100 psi." Preparation and use of the material are described in [26] and "the mix, per cubic yard, was 282 lb of cement (Type III), l16 lb water, and 23.7 cubic feet of water. The water cement ratio was 0.41."

The material was produced using specialized shotcrete equipment manufactured by Putzmeister-Neopor, Nurtingen, Germany. In the mixing process, cement was added to the water and, "after mixing, foam was added for 1 min 48 s from the foam generator. In addition, a recommended amount of  $CaCl_2$  was added to decrease the setting time." NEOPOR foaming agent was used. The specimen was prepared from a test block of lightweight aereated concrete which was produced in 1986.

#### RESULTS

## Drying experiment

Fig. 3 shows the weight losses of the concrete, microconcrete and shotcrete as a function of time under vacuum drying conditions. These results are presented as relative changes with the weight achieved after 4 hours of oven drying as the reference weight. Thus, the shapes of the curves show the amounts of water removed by vacuum drying and the position of the curves indicates the amount removed by heating. Reduction of the vapor pressure of water at the surface of the material provides an environment in which moisture would move under a concentration gradient. Diffusional processes such as this are characterized by exponential changes; for this reason the data are presented in a linear-log form.



FIG. 3 -- Moisture loss versus time for vacuum drying.

The simplest curve to understand in Fig. 3 is that for microconcrete. This shows a single exponential function with a correlation  $r^2$  = 0.997. At first glance, the other curves could be interpreted as straight lines on this linear-log plot, with deviations for drying times above 40 hours. However, since the specimen weight could be determined very accurately, the data presented warrants a closer inspection. When plotted on a linear-linear graph, the shotcrete shows a linear relationship for the first five points, with a  $r^2$  = 0.995, then approximates an exponential for the last four points. By similar use of a linear-linear plot and correlations, the concrete specimen exhibits two different mechanisms of moisture loss with different time constants. Although there is insufficient data for a detailed analysis, these observations suggest that more than one mechanism is involved. These mechanisms could be diffusional processes within the materials, but with the loss rate controlled by, say, an evaporative process at the surface in the initial period for some of the samples.

When this experiment was instigated, its ability to provide insight into the mobility and mechanism of movement of the water molecules within the porous matrix had not been considered; only after it was completed was this appreciated. Repetition of the experiment is planned with an emphasis given to close, evenly-spaced measurements throughout an extended period of time.

## Thermal conductivity

Figs. 4, 5, 6, and 7 show the measured thermal conductivity for the concrete, microconcrete, shotcrete, and lightweight shotcrete, respectively. For specimens with conductivity in the range 0.1 to 2 W/(m-K), the time for the stack to reach (thermal) steady state was 3 to 6 hours, the longer times for the poorer conductors. All thermal conductivity data given in these graphs were taken at 27.0  $\pm$  0.5°C.



FIG. 4 -- Thermal conductivity of concrete versus moisture content.



FIG. 5 -- Thermal conductivity of microconcrete versus moisture content.



FIG. 6 -- Thermal conductivity of shotcrete versus moisture content.



FIG. 7 -- Thermal conductivity of lightweight shotcrete versus moisture content.

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For the data on concrete, particular care was taken to obtain values at carefully spaced intervals as shown in Fig. 4. However, the data were not obtained in a single sequence from dry to soaked. Readings were begun at ambient moisture (1.43%). The specimen was then soaked (10.39%) and subsequently dried by warm air or wetted several times to complete the data sequence. Several of the points shown are in fact double points obtained on different cycles. The fact that this is very hard to discern indicates the exceptional consistency of the measurements obtained with this apparatus, the effectiveness of moisture content control obtained with the specimen containment system, and the absence of hysteresis effects in the specimen. Another striking feature of this thermal conductivity curve is the linearity of its dependence upon moisture content  $(r^2 = 0.999)$ . Above about 8% moisture content, there is a small deviation above linear behavior.

The data shown in Fig. 5 for microconcrete are very similar to those for concrete, but these data points show a very slight curvature. Again, data was taken during several cycles of drying and wetting; several points are again double points in this data set. Scatter of the data is so small that this slight curvature is believed not to be an artifact. Fig. 6 presents the data for shotcrete. Linear dependence is again evident in the moisture range 0 to about 6%. For the shotcrete, there is a strong increase in the thermal conductivity at higher moisture levels.

Measurements on the lightweight shotcrete were the most difficult to perform. The greatest challenge was keeping the fragile disk in one piece throughout the measurement sequence. Even using the head of smaller mass in the stiff-testing machine, the safety factor during specimen placement and loading was quite small. Based on experience gained during preliminary measurements, a scheme was developed whereby carefully measured amounts of water could be injected into the center of the specimen while it remained mounted, under load, in the stifftesting machine. This was accomplished by drilling a 0.17 mm diameter hole radially from the edge of the specimen to its center. A fine hypodermic needle could then be inserted to inject the water. Visual inspection of the technique on a sample outside the stack showed that it was successful for this highly porous and permeable specimen. The amount of water added was determined by weighing the syringe before and after each injection. Immediately following the 49% moisture point, the specimen was removed from the stack and weighed to check that there had not been errors or any significant water losses. Additional points were taken as the specimen was allowed to dry.

The response of the thermal resistance of the stack to the introduction of a "shot" of water was interesting. Immediately, the central portion of the specimen became wet, and shortly there afterwards the thermal resistance decreased. It was expected that this wet central region would thermally "short-circuit" the rest of the specimen, and then, as the moisture became more distributed, the resistance would increase somewhat. In fact, the opposite occurred. As the moisture became more evenly distributed, the stack resistance continued to decrease until a steady state was achieved. Thermal conductivity data for lightweight shotcrete shown in Fig. 7 are the steady-state values.

#### DISCUSSION AND CONCLUSIONS

Placing the specimens in a plastic bag had the advantage that, as the specimen was removed from the thermal conductivity stack, the specimen-bag interface could be inspected through the bag before the interfacial contact was disturbed. Although not quantitative, the lower (cooler) specimen surface appeared to have a somewhat higher concentration of moisture than the upper surface based on sample color. In view of the relationship between the vapor pressure of water and temperature, this was to be expected. This was also consistent with observed moisture distributions under a temperature gradient in porous materials as reviewed by Pratt [27].

It was also possible to see whether any liquid ("free") water was present at the specimen-bag interface. For steady-state measurements there was the concern that there would be progressive condensation at the cooler surface of the specimen as discussed by Woodside and Cliffe [28], and which would lead to dehydration of the interior material. Two pieces of evidence suggest, quite strongly, that this did not happen for the materials studied here. First, the thermal resistance of the stack did not increase with time after a steady state had been reached, even when the specimen remained under measurement conditions for periods of several days. Second, significant liquid water was never observed to collect at the specimen-bag interface, even when the specimen contained large amounts of water and had been in the apparatus for an extended period of time. The sensitive monitoring system, which allowed measurements to be made with small temperature differences across the specimen (usually 2 to  $5^{\circ}$ C), may be one reason for this. The absence of lack of liquid water also suggested that water in these materials was present as "adhesive" water, in de Marsily's terminology [25].

Values of thermal conductivity presented in the graphs and summarized in Table 1 are in agreement with previously reported values for similar materials. The value for the light-weight shotcrete in the dry state is consistent with cellular concrete data summarized by Valore (Fig. 20 of reference [12]). The concrete and microconcrete values are somewhat higher than the conductivity density curve suggested by Brewer [29]; they support Tye and Spinney's contention [6] that the conductivity is higher for higher density concretes. Our

	Densi Kg/m <sup>3</sup> [lb/c	ty u ft]	Conductivity W/(m-K)	Change $\underline{&\Delta(conduct)}$ $&\Delta(moisture)$
Concrete	1938	[120.9]	1.00	7.6
Microconcrete	2074	[129.4]	1.45	5.1
Shotcrete	2057	[128.4]	0.95	5.5
Lightweight	383	[ 23.9]	0.14	3.0

TABLE 1 -- Summary of Results: Thermal Conductivity as a Function of Moisture Content

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data is consistent with values reported by Campbell-Allen and Thorne [10]. Shotcrete values are difficult to compare, since this material contains metal fibres, which increase the density but may have little effect on the conductivity since they are oriented perpendicular to the heat flow; however, the values are in very close agreement with Sparks' data for concrete mortar with 4% and 10.4% moisture [7].

The apparent change in the thermal conductivity of the microconcrete specimen from 0.95 to 1.46 W/(m-K) for the moisture range 0 to 9% reported previously [12,13] to the values shown in Fig. 5 (1.47 to 2.04 W/(m-K) for 0 to 6.8% moisture) are interpreted in terms of an increase in bonding as the material has matured over the 15 year period; this reasoning is supported by the fact that not as much moisture could be added to the specimen in the current study. Percent increases in conductivity per percent increase in moisture values are calculated for the four materials and included in Table 1. They are in good aggreement with the values summarized by Valore [10].

The dependence of the thermal conductivity of concretes upon moisture content that has been observed has many different implications, from modeling detailed mechanisms of heat flow through porous wet media [30,31] to estimating heat loads for large projects [4]. The linear behavior shown by these concretes supports Campbell-Allen and Thorne's conclusion that "the conductivity of any given concrete varies approximately linearly with moisture content" [10]. In practical engineering terms and from the point of view of energy conservation in industrial applications and buildings, the data presented indicates the advantage of incorporating into the design of the structure mechanisms which will keep the concretes as dry as possible. This is particularly important for lightweight materials since heat transfer rates can at least double as the material becomes wet.

It is concluded that the test system and methodology reported here provide an accurate and effective (cost and time) means of evaluating the thermal performance of some materials of importance to the construction industry (surface and underground), the ventilation of hot mines, and the design of underground nuclear waste repositories. This includes a wide range of materials: those of low and intermediate conductivity, those with high porosity, and those with significant moisture content.

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THERMAL INSULATION UNDER FIRE

REFERENCE: Schultz, N., "Thermal Insulation Under Fire", <u>Insulation</u> <u>Materials: Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: This paper describes the different fire and flammability tests where thermal insulations are required to pass by various legal bodies, and all the proposed future test requirements. insulation manufacturers, engineers, architects and contractors should be aware of the current test requirements in this technology, thus enabling them to provide a thermal insulation system that can meet the total spectrum of requirements including the ones for fire and flammability. The standard used to test for fire resistance in building construction is ASTM E 119 "Fire Tests of Building Construction and Materials."

KEYWORDS: thermal insulation, fire resistance, ASTM E 119, flammability, U factors.

The cost of energy will likely increase in the years to come. The cost difference between the years will be the rate of escalation, which might be quite small during some periods and noticeably large during other times. In light of these thinkings, the United States and other major industrial countries are countering this volatility anteceded by implementing energy conservation measures (ECM) to reduce energy usage. The one ECM specified most often is the use of insulation to retard heat flow. This is applied to insulation that is used for equipment or building envelopes.

Architects and engineers constantly require and recommend insulation for new construction, retrofit, or renovation. Also, homeowners want to insulate their homes in most cases as cost effectively as possible. Many of these homeowners are do-it-yourselfers with just enough knowledge to install the insulation. Surprisingly, many professional applicators do not know very much more than the homeowners. If any of these people improperly installed the insulation or performed poor quality work, the best result would be reduced insulation value, thereby forfeiting some of the original objective. The worst scenario is potentially of a greater degree.

Apparently, what can happen in some cases is that unbeknown to the person applying the insulation, a fire hazard can be created. The results created could be anything from placing insulation over an electric lighting fixture to wrapping an insulation blanket too close to the open flame of a water heater. These unintentional dormant catastrophes are growing in occurrence, thus putting an onus on the government and the manufacturers. It follows that responsible professionals such as engineers and architects are also liable for their actions. Bearing this liability in mind, each of the above named parties have taken action to ensure nonculpability where there is a legal responsibility. The method used to

Neil Schultz, Exective Director, VTEC Laboratories Inc., 212 Manida Street, Bronx, New York 10474. indicate the relative fire characteristics of a material is to expose them to a series of tests.

Testing is the only method currently available that can be used to evaluate materials under different fire environments. Usually a discrete value is given as a test result, which then can be used as part of an index to evaluate numerous materials. The test results can be evaluated by a minimum rating required by a building code, which would be used to determine the suitability of the materials for that application.

#### FLAMMABILITY TESTING

Flammability is the behavior of a material when that material is exposed to a fire environment and it burns; flammability can measure (1) ignitability, (2) burning rate, (3) heat evolution, (4) smoke products, (5) products of combustion, and (6) endurance of burning. There are numerous tests which can be used to describe the properties of these materials. Many of them are ASTM standards.

Table 1 -- Flammability tests.

ASTM E 84	Test Method for Surface Burning Characteristics of Building Materials
ASTM E 162	Surface Flammability of Materials Using a Radiant Heat Energy Source
ASTM E 136	Behavior of Materials in a Vertical Tube Furnace at 750º C.
ASTM E 662	Specific Optical Density of Smoke Generated by Solid Materials

ASTM E 84 is probably the test most often specified to determine the flammability properties of insulation. The test data provide flame spread, fuel contributed, and smoke developed numbers. The flame spread index is classified as follows: Class A, 0-25, Class B, 26-75, and Class C, 76-200. The greater the risk of fire hazard occupancy, the lower the number specified. So a material with an A rating would be in a high risk category. ASTM E 84 is one of five test methods being validated in ASTM/ISR's International Fire Standards Project.

There are currently, however, serious shortcomings with this test method. That is, even though a material has a low-flame spread number, it still provides a large contribution to growth of fire in a room. There is also a question as to the accuracy provided where the subjects are low density materials. The smoke index number may not accurately describe the volume of smoke generated. Variation in construction of the test tunnel could affect the reproducibility of the smoke developed and flame spread. Since there is no other standard, the ASTM E 84 remains as the vanguard of flammability testing[1].

ASTM E 162

The tunnel specified by E 84 constitutes an expensive and cumbersome piece of apparatus. There is a need for a less expensive and more convenient flame spread test such as the E 162 radiant panel. The E 162 radiant panel test has numerous advantages over that of E 84. The sample size of the E 162 is 15.2

X 45.7 cm (6 x 18 in.), while E 84 requires a sample 7.6 m x 53.3 cm (25 ft X 21 in.), which is a rather cumbrous large size to deal with. Note, however, four samples are required for the radiant panel, whereas the tunnel uses only one sample. The ASTM E 162 could provide more accurate results in foam insulation that have ratings under 80 of the tunnel, but which prove to be as flammable as conventional materials with tunnel rating of several hundred[2].

The E 162 test method was originally published in 1960. Its unique feature is that a radiant heat source is used to expose the sample to the conditions mentioned previously and that reflects more accurately what might be found in a real fire environment.

A pilot burner is constantly left exposed on the sample to ignite any combustible gases that would be generated. The heat of this would be measured in the stack. Flame spread is observed visually and noted, based on time duration with distance. The E 162 method is typical of most flammability equipment that measures flame spread.

## ASTM E 136

The E 136 standard is the Behavior of Material in a Vertical Tube Furnace at  $750^{\circ}$ C. This test method is used to determine combustion characteristics of building material. A 7.6 cm (3 in.) inside diameter ceramic tube furnace is brought to  $750^{\circ}$ C. A sample measuring 3.0 wide X 3.0 cm. thick (1.5 x 1.5 in.) and 5.1 cm. long (2.0 in.) is inserted into the furnace. The temperature of the samples is recorded and a material passes the test if at least three out of the four samples demonstrate the following:

- The temperature of the thermocouples does not exceed 54°C above the furnace temperature at the beginning.
- 2. There is no flaming of the specimen after the first 30 s.
- 3. The weight loss of the sample does not exceed 50% of the initial weight.

# ASTM E 662

A test very frequently requested for insulation is ASTM E 662: Specific Optical Density of Smoke Generated by Solid Materials. This test procedure is used to determine the specific optical density of smoke generated by solid materials mounted in a vertical position in thicknesses not exceeding 2.5 cm. (1.0 in.). The photometric scale used closely relates to human vision. This method is fairly unique, and most called upon when a determined smoke density is needed.

## Other Flammability Tests

Numerous organizations and government agencies have other flammability test requirements for insulation. The U.S. Coast Guard has fire protection requirements for commercial vessels as well as mobile offshore drilling units. The relevant approval specifications are 46 CFR 164.006, 164.007, 164.008, 164.009, and 164.012.

- 164.006 Deck coverings
- 164.007 Structural insulation
- 164.008 Bulkhead panels

#### 164.009 Noncombustible materials

164.012 interior finish

The categories 164.007 and 164.009 are the most relevant to insulations. The 164.007 is a fire test which uses the ASTM E 119 time-temperature curve with samples that are conditioned and have specific sizes. The 164.009 flammability test uses an apparatus similar to that of E 119 but unfortunately, the test equipment and procedure are sufficiently different so that the E 136 equipment is not applicable.

# FIRE TESTING

The concept of ASTM E 119 fire testing is that the material or assembly being tested withstand fire or give protection from it. The most widely used fire test standard is the ASTM E 119 "Standard Method of Fire Tests of Building Construction and Materials".

The ASTM E 119 was the first established fire test standard and still remains the most prominent for fire testing.

The basic test procedure is that an insulation is exposed to a time-temperature curve as specified by the ASTM E 119 standard. The passing criteria, however, are different depending upon application. In the case of structural steel there are numerous thermocouples attached to the steel measuring the temperature of the steel. When the average of the thermocouples is  $538^{\circ}C$  ( $1000^{\circ}F$ ), or any individual thermocouple exceeds  $649^{\circ}C$  ( $1200^{\circ}F$ ), then the test failure criteria has been exceeded. For walls, partitions, and panels the failure criteria is when the average unexposed side temperature is  $121^{\circ}C$  ( $250^{\circ}F$ ) plus ambient (see Table 2).

The basis for failure for structural steel is that up to 50% of the strength of the steel is lost at  $538^{\circ}C$  (1000°F). The wall and floor failure criteria, as described above have unknown origins. It is known, however, that to prevent the spread of fire it is necessary to contain the heat as well as the fire and hot gases and that excessive transfer of heat can cause material on the unexposed side to combust.

There are conditions where the application of these and the other failure criteria under the ASTM E 119 are not properly covered. One such condition is a fire inside the building which would separate the curtain wall from the building exterior which is usually steel, aluminum or glass. The question is, "will the exterior skin be damaged?" Extensive fire testing indicated that these materials can last for long periods of time when rockwool or mineral wool insulation was placed between the fire source and the material.

Other fire test standards have been used to test insulation. The 1976 DOT-AAP Research Report and 1982 Department of Transportation (DOT) Federal Regulation describe a fire test, where insulation would be exposed to a torch fire for 30 min. The other part of the test is a pool fire test for 100 min. This test is intended to simulate fire conditions of a railroad tank car. Both tests are required by the DOT for using insulation on railroad tank cars.

The Coast Guard test standard identified by 164.007 "Structural Insulation", is the test for the fire resistance of Insulations. This fire standard specified by the Coast Guard is required for insulation in all ships flying the American flag. The test is based on the ASTM E 119, using the same end point criteria and method of measurment.

MANUFACTURER	PRODUCT	DESCRIPTION	COVER PL	ATE FIRE	RATING (h:min)
A	3 IN.(7.	6cm) (	5 PCF	ALUMINUM	1:00
Α	3 IN.(7.	6cm) 8	B PCF	STEEL	0:40
A	2 IN.(5.	1cm) 8	B PC	GLASS	0:30
В	2 IN.(5.	2cm) 8	B PCF	ALUMINUM	1:00
В	4 IN.(10	0.2cm) 4	4 PCF*	STEEL	3:15
8	4 IN.(10	0.2cm) 4	4 PCF	STEEL	1:50
B	3 IN.(7.	6cm) (	5 PCF	GLASS	0:40

TABLE 2 -- Fire test results for mineral wool insulation.

\*Fibers placed edge-wise

Another important fire standard is the high intensity fire test (HIFT), which is recognized by the petrochemical industry as a simulation of a petroleum spill fire. During this test, the temperature reaches approximately  $1093^{\circ}C$  (2000°F) in less than 3 min., and is maintained there for the duration of the test. The heat flux is 111.83 watts/in<sup>2</sup> (5000 Btu/ft<sup>2</sup>-hr) with a radiative component of 101.66 watts/in<sup>2</sup> (50000 Btu/ft<sup>2</sup>-hr), with the remainder of the component being convective. These fluxes have a tolerance of +/- 5%. The failure criteria for this test are the same as that for the ASTM E 119 standard.

There is a series of similar fire tests called room tests. A full size room test has approximate dimensions of 4.7 by 4.7m (12 by 12 ft), 3.9 by 3.9m (10 by 10 ft), or 3.9 by 4.7m (10 by 12 ft) with a ceiling height of 3.1m (8 ft). A standard size doorway is located in one wall. An ignition source is located in a corner opposite the door where its effect will be greatest. The walls or ceiling or both are lined with the insulation being tested. The room is constructed of an incombustible material such as millboard.

The room is set on fire by the ignition source, and the effects of the insulation on the fire are monitored by instrumentation of the insulation being noted by temperature rises. Further information on this test can be obtained from the ASTM E 603 "Standard Guide for Room Fire Experiments." Other versions of this test include half, third, or quarter room, where the room size is smaller and appropriate instrumentation is set up to make all necessary measurements.

## SPECIFYING FIRE AND FLAMMABILITY TESTS

In building construction the first place to check for the necessary fire/flammability requirement of insulation is the building codes. Most building departments clearly specify their minimum requirements for insulation. This does not mean that the architect or engineer or both (A&E) should settle for the minimum necessary requirements. The A&E is responsible to the owner for maximum safety with cost in mind. Also, the A&E could be able to realize additional savings for the owner by obtaining lower insurance costs, which are due directly from additional fire safety. Here again, the A&E must meet or exceed the minimum fire safety standards, while still following the cost versus return criteria of the owner.

The fire/flammability specification becomes more difficult when the code does not exactly define the requirement or there is no test standard for what is required. Safing insulation (mineral wool insulation used for building exteriors) is an example where there is not an exact requirement, yet fire protection is necessary.

Many A&Es find acceptable fire test data on a small scale ASTM E 119 test, (which indicates the integrity of the insulation during and after the test), and not a time-temperature requirement [3].

Another example of vague testing criteria is insulation for exhaust ducts from kitchens. After years of use a hazardous buildup of grease accumulates in the duct. A flash fire can cause combustion of the grease and result in high outside duct temperature, which could have the potential to combust materials outside the duct or even have flames drip out or extrude from the seams. The reverse is also true of a fire outside the duct which can cause the grease inside to ignite and spread throughout the duct system. There is only one test standard for an exhaust duct in this country and that fire test standard is by the Wisconsin Building Department. Lacking in any other standard test, other building departments are accepting this standard. insulation would be a perfect material to place on the outside of the duct to prevent fire spread. Without a more defined standard, it will be difficult to specify fire protection for exhaust ducts. Yet, the A&E must decide on a type of fire protection.

There are numerous other examples of incomplete standard specification, which are contributable to many fatalities and property loss.

## THERMAL INSULATIONS

During the last decade, the thermal insulation industry has seen a large boom and then a slump with a recovery eminent. New applications and materials for thermal insulations are appearing which have to be carefully scrutinized especially in potentially fire hazard environments. Therefore, it is necessary for the A&E to know which thermal insulation to use and how to provide adequate fire ratings. Almost all thermal insulations are usable in similar applications for building construction when the proper modifications are made to increase their fire resistance or flammability properties. The three main materials used as insulators are fiberglass, rockwool (mineral wool), and foams.

## FIBERGLASS

Fiberglass is composed of glass fibers bonded with a thermosetting resin. It is resistant to bacteria or fungus attack. The fibers retain their thickness under operating conditions, and are nonreactive with copper, steel, or aluminum. Standard fiberglass has an operating temperature of  $177^{\circ}C$  ( $350^{\circ}F$ ). Temperatures above that tend to make it shrink. At about  $538^{\circ}C$  ( $1000^{\circ}F$ ), a color change occurs. A yellow fiberglass, for instance, turns white. When the temperature reaches  $816^{\circ}C$  ( $1500^{\circ}F$ ), the fiberglass melts into globs of glass, losing all thermal protection, (before fiberglass loses its physical characteristics, it can contribute to the fire).

A typical flame rating for a fiberglass sample developed per ASTM E 84 specification is flame spread 20, fuel contributed 15, and smoke developed 20. High temperature fiberglass materials are available which will give better fire
performance, however, they are more expensive and not cost effective on a production basis. Generally speaking, fiberglass does not provide adequate protection in a fire. A panel composed partly of fiberglass can lose its physical properties within 10 min, depending on the extent of the fire.

### MINERAL WOOL

Mineral wool is another fire retardant material. The mineral wool products, like the fiberglass products, come in different densities. Maximum densities for the mineral wool obtainable in a rigid, semiflexible or flexible 144.18 kgs/m<sup>3</sup> (9 lb/foot <sup>3).</sup> Operating temperatures and form are about melting temperatures are much higher than that of fiberglass. At 649°C (1200°F), there are no signs of shrinkage or fusing. They can withstand 704°C (1300°F), and will not melt until surface temperature up to temperatures reach 1204°C (2200°F). The density of the mineral wool controls the maximum surface temperature. A 40.05 kgs/m<sup>3</sup> (2.5 lb/ ft<sup>3</sup> ) density is rated at 204°C (400°F) maximum, while 144.18 kgs/m<sup>3</sup> (9.0 lb/foot <sup>3</sup>) mineral wool is rated at 649°C (1200°F) (hot-surface temperatures). Α typical flame rating of a mineral wool is flame spread 15, fuel contributed 0, and smoke developed 0. The price of mineral wool or fiberglass depends highly on geographical location, and some manufacturers prefer one over the other strictly for shipping cost reasons. Extremely high conditions require a ceramic fiber blanket which costs 20 times as much as fiberglass or mineral wool. When considering fire resistance, mineral wool is naturally better than fiberglass in fire applications, but fiberglass can be used in a fire resistance design. In future applications, the fire resistance of the insulation material should be one of the major considerations.

#### CELLULAR PLASTICS

Foams usually are the least fire resistant of all the insulation materials. Expanded polystyrene, polyurethane(PUR)/polyisocyanurate, phenolformaldehyde, and urea formaldehydes are the most widely used in the insulation industry. Even though the actual action of foams in a real fire is not completely understood, manufacturers sell foams with labels for different flammability ratings, all of which should be researched by the specifier.

Some foams can have a high level of toxics such as carbon monoxide (CO), hydrogen cyanides (HCN), and nitrogen oxides, and thus could directly contribute to a fire as a fuel source. To prevent the combustion of some foams, coatings are applied to delay the burning. A thin coat of fire protective material 0.381mm (0.015 in.) thick or less is applied to the surface. Once the fire penetrates the protective cover, the foam will burn. Work is being completed on foams which causes the foam to develop a protective char that stops combustion, once the ignition source is removed. Some of the new foams do not generate smoke or toxic gases in a fire. The other possible way to use foams is when sufficient protection is afforded to the foam by means of approved fire resistive thermal barriers with or without automatic sprinklers. The minimum rating on any protective material for foam is 15 minutes. When using foams, seek knowledgable advice, use reliable materials, and consult local codes and regulations before making a final commitment.

A new generation of polyimide, polystyrene and phenolic cellular plastics are demonstrating very favorable fire resistant and flammability properties. They are innately fire retardant by virtue of their chemical structure.

### OTHER THERMAL INSULATIONS

A new generation of calcium silicate materials is evolving which will provide insulation and act as an effective fire barrier for up to 3 hours. Most of these products come in rigid boards or panels, with sizes ranging from 1.2 by 3.0m to 0.6 by 6.0m (4 by 10 ft to 2 by 20 ft). Thicknesses vary from 19 to 64mm (3/4 to 2 1/2 in.) with densities starting at 128.16 kgs/m<sup>3</sup> (8 lb/ft<sup>3</sup>) and going to 817.02 kgs/m<sup>3</sup> (51 lb/ft<sup>3</sup>). Their thermal resistivity properties are not good as the conventional insulations, yet they serve as an excellent compromise when fire protection insulation is the primary concern. Conventional glass is being reworked into cellular glass cells and the ASTM E 84 results are 5 flame spread and 0 smoke density. Their limiting operational temperature, however, is  $649^{\circ}$ C (1200°F), which is quickly exceeded in a fire. Note here also that their high temperature performance is analogous to conventional fiberglass.

#### FUTURE FIRE AND FLAMMABILITY REQUIREMENTS FOR THERMAL INSULATIONS

Future fire requirements for thermal insulation can only change as quickly as the fire technology with its associated testing does. The complete understanding of fires is still under investigation by scientists, with no new major breakthroughs seen as this time. However, the National Institute of Standards & Technology (NIST) is developing new methods of fire and flammability testing. Their programs are interwound with computer models producing results that can be directly applied to real life conditions.

The most advanced piece of flammability equipment developed by Dr. Vytenis Babrauskas [4] at National Institute of Standards and Technology is the Cone Calorimeter. This piece of equipment is used to measure rate of heat release (RHR) in samples measuring 10 by 10cm (4 X 4 in.). The primary purpose of the calorimeter is to measure RHR of products in order to estimate their contribution to a room fire (as opposed to characterizing in detail their combustion chemistry behaviors). The rate of heat release is determined by measuring combustion product gas flow and oxygen depletion, while mass loss is simultaneously recorded (directly). The Cone Calorimeter was approved April 27, 1990 as ASTM E 1354, Test Method for Heat and Visible Smoke Release Rates for Material and Products Using an Oxygen Consumption Calorimeter.

The ASTM has a subcommittee E5.11, which is reviewing the ASTM E 119 test method. One of the proposed revisions is to modify the time-temperature curve, so that it is more severe.

Other proposed changes include furnace calibration and procedural methods which do not directly alter the fire environment. They are more for the furnace operator[5].

#### CONCLUSION

It is necessary for engineers, architects, designers, and manufacturers of thermal insulation to understand the requirements for (a) resistance to ignition, and, once combustion has begun, (b) resistance to the spread of flame. As a start, this group and any others concerned with insulation should understand the primary differences between (a) and (b). The insulation that is specified should be tested according to the appropriate method(1). Very often, the limiting factor for fire protection is specified by building codes. However, fire protection can go beyond this minimum requirement. Manufacturers should become aware of changing fire/flammability tests so that their products are designed with minimum fire hazard and can pass tests when these tests become the required minimum. Professionals specifying thermal insulation should use applicable fire/flammability test standards that provide optimum information about the combustibility of the insulation.

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**Innovative Insulations** 

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# GAS-FILLED PANEL HIGH-PERFORMANCE THERMAL INSULATION

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ABSTRACT: This paper introduces a new high performance non-CFC based insulating material with primary applications for refrigerator/freezer and building walls. Characteristics of these gasfilled panels, projected and experimental thermal performance levels, cost estimates, and manufacturing/installation issues for this material are discussed. Independent testing of prototypes has yielded R-values of 36 m-K/W (5.2 hr-ft<sup>2</sup>-F/Btu-in) for air filled panels, 49.3 m-K/W (7.1 hr-ft<sup>2</sup>-F/Btu-in) for argon filled panels, and 86.8 m-K/W (12.5 hr-ft<sup>2</sup>-F/Btu-in) for krypton filled panels. Target R-values values are 36 m-K/W (5.2 hr-ft<sup>2</sup>-F/Btu-in), 55 m-K/W (8 hr-ft<sup>2</sup>-F/Btu-in), and 105 m-K/W (15 hr-ft<sup>2</sup>-F/Btu-in) for air, argon, and krypton filled panels, respectively. Manufacturing costs depend strongly on gas-fill and structural requirements and are estimated at 42 to 850 /m<sup>3</sup> (0.10 to 2.00 \$/ft<sup>2</sup>-in). This insulation system can be fabricated using commercially available materials and equipment.

KEYWORDS: non-CFC insulation, thermal insulation, multilayer, argon, krypton, low emissivity, gas-filled

# INTRODUCTION AND MATERIAL DESCRIPTION

Recent research efforts at Lawrence Berkeley Laboratory's Windows and Daylighting Group have focused on the development of low to moderate cost, highly insulating gas-filled panels (GFPs). Development of this and other high-performance insulating materials is motivated by the

Brent T. Griffith, Research Associate; Dariush Arasteh, Staff Scientist; Stephen Selkowitz, Group Leader; Windows and Daylighting Group; Lawrence Berkeley Laboratory; Berkeley, CA 94720. need to replace CFC blown foam insulation (currently the highest performance insulation available) while also building more energy efficient buildings and appliances.

Gas-filled panels (GFPs), an outgrowth of superinsulated window technology [1], insulate by encapsulating a low thermal conductivity gas or gas mixture at atmospheric pressure within sealed panels. Low emissivity baffles suppress convective and radiative heat transfer. A schematic of one possible GFP is given in Figure 1. However, unlike foams or fibrous insulations, GFPs are not a homogeneous material; rather they are an assembly of specialized components. The wide range of potential applications (appliances, manufactured housing, site built buildings, refrigerated transport, etc.) lead to several alternative embodiments. GFPs are as much a new approach to insulating as they are a new insulating material. While the materials used for prototype GFPs are commercially available, fine tuning of components may be necessary for a commercial product. This material is the subject of a patent application by Lawrence Berkeley Laboratory. With the exception of a description of the panels that were independently tested, specifics of developed panel designs and materials are omitted for patent reasons.



Figure 1: Gas-filled panel schematic cross-section. This figure shows a random orientation of baffle layers; other configurations are possible.

GFPs utilize interior baffles to minimize heat transfer and to provide structure. Convection suppression is achieved by constructing baffles from multiple non-permeable layers. These layers can be either flexible or structural and can take on various geometric forms. Baffles are constructed to create interior cavities optimized in thickness (direction of heat flow) for the specific gas and application. Typical thicknesses range from 5 to 12 mm (0.2 -0.5 in.). Baffle surfaces are pre-coated with a low emissivity surface, typically a thin layer of aluminum 200 to 500 angstroms thick, in order to minimize radiative heat transfer across the cavities. To limit solid conduction, baffles are constructed of low conductivity materials such as thin plastics or paper and are arranged to create long solid conduction paths. For most GFP embodiments, the baffle is self-supporting and helps define the shape of the panel. The baffles can be made with stiff materials to create a strong supportive structure or they can be made flexible and resilient. Depending on the baffle, a continuous variation between structural panels and merely selfsupporting panels is possible. While refrigerator and freezer applications will require structural panels, building wall cavity applications are best served with non supportive panels which can be collapsed for transport.

Given an effective baffle, solid conduction through the gas is the only remaining means of heat transfer. Air is a good insulator and low-cost air GFPs are expected to have many uses in building applications. However, other gases, such as argon (Ar), carbon-dioxide ( $CO_2$ ), sulfur hexafluoride (SF<sub>6</sub>), krypton (Kr), and xenon (Xe) have significantly lower thermal conductivities. These gases are all non-toxic and either non-reactive or inert. We focus on the use of argon and krypton in GFPs since these gases are inert, have the proper thermophysical properties, and are readily available from the atmosphere. While xenon offers the potential for superior thermal performance, it is currently too costly for such applications. It should be noted, however, that as air separation techniques have improved over the past two decades, the prices of specialty gasses have dropped.

External barriers which serve to contain both the baffle and gas are the final critical component of GFPs. High barrier, multilayer polymer films developed for the food packaging industry have been used successfully for prototypes. Such films, which use gas barrier resins such as ethylene vinyl alcohol (EVOH) and polyvinyl alcohol (PVOH), are durable, puncture resistant, have very low gas transmission rates, and are heat sealable. Other barrier materials under investigation include aluminum and silicon oxide coatings.

Panel geometries will vary depending on the intended application. In order to avoid changing existing foam-in-place manufacturing methods, refrigerators and freezers will initially use thin (approximately 25 mm (1 in)) modular krypton filled panels in composite with a non-CFC foam. For building applications, the panels will initially fit into stud wall cavities with sealing flanges extending over studs for stapling in a manner similar to fiberglass insulation. Multiple layers of individual panels can be used for greater flexibility in sizing thickness and for greater insurance against punctures. Panel shapes, sizes, and stiffness can be adjusted for numerous other applications including HVAC insulation, hot water heater insulations, swimming pool and spa covers, refrigerated transport walls, and airplane walls.

### PROJECTED PERFORMANCE AND COST

The theoretical performance levels for GFPs are based on eliminating convection, infrared radiation and solid conduction, with only conductive heat transfer through the still gas remaining. Still gas conductivities, at atmospheric pressure and 273 K (32 F), are 0.0241 W/m-K (0.0139 Btu/hr-ft-F) for air, 0.0164 W/m-K (0.0095 Btu/hr-ft-F) for argon, and 0.0087 W/m-K (0.0050 Btu/hr-ft-F) for krypton [2]. Table 1 presents theoretical maximum R-values based on these values and conductivities at 300 K (80 F). A temperature of 273 K (32 F) is representative of the temperature of a typical GFP in a refrigerator/freezer or building wall while the higher temperature of 300K (80 F) is close to the temperature under ASTM C 518 test conditions. While convection may be effectively eliminated, heat transfer by solid conduction and minimal radiation will degrade these values slightly in real panels. Values given for projected performance are estimates based on testing and computer simulations.

Cost estimates for air, argon, and krypton GFPs shown in Table 2 are based on prices of material components and the assumption that roughly 80% of the cost is materials. This assumption is typical for the high throughput assembly of plastic products from preprocessed, roll stock materials. These cost numbers are preliminary and do not reflect a detailed examination of manufacturing and marketing issues as well as specific material's economies. For comparison, cost data for fiberglass and CFC foams is also given [3]. Costs given for all products are costs for the assembler to produce the materials. Note that costs given in the literature often vary substantially depending on the degree of indirect overhead costs incorporated into the "manufacturers costs." GFPs with R-values anywhere in between those given for air and those given for krypton can be manufactured using the appropriate mixtures of air, argon, and krypton. Both performance and cost are roughly linear with gas composition.

	Theoretic 273 K (32 F)	Projected R-value 273 K (32 F)	
Air GFP	41 (6.0)	38 (5.5)	36 (5.2)
Argon GFP	61 (8.8)	56 (8.1)	55 (8)
Krypton GFP	115 (16.6)	106 (15.3)	105 (15)

Table 1 -- GFP Theoretical and Projected Thermal Performance R-values in m-K/W (hr-ft<sup>2</sup>-F/Btu-in)

	l-value	Estimat	ed Costs
m	-K/W	\$/m <sup>3</sup>	\$/m <sup>2</sup> -R
(hr-ft	<sup>2</sup> -F/Btu-in)	(\$/ft <sup>2</sup> -in)	(\$/ft <sup>2</sup> -R)
Fiberglass	24.3	17-21	0.60-0.90
	(3.5)	(0.04-0.05)	(0.01-0.015)
CFC Blown Foam	50.7	85-210	1.8-4.3
	(7.3)	(0.20-0.50)	(0.03-0.07)
Air GFP	36	42-85	1.2-2.4
	(5.2)	(0.10-0.20)	(0.02-0.04)
Argon GFP	55	127-212	2.4-3.7
	(8)	(0.30-0.50)	(0.04-0.06)
Krypton GFP	105	635-850	6.1-7.9
	(15)	(1.50-2.00)	(0.10-0.13)

 Table 2 -- Performance and Manufacturing Cost Estimates

# **PROTOTYPE EVALUATION**

During 1990, over one hundred prototypes were built and their thermal performance evaluated using an infrared imaging system. Prototype samples, typically 200 or 300 mm square (8 or 12 in. square), were placed in a rigid foam board of a known resistance. A temperature difference was generated across the insulation by placing the sample between ambient temperature and a cold chamber. The infrared imaging system was then used to compare warm side surface temperatures of the prototype to that of the surrounding foam. This setup is shown schematically in Figure 2. Warm side temperatures are directly correlated with thermal resistances: the warmer the room side surface temperature, the better the insulator. Such side by side testing allows for quick, visual, and accurate evaluation of prototype samples. A versatile postprocessing system provides quantitative information on the prototypes. Figures 3, 4, and 5 are samples of this post-processed data. These figures show that air filled panels perform as well as rigid styrene foam board (assumed at R35 m-K/W (R5 hr-ft<sup>2</sup>-F/Btu-in), argon panels perform slightly better than CFC blown polyiso-cyanurate foam board (assumed at R50 m-K/W (R7.2 hr-ft<sup>2</sup>-F/Btu-in)), and krypton filled panels perform significantly better than CFC blown polyiso-cyanurate foam board. The infrared thermagrams of Figures 3,4, and 5 generally show that temperatures are roughly the same for different areas. This, in itself, is useful information.





Figure 2: Schematic cross-section of infrared radiometer and cold box facility. The infrared camera records the warm side temperature distribution of a sample placed between the cold box and ambient. The closer the sample's (or part of the sample's) warm side temperature is to ambient, the better an insulator it is. A computer, attached to the infrared radiometer, allows for quick and versatile post-processing.





Figure 3: Infrared image of the warm side of a 5.1 cm (2 in) thick rigid styrene board with an insert containing a 5.1 cm (2 in) thick prototype air GFP. The back of the panel faces a cold box at -12.1 C (10.1 F); ambient temperature is 22.5 C (72.5 F). The warm side temperature of the styrene board averages 20.3 C (68.5 F) with a maximum of 20.6 C (69.1 F) and a minimum of 20.1 C (68.2 F) while the warm side of the air GFP insulation averages 20.5 C (68.9 F) with a maximum of 20.9 C (69.6 F) and a minimum of 19.8 C (67.6 F). The lack of contrast in this thermograph indicates uniform temperatures. A temperature grey-scale is shown at the bottom of the figure. Since surface temperatures correspond to heat loss rates, a higher warm side temperature.implies a lower heat loss rate. Given an Rvalue of 35 m-K/W (R 5 hr-ft<sup>2</sup>-F/Btu-in) for styrene, the R-value for the air GFP is calculated at 37 m-K/W) (R 5.4 hr-ft<sup>2</sup>-F/Btu-in).



Figure 4: Infrared image of the warm side of a 2.6 cm (1 in) thick sample of CFC-blown foam with an insert containing a 2.6 cm (1 in) thick prototype argon GFP. The back of this assembly faces a cold box at approximately -18.6 C (-1.5 F); ambient temperature is approximately 22 C (71.6 F). The warm side temperature of the CFC blown foam averages 19.2 C (66.6 F) with a maximum of 19.4 C (66.9 F) and a minimum of 18.9 C (66.0 F) while the warm side of the GFP insulation averages 19.6 C (67.3 F) with a maximum of 20.1 C (68.2 F) and a minimum of 19.1 C (67.3 F). In this figure, warmer areas are lighter and colder areas are darker. A temperature grey-scale is shown at the bottom of the figure. Since surface temperatures correspond to heat loss rates, a higher warm side temperature implies a lower heat loss rate. If the R-value of the CFC blown foam is taken as R 50 m-K/W (R 7.2 hr-ft<sup>2</sup>-F/Btu-in), the R-value of this argon filled GFP is calculated at R 55 m-K/W (R 7.9 hr-ft<sup>2</sup>-F/Btu-in).



20.6 22.0 24.5 27.2 29.5'C

Figure 5: Infrared image of the door on a real freezer. The freezer is operating at about -20.5 C (-4.9 F) with an ambient temperature of 26.7 C (80 F). Half of the freezer door was left as manufactured (with 6 cm (2.4 in) of CFC blown foam); the other half was retrofitted with 4.3 cm (1.7 in) of krypton gas-filled panels. In this figure, warmer areas are lighter in shade and colder areas are darker. A temperature grey-scale is shown at the bottom of the figure. Since surface temperatures correspond to heat loss rates, a higher warm side temperature implies a lower heat loss rate. The infrared photo shows no significant difference (the resolution of the camera is 0.1 C) between the warm side temperature of both sides of the freezer door, indicating that 4.3 cm (1.7 in) of GFPs are as good an insulator as 6 cm (2.4 in) of CFC blown foam. (The average surface temperature is 24.8 C (76.6 F) across the solid white line. A second line at a temperature of 21.7 C (71.1 F) is used to define the scale.)

While infrared thermography is excellent for a quick comparison of the thermal performance of different specimens, it is not presently a developed technique to determine R-values. For this reason, several samples were fabricated and sent to Oak Ridge National Laboratory (ORNL) for independent testing. The GFP specimens were tested in the ORNL Advanced R-matic Apparatus which was designed to meet ASTM C 518, Configuration B (two transducers, both faces) [4]. Vertical heat flow conditions were tested with both heat flow up and heat flow down. The mean temperature was approximately 24 C (75 F) with a temperature difference of approximately 22.2 C (40 F). The apparatus is calibrated as specified by ASTM C 518 with an estimated uncertainty of  $\pm 3\%$  for homogeneous specimens. The specimens measured 40.6 x 40.6 x 2.5 cm (16 x 16 x 1 in.) with a metering area of 25.4 x 25.4 cm (10 x 10 in.) to insure one dimensional heat transfer measurement and minimal edge effects. Note that this standard advises against its use for measuring inhomogeneous and/or anisotropic material. Because of the nature of the baffle used in these samples, they could be considered inhomogeneous. However, IR thermography and finite element modeling indicate one dimensional heat transfer. Given this and the smaller metering area, the heat flux measurements should be an appropriate evaluation of thermal resistance.

The specimens tested at ORNL were intended to demonstrate the general gas filled panel approach and were not optimized or designed for mass production. The one inch specimens were encased in a rigid styrene foam bivalve for a total test thickness of two inches. "Blank" styrene was also measured at ORNL and the effect of the mask was backed out by ORNL to arrive at the final results. The GFPs were constructed with one primary. barrier comprised of two films sealed around the perimeter. The inside was split into two cavities by a heat sealed layer which served to limit mass transfer but was not hermetically sealed. Each cavity was filled with a baffle pile that consisted of three layers of 13 micron (0.5 mil) two sided metallized polyester film and two layers of "clear" 13 micron (0.5 mil) polyester film. The clear film was oversized (60 x 60 cm (24" x 24")) and crumpled up in an even but random fashion to create alternating clear and metallized layers. This produced a panel with eleven layers in one inch and with an average cavity size of less than 2.5 mm (0.1"). It is difficult to exactly quantify cavity scale due to the nature of the "crumpling". The intent with these panels was to effectively eliminate convective and radiative heat transfer. Except for the use of ultra thin films, solid conduction minimizing was not attempted.

Results from ORNL [5] are summarized in Table 3 and indicate prototype performance levels close to predicted levels. These tests found that the difference between heat flow up and heat flow down was less than 1%; this is within the 2% reproducibility of the R-matic. This finding indicates that the contribution of convection to heat transfer has been effectively eliminated. The differences between measured and projected R-values for the argon and krypton GFPs is primarily attributed to solid conduction through the large numbers of baffle layers. In addition, decreased performance may be attributed to fill concentrations less than 100%. However, oxygen concentration measurements (a crude measurement of gas fill) indicate that fill concentrations are better than 98%.

	ORNL Measured	Projected
Air GFP	36.1 (5.2)	36 (5.2)
Argon GFP	49.3 (7.1)	55 (8)
Krypton GFP	86.7 (12.5)	105 (15)

# Table 3 -- Measured R-values from the ORNL R-Matic and Projected R-values in m-K/W (hr-ft<sup>2</sup>-F/Btu-in)

### MANUFACTURING AND APPLICATIONS

The large scale manufacture of GFPs will not require the development of any substantially new materials processing technologies. The use of finished, roll stock material components makes the assembly of the panels relatively simple. Existing machinery from the food packaging industry such as thermoformers, impulse heat sealers, and bag making and wrapping machines can be used to manufacture GFPs at high line rates. Complete machines, known as form, fill and seal equipment, routinely used in the food packaging industry, can rapidly encapsulate the baffle with a barrier material, vacuum flush, gas back fill, and seal the panel into a final product.

Prototypes tested to date have been filled with a simple gas-filling apparatus. Fill percentages using this apparatus are generally in the 90-98% range. Advanced gas-filling methods using vacuum chambers are expected to yield GFPs with fill fractions of 98%-100%; these gas fill percentages have been met in both the window and food industries using vacuum chamber equipment.

Product lifetimes are a function of barrier material gas transmission rates and sealing quality. Barrier materials used in prototypes to date are taken from applications in the food packaging industry and have  $O_2$  transmission rates of 0.79 cc/m<sup>2</sup>-day-atm (0.05 cc/100in<sup>2</sup>-day-atm) at 296 K (73.4 F) and 0% R.H. We expect further development of these barrier materials to produce barriers with even lower transmission rates which will be acceptable for use in GFPs. While GFPs should be designed for high lifetime gas retention rates, it should be noted that a failure of the barrier material will degrade the performance of a GFP to no less than that of an air GFP, R 36 m-K/W (R 5.2 hr-ft<sup>2</sup>-F/Btu-in). On the other hand, the failure of the barrier material in some vacuum insulations may degrade performance to significantly lower R-values.

One of the challenges in developing GFPs is to create a structural baffle which can be substituted for and/or used in conjunction with foam-in-place applications (i.e., refrigerator/freezers). Work on this task has only recently begun and initial attempts have been encouraging. Figure 6 shows a recently



Figure 6: This photograph shows a first generation structural GFP prototype carrying a load of six standard bricks. The mass of the load is 13 kg (28.6 lb). The panel measures  $20 \times 20 \times 5$  cm. (8 x 8 x 2 in) and has a density of approximately 38 kg/m<sup>3</sup> (2.4 lb/ft<sup>3</sup>).

developed first generation structural GFP with a density of only  $38 \text{ kg/m}^2$  (2.4 pcf) supporting six bricks. The bricks exert a force of 700 N (1 psi) onto the panel. Under this load, this 50 mm (2 in.) thick GFP elastically deflects approximately 0.006 m (0.25 in). This sample is also exceptionally stiff in torsion. Note that this panel is constructed differently than the flexible panels sent to ORNL. Preliminary infrared testing shows slightly lower performance due to solid conduction. Development is continuing with a focus on optimizing the tradeoffs between structural and thermal performance.

The immediate applications for GFPs include krypton GFPs in refrigerator/freezer walls and argon GFPs in manufactured housing wall panels. In both of these applications, wall thickness and energy use are a premium and GFP's offer R/thickness values significantly higher than standard practice. In addition to saving space, higher R/thickness values also conserve costly structural lumber in building applications.

# CONCLUSIONS

Preliminary research efforts aimed at developing prototype gas-filled panel insulations indicate that such materials can perform significantly better than conventional CFC-blown foams and can be built at a reasonable cost using a novel configuration of commercially available materials. Performance values of approximately R36 m-K/W (5.2 hr-ft<sup>2</sup>-F/Btu-in), R55 m-K/W (8 hrft<sup>2</sup>-F/Btu-in), and R105 m-K/W (15 hr-ft<sup>2</sup>-F/Btu-in) at 273 F (32 F) are expected for air, argon, and krypton GFPs, respectively. The measured thermal performance of prototypes tested at ORNL approached these values with R36.0 m-K/W (5.2 hr-ft<sup>2</sup>-F/Btu-in), R49.3 m-K/W (7.1 hr-ft<sup>2</sup>-F/Btu-in), and R86.7 (12.5 hr-ft<sup>2</sup>-F/Btu-in), respectively. Infrared thermography has also verified performance levels superior to those of CFC-blown foams.

A continuing research and development effort is underway to optimize designs for added thermal performance improvements, improve component materials, and compatibility with fabrication technologies. Issues of structure, lifetime performance, and cost effectiveness will be addressed. In conjunction with the building and appliance industries and interested utilities, prototypes will be built and tested in refrigerator/freezer and building walls. Particular attention will be focused on the manufacturing processes involved. Further heat transfer measurements and other building code related tests (e.g., flame spread) are planned.

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# AEROGEL - A HIGH PERFORMANCE INSULATING MATERIAL AT 0.1 BAR

**REFERENCE:** Hunt, A. J., Jantzen, C. A., and Cao, W., "Aerogel - A High Performance Insulating Material At 0.1 Bar" <u>Insulation</u> <u>Materials: Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Silica aerogel is a low-density, transparent and porous solid that achieves vacuum-like insulating performance at a very modest vacuum (0.1 bar). Research is under way to further improve the thermal performance of this already highly insulating material by reducing the internal radiative heat transfer. The thermal resistance of aerogel is increased by adding infrared absorbing compounds to reduce the radiative heat transfer within the aerogel. Aerogel samples were prepared and their infrared transmission spectra were measured. After suitable materials and preparation methods were evaluated, an evacuated aerogel panel was prepared for thermal conductivity measurements. The measurements performed on this panel demonstrated significant increases in thermal performance over similar panels without opacification agents. Enhanced aerogel insulation has the potential to displace the current use of chlorofluorocarbons (CFCs) in refrigerator, and other appliances, thereby reducing the CFC damage to the ozone layer while producing significant gains in energy efficiency.

KEYWORDS: aerogel, silica, non-CFC insulation, evacuated insulation, thermal insulation, supercritical drying, microporous

# INTRODUCTION AND BACKGROUND

Aerogel is an unusual open-cell, transparent superinsulator first discovered over 60 years ago by S.S. Kistler [1]. It has unique properties resulting from the chemical preparation method that produces an extremely low density transparent solid. The low density and small pore size combine to produce a material that has

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extremely high thermal resistance. Caps and Fricke [2] showed that when silica aerogel is evacuated to a partial vacuum (0.1 bar) the effective thermal conductivity is reduced by a factor of three to about 0.007 W/m-K (R-20/inch in British units) at 280 K making it one of the best insulating solids known.

Research and development has been carried out since 1982 at Lawrence Berkeley Laboratory (LBL) under DOE sponsorship in order to understand and improve the optical and thermal properties of aerogel, develop production methods suitable for large scale manufacture, and evaluate the material for various applications. In particular, advances at LBL include the discovery of alternative preparation methods that avoid toxic starting compounds [3], analysis and reduction in the residual light scattering [4,5], development of a method to significantly reduce the temperature and pressure of the drying process [6], and understanding of the thermal and structural properties [7,8]. In addition progress has been made towards increasing the aerogel tile size and estimating manufacturing costs.

Aerogel has several unique advantages for many insulation applications. Because of the size of its pores (0.01 to 0.02 microns, or about 1/40 of the wavelength of visible light) it is transparent, which makes it an excellent candidate for applications such as windows, skylights, and freezer display doors. A further consequence of the small pore size is the excellent thermal performance at modest vacuum. Minor gas permeation or out-gassing have virtually no effect on its performance, meaning a long useful life - an important consideration in its use in commercial products. Other vacuum panel insulations typically have much more stringent vacuum requirements ( $10^{-3}$  to  $10^{-9}$  bar) making a long-life seal much more difficult. The small pore size removes the need for high evacuation because the air molecules interact with the aerogel structure more than with each other.

Evacuated aerogel insulation, provided that it can be made at a low enough cost, is an energy-efficient alternative to polyurethane foam insulations now used in refrigerators and freezers. Polyurethane foams achieve their performance through the use of CFC as the blowing agent. It is widely understood that CFCs threaten the earth's ozone layer. The Environmental Protection Agency estimated that in 1985 the production of plastic foam and foam insulation products was responsible for over 28% of the fully halogenated CFCs used in the United States. Refrigerator/freezer usage represents a substantial fraction of this use. Vacuum evacuated aerogel represents a leading technology option for eliminating CFCs in this application.

The National Appliance Energy Conservation Act of 1987 established efficiency standards for refrigerators and freezers that took effect on January 1, 1990 with more stringent standards adopted for 1993. Meeting these standards with existing foamed insulation will be very difficult. It will require substantial increases in insulation thickness if conventional foams are used. Increased costs and loss of useful cooled volume or increases in outside dimensions will occur unless alternative insulating materials are developed. (An even less favorable outcome, though unlikely, would be that the efficiency standards be significantly relaxed because cost effective alternative insulation was unavailable.) The present paper describes the preparation, characterization, and thermal performance of silica aerogels that have been modified to increase their thermal resistance by adding infrared absorbing and scattering compounds. In evacuated silica aerogel this radiative component is responsible for half or more of the thermal conductivity at room temperature [9]; therefore its reduction by infrared opacification can increase the thermal resistance of silica aerogel by 50% or greater. It is anticipated that opacified aerogels will have four or more times the thermal resistance of the best existing CFC-containing polyurethane insulation. Thus, replacing polyurethane insulation with aerogel of the same thickness will provide substantial gains in thermal efficiency in refrigerators and freezers and reduce heat losses in piping and in a variety of other industrial applications. Opacifying aerogel also substantially reduces its cost by reducing the amount of raw materials used in its production for the same thermal performance, thereby increasing the likelihood of its adoption on a commercial scale.

# AEROGEL PREPARATION AND PROPERTIES

Aerogel is produced using sol-gel processing techniques followed by a supercritical solvent extraction process. Briefly, a liquid organic silicon compound is mixed with water, alcohol, and a combination of catalysts. The water acts to hydrolyze the organic silicon compound tetraethylorthosilicate (TEOS) and the resulting silica ester condenses to form a solution of silica particles or polymer called an alcosol. The silica particles grow until they join together to form an open cell, cross-linked structure with a high fraction of voids and extremely fine pore size. This gel is referred to as an alcogel. To make alcogel useful for an insulator, the liquid solvent filling the void space must be removed. Simple air or oven drying leads to densification and collapse of the material to a dense glassy state approaching that of bulk silica. However, if the alcogel is placed in a pressure container and the pressure and temperature are raised above the critical point of the solvent, the pressure can be released slowly allowing the solvent to leave the silica microstructure intact. The resulting low density solid material is called aerogel. To extract the alcohol directly involves raising the temperature to approximately 550 K and a pressure of approximately 90 bars [10]. An alternative process was developed at LBL in which liquid carbon dioxide is substituted for the alcohol and the supercritical extraction process is carried out at approximately 310 K and 70 bars. The reduced temperature makes possible the use of less expensive equipment, provides more rapid temperature cycling of the equipment, and considerably reduces the risk of alcohol vapors exploding [11].

As a result of this processing aerogel has very unusual properties. Its bulk density can vary from less than 0.02 to over 0.4 g/cm<sup>3</sup> making it one of the lowest density solid materials. It is transparent to visible light and has a very high surface area (900-1000 m<sup>2</sup>/gm). The sound velocity is as low as 100 m/sec [12] and its mechanical impedance is roughly a mean between most solids and air. The silica composition, low density, and microstructure combine to make it an excellent thermal insulator. However, the infrared transmission spectrum given in Figure 1 shows significant transparency between the wavelengths of 3 and 8 microns. Even



Figure 1. Infrared transmission of undoped silica aerogel sample of 2 mm thickness.

at room temperature the overlap between the black body curve and these partially transmitting bands accounts for a significant portion of the thermal conductivity in evacuated aerogel [12].

To reduce this heat transfer in partially infrared transparent porous materials it is necessary to find suitable materials to absorb or scatter radiation in the 3 to 8 micron wavelength region. Criteria were developed to identify suitable candidates for the opacifying agents: (1) they should have strong absorption or scattering in the 3 to 8 micron region to minimize the amount of material required, (2) they should be available in small particle sizes and form good dispersions in alcosols, (3) they should not interfere unduly with the sol-gel process, and (4) they should be nontoxic and low cost. The search was confined largely to inorganic compounds. Various sources of infrared, physical, and chemical data were used to find likely materials.

Experiments were carried out to find methods to incorporate an opacifying agent into the aerogel. This was accomplished by either adding fine particulates or introducing soluble or reacting compounds into the early stages of aerogel preparation. The opacifiers are added after the initial hydrolysis reaction at a predetermined viscosity before gelation takes place. The task is difficult because the opacifiers tend to interfere with gelation, settle out, or form an incomplete dispersion because of interactions with the silicon alkoxide. After gelling and aging, the gels in the form of coupons are supercritically dried using the carbon dioxide substitution process. Measurements the infrared spectra were performed on aerogel coupons doped with various amounts of carbon, a transition metal oxide, and a carbonate. The aerogel coupons doped with the oxide or carbonate opacifiers proved unsatisfactory by the infrared characterization. They produced a much lower opacity than was predicted using Mie calculations indicating a large amount of agglomeration was occurring. Several types of carbon black were dispersed into solutions that exhibited sufficiently slow settling so as to produce a uniformly doped alcogel. The carbon was found to be the best candidate to block the 3-8 micron infrared window and came closest to its theoretical performance. It was decided to concentrate on carbon for thermal measurements but the search for alternatives is under further consideration. The experimental work then concentrated on determining the optimum doping level and dispersion methods for the opacifier particles. When suitable dispersions of dopant in the alcogel were obtained, larger samples were prepared and formed into evacuated panels for thermal conductivity measurements.

A panel for thermal testing was made from a matrix of eight 15 cm square by 1.3 cm thick tiles. The tiles were assembled and trimmed to a 25 cm x 25 cm panel 2.5 cm thick. The tiles were placed into a plastic preform, evacuated, and sealed by a plastic welding technique. The apparent thermal conductivity of the panel was measured by Holometrix Inc. in accordance with ASTM C 518-85 utilizing a heat flow meter. The specimen was installed horizontally between 30 cm square aluminum surface plates with a hemispherical emittance of 0.82 at 297 K. Due to irregularities in the panel a packing agent was used to obtain good thermal contact with the plates. The sensing area for the transducer was 10 cm square located in the center of the 30 cm square area. The instrument was calibrated using the National Institute of Standards and Technology Standard Reference Material 1450b.

# ANALYSIS

Heat transfer in aerogel materials is complicated by the infrared transparency of the material because the conductive and radiative heat flows are coupled. This paper considers only evacuated aerogel so that contributions to gaseous conductivity are specifically neglected. The heat transfer analysis is based on the radiation diffusion model [7]. For small temperature differences the wavelength-dependent heat flux is given by,

$$q_{r\lambda}(T) = \frac{4n_{\lambda}^{2}\sigma T_{r}^{3} \Delta T}{\frac{1}{\epsilon_{1\lambda}} + \frac{1}{\epsilon_{2\lambda}} - 1 + 0.75 \tau_{o\lambda}} \frac{\partial e_{\lambda b}(T_{r})}{\partial e_{b}(T_{r})}$$
(1)

where  $\frac{\partial e_{\lambda b}(T_r)}{\partial e_b(T_r)}$  represents the Rossland weighting function,  $T_r$  is the mean radiative temperature,  $n_{\lambda}$  is the real index of refraction,  $\varepsilon_{1\lambda}$  and  $\varepsilon_{1\lambda}$  are the spectral surface emissivities, and  $\tau_{o\lambda}$  is the optical depth.

Hartmann's results [7] demonstrated the ability of this modified solution to provide an accurate, simple approximation to the full integro-differential energy equation describing the coupled radiative-conductive heat transfer. Equation (1) describing the coupled radiative-conduction heat transfer was evaluated from infrared transmission data for both carbon doped and unmodified aerogel.

This attenuation of radiant heat transfer greatly simplifies the heat transfer analysis of doped aerogel. Measurements for a 2 mm thick carbon doped sample indicate a peak transmission of 1.8% at a wavelength of 4.7 microns (Fig. 2), which corresponds to an absorptivity of 20 cm<sup>-1</sup>. This represents a 1/e attenuation length of only 0.05 cm. This length is much smaller than the panel thickness of 2.5 cm, therefore radiation effects can be neglected leaving heat transfer as a function of the solid conduction only. Figures 1 and 2 were used to calculate the dimensionless spectral flux shown in Figure 3 for a mean radiative temperature of 288 K. The effectiveness of the opacifying agents was analyzed using Mie scattering theory. Mie calculations were carried out for the chosen doping level and particle size range using the complex index of refraction for the material. The calculation was carried out assuming complete dispersal of the primary particles. The results were compared with the measurements to determine the state of aggregation of the particles. Significant discrepancies indicated the degree of agglomeration.



Figure 2. Infrared transmission spectrum of carbon opacified silica aerogel sample of 2 mm thickness.



Figure 3. Dimensionless spectral flux for both carbon opacified and unmodified aerogel samples.

## **RESULTS AND DISCUSSION**

The infrared transmission spectrum of carbon opacified silica aerogel sample with a thickness of approximately 2 mm is given in Figure 2. This sample had a bulk density of  $\sim 0.108 \text{ g/cm}^3$  and was doped with a carbon loading of 0.06% of the gel volume. It can be seen that the transmission has been reduced from about 50% in the undoped aerogel (Fig. 1) to less than 2%. The transmission of the sample at 4.7 microns is within a factor of 5 of that predicted by Mie calculations. This factor indicates that there is still a significant degree of agglomeration of the primary carbon particles; but this was a much closer agreement than for the other doping agents, and was only achieved after many different dispersion techniques were tried. Results shown in Figure 3 for  $T_r = 288K$ ,  $\varepsilon_1 = \varepsilon_2 = 1$ , and n = 1.02 demonstrate the ability of carbon to block infrared radiation effectively.

The alcogels used to make the panels for the heat flow testing were prepared in a similar fashion to that used to make the coupons for infrared measurements except for the size. The supercritical solvent extraction for the larger tiles was carried out and post-processed in a similar way. The thermal conductivity of the 25.4 by 25.4 cm. carbon-doped panel was measured to be 0.00654 W/m-K under vacuum (R-22.1 per inch). This represents the conductivity of the entire package including the plastic preform and is probably a conservative value because of lateral conductivity of the plastic film and finite panel size. In another series of tests, the nondoped aerogel measured a value of 0.00964 W/m-K (R-15 per in.) using a similar package. However, when the undoped aerogel was measured without the plastic packaging, a value of 0.00719 W/m-K (R-20.1 per in.) was achieved under vacuum conditions. Thus, the true thermal resistance of the aerogel is higher than the number quoted above. Further tests are underway to confirm this result. In Figure 3 it can be seen that the effect of the doping agent is to reduce the peak spectral flux by about half. Since the heat flow is proportional to the integral of the spectral flux it is clear that the doping level should be increased to further reduce the radiative component of heat transfer. Future samples will use higher doping levels to further lower the thermal conductivity.

The preliminary results of this research are promising and indicate a definite improvement in the thermal properties of opacified aerogel over the pure silica aerogel. The loading of carbon was only about 0.06% of volume of the alcogel. The result is encouraging because a substantial increase in thermal performance was gained at the negligible cost. Even with somewhat higher doping levels than indicated above, this conclusion should remain substantially unchanged. Thus, this performance increase translates into reduced costs for raw materials to achieve a specified thermal performance and thereby improves the economic viability of using aerogel insulation for non-transparent applications.

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REDUCTION OF HIGH TEMPERATURE THERMAL CONDUCTIVITY OF THIN-WALL CERAMIC SPHERES

REFERENCE: Chapman, A. T., Cochran, J. K., Ford, T. R., Furlong, S. D., and McElroy, D. L., "Reduction of High Temperature Thermal Conductivity of Thin-Wall Ceramic Spheres," <u>Insulation Materials: Testing and Applications,</u> 2nd Volume, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Within the last five years, a method for fabricating millimeter sized thin-wall hollow spheres from conventional powders has been developed at Georgia Tech. On firing, the spheres develop good strength as the walls are sintered to near theoretical density. These spheres have reasonably low thermal conductivities at low temperatures but conduction increases more rapidly than desired with increasing temperature due to transparency of the sphere wall. This work presents the results of opacification of the sphere walls with thermal pores to reduce the radiative component of conductivity at high temperatures. Mie scattering was employed in an effort to select an opacifying pore size distribution with a high thermal radiation scattering efficiency. It was shown that incorporation of pores in the micron size range could significantly reduce high temperature radiative conductivity.

KEYWORDS: thermal conductivity, hollow spheres, opacificaton, radiation scattering, low density

A process has been developed which produces monosized, thin-wall, polycrystalline, hollow, ceramic spheres typically with sphere diameters between 2 and 4 millimeters, Figure 1, and wall thickness between 50 and  $100\mu m$ , Figure 2, resulting in sphere bed (bulk) densities between 10 and 60 lb/ft<sup>3</sup>. Because the fabrication is a powder process, virtually any composition is selectable and a variety of

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Figure 1. Monosized Thin-Wall Hollow Alumina Spheres 3.0 mm in Diameter.



Figure 2. Hollow Spheres with Thin  $(50-100 \ \mu m)$  Uniform Thickness Walls.

phases can be mixed in the sphere wall. Hollow spheres of more than thirty compositions have been fabricated but most of the effort has been on either alumina or mullite as these are excellent materials for high temperature applications. After forming, the powder shells are conventionally fired to sinter the walls, leaving the shells impervious or porous as Thermal conductivities of beds of these spheres desired. were measured up to 600°C by Shapiro<sup>(1)</sup> and McElroy<sup>(2)</sup> at Oak Ridge National Laboratory in a radial heat flow apparatus labeled ORNL-8. Shapiro reported room temperature values of 0.21 to 0.29 W/mK for alumina sphere beds ranging in density from 0.26 to 0.90 g/cc, respectively. McElroy reported values of 0.13 W/mK for mullite spheres at a density of 0.30 g/cc and 0.08 for fused silica spheres at 0.32 g/cc. These values were in air at one atmosphere. Both researchers noted that due to the transparency of the single phase walls, thermal conductivity increased rapidly with temperature.

Thus, low temperature thermal conductivities are adequate for many insulation applications but it would be helpful to reduce the high temperature conductivities by introducing infrared radiation scattering centers into the sphere walls. This could be accomplished by adding graphite powder of the proper particle size to the sphere composition so that on firing the graphite would be removed by oxidation leaving a pore structure related to the size distribution of the graphite. Obviously, the resultant pore size would be smaller than the particle size of the burnout material due to firing shrinkage and sintering characteristics of the oxide powder shell wall. The object of this effort is to design and test a high reflectivity infrared scattering system by considering scattering parameters to help select the proper pore size distribution and then to produce that pore size distribution in a sphere wall.

### SCATTERING THEORY

For energy incident on a transparent medium containing small particles of different index of refraction, part of the energy is scattered, part is transmitted, and part is absorbed. If both phases are sufficiently transparent (i.e. absorption is zero), then the fraction of radiation transmitted,  $\tau$ , and the fraction reflected, R, sum to unity or

$$\tau + R = 1 , \qquad (1)$$

and the quantity of radiation reflected is then R = 1 -  $\tau$  where

$$\tau = \exp\left(-3KV_{r}x/4r\right), \qquad (2)$$

assuming independent scattering by the particles <sup>(3)</sup>. In equation(2), K is the scattering factor which varies in value from 0 to 4 as presented below,  $V_p$  is the volume fraction of scattering particles, x is the thickness of the scattering matrix (sphere wall thickness in this case), and r is the particle radius. The scattering factor, K, is a strong function of the ratio of particle diameter, d, to scattered wavelength  $\lambda$ , or

$$K = f(d/\lambda) \quad . \tag{3}$$

Scattering effects can be considered to fall into three ranges based on  $d/\lambda$ . If  $d/\lambda$  is less than 0.2, K is described by Rayleigh scattering <sup>(4,5)</sup> and K is proportional to  $(d/\lambda)^4$ . Near the upper limit of Raleigh scattering (i.e.  $d/\lambda = 0.2$ ), K values of 0.1 or less are typical. As  $d/\lambda$  decreases below 0.2, K decreases rapidly and scattering becomes ineffective. In the range 0.2 <  $d/\lambda$  <5, Mie theory<sup>(6,7)</sup> predicts K will increase from ~0.1 near  $d/\lambda = 0.2$  to a maximum of ~4 near  $d/\lambda = 1$  and then decrease to a constant value of ~2 near values of  $d/\lambda = 5$ . For values of  $d/\lambda > 5$ , K maintains a constant value of ~2. In the Mie scattering range, the maximum of the K vs.  $d/\lambda$  curve occurs at  $d_{max}/\lambda$  where <sup>(7)</sup>

$$\frac{d_{\max}}{\lambda} = \frac{4.1}{2\pi (m-1)}$$
 (4)

and m, the relative index, equals the ratio of the index of the particle  $(n_p)$  to the index of the continuous media  $(n_m)$  or  $m = n_p/n_m$ .

The value of the scattering factor also depends on relative index of refraction, m. Kingery<sup>(3)</sup> states that the greatest scattering effect in ceramic systems is related to the relative index, m, with significantly increased scattering for systems with the greater m values. Thus, in addition to selecting an opacifier with the optimum size, it

is important to select the largest relative index from available systems. In many cases, the largest relative index is afforded by the introduction of pores into the system where thermal stability is consistent with maintaining the optimum pore diameter.

It should be noted that to discuss scattering theory properly, the complex index of refraction should be taken into account rather than the simple index. However, inclusion of the complex index leads to complications of the scattering models. In cases where absorption is near zero (as is the case here for alumina and mullite), the complex index is approximately equal to the simple index. In those cases, use of the simple index is a reasonable assumption and can be used for estimation of parameters<sup>(8)</sup>.

It is common to calculate the scattering coefficient, S, to compare relative scattering efficiencies. Here, S is extracted from Equation(2) as,

$$S = 3KV_{\gamma}/4r .$$
 (5)

When S is plotted vs. particle diameter, a dramatic effect of K on S can be seen as sharp scattering peak in the range of  $d/\lambda \sim 1 \pm 0.5$ . Outside of these particle sizes, an added "scattering" phase would have relative small effects in blocking transmission of radiation.

#### SCATTERING SYSTEMS

The systems selected for opacification were alumina and mullite. Parameters of interest are presented in Table 1. The index of refraction of both alumina and mullite is relatively constant for wavelengths from 0.3 to 5  $\mu$ m<sup>(3)</sup>. Absorption coefficients for both are low over this wavelength range (3). Because the opacifier is a pore system and the index of refraction of air is 1.0, the relative index is the same as the sphere wall matrix index. Thus,  $d_{max}/\lambda$  for both systems is ~ 1. The temperature range of interest is 1000 to 1500C and the peak radiation intensity from Wein's Law is from 1.6 to 2.2  $\mu m$  over this range. Actually, from Plank's Law, ~90% of the radiation energy is between 0.7 and 5.0  $\mu$ m for this temperature range. Thus the ideal pore size distribution would range from 0.7 to 5.0  $\mu$ m with a peak at 2.0  $\mu$ m, Figure 3.

To produce such a pore size distribution, three graphites were selected for incorporation into the sphere walls. These were labeled as L,S, and A graphites which had average particle sizes of 3.1, 6.9, and 10.0  $\mu$ m respectively with particle size distributions as shown in Figure 3. It is apparent from Figure 3 that L graphite matched the desired particle size distribution most closely and conceivably could produce a pore size distribution that substantially overlapped if 10-20% linear shrinkage occurred during firing.

Wall Material	Index	d <sub>max</sub> /λ	$\lambda_{\max} Range^*$	d <sub>max</sub> Range
	(n)	(Eq.4)	( $\mu$ m)	(µm)
Mullite Alumina	1.64 1.76	1.02	1.6-2.2	1.6-2.2 1.4-1.9

Table 1. Scattering Systems and Parameters Using Pores as opacifiers for Infrared Reflectance.

\* Wavelength of maximum radiation intensity from 1000 to 1500C as calculated from Wein's Law.



Distributions for Opacifying Graphites L, S, and A.

#### EXPERIMENTAL PROCEDURE

Hollow ceramic spheres were fabricated from alumina or mullite composition powders with and without additions of graphites A, S, and L as specified in Table 2. Where quantity of graphite addition was not stated, the amount added was 5 w/o based on the dry weight of either the alumina or mullite powders. A minimum of four liters of spheres was fabricated for each composition as required for thermal conductivity measurement. After forming, the polymer bonded powder shells were fired to 1500, 1550, or 1600C in air for three hours to densify the sphere walls and burnout the graphite to create the thermal pores.

Pore systems were characterized from SEM photomicrographs of fracture surfaces of fired sphere walls. In addition, selected samples were subjected to mercury Characteristics of Hollow Ceramic

	Spheres	Character			
Sample	Matrix	Graphite Opacifier	Diameter (mm)	Density <sup>*</sup> (g/cc)	FT <sup>!</sup> (°C)
018	Alumina	None	3.6	0.3	2 1550
017°	Alumina	None	3.6	0.4	5 1550
012	Alumina	None	3.8	0.5	3 1550
003	Alumina	5% A	2.5	0.3	8 1550
004	Alumina	5% S	3.2	0.3	8 1550
019°	Alumina	5% L	3.2	0.3	8 1550
020	Alumina	5% L	3.1	0.2	6 1500
023	Alumina	10% L	3.0	0.3	4 1550
014°	Mullite	None	3.3	0.2	2 1600
024°	Mullite	5% L	3.2	0.3	2 1600

\* Density is bed or bulk density. ! FT is firing temperature.

\* Thermal Conductivity also measured at ORNL.

intrusion porosimetry to determine pore distributions and volumes using automated commercial porosimetry up to pressures of 30 ksi.

Thermal conductivities were measured on loose beds of spheres using a radial heat flow apparatus (Unitherm Model 4141, Anter Laboratories, Pittsburgh, PA) from 400 to 1200°C. This apparatus measures conductivity on a cylindrical volume ~ 11.5 cm in diameter by 45 cm in length. The temperature gradient is measured across a gap of 3.2 cm or approximately 10 times the sphere diameters which provided good homogeneity and measurement reproducibility.

In addition, thermal conductivity of samples 014, 017, 019, and 024 were measured at Oak Ridge National Laboratory using apparatus ORNL-8<sup>(1)</sup> which is a radial heat flow device also. ORNL-8 is capable of measurement from ~ 30 to ~ 500 C using a cylindrical measurement volume 5.4 cm in diameter by 90 cm in length. The thermal gradient was measured across a gap of 2.5 cm.

### RESULTS AND DISCUSSION

### Pore System Analysis

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Average pore diameters and pore volumes from mercury porosimetry are compared to average pore sizes measured from SEM micrographs in Table 3. Micrographs of alumina wall microstructures without and with opacification are presented

Sample No.	Sphere Wall	Graphite Opacifier	<u>Mercury Po</u> Diameter	<u>prosimeter</u> Pore Vol	SEM Pore
	Material		(µm)	(%)	(µm)
018	Alumina	None	0.28	15	1.0
019	Alumina	5% L	0.40	36	3.7
020	Alumina	5% L	0.45	35	NA
004	Alumina	5% S	0.55	26	NA
023	Alumina	10% L	0.8 & 1.2	55	NA
014	Mullite	None	0.9	29	1.3
024	Mullite	5% L	1.5 & 1.8	43	1.3

Table 3. Pore Analysis in Pore Opacified Alumina and Mullite Shells.

in Figures 4 and 5 respectively and mullite microstructures without and with opacification in Figure 6 and 7 respectively.

For alumina spheres, the pore diameters measured by Hg intrusion are considerably smaller than estimated from micrographs. This is common for Hg intrusion because the pore size is calculated from the pressure required to force Hg through the necks of interconnected larger pores whereas from micrographs the tendency is to measure the largest pore diameters. Nevertheless, mercury intrusion pore diameter did vary in the order expected. Additions of 5% L graphite increased the pore size from 0.28 to 0.40  $\mu$ m at 1550°C while additions of 10% L increased pore diameters to a bimodal pore distribution with peaks at 0.8 and 1.2  $\mu$ m. When a larger size graphite was added (i.e. sample 004, 5% S graphite), the pore size increased from 0.40  $\mu$ m for 5% L graphite to 0.55  $\mu$ m for S graphite. The same relative effect for pore diameter is seen in SEM diameters. Without opacification, average pore size was 1.0 µm and addition of 5% L increased diameters to 3.7  $\mu$ m. Note that this is approximately the same as estimated to be needed for infrared scattering.

When pore volumes are considered, the pore volumes are reasonable. The unopacified alumina had a pore volume of 15%. Addition of 5% L graphite increased porosity by ~20% to 35%. The density of alumina is 4.0 g/cc and that of graphite is 1.8-2.0 g/cc. Burnout of the 5% graphite would add a minimum of 10% porosity to the shell wall and if the graphite was platty as is true of L graphite, porosity would

be increased even more due to inefficient packing of alumina grains around the graphite particles during formation of the spheres. Addition of 10 v/o L graphite, increased porosity 40 v/o which is consistent with the 20% porosity increase provided by 5% L additions.



Figure 4. Fracture Surface of Alumina Sphere Wall without Opacification.



Figure 6. Fracture Surface of Mullite Sphere Wall without Opacification.



Figure 5. Fracture Surface of Alumina Opacified with L Graphite.



Figure 7. Fracture Surface of Mullite Opacified with L Graphite.

Mullite pore diameters were more consistent between Hg intrusion and SEM observation, Table 3, indicating that neck formation is not as prevalent in the mullite. Here the Hg intrusion data is probably more meaningful as addition of 5% L graphite increased pore diameters from 0.9  $\mu$ m to a bimodal pore distribution peaking at 1.5 and 1.9  $\mu$ m. This is certainly in the correct scattering range calculated from scattering theory. Pore volume increases of 14% (from 29 to 43%) on 5% L addition to mullite was also more consistent with the volume of graphite added as 8% porosity increases would be expected based on densities alone.

### Thermal Conductivity

<u>Alumina Hollow Spheres.</u> The effect of bed density on thermal conductivity for alumina spheres is shown in Figure It can be seen that for spheres fired at 1550°C, sphere beds with densities of 0.32 and 0.45 g/cc had conductivities almost identical and for a bed density of 0.53 q/cc, conductivity increased ~0.08 W/mK at low temperatures due to increased conduction. At high temperatures, the 0.53 g/cc spheres would be expected to have a reduced conductivity due to a larger number of radiation scattering centers. This was not the case as conductivity increased even faster with temperature for the higher density spheres compared to the lower density spheres. Possibly variations in firing temperature may have changed the pore structure sufficiently to produce this effect. Overall, the changes produced by the density variation were relatively small.

Opacifying alumina with thermal pores certainly was effective in reducing high temperature thermal conductivity, Figure 9, when the correct pore size range was produced. The larger particle size A and S graphites had relatively little effect on conductivity but conductivity reductions of 35% were produced by L graphite. It was noted earlier that if the diameter of the scattering phase was varied from the optimum size by as much as 50%, the scattering effectiveness would be reduced dramatically. That is apparently the case here. Consequently, L graphite was used for all other opacification studies.



Figure 8. Thermal Conductivities of Unopacified Alumina Hollow Spheres of Various Bulk Densities.



Figure 9. Thermal Conductivities of Alumina Spheres Hollow Opacified 5% with A, s, and  $\mathbf{L}$ Graphites.
Bed density variations did produce predictable conductivity variations as a function of temperature in well opacified alumina spheres, Figure 10. At lower temperatures, the higher density (0.38 g/cc) opacified spheres had a 20% higher conductivity than lower density (0.26 g/cc) spheres. As temperatures increased, the increased number of radiation scattering centers in the higher density bed reduced the conduction below that of the lower density bed. In both cases, the opacification was effective in reducing conductivity compared to no opacification.

Increasing the quantity of L graphite from 5 to 10 w/o produced relative little effect on conductivity, Figure 11. As the graphite content increased, the pore size increased along with the volume of pores and moved the pore size distribution to one with a lower scattering efficiency. It is likely that there is an optimum quantity of graphite for maximum scattering efficiency and determining that optimum could be very beneficial.

Finally, to provide independent confirmation that the opacification effect is substantial and to provide an indication of accuracy of measurement, conductivity comparisons were made between values on the Anter Unitherm 4141 and ORNL-8, Figure 12, for the identical set of samples, Nos. 017 and 019. It was confirmed that the pore opacified 019 had a conductivity -0.1 W/mk lower than 017 at -500°C. However, there was a difference in the values of -0.1 W/mk for the same samples between the two units or -20-25% at 500°C and in the 0.3-0.4 W/mK range. Note that at room temperature, the effect of opacification disappears as would be expected as radiation contributions become a small portion of conductivity.

<u>Mullite Hollow Spheres</u> - At this time only a limited quantity of data has been generated on opacification of mullite hollow sphere walls, Figure 13. Mullite was opacified with 5% L graphite and the effect appears to be as significant as for alumina. A conductivity reduction of ~30% was produced at 1200C by opacification. However, the opacified mullite did have a significantly higher density which may have produced some of the conductivity reduction at high temperature. As can be seen from the ORNL data for these two samples, the effect of opacification again disappears at low temperatures as radiation effects are minimized.

Agreement between the Anter 4141 data and ORNL-8 data was much better for the mullite data as the values were within 15% of each other at  $500^{\circ}$ C.



Figure 10. Thermal Conductivities of 5% L Graphite Opacified Alumina Spheres of Different Bulk Densities.





Figure 12. Comparison of Thermal Conductivity Measurements on Unitherm 4141 and ORNL-8 For Alumina Opacified Spheres.



Figure 13. Comparison of Thermal Conductivity Measurements on Unitherm 4141 and ORNL-8 For Mullite Opacified Spheres.

# CONCLUSIONS

Opacification of alumina and mullite hollow spheres with a pore system in the correct scattering size range resulted in thermal conductivity reductions of 30-35% at 1200°C. The results are encouraging to the extent that further opacification studies are in progress both for pore

opacified systems and particle opacified systems.

# ACKNOWLEDGEMENTS

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**Test Methods and Comparisons** 

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AUTOMATED LOW-TEMPERATURE GUARDED HOT PLATE FOR MEASURING APPARENT CONDUCTIVITY<sup>2</sup>

**REFERENCE:** Smith, D. R., Dube, W. P., and Filla, B. J., "Automated Low-Temperature Guarded Hot Plate for Measuring Apparent Conductivity," <u>Insulation Materials: Testing and Applications,</u> <u>2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: A guarded-hot-plate apparatus has been developed for measuring the apparent thermal conductivity of flat-slab thermal insulation materials at low temperatures. Relevant ranges of physical variables which can be accommodated (under different conditions) are: diameter, 203 mm (fixed); thickness, 5 to 30 mm; mean specimen temperatures, 100 to 400 K; temperature differences, 3 to 150 K; temperature gradients, 0.1 to 10 K/mm; and thermal resistance, 0.02 to 1.5  $\text{K} \cdot \text{m}^2/\text{W}$ . Conductivity can be measured in dry air, gaseous nitrogen, helium, argon, or neon, or in vacuum. Apparent conductivity of an insulation material can be studied as a function of temperature, fill-gas pressure, or species, giving insight into the heat-transfer processes present in the material. Control of the main heater may be stabilized either at constant heater power or constant heater temperature. This apparatus will be useful in development of low-temperature Standard Reference Materials, and is being used to study heat transfer in closed-cell foam insulation containing HCFC blowing agents.

**KEY WORDS:** apparent thermal conductivity, automatic control, guarded hot plate, low-temperature, thermal insulation

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# 1. Introduction

There are no primary physical standards of thermal conductivity,  $\lambda$ , for insulation materials, over any range of conductivity. Thermal conductivity can be obtained by use of Fourier's law of conduction,

$$Q = -\lambda \cdot \mathbf{A} \cdot (d\mathbf{T}/d\mathbf{x}), \qquad (1)$$

where Q is the rate of flow of heat through a specimen of cross-sectional area A, under the imposition of a temperature gradient Dt/dx along the thickness of the specimen. To obtain  $\lambda$  in practice this relation takes the form

$$\lambda = (Q/A) / (\Delta T/L), \qquad (2)$$

where the temperature gradient Dt/dx has been approximated by the ratio  $\Delta T/L$ ; as is usual in practice,  $\Delta T$  is the magnitude of the temperature difference across a specimen of thickness L. Some apparatus determine the conductivity of a test specimen in terms of the conductivity of a standard, or reference specimen; however, the conductivity of the standard reference specimen must be determined by primary, or absolute, measurements. A guarded hot plate (GHP) can in principle determine  $\lambda$  by absolute measurements under stable and reproducible conditions; the geometry (ratio of area to thickness of specimen) of the GHP is designed for measurement of conductivity of poor thermal conductors in the shape of planar slabs.

In some types of thermal insulation materials, radiative or even convective heat transfer (due to the fill gas) may also be present, in addition to conduction. Because of the effects of combined modes, the result of a measurement in a conductivity apparatus is termed an "apparent" conductivity. In this work the symbol  $\lambda$  is to be understood as apparent conductivity unless further qualified.

The information presented in this work complements that given for a square guarded-hot-plate apparatus built according to the new international standard, ISO DIS 8302, and described in detail by De Ponte, Langlais, Boulant and Klarsfeld [1]. Much additional information on the design of different GHP apparatus and on the analysis of errors that may arise is available from various sources, but especially reference [2].

# 2. Brief Description of GHP Apparatus

The GHP and associated control system described here are a highly modified version of a commercial GHP apparatus, originally designed for operation at temperatures above  $0^{\circ}C$ (273 K). Significant changes were made [3] to include the possibility of operation at temperatures from 100 to 273 K, and to improve thermal anchoring of thermocouple leads, especially when operating the system under vacuum. The dataacquisition and control systems have been completely redesigned and implemented since Ref. [3] was first distributed.

The purpose of the stack heaters and associated guarding is to create stable one-dimensional flow of heat under defined conditions within the central, metered portion of the specimens. Under these conditions the apparent conductivity can be obtained according to Eq. (2). The layout of the central measurement stack of this GHP, is consistent with the specifications of ASTM C 177, "Steady State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded-Hot-Plate Apparatus".

The measurement stack (Fig. 1) denotes the metered, inner-guard, and auxiliary heater plates, together with the specimens, cold plates and offset insulation. Each stack element has a circular cross section, perpendicular to the vertical symmetry axis. Heat flows from the metered heater, through the specimens, parallel to this vertical axis. The circular main (metered) heater (MH) plate provides the meas-The ured quantity of heat flowing through the specimens and thereby develops a temperature gradient within the specimens. Concentric and coplanar with this MH plate, but separated from it by a narrow gap (G), is the annular inner-guard (IG) heater plate. The purpose of the gap is to inhibit interchange of heat between the two plates. The MH/IG assembly is sandwiched between a pair of specimens (S, S'; matched, where possible, in thickness and density) with diameters equal to the outer diameter of the MH/IG unit. Offset insulation (OI) between the system's heat sink (cold plates, CP) and auxiliary heater (AH) plates provides a thermal resistance which allows the temperatures of the AH plates to be raised above the (uncontrolled) temperature of the heat sink. The AH plates allow the cold-side temperatures of the specimens to be controlled at values independent of those of the warmer As a consequence, the arithmetic sides of the specimens. mean surface temperatures of the specimens can be controlled independently of the temperature gradient.

Interaction between the outer edges of the measurement stack and the surrounding environment is reduced by the presence of the cylindrical, isothermal outer-guard heater (OG, Figs. 1 and 3). Heat exchange between the edges of the stack and its environment, which would produce unwanted radial flow of heat within the specimens, is further reduced by surrounding the stack and outer guard with loose-fill (vermiculite: exfoliated mica) insulation (IF, Fig. 1). The stack, outer guard, and loose fill are all contained within the fluidcooled stainless-steel shroud (FCS), which is in turn contained within a vacuum-insulated stainless-steel environmental chamber (EC) This chamber permits the ambient air to be removed from the GHP system. The chamber may then be left evacuated or may be back-filled with any fill-gas such as dry air, argon, helium, neon, or nitrogen, that does not condense under the chosen experimental conditions. Liquid nitrogen  $(76 \text{ K}; -197^{\circ}\text{C})$  or chilled alcohol  $(-50 \text{ to } +40^{\circ}\text{C})$  is used to



Figure 1. Schematic diagram of guarded-hot-plate stack and terminal strip together with supporting plates and environmental shielding.

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- RL rigid fluid line from BCP to TCP incoming fluid
- BCP bottom cold plate
- TCP top cold plate
- FL flexible line to TCP
- RL' rigid fluid line from TCP to
- FS fluid lines on shroud
- vacuum-insulated line P - pump-out port on



cool the cold plates in the stack, and together they provide a wide range of choices for mean specimen temperatures.

Due to the space occupied by identifying legends within Fig. 1, it is not possible to show dimensions of the important parts of the measurement stack within the figure (which is only approximately to scale). The heater and thermocouple leads have also been omitted for clarity. The main heaterinner guard (MH/IG) assembly, specimens (S, S'), auxiliary heaters (TAH, BAH), and cold plates (TCP, BCP) are each 203 mm (8.00 in) in diameter, and occupy a horizontal cross-sec-tional area of 324.3  $cm^2$ ). The main heater itself has a diameter of 101.6 mm (4.00 in), and an area of  $81.07 \text{ cm}^2$ . The MH/IG assembly is 19 mm thick, while the TAH and BAH are 16 mm thick. The apparatus can handle specimens with thicknesses ranging from 2 to 30 mm. When 25-mm specimens are mounted the vertical thickness of the stack from the surface of the bottom cold plate (BCP) to the surface of the top one (TCP) is about 102 mm. The clearance between the outer margin of the stack and the outer guard is about 20 mm, except where the strain-relief platforms (SR; for protecting leads) extend from the heater plates. The diameter of the fluid-cooled shroud (FCS) is about 425 mm, and the inner diameter of the environmental chamber (EC) is 432 mm.

Each cold plate is offset 85 cm from its supporting baseplate by a composite spacer (CS); and the bottom baseplate (BP), which also supports the fluid-cooled shroud, is offset from the system base by support posts (SP) 110 mm long. This system base supports the environmental chamber as well as the thermally insulated thermocouple terminal strip (TS). This location for the terminal strip reduces the effect of nearby coolant lines on the junctions of the thermocouple wires with the copper leads to the voltmeters.

# 3. Detailed Description of Apparatus Elements

# 3.1 Main Heater-Inner Guard Assembly

The MH/IG unit (Fig. 1) is constructed of two aluminum "half-plates", 9 mm thick and 203 mm in diameter, sandwiching a silicone-rubber heater element (RH) of the same diameter and about 1.5 mm thick. The plate surfaces are black anodized to achieve high emittance (about 0.85). This increases the emittance of the surfaces adjacent to the specimens and thereby promotes radiative heat transfer to the specimens and simulates common in-use conditions to which the specimens are subjected.

To reduce exchange of heat between the central MH plate and the surrounding annular IG plate, there is a gap between the two plates. The gap is 1.6 mm (0.063 in) wide and is filled with fibrous mineral wool to inhibit both radiative and convective exchange of heat between the MH and IG plates. The central MH plate is 101.6 mm in diameter, and the radial



Figure 3. Details of outer guard with heater wire and tempering posts.



Figure 4. Layout of heater wire within silicone-rubber main heater element. Four vertical leads at bottom are potential and current leads.

thickness of the surrounding IG plate is 49.2 mm. Although the actual area of the MH plate is 81.07 cm<sup>2</sup>, the area used to compute thermal conductivity includes half the area of the gap, giving a total effective area of 83.65 cm<sup>2</sup> as the area through which metered heat is assumed to flow.

The opposing half-plates of the MH/IG assembly are held together with one countersunk flathead screw (shank diameter: 3 mm) through the MH plate, and three through the IG plate, at locations spaced  $120^{\circ}$  apart. The surface of the MH plate proper is flat across its 102 mm diameter within about 25  $\mu$ m. The surface of the total MH/IG assembly (203 mm diameter) has about the same relative flatness.

The silicone-rubber heater element is also configured as two concentric, electrically separate heater elements. The resistance of the central, metered heater is 7.5  $\Omega$ , and that of the electrically separate IG heater is 19  $\Omega$ . Two wires supply current to the MH heater, and two more allow the potential difference across the heater element to be measured. These four wires are carried across the gap between the MH and IG heater elements by a bridge in the rubber sheet; this bridge fills 20% of the gap perimeter.

To provide for sensing of plate surface temperatures by thermocouples, two shallow grooves are machined into the plate surface facing the specimen. One groove follows a direct radial path across the gap to a point near (10 mm from) the center of the plate, stopping short of the central screw, and the second groove extends radially in to a point 45 mm from the edge of the plate. The grooves have a depth equal to the diameter (0.2 mm including electrical insulation) of the thermocouple leads. These grooves support thermocouple leads cemented flush with the surface of the heater plate.

Uniformity of power generation in the <u>MH element</u>: Rubbing a soft graphite pencil across a sheet of paper laid on the surface of a rubber MH element revealed the layout (Fig. 4) of the heater wire within the MH region, where the compressibility was less than that of the rubber alone. [Note: in the region of the circumferential arcs between the entrance and exit leads and the zigzag pattern, the wire segments are not actually as close together as drafted in Fig. 4. Assume they are spaced as uniformly as in the zigzag region.] Over most of the MH element the wire generally follows parallel paths 5 mm (0.5 cm) apart, giving a mean heater-wire density of 20 mm of length per cm<sup>2</sup> of area. However, in the center of the element the pattern changes in order to make room for the flathead screw used to fasten the upper and lower half-plates together. The dotted rectangle in the center of the MH element (Fig. 4) shows the smallest area (5.5 cm<sup>2</sup>; 6.7 % of the total area), that contains all the non-uniformly distributed wire (61 mm) in the center. Within the rectangle the mean heater-wire density is only 11 mm per cm<sup>2</sup> of area, 55% of the mean wire density outside the rectangle. The power density developed by a given current is directly proportional to the

heater-wire density. Thus within the dotted rectangle at the center of the MH element the generated power density is only 55 % of the value outside the rectangle. This layout applies to the design as we presently have it.

Assuming a value of 236 W/( $m \cdot K$ ) for the thermal conductivity of the two 9 mm aluminum heater plates sandwiching the rubber heater element, a back-of-the-envelope calculation indicates that the radial uniformity in temperature at the outer surfaces of the aluminum plates of the MH assembly is probably acceptably uniform. It could be thought prudent to improve the uniformity of the heat generation by closer spacing of the wire layout in future designs of the heater elements. This would have to be balanced against the increased total resistance resulting from use of the required greater length of heater wire, in turn affecting the required output impedance of the power supply.

# 3.2 Auxiliary Heater Units

Each AH unit (Fig. 1) is composed of two black anodized aluminum half-plates, of the same diameter (203 mm) and emittance as those used for the MH/IG assembly, but lacking a gap. The mounting of thermocouples in 0.2 mm grooves in the surfaces facing the specimens is practically identical to that used for the MH/IG assembly. A rubberized heater element having a resistance of 15  $\Omega$  is sandwiched between the two plates. The total thickness of an AH assembly is about 16 mm. Because the power dissipated in these AH units need not be known, the heater elements are supplied by only two current leads (no potential leads).

Our power supplies, commonly available models, have a maximum output of 55 V and 5 A, most effectively matched to a load with a resistance of 11 ohms.

It may be of minor, yet practical, interest to note that one heater element failed and had to be replaced with another home-made element with a resistance of 56  $\Omega$  (almost four times that of the other, original AH element). The effect of this higher resistance was to increase by about 30 % the time needed by this heater plate to reach its control temperature. However, the additional time required is small compared to the total time required for a complete run, and causes no problems. No degradation of final control stability has been observed in comparison to the stability of the other AH plate with the original value of heater resistance.

# 3.3 Cold Plates

These copper plates (Figs. 1,2), 9.5 mm thick, are cooled by fluid flowing in 6.4-mm (nominal 1/4 in) copper refrigerator tubing soft-soldered to the cold-plate surfaces facing away from the stack. Each cold plate is insulated from its supporting baseplates by four spacers. The central spacer is a solid cylinder 12.5 mm in diameter to which the plate is bolted. Three other spacers are hollow cylindrical shells, equidistantly distributed around the perimeter. All spacers are 90 mm long and made of plastic composite. As coolants, chilled alcohol is circulated from a commercial refrigerated bath, or liquid nitrogen (LN) is supplied from a 160-liter vessel. The flow rate of LN is controlled by adjusting a needle valve in the input line. Loss of cooling power from the circulating coolants is greatly reduced by the use of tubular foam insulation fastened around the flexible input fluid lines, about 50 cm long, connecting the source of coolant to the vacuum-insulated stainless-steel fluid lines at the base of the apparatus.

# 3.4 Offset Insulation

Typically one to three layers of woven fibrous glass fabric are used as offset insulation (Fig. 1) between each AH plate and the adjacent cold plate.

## 3.5 Outer Guard

The outer guard (Fig. 3) is made of 1.6 mm copper sheet. It extends from the lower to the upper cold plate. The guard has an oblong shape to allow room for strain-relief platforms on each heater plate. It consists of two halves, split by a vertical plane. The halves are bolted together around the stack during system operation. Resistive heater wire is wound noninductively on each half of the guard. Copper tempering lugs similar to those used on the edges of the MH/IG and AH plates are bolted to this outer guard. The thermocouple leads and current leads to the central heater plates are wound around and cemented to these lugs on the outer guard to provide further thermal anchoring before the leads meet the stack elements. This precaution is taken to reduce exchange of heat between the metered heater plate and its environment.

# 3.6 Fluid-Cooled Shroud

The shroud (Figs. 1,2) is a cylindrical stainless-steel (about 0.8 mm thick) sheet to which 6.4 mm (nominal 1/4 in) copper tubing is soldered; adjacent coils are spaced about 50 mm apart and spiral around the inner circumference from bottom to top. The shroud's coolant line is connected in series with the lower and upper cold plates. This permits coolant leaving the cold plates to make the shroud an environment at a temperature approximately equal to that of the mean stack temperature. To reduce heat leak from the shroud (and from its coolant) to its supporting baseplate, a sheet of composite fiberglass-epoxy 1 mm thick and 6 mm high was formed into a cylinder and bolted to the bottom of the shroud.

#### 3.7 Specimen Thickness Spacers

When compressible specimens are measured, tubular spacers (Fig..1) between the MH and AH plates fix the thickness of the specimens at a definite value. While ceramic spacers

would have the advantage of a desirably low thermal conductivity, they suffer from two disadvantages: they are brittle (and so the ends chip easily) and they can be sized to the desired length only by grinding. In fact, 5-mm silica tubes (wall thickness of about 1 mm), ground to a length of 25 mm, were used for a time, but they were easily chipped during the process of mounting between the two heater plates. We replaced them with thin-walled (0.25 mm) stainless steel tubing. Although its conductivity is appreciably higher than that of silica or other ceramics, the much thinner wall of the metal spacer compensates in practice to yield an acceptably low total shunt conductance. This fact allows one to exploit the easy machinability of the stainless steel tubing, which can be easily and relatively quickly sized on a lathe to match the thickness of any given specimen.

The stainless-steel spacers are cut to a length slightly less than the free-standing thickness of the specimen. In use, the spacers are mounted within three equally spaced vertical slots cut into the perimeter of each specimen. To inhibit convective and radiative heat transfer within the interior of the spacer tubes, they are filled with fibrous glass or mineral wool insulation. After the stack is assembled with specimens and spacers in position, the stack elements are lightly compressed by means of a screw centered on the upper base plate until all spacers are under light compression. This establishes the specimen thickness as the length of the spacer tube.

## 4. Temperature Sensors

#### 4.1 Thermocouples

Thermocouples sense the temperatures of the central portions of the plate surfaces (TAH, BAH, and MH/IG) adjacent to the specimens, with minimal disturbance of the thermal field of the stack. Type E (Chromel-Constantan4) thermocouples were chosen because this type has the largest sensitivity (about 45  $\mu$ V/K at 173 K (-100°C), and about 60  $\mu$ V/K around and above 300 K, or 27°C) compared to other types (J, K, T) commonly used in the range from cryogenic to room temperatures. Thermocouples were made from wire 0.13 mm (0.005 in) in diameter. Thermocouple leads were cemented into the grooves machined into the plates, flush with the surface.

To increase the thermal resistance of the metallic thermocouple leads, and thereby to reduce unwanted exchange of heat between the stack and its environment, generous lengths of thermocouple wire were used, and the excess neatly coiled in the space between the edge of the stack and the thermocouple terminal strip (TS in Fig. 1). This terminal strip is where the thermocouple leads are connected to copper leads running to the voltmeters. The terminal strip is located on the system base (Fig. 1).

# 4.2 Differential Thermopile

To allow the temperature of the annular IG heater plate to be brought into balance with that of the central MH plate, by sensing the mean temperature difference between them, a differential thermocouple (thermopile) was constructed from the same Type E wire as that used for the absolute thermo-One thermopile was made with a total of 20 junccouples. tions (10 pairs) evenly spaced approximately every 18° around the circumference of the gap, and mounted adjacent to the top half-plate. Adjacent junctions are located on opposite sides of the MH/IG gap, about 12 mm radially from the edge of the gap. A second, identical thermopile was mounted in a similar fashion adjacent to the bottom half-plate. The two 10-pair thermopiles are connected electrically in series, allowing a single thermocouple lead (of identical material to the other two "outer" leads) to be brought out from this common junction; this permits each of the top and bottom thermopile signals to be read separately for the top and bottom half plates of the MH/IG assembly. The control signal for the IG heater plate is derived from the total (series) signal from the two thermopiles. Each thermopile is electrically insulated from its adjacent half-plate by a thin polymer film which provides thermal contact with the plate. The differential thermopile is held in position between the MH and IG plates and the rubberized heater by small pieces of adhesive polymer tape. Leads of the thermopile are also connected to their copper extension leads at the insulated terminal strip on the system base.

# 4.3 Platinum Resistance Thermometers (PRTs)

As a backup to thermocouples in case of failure, and to provide cross-checking of the temperature measurements, platinum resistance thermometers (PRTs) are also used to measure plate temperatures. They are cemented within cavities at the centers of the plate interiors. PRTs provide the input signals to the temperature controllers for the MH and both AH heater plates. To sense the temperatures of these plates, a PRT is mounted in a shallow interior cavity near the center of each heater plate, within the space between the halfplates where the rubberized heater is located. Each PRT and its leads are cemented to the half-plate whose temperature it The PRT for the MH plate is mounted off-center to measures. provide a greater length of wire for thermal anchoring before the wire leaves the stack. Each PRT is a rounded rectangular prism, 11.5 mm long by 4.0 mm wide by 1.4 mm thick. The icepoint resistance of each PRT is about 500  $\Omega$ .

Each PRT was individually calibrated. A polynomial function was fitted to the calibration data of each PRT to permit temperature to be calculated from the measured resistance. PRTs are also mounted on the outer guard, as part of its control system, and on the isothermal thermocouple reference box (details below). All PRTs in the system are connected in series with a standard resistor,  $R_{\rm std}$ , and the series of PRTs and standard resistor is powered by a constant 2 mA current from a dc power supply of high stability. The potential difference across  $R_{std}$ , divided by this resistance, yields the PRT current. The individual potential differences across the PRTs divided by the current gives the resistance of the PRT, from which the temperature is obtained from the calibration relation. The 2 mA current dissipates a "self-heating" power of 2 mW in each PRT, compared to typical MH metered powers of 50 to 1000 mW. This value of current was chosen to take full advantage of the 6-digit resolution of the digital voltmeter measuring the potential differences.

Thermal Anchoring Thermal tempering or anchoring has two purposes: to help a thermocouple junction to come to equilibrium at the temperature of the surface it is to measure, and also to impede the flow of heat along heater and/or sensor leads between the MH/IG heaters and surrounding elements which are at temperatures different from that of the metered MH element. To provide such anchoring, approximately 150 mm lengths of each of the leads of thermocouples, thermopile, and PRTs were wrapped and cemented with varnish around individual cylindrical copper lugs bolted to the outer circumference of the half-plate (on MH or AH) facing the specimen. Both the length and the diameter of the lugs are 13 mm. Similar copper lugs are also bolted to the outer guard, which is maintained at the mean temperature of the specimens in the measurement stack. The leads are also thermally anchored to these lugs before they pass down to the terminal strip. By this means the attempt is made to redirect any parasitic flow of heat either to (or from, depending on the temperature of operation) the outer quard or to the edge of the MH/IG assembly.

There is a disadvantage accompanying the extensive thermal anchoring: it is a lengthy and time-consuming process to remove the varnish anchoring the leads to the various tempering lugs in order to reposition, repair or replace damaged or broken elements (sensors and leads; heater elements) of the heater plates. Cement must be removed from leads on all tempering lugs and plate surfaces, endangering elements still operational, in order to replace just one faulty element. This situation has seriously interfered with our ability to make adjustments that might improve operation.

# 4.4 Thermocouple Reference Junctions

Proper use of an ice reference bath for thermocouples requires great care; establishment around reference junctions of an environment at the temperature of the ice point of pure, air-saturated water is not a trivial task. The use of such a reference bath was avoided, because of the associated problems. Also, an electronic ice reference was not used because of unacceptable imprecision (generally giving an uncertainty of about 0.25 K). Instead, all thermocouple leads are connected to copper leads within an isothermal junction box (the reference; TS in Fig. 1) located within the environmental chamber on the system base, a 12-mm thick plate of aluminum. The box is made of heavy aluminum sheet and is large enough to permit use of about 50 mm of fibrous glass batt insulation inside the box and surrounding the junction strip. This location and mode of thermal protection of the terminal strip tends to reduce the possibility of thermal gradients between pairs of junctions on the same thermocouple. Such gradients would generate parasitic thermal emfs that would degrade the accuracy of temperature measurements.

Locating the isothermal junction box within the chamber also allows the use of a minimal length of thermocouple wire between the stack and the reference point. This reduces the chances of damage (kinks, strains) to the wire, which would lead to errors in measurement. This location also tends to reduce the difference in temperature between opposite ends of the thermocouple leads, and reduces the magnitude of the total thermoelectric signal. Then small variations in temperature of the sensing junction generate a greater fraction of the total signal; this provides greater sensitivity to variation in temperature at the sensing junction.

The Celsius temperature  $T_{ref}$  of the junction strip within the isothermal reference box is measured with a calibrated PRT. This temperature is then converted to the corresponding thermocouple emf  $E(T_{ref}, 0)$  that would be generated by a thermocouple of similar type referenced to the ice point and measuring the same temperature. This emf is added to the emf  $E(T,T_{ref})$  of each thermocouple referenced to the box, to obtain the emf

$$E(T,0) = E(T,T_{ref}) + E(T_{ref},0)$$
 (3)

that the sensing thermocouple would generate if it were referenced to an ice-point bath. From that corrected emf, E(T,0), the temperature T of the measuring junction is obtained. This simple but lengthy procedure is easily carried out within the computer program controlling the apparatus.

# 5. Data Acquisition and Control

# 5.1 The Low-Noise Scanner and DVMs

The copper leads from the isothermal reference box are connected to a commercial low-noise electronic scanner (Fig. 5). This scanner was designed specifically for use with thermocouple circuits and contains two 8-channel switching modules. Each channel is an electronic 2-pole switch that connects both leads of a selected input channel to a single 2-lead output channel. The output is read by a digital microvoltmeter (DVM). All DVMs in the system have a leastcount sensitivity of 10 nV. The present system uses 14 of the 16 available channels. One of the 14 channels is shorted. Readings on this shorted channel serve as an estimate of the combined bias in the readings of the DVM and scanner.





This bias is subtracted from all other channel readings. A second scanner channel reads the voltage across the reference PRT so that the correct temperature can be obtained from the thermocouple outputs (Sect. 4.4 above). Of the twelve remaining scanner channels that are used, six are for thermocouples that read temperatures on the heater plates, and another six are for differential thermocouples that read the temperature difference between two different points on the surfaces of the heater plates. All 14 channels are scanned under the direction of a computer, at regular times and in a fixed order.

Three DVMs (Fig. 5) are used in the control system. A computer reads the encoded output of the scanner DVM over a standard bus (general-purpose interface bus: GPIB) for realtime processing. As the thermocouple signals are converted to thermocouple temperatures by the computer, a temperaturetime plot of any selected heater plate is displayed to inform the operator of the current thermal conditions within the stack.

The PRT current is vital to the operation of five different PRTs on the heater plates, especially the PRT from which the temperature of the MH plate is obtained. The PRT current is read with no interposed relays or switches by a second DVM that has no other tasks assigned. The output of this DVM yields the PRT current as described above in Sect. 4.3.

The third DVM reads the potential difference across the PRT located on the MH plate so that the temperature of the hot-side surface of the specimens can be determined.

# 5.2 The Data-Acquisition and Control Computer

Control of the measurement process is actually divided between two computers (Fig. 5): (a) the master, a scientific desk-top computer; and (b) a slave, a commercial data-acquisition and control (DAC) unit which is a control computer that uses a limited dialect of a BASIC control language. The DAC unit holds up to eight user-selected plug-in modules in rear-panel slots. In this system, three rear-panel modules are DVMs. A fourth module is a 20-channel multiplexer (MUX) that selects voltages to be read by one of the three DVMs. The voltages read are (a) that across a shorted channel, (b) the output of the differential gap thermopile, and (c-f) voltages across the PRTs on the outer guard and on the main and auxiliary heater plates. The voltage across the shorted channel in the MUX serves the same function as that in the scanner. The other two modular DVMs read the voltage across the main heater and across a standard resistor in series with it; from these voltages and the value of the standard resistor the metered power is calculated.

The last two rear-panel modules in the DAC unit are each 4-channel digital-to-analog (D/A) converters. The DAC unit computes the control signal that should be sent to the control input of the power supply for each of the five heaters (outer guard plus the MH, IG, and two AH plates); these control signals are then directed to their respective power supplies through the D/A converters.

Because each D/A converter uses only 12-bit precision, and  $2^{12}$  = 4096, the full-scale output of each controlled 55 V power supply is divided into steps of 55/4096 V, or approximately 13 mV. This step size is adequate for control of all but one of the heater plates. Such a step in the output voltage fed to the 7.5  $\Omega$  main-heater element leads to a power step at mid-range voltages that is unacceptably large for the fine precision required for stable control of the MH plate. For this reason the MH power supply is controlled by two D/A signals in series. One signal coarsely controls the output of the power supply, with 12-bit precision, from zero to full scale, and the other D/A signal provides fine control, within the 13 My coarse step, with another 12 bits of precision. Α 10-turn voltage divider was adjusted so as to add the two (coarse and fine) D/A signals together in the proper ratio, and simulate a 24-bit D/A converter from two 12-bit convert-Because of the imprecision of the step size in the ers. coarse signal from the D/A controller, as well as the imprecision of response in the controlled power supply, the effective precision of this combination is really only about 14 or However the 2- to 3-bit improvement in precision 15 bits. over the original 12-bit level is real, useful, and accept-able. It is equivalent to about a half to a full order of magnitude.

The master computer reads the output of the DAC unit over the GPIB for real-time processing. As the PRT readings are converted to PRT temperatures by the computer, a temperature-time plot of these PRT readings is displayed.

### 6. The Control Program

The guiding purposes behind the development of the control software were (a) to measure, control, and stabilize the heater plate powers and temperatures automatically; (b) to protect the apparatus, specimens, and user from damage or injury; (c) to allow the user to interact with the experiment, by observing or even changing the experimental conditions, as desired.

The computer program used with this system contains more than 4100 lines of BASIC code and so can be neither listed nor completely discussed here. A complete listing is available from the authors. Some features of the control program will be briefly outlined.

# 6.1 Control Computers

Experience with a similar software program controlling another (high-temperature) GHP apparatus [5] showed that the

computational burden of real-time control by means of a single computer left little time for user interaction. The output of a PRT is measured every 5 s, requiring 25 s for all five PRTs; the output of a thermocouple is measured after sufficient time (10 s) is allowed to elapse for the reading on the DVM to stabilize, requiring 160 s for the total of 16 thermocouples. Data conversions are performed for all measurements, real-time displays are updated, and all controller outputs are adjusted during each 5 s control cycle. If a single desk-top computer had been used with this system, the time available for user interaction within each control cycle would have been unacceptably small, or else the frequency of data logging would have to be greatly reduced. The addition of the second, slave, computer allowed control and userinterface tasks to be separated. This enhanced both system efficiency and accessibility because of the parallel operation of the two computers.

The master computer is loaded with the complete package of software that controls the experiment and displays acquired information. At the beginning of a programmed series of runs, the modules of control software are downloaded to the slave computer (the DAC). The portion transferred is that time-intensive part of the control software having to do with frequent, regular measurement of the metered power passing through the specimens as well as measurement and control of temperatures of the heater plates.

### 6.2 PID Control of Heater Plates

A second point of interest concerns the proportionalintegral-derivative (PID) controllers used in this system to control the heater plates. Analog controllers are not used. Instead, PID control algorithms [6] are written in software and executed by the DAC computer. This allows much greater flexibility in the type of control algorithm used, because the controller gain parameters may be made variable through the use of a gain function rather than a gain constant. The gain parameters can be made to vary with magnitude of the error signal, as well as with temperature of operation, which may affect physical time constants that affect the approach to stability.

In particular, for PID control of the power developed in each heater plate, a proportional gain which is greatly reduced at operation near the setpoint temperature is used. The gain function follows an inverted Gaussian (bell curve) function which is near maximum at operation far from the setpoint, but drops to a minimum (near, but not zero) at the setpoint. This makes the response of the controller much less sensitive to random fluctuations in the readings of the heater temperature, and leads to very quiet operation of the heater power at stability. Conversely, for the integral gain, a Gaussian function which rises to its maximum at the setpoint is used. This has the effect, in integrating the power to obtain the average, of heavily weighting the values of power obtained near the setpoint. Derivative control is not even used and its absence is not noticed. Examples of typical operation of a GHP under this PID control algorithm are shown in reference [4].

# 6.3 Features of Measurements

At the beginning of a set of experimental runs, the operator enters through the keyboard the experimental control parameters. These include hot- and cold-plate temperatures as well as specimen areal density (mass per unit area) and thickness, and environmental gas species and pressure. At this time the operator may specify the usual double-sided mode of operation, or may choose to measure only one specimen, with the other specimen and auxiliary heater used as a back guard. She may also specify operation at constant temperature or at constant power. Operation at constant tem-perature is much more efficient initially, in terms of time required to reach stable operation. However, when all heater plates are operating at stable temperatures, it is possible to change control of the MH plate to constant-power operation, at a power equal to the average level of power used to obtain stability. Then the standard deviation of MH power and  $\lambda$  (computed from Eq. 2) is much improved, often by an order of magnitude.

When unusual experimental conditions are being encountered, as for example in the early stages of a new investigation, the user can interact with the program to change the control parameters (gain, reset time, and other time constants, as well as control temperatures) for any individual plate controller. However, when the computer has determined that stability has been reached, then user interaction that would change the conditions of the experiment is locked out until the next run begins.

The operator may enter a series of up to nine experimental runs, which the computer will automatically perform over a period of several days. The operator examines the operation two or three times per day to check for malfunctions, but usually there is no need for continuous attention. As a result, continuous operation is possible, with once- or twice-daily visits by the operator being sufficient.

# 6.4 Safety

The control code contains several tests to improve safety and ensure correct choices for operating conditions. At the time the operator enters experimental parameters, software tests reject input that is obviously wrong, but allow an attempt to re-enter correct parameters. Examples of such rejected input are: a cold-side temperature that is higher that the hot-side temperature, temperature setpoints that are lower than possible with operation with LN coolant, or setpoints that are higher than can be safely allowed.

During an experiment the computer will shut down the run under certain conditions of fault. These conditions include: (a) the program modules for calculating thermocouple or PRT temperatures return values that are unrealistic or unsafe, which could call for dangerously high heater powers; (b) temperatures are too high in the MH plate.

The operator can also abort a run from the keyboard if desired; this directs the computer to shut down all operations in an orderly manner, starting with the power supplies. If the computer should fail to carry out this command, the operator can turn off the heater power manually; the operator's console is next to and within easy reach of the electronics rack containing the power supplies.

# 7. Precision, Accuracy and Repeatability.

For this apparatus the resolution and stability of temperature control of the MH, IG, and AH plates is usually within 10 mK of the final mean value of temperature. The stability of control of the power to the metered heater, under "constant-temperature" control, is typically within  $\pm 0.5\%$  and is often within  $\pm 0.2\%$  of the final mean value of power. The measured values of conductivity are unaffected if the OG heater is controlled within 1 K, as is the usual case.

Measurements of a pair of specimens of SRM 1450b [7] indicated that the uncertainty of conductivity measurements at and near room temperature is about  $\pm 5\%$ . Conductivity values are repeatable within  $\pm 1\%$  over the complete range of temperature from -150 to  $\pm 60^{\circ}$ C (123 to 333 K).

As a further check on the operation of this apparatus, the apparent conductivity (combining conductive and radiative heat transfer) of air at three different thicknesses was By using the one-sided mode, with the top heater measured. held at the same temperature as that of the main heater, heat flowed only downward from the main heater to the bottom plate, to produce thermal stratification of the air specimen and avoid convection. Two annular circular strips, with inner and outer diameters of 190 and 203 mm, respectively, were made from polystyrene plastic, with thicknesses of 3.0 and 5.8 mm. The layer of air enclosed within the interior of an annulus acted as an air specimen, with a thickness given by that of the annulus. A third air specimen, 8.8 mm thick, was formed by stacking one annulus on top of the other. The total rate of heat transfer through an air specimen by conduction and radiation is given by combining Fourier's law of conduction (Eq. (2)) with the Stefan law for radiative heat transfer:

$$Q = \mathbf{A} \cdot \mathbf{\lambda} \cdot \Delta \mathbf{T} / \mathbf{L} + \sigma \cdot \mathbf{A} \cdot (\mathbf{T}_{\mathbf{b}}^{4} - \mathbf{T}_{\mathbf{c}}^{4}) / (2/\varepsilon - 1), \qquad (4)$$

where  $\sigma$  is the Stefan-Boltzmann constant,  $\varepsilon$  is the emittance of the two plates, and  $\Delta T$  is  $(T_h - T_c)$ . The second, radiative, term is independent of thickness L between the plates. By simple algebraic factoring and collection of similar terms



Figure 6. "Apparent" conductivity, including radiative transport, of air at 20°C (crosses) and 25°C (plusses), as a function of thickness of air specimen. The intercept of a line with the conductivity axis gives the conductivity alone, without radiative transport (see text). For comparison, the accepted value of thermal conductivity of air (25.76 mW/m·K) is shown as a circle on the conductivity axis.

## (A, $\Delta T$ and L) this expression can be rewritten as

$$\mathbf{Q} = \mathbf{A} \cdot \left[ \lambda + \sigma \cdot \left( \mathbf{T}_{\mathbf{h}}^{3} - \mathbf{T}_{\mathbf{h}}^{2} \mathbf{T}_{\mathbf{c}}^{-} \mathbf{T}_{\mathbf{h}}^{2} \mathbf{T}_{\mathbf{c}}^{2} - \mathbf{T}_{\mathbf{c}}^{3} \right) \cdot \mathbf{L} / \left( 2/\epsilon - 1 \right) \right] \cdot \left[ \Delta \mathbf{T} / \mathbf{L} \right], \quad (5a)$$

$$= \mathbf{A} \cdot \lambda_{app} \cdot \Delta \mathbf{T} / \mathbf{L} \,. \tag{5b}$$

The expression (5a) combines the term for radiative transport with the conductive term in the first set of square brackets; the combination then defines an "apparent" conductivity through (5b) by

$$\lambda_{app} = [\lambda + \sigma \cdot (T_{h}^{3} - T_{h}^{2}T_{c} - T_{h}T_{c}^{2} - T_{c}^{3}) \cdot L/(2/\epsilon - 1)], \qquad (5c)$$

which is made up of the conductivity of air,  $\lambda$ , plus a second term ("radiative conductivity") now linearly dependent on L, the specimen thickness.

When the expression for  $\lambda_{app}$  given by Eq. 5c is plotted as a function of L, the combination of radiative constants acts as the coefficient of L and gives the slope of a straight line; the intercept of the line is the conductive term,  $\lambda$ , for air. Figure 6 shows three measurements (X) of the apparent conductivity of air for the three different air specimen thicknesses: 3.0, 5.8 and 8.8 mm, measured at a mean temperature of 20°C; another three measurements (+) are shown for 25°C. The intercepts (X, +) of the two fitted lines with the conductivity axis are 24.69 and 24.56 mW/(m·K). Solving for  $\varepsilon$ , by equating the experimental values of the slopes to the coefficient of L in Eq. (5c), gave values of 0.85 and 0.86. The emittance of anodized aluminum is not accurately known, but the mean of our two values, 0.85, is in good agreement with the available data and with a previous measurement of the intercepts, 24.62, is lower than the value of 25.76 (0 on the conductivity axis of Fig. 6) for the conductivity of air at 22°C, by 4%.

Work to improve the accuracy of measurements of conductivity with this apparatus is continuing.

Note added in proof: One reviewer pointed out that the 2 mA current powering the PRTs causes a self-heating of 2 mW. The self-heating of the PRT on the MH plate constitutes a small systematic error which has the correct sign, but too small a magnitude, to be responsible for more than at most 1% of our values for the conductivity of air. This is insufficient to account for the above 4% difference between our values and the accepted value. But the oversight is being corrected.

### 8. Acknowledgements

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# Robert R. Zarr<sup>1</sup>

# INTRA-LABORATORY COMPARISON OF A LINE-HEAT-SOURCE GUARDED HOT PLATE AND HEAT-FLOW-METER APPARATUS

**REFERENCE:** Zarr, R. R., "Intra-Laboratory Comparison of a Line-Heat-Source Guarded Hot Plate and Heat-Flow-Meter Apparatus," <u>Insulation Materials: Testing and Applications. 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: The apparent thermal conductivity<sup>2</sup> and thermal resistance of several building insulations were determined at the National Institute of Standards and Technology (NIST) using three apparatus. Reference values were determined with NIST's 1-metre Line-Heat-Source Guarded Hot Plate (GHP). The other two apparatus were heat-flow-meter (HFM) apparatus calibrated with a Standard Reference Material of fibrous-glass board (138 kg/m<sup>3</sup>). The apparatus - NIST's 1-metre GHP; the 610 mm HFM; and the 305 mm HFM - were operated at a mean temperature of 24°C and temperature difference of 22°C across the thickness of the specimen. Measurements of thermal conductivity were determined for individual specimens nominally 25 mm thick, covering a range of thermal conductivity from 0.02 to 0.16 W/(m • K). The thermal resistance of thick specimens (1 to 7  $m^2 \cdot K/W$ ) of cellular plastic, fibrous and loose-fill insulation was also determined. Deviations from reference values of the GHP were calculated for the HFM apparatus. For the 610 mm HFM, the majority of deviations were within  $\pm 3\%$  of the GHP. Substantial improvement was achieved by calibrating the 305 mm HFM as a function of thickness; however, deviations ranged from +1.5 to +9.2% from the GHP. The apparatus, test procedure, and analysis of the deviations are described in the paper.

**KEYWORDS:** heat flow meter, heat flux transducer, intra-laboratory comparison, guarded hot plate, line heat source, precision, thermal conductivity, thermal insulation, thermal resistance

Two methods to determine the thermal conductivity ( $\lambda$ ) or thermal resistance (R) of flat specimens of insulation are the guarded hot plate and the heat-flow-meter apparatus (ASTM Standard Test Method C 177 and C 518, respectively). Both methods use parallel plates to induce a steady temperature gradient across the thickness of the specimen. Whereas, the guarded hot plate method is considered absolute for determining thermal conductivity, the heat-flow-meter apparatus requires calibration. A specimen having a known thermal conductivity is required to determine the calibration coefficient of

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<sup>&</sup>lt;sup>2</sup>For brevity, the term "thermal conductivity" will be used in the remaining text.

the apparatus. For this purpose, Standard Reference Materials (SRMs) and Transfer Standards (TS) of thermal resistance are available from the National Institute of Standards and Technology (NIST).

As new insulations gain acceptance, industry and NIST continually encounter the question: How suitable are Reference Materials currently offered by NIST for calibrating HFMs for these new insulations? At issue is whether a calibration coefficient determined using fibrous-glass insulation is appropriate for another material such as cellular-plastic foam. To provide insight, NIST proposed calibrating industry's test equipment using NIST's Reference Materials (i.e., fibrous-glass board SRM) and conducting a laboratory comparison with several building materials. Assisting NIST in planning the comparison was an Advisory Panel<sup>3</sup> from industry.

Three apparatus were selected for comparison; NIST's 1-metre Line-Heat-Source Guarded Hot Plate (GHP) [1], and two commercial heat-flow-meter apparatus. The commercial heat-flow-meter apparatus were recommended by the Advisory Panel as representative of industry's test equipment. The large apparatus ( $\text{HFM}^{\#}1$ ) is generally used for precision laboratory measurements; and the small apparatus ( $\text{HFM}^{\#}2$ ), as a quality-assurance instrument. The heat-flow-meter apparatus,  $\text{HFM}^{\#}1$  and  $\text{HFM}^{\#}2$ , have a plate size of 610 and 305 mm square, respectively. Details of the apparatus are provided later.

Specimens of building materials were selected to provide a range of thermal conductivity from 0.02 to 0.16 W/( $\mathbf{m} \cdot \mathbf{K}$ ) and thickness from 25 to 184 mm. The materials were obtained from four sources: NIST; locally from distributors; directly from members of the Advisory Panel; or, from manufacturers participating in the 1990 ASTM C 687 Round Robin [2]. Specimens were tested sequentially in the GHP, HFM<sup>#</sup>1, and if suitable, HFM<sup>#</sup>2. Replicate measurements for some specimens were conducted with the HFM apparatus. This paper describes the materials and apparatus, evaluation of NIST's fibrous-glass board as a Transfer Standard of thermal resistance, calibration of the HFM apparatus, and results of the intra-laboratory comparison.

# DESCRIPTION OF TEST MATERIALS

A description of test materials, their physical properties, and source is presented in Table 1.

#### Specimen Preparation

The materials described in Table 1 were placed in an environment of  $22^{\circ}$ C and  $40 \pm 10\%$  relative humidity (RH) after delivery. Rigid and flexible materials were obtained in one of two forms; board or roll. Boards of XPS and PIR obtained from the manufacturer (Table 1) had been aged a minimum of 6 months prior to delivery. Specimens were prepared by cutting a disk (nominally one-metre in diameter) from the center of a half-board (1.2 by 1.2 m) or slab. The disk was placed on a large granite plate having a flatness of 0.0152 mm (unilateral tolerance). Nine individual measurements of thickness were made using a caliper (0.1 mm resolution) and averaged. A variation of two times the standard deviation (s) is shown in Table 1 for rigid and flexible materials.

Specimens of loose-fill insulation (Table 1) were prepared from material supplied by manufacturers participating in the 1990 ASTM C 687 Round Robin [2]. Specimens were blown into a test frame (610 by 610 by 150 mm) using a pneumatic blowing machine. A thin, non-metallic screen stretched across

<sup>&</sup>lt;sup>3</sup>The Advisory Panel was formed at NIST in December 1987. Subsequent meetings were held in conjunction with ASTM Committee C-16. Members consist of (in alphabetical order): Amoco Foam Products Co., Armstrong World Industries, Inc., DOW Chemical USA, Jim Walters Research Corp., Holometrix, Inc., Huntsman Chemical Corp., Manville Sales Corp., Mobay Corp., Oak Ridge National Laboratory, and UC Industries, Inc.

_	Material	Bulk Density <sup>a</sup> (kg/m <sup>3</sup> )	Thickness <sup>ab</sup> (mm)	Source and ASTM Specification
1.	Elastomeric cellular foam, flexible, skin one side	54	25 ± 0.5	Distributor (C 534)
2.	Extruded polystyrene (XPS)	27 28 30	$26 \pm 0.5 \\ 38 \pm 0.9 \\ 51 \pm 0.1$	Manufacturer (C 578)
3.	Fibrous-glass board	134-144	26 ± 0.4	NIST (SRM-internal)
4.	Fumed-silica board	323	26 ± 0.2	NIST (SRM-1449)
5.	Gum rubber, 35 durometer $\pm$ 5	1003	26 ± 0.3	Distributor (D 2000)
6.	Molded expanded polystyrene, from beads (MEPS)	26	24 ± 0.4	Distributor (C 578)
7.	Polyisocyanurate (PIR) with foil facers	32	$26 \pm 0.5$	Distributor
8.	Polyisocyanurate (PIR) with permeable facers	45	40 ± 1.3	Manufacturer
9.	Cellulose loose-fill	29	178-184	Manufacturer (C 739)
10.	Fibrous-glass batt, phenolic binder	11	152	Manufacturer
11.	Mineral wool loose-fill	31-36	158-164	Manufacturer (C 764)
12.	Unbonded fibrous-glass loose-fill	7.1	167	Manufacturer (C 764)

#### TABLE 1 -- Specimens of Building Materials

<sup>a</sup>Range provided for multiple specimens

<sup>b</sup>Variation of 2 times the standard deviation provided for rigid/flexible specimens

the bottom of the frame supported the loose-fill insulation. The nodule size (i.e., un-opened clumps of fibers) of the mineral wool and unbonded fibrous-glass loose-fill was not measured, but estimated to be less than 25 mm. Values of bulk density for fibrous-glass batt and loose-fill insulation reported in Table 1 were computed for the metering area of a "cookie cutter" 406 mm in diameter.

# DESCRIPTION OF TEST PROCEDURE

#### Apparatus [Variable]

Characteristic features of the apparatus used in this study are shown in Figure 1 and Table 2. Further details of NIST's 1-metre GHP are available in Reference [1]. The heat-flow-meter apparatus were selected as representative of industry's test equipment. HFM<sup>#</sup>1 was purchased by NIST in 1978, but was recently up-graded by the manufacturer in 1989. HFM<sup>#</sup>2 was loaned to NIST by a member of the Advisory Panel and delivered in January 1990. Both were built by the same manufacturer. All apparatus were located in the same laboratory having ambient conditions of  $21 \pm 1^{\circ}$ C and  $40 \pm 2\%$  RH.



FIG. 1 -- Schematic configuration of test apparatus - (1) P<sub>1,2</sub>, heating and cooling plates, respectively;
(2) S<sub>1,2</sub>, specimen and "dummy" specimen, respectively;
(3) H, heat-flux-transducer (HFT);
(4) M, substrate material between heat-flux-transducer and heating plate

TABLE	2		Test	Apparatus
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	GHP	HFM#1	HFM <sup>#</sup> 2
1. ASTM Test Method:	C 177. C 1043	C 518	C 518
2. Plate Geometry:	Round	Square	Square
3. Plate Dimensions A,B (mm):	406, 1016	254, 610	100, 305
4. Max. Specimen Thickness (mm):	380	200	100
5. Primary Guard:	36-junction thermopile	64-junction thermopile	
6. Secondary Guard:	Environmental Chamber	Environmental Chamber	Edge Insulation
<ol> <li>Plate Emittance (Normal) Cold-face (top): Hot-face (bottom):</li> </ol>	0.89 ± 0.01 0.89 ± 0.01	0.82 <sup>*</sup> 0.80 <sup>*</sup>	Unknown Unknown
8. Plate Sensors:	PRT 0.01°C	Type K Special limits $\pm 1.1$ °C	Type T
9. Direction of Heat Flow:	Up, 1-sided	Up	Up
10. HFT, Type of Junctions:	-	Co-planar [3]	Spiral-wound
11. HFT Substrate Material and Thickness (mm):		Acrylic Plastic, 1.6.	Silicone Rubber, 4.8
12. References:	[1,4-6]	[7]	

\*Value reported by manufacturer of HFM

#### Procedure

For the comparison, the apparatus were operated at a mean specimen temperature  $(T_M)$  of 24°C and a temperature difference ( $\Delta T$ ) of 22°C. Each apparatus was operated with heat flow in the upward direction. For the GHP, the bottom plate was maintained at the same temperature as the hot plate and a "dummy" specimen was placed between them. Specimens of materials described in Table 1 were tested sequentially in the apparatus by order of size; GHP, HFM<sup>#</sup>1, and, HFM<sup>#</sup>2. However, the majority of materials were tested only in the GHP and HFM<sup>#</sup>1 because the small HFM apparatus was unavailable when testing began in 1989.

Rigid and flexible specimens were prepared for the apparatus as illustrated in Figure 2. First, a large disk was tested in the GHP. Subsequently, a 610 by 610 mm square was excised for HFM<sup>#</sup>1; and finally, a 305 by 305 mm square for HFM<sup>#</sup>2. For these materials, the variation of average density for each specimen was no more than 2%; and in most cases, less than 1%. Variations in  $\lambda$  or R would be expected to be less than 0.1%. Specimens of XPS were stacked for thicknesses of 77, 102, and 159 mm; specimens of PIR w/permeable facers, for thicknesses of 80, 121, and 162 mm. Specimens (and stacks) of XPS and PIR w/permeable facers were tested sequentially in all apparatus within 4 days to minimize effects (if any) of thermal aging of the material.





FIG. 2 -- Relative size of the meter areas for rigid and flexible specimens

Specimens of loose-fill insulation and fibrous-glass batt were tested in the GHP and  $\text{HFM}^{\#}1$ , in that order. The specimens of loose-fill insulation were tested in their test frames and the effect of the thin, support screen was neglected during tests. As mentioned, the density of the loose-fill specimens was determined by a "cookie-cutter" that removed a metering area 406 mm in diameter from the center of the specimen. Consequently, the density of the 254 mm square metering area was not determined. Previous work [8] has shown that on average, the frame density is 5% less than a 254 mm metering area.

# EVALUATION OF TRANSFER STANDARDS OF THERMAL RESISTANCE

Transfer Standards for calibrating the HFM apparatus were selected from an SRM lot of fibrousglass boards (Lot 1970) previously described by Siu [9]. The material was the same as the three lots [9,10] supplied later to the Standard Reference Materials Program as SRMs 1450, 1450a, and most recently, 1450b. The thermal conductivity ( $\lambda$ ) of Lot 1970 was predicted by the following equation [9]:

$$\lambda = 0.02052 + 1.303 \times 10^{-5} \cdot \rho + 4.015 \times 10^{-10} \cdot T_M^{-3} \tag{1}$$

where:

 $\rho$  = bulk density of the specimen (kg/m<sup>3</sup>) [117  $\leq \rho \leq 129$  kg/m<sup>3</sup>], and  $T_M$  = mean specimen temperature (K) [255  $\leq T_M \leq 330$  K].

Regression coefficients for Eq 1 were determined from a fit of 37 measurements of thermal conductivity of Lot 1970 using NIST's 200 mm square Guarded Hot Plate [9]. Specimens were dried at 100°C for 24 hours prior to placement into the apparatus. At 24°C, the uncertainty in  $\lambda$  was estimated to be  $\pm 2\%$  which includes both material variability and measurement uncertainty [9].

#### Measurement as Function of Temperature

To investigate the uncertainty in the value of  $\lambda$  predicted by Eq 1, the thermal conductivity of an individual specimen was determined using NIST's 1-metre GHP. Prior to testing, the specimen was removed from an environment of 21°C and 40 ± 10% RH where it had been placed for several years. Nineteen measurements of thermal conductivity were conducted over the temperature range of 10 to 50°C. Measurements progressed in-sequence starting at 10°C, proceeding to 50°C, and then returning again to 24°C. The sequence of measurements required 35 days. A repeat measurement was conducted at 24°C for 24 hours one year later.

Measurements of thermal conductivity as a function of  $T_M$  are illustrated in Figure 3 as individual data points. Values of  $\lambda$  predicted by Eq 1 are shown as a solid line. The agreement between predicted values of  $\lambda$  and measured values ( $\lambda_r$ ) was -1.2% at 10°C and less than 1% from 24 to 50°C. At 24°C, the deviation ranged from +0.2 to -0.3%. The lower value of  $\lambda_r$  at 24°C was obtained at the end of sequential testing after a  $T_M$  of 50°C was achieved. The difference was attributed to moisture migrating from the specimen during measurements at  $T_M$  above 40°C.

#### Measurements of Stacked Specimens

To provide Transfer Standards of high thermal resistance, individual boards of Lot 1970 were stacked and their thermal resistance determined with NIST's 1-metre GHP. Six measurements of thermal resistance ( $R_r$ ) at  $T_M$  of 24°C and  $\Delta T$  of 22°C were conducted for thicknesses of 26, 53, 79, 106, 132, and 158 mm. Prior to testing, the specimens were removed from an environment of 21°C and 40  $\pm$  10% RH where they had been placed for several years. The density of the stacked specimens ranged from 135 to 140 kg/m<sup>3</sup>.



FIG. 3 -- Thermal conductivity of fibrous-glass board (26 mm) as a function of temperature

Measurements of  $R_r$  as a function of thickness ( $\ell$ ) are illustrated in Figure 4 as individual data points. A linear-regression equation,

$$R_r = -0.00878 + 0.0309 \cdot \ell$$

is shown as a solid line. At 24°C, the uncertainty of  $R_r$  for the Transfer Standards was estimated to be less than  $\pm 1\%$  [6]. Values of  $R_r$  were subsequently used to calibrate the HFM apparatus at thicknesses greater than 26 mm.

#### CALIBRATION OF HEAT-FLOW-METER APPARATUS

Calibration of a HFM apparatus for a specific test condition is determined by the relationship, S = q/E, where q is the heat flux through the specimen and E is the emf output of the heat-flux-transducer. For the comparison, the HFM apparatus were calibrated at  $T_M$  of 24°C and  $\Delta T$  of 22°C<sup>4</sup> in accordance with ASTM Test Method C 518-85. A discussion of HFM apparatus and calibration errors is available in References [11-15].

<sup>&</sup>lt;sup>4</sup>For HFM<sup>#</sup>2, the  $\Delta T$  increased slightly with specimen thickness ( $\ell$ ). At 106 mm, the  $\Delta T$  was 24°C (Figures 5 and 10). The plate temperatures of HFM<sup>#</sup>2 were set with small potentiometers that required 1 to 2 days to adjust for final temperatures. To facilitate testing of thick specimens ( $\ell > 26$  mm), the potentiometers were not adjusted.



FIG. 4 -- Thermal resistance of fibrous-glass board (135  $\leq \rho \leq$  140 kg/m<sup>3</sup>) as a function of specimen thickness

#### Calibration Coefficient (S)

Calibration coefficients  $(S_{1,2})$  for HFM<sup>#</sup>1 and HFM<sup>#</sup>2 were determined from the following equation:

$$S = (T_H - T_C)/(E \cdot R_r) \tag{2}$$

where:

$$\begin{split} S &= \text{calibration coefficient at the specimen thickness and mean} \\ & \text{temperature of the transducer ((W/(m^2 \cdot mV)),} \\ T_H &= \text{hot plate temperature (°C, or K),} \\ T_C &= \text{cold plate temperature (°C, or K),} \\ E &= \text{emf of the heat flux transducer (mV), and,} \\ R_r &= \text{specimen thermal resistance of Transfer Standard (m^2 \cdot K/W).} \end{split}$$

The calibration coefficient (S) is sensitive to changes in temperature and edge heat losses of the transducer (i.e., lateral heat flow due to temperature gradients across the face of the transducer in contact with the specimen). For measurement of thick specimens, S was determined using the stacked Transfer Standards of high thermal resistance described above. Additional edge insulation was not used for these calibration tests.

#### Calibration as a Function of Thickness

The two HFM apparatus were calibrated using the identical stack of fibrous-glass boards described above. Values of S were determined for HFM<sup>#</sup>1 at thicknesses of 26, 53, 79, 106, 132, and 158 mm, and for HFM<sup>#</sup>2 at 26, 53, 79, and 106 mm. Values of S as a function of thickness  $(\ell)$  are plotted in Figure 5 as individual data points. For HFM<sup>#</sup>1, the variation of S<sub>1</sub> was about 4% for  $26 \le \ell \le$ 158 mm. Because of this small effect and the single replicate of data points, S<sub>1</sub> was treated as a constant for all measurements. For HFM<sup>#</sup>2, however, S<sub>2</sub> increased 90% as  $\ell$  varied from 26 to 106 mm. A quadratic equation in  $\ell$  was fit to the inverse of the calibration coefficient (S<sub>2</sub>):

$$1/S_2 = 3.480 \cdot 10^{-2} - 1.128 \cdot 10^{-4} \cdot \ell - 5.580 \cdot 10^{-7} \cdot \ell^2. \tag{3}$$

The form  $1/S_2$  was selected to linearize the data. The largest residual of the fit was -0.7%. Equation 3 was used to predict values of  $S_2$  for measurements of thick specimens ( $26 < \ell \le 106$  mm). The large dependence on thickness of HFM<sup>#</sup>2 was believed to be caused by lateral heat loss at the edges of the heat-flux-transducer.



FIG. 5 -- Calibration coefficient of the HFM apparatus as a function of specimen thickness

# Repeatability of Calibration

A running record was maintained for both HFM apparatus. Calibration measurements were conducted at intervals varying from one day to one month using the same specimens of fibrous-glass board (26 mm thick). The specimens were heated in a hot-air oven at 90°C for a minimum of 15 hours prior to placing into their respective apparatus. The calibration coefficients  $(S_{1,2})$  are plotted as a function of time (in days) in Figures 6a and 6b, respectively. Values of  $S_1$  decreased about 0.8% over a period of one year (see Figure 6a). The reason for the decrease is not known at present. Values of  $S_2$  have a greater variability than  $S_1$ . The average of the  $S_2$  data is  $31.7 \pm 0.4 \text{ W/(m}^2 \cdot \text{mV})$  ( $\pm 2s$ ).


FIG. 6 -- Repeatability of calibration coefficient: a) HFM<sup>#</sup>1; b) HFM<sup>#</sup>2

### COMPARISON OF GHP AND HFM MEASUREMENTS

For the comparison, reference values of thermal conductivity  $(\lambda_r)$  and thermal resistance  $(R_r)$  were determined by NIST's 1-metre GHP. The deviation ( $\delta$ ) of measurements determined by HFM<sup>#</sup>1 and HFM<sup>#</sup>2 was defined by:

$$\delta = (y_{1,2} - y_r)/y_r \cdot 100 \tag{4}$$

where the term, y, represents either  $\lambda$  or R, and the subscripts 1, 2, and r identify data from HFM<sup>#</sup>1, HFM<sup>#</sup>2, and GHP, respectively.

For several specimens, measurements were repeated in the HFM apparatus. Replicate measurements were conducted at daily intervals by the same individual. At the conclusion of a test, the specimen was removed from the HFM apparatus and placed in an environment of 22°C and 40% RH (unless otherwise noted) for a minimum of 15 hours until the next test. On the following day, the specimen was re-installed in the HFM apparatus in the same orientation as the previous test. Heat flow was in the upward direction.

## Data Collected from HFM#1

Two groups of data were collected from HFM<sup>#</sup>1; thermal conductivity ( $\lambda_1$ ) and thermal resistance ( $R_1$ ). Measurements of  $\lambda_1$ , their average, and variability (3s) are summarized in Table 3 in order of increasing variability. Measurements of fumed-silica board (SRM-1449) were normalized to standard atmospheric pressure (STP=101.325 kPa) using a pressure-dependent equation [16]:

$$\lambda_{STP} = \lambda_P + 7.727 \cdot 10^{-5} \cdot (101.325 - P)$$

where P = atmospheric pressure in kPa, and  $\lambda_p$  = the thermal conductivity in W/(m · K) determined at P.

		Measure	ment <sup>a</sup> of λ <sub>1</sub> (V	V/(m • K))			
Material	#1	#2	#3	#4	#5	Average	3s(%)
Gum rubber	0.1559	0.1560	0.1558	0.1557	0.1558	0.156	0.2
MEPS	0.03289	0.03292	0.03295	0.03297	0.03292	0.0329	0.3
Fumed-silica board, <sup>bc</sup> SRM-1449	0.02053	0.02055	0.02057	0.02057	0.02058	0.0206	0.3
Fibrous-glass board <sup>c</sup>	0.03276	0.03279	0.03273	0.03278	0.03282	0.0328	0.3
Fibrous-glass board <sup>d</sup>	0.03278	0.03281	0.03291	0.03285	0.03282	0.0328	0.5
PIR, foil-facers	0.02510	0.02510	0.02507	0.02509	0.02493	0.0251	0.9
Elastomeric cellular foam	0.03606	0.03619	0.03617	0.03591	0.03589	0.0360	1.2
PIR, permeable facers	0.02267						
XPS #1	0.02801						

#### TABLE 3 - Repeatability of HFM#1

<sup>a</sup> Extra-digit included for calculation of standard deviation (s).

<sup>b</sup>  $\lambda_1$  normalized to atmospheric pressure of 101.325 kPa.

<sup>c</sup> Specimen conditioned at 90°C prior to each measurement.

<sup>d</sup> Specimen conditioned at 22°C prior to each measurement.

As shown in Table 3, the variability is dependent on material and specimen conditioning. A variability less than 0.5% was observed for specimens of gum rubber, MEPS, fumed-silica board, and fibrous-glass board conditioned at 90°C prior to each measurement. Larger variability was observed for specimens of PIR w/foil facers, elastomeric cellular foam, and fibrous-glass board conditioned at 22°C prior to each measurement. Note that measurement #5 for PIR w/foil facers was low in comparison to the previous four measurements and was considered a possible outlier. Re-examining the test data proved the value valid; however, if excluded, the variability was lowered to 0.2% (3s). The source of variability for elastomeric cellular foam was not investigated.

# Deviations of $\lambda_1$ and $R_1$ (HFM<sup>#</sup>1)

The mean deviation of  $\lambda_1$  for each material is plotted in Figure 7 for a range of 0.0205 <  $\lambda_r$  < 0.156 W/(m · K). Error bars represent the maximum and minimum values of the deviations of  $\lambda_1$  for the given material. The deviations were within  $\pm 3\%$  of the GHP measurements with the exception of  $\delta_1$  for PIR w/foil facers (+5.6%). The source of error for this deviation was believed to be the highly conductive metal-foil facers which caused significant lateral heat flow along the hot-face of the heat-flux-transducer in contact with the specimen. Note that  $\delta_1$  for PIR w/permeable facers was for a 40-mm-thick specimen. All other specimens were nominally 25 mm thick.

Deviations of  $R_1$  for fibrous-glass batt, loose-fill insulation, and specimens of PIR w/permeable facers and XPS are plotted in Figure 8 for a range of  $0.9 < R_r < 7 \text{ m}^2 \cdot \text{K/W}$ . Specimens of PIR w/permeable facers and XPS were stacked to achieve R-values greater than  $2 \text{ m}^2 \cdot \text{K/W}$ . The same stacks were tested previously in the GHP. Again, the deviations were within  $\pm 3\%$  of the GHP measurements with the exception of one specimen of unbonded fibrous-glass loose-fill (-6.1%). A possible source of error for this specimen of unbonded fibrous-glass loose-fill was the variation in specimen density between the metering areas of the GHP and HFM<sup>#</sup>1 apparatus [8]. Evidence of bias was observed in deviations of PIR w/permeable facers and XPS and will be discussed later in the paper.



FIG. 7 -- Deviations of thermal conductivity ( $S_1 = constant$ )



FIG. 8 -- Deviations of thermal resistance ( $S_1 = constant$ )

# Data Collected from HFM#2

Similarly, two groups of data were collected from HFM<sup>#</sup>2: thermal conductivity ( $\lambda_2$ ) and thermal resistance (R<sub>2</sub>). Measurements of  $\lambda_2$ , their average and variability (3s) are summarized in Table 4 in order of increasing variability. Values of  $\lambda_2$  were determined for S<sub>2</sub> based on calibration at 26 mm (S<sub>2</sub> = constant). In general, the variability of HFM<sup>#</sup>2 was greater than HFM<sup>#</sup>1.

TAB	LE 4		Repeatability	of	HFN	1#2
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		Measur	ement <sup>a</sup> of $\lambda_7$ (	W/(m • K))			
Material	#1	#2	#3	#4	#5	Average	3s(%)
MEPS	0.0340	0.0340	0.0342	0.0342	0.0340	0.0341	0.9
Fibrous-glass board <sup>b</sup>	0.0323	0.0325	0.0324	0.0326	0.0324	0.0325	0.9
XPS #2 XPS #1	0.0285 0.0277	0.0282	0.0286	0.0283	0.0286	0.0285	2.0

<sup>a</sup> Extra-digit included for calculation of standard deviation (s).

<sup>b</sup> Calibration specimen conditioned at 22°C prior to each measurement.

# Deviations of $\lambda_2$ and $R_2$ (HFM<sup>#</sup>2)

The mean deviation of  $\lambda_2$  for each material is plotted in Figure 9 for a range of 0.028  $< \lambda_r < 0.033 \text{ W/(m \cdot K)}$ . Error bars represent the maximum and minimum values of the deviation of  $\lambda_2$  for the given material. The deviations were within  $\pm 3.5\%$  of the GHP measurements; however, fewer materials were tested in HFM<sup>#</sup>2 due to its late availability. All specimens had a nominal thickness of 25 mm.



FIG. 9 -- Deviations of thermal conductivity ( $S_2 = constant$ )

Deviations of  $R_2$  for specimens of PIR w/permeable facers and XPS are plotted in Figure 10 as individual data points for  $0.9 < R_r < 3.5 \text{ m}^2 \cdot \text{K/W}$ . The specimen stacks were the same as those tested previously in the GHP and HFM<sup>#</sup>1.  $R_2$  was determined for two cases: 1)  $S_2$  constant (based on calibration at 26 mm); and, 2)  $S_2$  computed using Eq 3. For  $S_2$  constant, deviations ranged from +11 to +63% for PIR w/permeable facers, and +3.1 to +85% for XPS. Computing  $S_2$  with Eq 3 reduced the range of deviations to +5.2 to +9.2% for PIR w/permeable facers, and +1.5 to +5.5% for XPS. Evidence of bias was observed for deviations of both materials.

## ANALYSIS OF HFM DATA

The deviations shown in Figures 7 to 10 are the percentage difference between the GHP and HFM measurements. Of particular interest is whether measurements made on a given material are biased. As mentioned, evidence of bias was observed for multiple measurements of PIR w/permeable facers and XPS for both HFM<sup>#</sup>1 and <sup>#</sup>2. It is helpful to think of the "true" bias for measurements made by a given instrument on a given material as the long-run average of deviations computed from measurements made on a large number of specimens of the given material. The long-run average will be zero if the instrument is not biased for the given material. In practice, the true bias is rarely, if ever, known.



FIG. 10 -- Deviations of thermal resistance. (1) Raw data (solid data points);  $S_2 = \text{constant}$ ; and (2) corrected for thickness using Eq 3 (open/solid data points):  $1/S_2 = 3.480 \cdot 10^{-2} - 1.128 \cdot 10^{-4} \ell - 5.580 \cdot 10^{-7} \ell^2$ 

An estimate of bias for a given material is available when measurements have been taken on more than one specimen of the given material. For these materials, an interval may be computed that contains the true bias with high probability, for example 95% confidence intervals. Confidence intervals of 95%  $(v_1, v_2)$  for the true bias were computed from the following formula:

$$(v_1, v_2) = m \pm t \cdot s \cdot n^{-\frac{1}{2}} \tag{4}$$

where:

- n = number of deviations ( $\delta$ ) for the given material,
- $m = average of deviations (\delta)$  for the given material,
- = t-value from Student's t having a tail probability of 0.025 and t degrees of freedom = n - 1, and
- = standard deviation of deviations ( $\delta$ ) for the given material. s

Confidence intervals of 95% for materials having three or more measurements (n > 3) are given in Table 5.

Apparatus	Material	n	m (%)	s (%)	(v <sub>1</sub> , v <sub>2</sub> ) (%, %)
HFM <sup>#</sup> 1	PIR, permeable facers	4	-1.73	0.891	(-3.15, - 0.31)
HFM <sup>#</sup> 1	XPS	6	+2.00	0.483	(+1.49, +2.50)
HFM <sup>#</sup> 1	Cellulose loose-fill	3	-0.84	0.584	(-2.29, +0.61)
HFM <sup>#</sup> 1	Mineral wool loose-fill	4	-0.46	1.78	(-3.28, +2.37)
HFM <sup>#</sup> 2	XPS	5	+3.55	1.55	(+1.39, +5.70)

If a particular interval does not contain zero, then the measurement made on the given material by the given instrument are biased. From Table 5, we see that measurements for PIR w/permeable facers and XPS were biased for HFM<sup>#</sup>1, while those for cellulose and mineral wool loose-fill were not. Although small, the bias in measurements of PIR w/permeable facers (-1.73%) and XPS (+2.00%) was opposite in direction (-/+) suggesting that the bias was material dependent. This implies that the source of bias may depend on the type of specimen used to calibrate the HFM apparatus. Further work has been proposed to investigate this effect.

From Table 5, we conclude that measurements made on XPS measured with  $HFM^{\#2}$  were also biased with the bias being positive (+3.55%). Furthermore, both deviations of PIR w/permeable facers were also positive (see Fig. 10) suggesting the source of bias includes instrument error; i.e., lateral heat losses at the edges of the heat-flux-transducer.

### SUMMARY AND CONCLUSIONS

An intra-laboratory comparison was conducted using NIST's 1-metre Line-Heat-Source Guarded Hot Plate (GHP) and two commercial heat-flow-meter apparatus (HFM<sup>#</sup>1 and <sup>#</sup>2). The large HFM apparatus, HFM<sup>#</sup>1, had a plate size of 610 by 610 mm with a 254-mm-square heat-flux-transducer (HFT). HFM<sup>#</sup>2 was one-half the size, 305 by 305 mm, with a 100-mm-square HFT. The HFM apparatus were calibrated at  $T_M$  of 24°C and  $\Delta T$  of 22°C with individual and stacked specimens of fibrous-glass board from an internal lot of Standard Reference Material. For the comparison, all apparatus were operated at the same conditions of  $T_M$  of 24°C and  $\Delta T$  of 22°C. Heat flow was in the upward direction.

Individual specimens (nominally 25 mm) of elastomeric cellular foam, fibrous-glass board, fumedsilica board, gum rubber, MEPS, PIR w/foil facers, PIR w/permeable facers (40 mm), and XPS  $(0.02 < \lambda < 0.16 \text{ W/(m \cdot K)})$  were tested sequentially in the GHP and HFM<sup>#1</sup>. Additionally, specimens of loose-fill insulation, fibrous-glass batt, and stacked specimens of PIR w/permeable facers and XPS (0.9 < R < 7 m<sup>2</sup> · K/W) were also tested sequentially. The majority of deviations of  $\lambda_1$  and  $R_1$  were within ±3% of the GHP measurements. Bias was observed for the deviations of PIR w/permeable facers (-1.73%) and XPS (+2.00%). The opposite direction suggests that the bias was material dependent. Further work has been proposed to investigate the source of bias.

Similarly, individual specimens (nominally 25 mm) of fibrous-glass hoard, MEPS, and XPS (0.028 <  $\lambda$  < 0.033 W/(m • K)) were also tested sequentially in HFM<sup>#</sup>2 after the GHP and HFM<sup>#</sup>1. Deviations of  $\lambda_2$  were within  $\pm 3.5\%$  of the GHP measurements. Considerable errors occurred, however, when thick specimens ( $\ell$  > 26 mm) of stacked PIR w/permeable facers and XPS (0.9 < R < 3.5 m<sup>2</sup> • K/W) were measured with HFM<sup>#</sup>2. Deviations ranged from +11% to +63% for PIR w/permeable facers for 40  $\leq \ell \leq$  77 mm; and from +9.1 to +85% for XPS for 38  $\leq \ell \leq$  102 mm. Calibrating HFM<sup>#</sup>2 as a function of thickness reduced the deviations substantially, but did not eliminate the hias (see Figure 10).

Historically, thermal testing laboratories have conducted *inter*laboratory comparisons to determine precision and bias statements for ASTM Test Methods C 177 and C 518. These comparisons have been conducted under the auspices of ASTM, ISO, and other organizations using both the guarded hot plate and heat-flow-meter apparatus [17-21]. Generally, most of the interlaboratory comparisons have demonstrated a high precision with some notable exceptions [8,20]. In the latter case [20], 16 laboratories using a small HFM apparatus similar or the same as HFM<sup>#</sup>2 obtained a reproducibility of 8.4, 11.7, and 20.2% for specimens of PIR at thicknesses of 25.4, 50.8, and 76.4 mm, respectively. The trend is similar to results obtained in this intra-laboratory comparison for HFM<sup>#</sup>2. The results of these two studies suggest caution in using small HFM apparatus (plate size 305 mm) at specimen thicknesses greater than 26 mm.

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A ROUND-ROBIN COMPARISON OF AUSTRALASIAN THERMAL LABORATORIES

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ABSTRACT: This paper describes a comparison of seven laboratories in Australia and New Zealand, and one US laboratory, involved in testing thermal properties of materials. The primary objective of this Round Robin was to establish or confirm what degree of confidence can be placed in measurements by the various laboratories in this region. It was carried out using two industrial insulant products which were premeasured. Three samples were sent to each participant, who each conducted four thermal resistance measurements.

Results were typically within  $\pm 6\%$  of the mean value, with a worst-case difference of -15.5\% from the mean value.

KEYWORDS: Confidence, interlaboratory comparison, measurement, Round-Robin, R-Values, standardisation, survey, thermal insulants.

# INTRODUCTION

This paper describes a comparison of laboratories in Australia and New Zealand, and one US laboratory, involved in testing thermal properties of materials. At the time this work was planned there were about 10 Australasian laboratories involved in work of this type, about half as independent laboratories and half testing mainly their own products. Seven of these eventually participated, giving a total of eight laboratories.

The primary objective of this Round-Robin was to establish or confirm what degree of confidence can be placed in measurements by the various laboratories in this region. The willingness of a US

H. Trethowen is Senior Research Engineer at Building Research Association of New Zealand, Private Bag, Porirua, New Zealand; A. Desjarlais is Manager of Thermal Insulation Evaluation, Holometrix, 99 Erie Street, Cambridge, MA, 02139. laboratory to join in this Round-Robin also allows comparison with one other important region. No attempt is included to probe any deficiencies in standard or actual procedures - indeed this survey includes laboratories using four different procedures. Part of the desired confidence-establishing goal includes the question of whether the choice of method need concern laboratory clients.

#### DESCRIPTION OF SURVEY

The survey involved each participant measuring three sample pieces of two insulant board materials in four measurements. The thermal resistances ranged from about  $0.2 \ m^2 \, ^\circ C/W$  to  $3.0 \ m^2 \, ^\circ C/W$ . In a preliminary series, the coordinating laboratory (Building Research Association of New Zealand, Laboratory Z) measured the thermal resistance of every board. Individual samples were then cut, arbitrarily from various boards, to sizes to suit each participant. Participants received one low resistance sample of 14mm high density mineral fibre board, and two nominally identical samples of 50mm medium density glass fibre board. They were asked to measure the R-value of each of these separately, and then of the two glass fibre samples together with no septa. No results were revealed to any participant until all participant results were received for each board type.

The two test materials were chosen from regular industrial products. The mineral fibre board was 14-15mm thick, density 421 kg/m<sup>3</sup>. The fibre diameter was estimated as 4 - 6 micron. The surface was nominally flat one side, and dimpled 0.7mm on the other, and this surface condition was left in place. The glass fibre board was 50mm thick, density 50 kg/m<sup>3</sup>, with fibre diameter 8-9 micron. The surface was nominally flat both sides. These materials were chosen partly for availability, partly because they were expected to have relatively small variation in conductivity with density variation.

Figure 1 shows typical previously published values from ASHRAE [1] for the thermal conductivity of glass fibre. Values for mineral fibre board are less well reported, ASHRAE reports about 0.06 W/m°C in this density range.

The test conditions were left open to the participating laboratories. This was decided in order not to exclude any of the candidate participants, some of whom have (different) restrictions on the temperatures at which they can test. Instead, it was planned to normalise all results to a mean temperature of  $24^{\circ}$ C. This, after all, is the likely approach for typical routine commercial test work which these laboratories might undertake. Many of the laboratories did in fact test close to  $24^{\circ}$ C mean temperature, though at different surface temperatures, but one laboratory (Z) is restricted to around  $20^{\circ}$ C and  $6^{\circ}$ C surface temperatures, mean  $13^{\circ}$ C.

A survey like this expects to encounter variance in result due to a number of factors including:-

- inter-sample variance
- within-lab variance
- between-lab variance
- between-method variance
- temperature effects



(Reproduced with permission from ASHRAE from Fig.2, Ch 20, Handbook of Fundamentals 1989 This corrected version supplied by ASHRAE)

FIGURE 1, TYPICAL THERMAL CONDUCTIVITY, FIBRE GLASS

Any of these variations may be different at different insulation resistances. In this survey the first two items were expected to be indistinguishable. The initial pretesting in which the coordinating laboratory (Z) measured all boards, was expected to provide information on the combined inter-sample and its own in-house variability. For the survey to be useful, this combined variability has to be sufficiently small. The between-lab variance and between-method variances would also be likely not to be separable, unless one of the variances was dominant, as each laboratory used only one method. To determine this combined between-laboratory and between-method variance was in fact an object of this survey. Finally, the effects of test temperature were expected to be present but not intended to be part of the survey.

# PARTICIPATING LABORATORIES

Table 1 lists the eight participating laboratories. There were three New Zealand laboratories including the coordinating laboratory, four Australian, and one United States of America. One New Zealand and three of the Australian laboratories operate principally to support their own insulation products, the remainder are independent.

TABL	E 1 SUMMARY OF PARTICIPATING LABORATORIES AND EQUI	MENT
	(a) - Schedule of Participants, in alphabetic order	

LABORATORY	
Australian Consolidated Industries	Dandenong, Melbourne, Australia
Building Research Association of New Zealand	Judgeford, New Zealand
Bradford Insulation	Nunawading, Melbourne, Australia
Commonwealth Scientific & Industrial Research Organisation	Highett, Melbourne, Australia
Concord Research & Development Centre	Concord West, Sydney, Australia
Holometrix Inc	Cambridge, Mass, United States of America
New Zealand Fibre Glass	Penrose, Auckland, New Zealand
Wool Research Organisation of New Zealand	Lincoln, Christchurch, New Zealand

# (b) Schedule of Equipment and Laboratory Code Names

Laboratory Code Reference (Randomized)	Equipment Type	Metered Area (mm)	Max Sample Size (mm)	Equipment Origin	Equip Tempera	oment ture Limits
		_			Warm Side (max)	Cold Side (min)
Z	Guarded hot box ASTM C236 type (with mounting frame)	1000 x 1000	1600 x 1200 x 100	Custom made	+20°C	+6°C
A	Series Comparat <b>o</b> r, BS 5335	1000 x 600 or 300 x 300	2000 x 1500 x 170	Custom made	+36°C	+19°C
В	Heat flow meter	100 x 100	305 x 305 x 100	Proprietary "B"	+70°C	+10°C
с	Heat flow meter ASTM C518	250 x 250	610 x 610 x 200	Proprietary "A"	+37°C	+13°C
D	Heat flow meter ASTM C518	250 x 250	610 x 610 x 200	Proprietary "A"	+45°C	+5°C
E	Heat flow meter ASTM C518	460 x 460	1000 x 1000 x 200	Custom made	+45°C	+5°C
F	Heat flow meter ASTM C518	250 x 250	610 x 610 x 200	Proprietary "A"	+45°C	+0°C
G	Heat flow meter ASTM C518	250 x 250	610 x 610 x 150	Proprietary "A"	+220°C	-180°C

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Five laboratories used commercial guarded hot-plate insulation testing equipment. One used custom-made heat-flux meter equipment to ASTM C518-85 [2] with 460mm x 460mm metering area. Another laboratory used custom-made comparative equipment to BS 5335 [3] intended for use on quilting fabrics. The temperature drop is measured relative to that across an adjacent reference sample (of R ~ 1.4 m<sup>2</sup>°C/W). The equipment is periodically checked against a sample of reference material known as NBS 1450b, supplied by National Bureau of Standards. Finally, laboratory Z used a guarded hot box normally operated as in ASTM C236 [4], with a special holding frame to generate one-dimensional heat flow in a single-layer sample of insulant at the required thickness. This is prewired with a set of differential thermocouples which indicate the surface-to-surface temperature difference across the insulant sample.

#### RESULTS

#### Laboratory Z Pre-measurements

The properties of the five  $1600 \times 1200$  sheets of mineral fibre board, and the eight  $1200 \times 1200$  sheets of medium density fibre glass, as measured by Laboratory Z, are shown in Table 2 and Figures 2 and 3.



These results were corrected from the mean test temperatures listed in Table 2 to 24.0°C using linear temperature coefficients of 0.38%°C for the mineral fibre board, and 0.34%°C for the glass fibre. eg. (resistance at 24°C) = (resistance at 13°C)/(1 + 11 x 0.38/100). These coefficients were found as follows: Data in ASHRAE [1] Handbook of Fundamentals (Chapter 22) group mineral fibre and glass fibre together, and at a density of  $48 \text{kg/m}^3$  over the  $10-30^\circ$ C range, indicates a temperature coefficient of  $0.38\%/^\circ$ C. Measurements by laboratory E indicated a temperature coefficient of  $0.34\%/^\circ$ C for the glass fibre material used in this series, over the range  $20^\circ-30^\circ$ C. Lack of time prevented similar measurements for the mineral fibre board.

The mineral fibre board thickness varied from 14 to 15mm. The surface was dimpled on one side, and the thickness variations occurred on both small scale (a few mm) and large scale (over the whole board). The board density was determined as  $421 \pm 7 \text{ kg/m}^3$ .

The thermal resistance of the five boards averaged 0.247  $m^{2}$ °C/W when measured at a mean temperature of 13°C, with a standard deviation<sup>a</sup> of 0.0023  $m^{2}$ °C/W (0.9%). When corrected at the assumed temperature coefficient of 0.38% per °C, to a mean temperature of 24°C, this translates to a mean of 0.237  $m^{2}$ °C/W. See Figure 2.



The fibre glass boards were found to average 49.7mm thickness, with a standard deviation of 0.49mm ( $\pm$ 1% of mean) on the board averages. The extreme thickness ranged at any point from 47.5mm to 52.0mm. The density of these boards was determined as 50  $\pm$  3 kg/m<sup>3</sup> ( $\pm$ 6% of mean).

The thermal resistance of these eight boards averaged 1.532 m<sup>2</sup>°C/W, with standard deviation 0.026 ( $\pm$ 1.7% of mean) at a mean temperature of 13°C. When corrected to 24°C mean using the temperature coefficient of 0.34°C the average resistance becomes 1.475 m<sup>2</sup>°C/W. See Figure 3.

<sup>a</sup> throughout this paper standard deviations apply to the sample, not to an assumed parent population: they do not contain "Bessels correction".

#### Results from all Participants

The results of measurements from all laboratories are summarised in Tables 3 and 4, with the mean temperatures at which those results were obtained. These values are then converted to a mean temperature of 24°C, using the same temperature coefficients as above.



<u>14mm Samples:</u> Table 3 indicates the mean resistance reported for the 14mm mineral fibre boards, before temperature correction, as 0.229  $m^{2}$ °C/W, with standard deviation 0.007  $m^{2}$ °C/W (3.1%). After adjustment to 24°C, this became 0.228  $m^{2}$ °C/W, with standard deviation of 0.006 (2.6%). The thickness reported for these samples ranged from 13mm to 15mm (±7%), three laboratories of six reporting to the nearest whole mm. No individual result differed from the mean by more than 1.3 times the standard deviation, whilst the mean result from proprietary equipment "A" differed from those of custom-made by only 0.6 of the common standard deviation. Figure 4 illustrates these results.

<u>50mm Samples:</u> Table 4 shows the mean resistance reported for 50mm and 100mm glass fibre board. The mean as-measured value for 50mm boards was 1.488 m<sup>2°</sup>C/W, with standard deviation of 0.050 m<sup>2°</sup>C/W (3.4%). After adjustment to 24°C, the mean becomes 1.487 m<sup>2°</sup>C/W, with standard deviation 0.049 m<sup>2°</sup>C/W (3.3%). The reported thickness ranged from 47.4mm to 52.3mm. The highest and least individual results differ from the mean by +1.3 and -1.5 standard deviations respectively.

The mean of results from proprietary equipment "A" differed from that of custom-made by only 0.32 of the standard deviation. Figure 5 illustrates these results.



<u>100mm Samples:</u> The mean as-measured resistance for 100mm boards was 2.883  $m^2$ °C/W with standard deviation of 0.173  $m^2$ °C/W (6.0%). After adjustment to 24°C the mean becomes 2.882  $m^2$ °C/W with standard deviation of 0.161  $m^2$ °C/W (5.6%). The highest and lowest individual results differ from the mean by +0.6 and -2.5 standard deviations respectively. The individual 100mm results differ from the sum of the two contributing 50mm results by extremes of +0.8% and -28.4%, or -1.1% excluding laboratory A.

#### DISCUSSION

The results of this survey indicate inter-laboratory differences of 0% -6% with guarded hotbox (GHB) and heatflow meter (HFM) facilities. The largest individual difference was -15.5%. Previous surveys reported (e.g. Bales, [5]) have shown a similar variation. Table 5 lists some basic comparisons between the various results, for reader convenience.

The thermal resistance values have been taken as the primary parameter in this study. In practice all laboratories used procedures that determine resistance, from which the thermal conductivity is derived. This conductivity is in all cases an apparent conductivity. In the typical industrial samples used here, the thickness varies over the sample on both small and large distance scales. There is an uncertainty as to the "true" thickness, but the choice of plate spacing is an important one. Insufficient plate spacing will tend to increase the observed resistance pro-rata with thickness, ie, about 7% per mm for the 15mm sample, 2% per mm and 1% per mm for the 50mm and 100mm

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			-	7	ę	4	ŝ	MEAN	SD	
	thickness	E	14.5	14.5	14.5	14.5	14.5			
	R m²₀C/M k W/m⁰C	/}@ }13°C	0.248 0.0585	0.244 0.0594	0.249 0.0582	0.249 0.0582	0.244 0.0594	0.247 0.0587	0.0023	
	R m²°C/W k W/m°C	} @ } 24°C	0.238 0.0609	0.234 0.0620	0.239 0.0607	0.239 0.0607	0.234 0.0620	0.237 0.0613	0.0022	
		(a) Labo	oratory Z test:	s on all boar	ds, 14 mm I	nineral fibre	board, 420	kg/m³		
				8	OARD NO					
	-	2	e	4	2	9	2	8	MEAN	8
thickness mm	49.25	49.34	49.13	49.46	49.50	49.87	50.15	50.67	49.7	0.49
R m <sup>2</sup> C/W }	1.533	1.517	1.538	1.535	1.473	1.567	1.553	1.540	1.532	0.026
k W/m°C } 13°C	0.0321	0.0325	0.0319	0.0322	0.0336	0.0318	0.0323	0.0329	0.0324	0.0005
R m <sup>2</sup> C/W }	1.476	1.460	1.480	1.478	1.418	1.508	1.495	1.482	1.475	0.025
k W/m°C }24°C	0.0334	0.0338	0.0332	0.0335	0.0349	0.0331	0.0335	0.0342	0.0336	0.0006
			(h) I ahorato	nv 7 Tests c	on all board	s. 50 mm of	ass fibre			
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thickness	14.5	:	15	4	:	14.35	13	14.35	14.2	0.61
Temp °C R m²C/W k W/m °C	13 0.247 0.0587	: : :	23.85 0.223 0.0673	23.55 0.235 0.0596		26.95 0.218 0.0658	25.1 0.220 0.0591	24.35 0.231 0.0621	0.229 0.0621	0.007 0.003
R m²C/W	0.237 0.0610	::	0.223 0.0673	0.235 0.0596	: :	0.220 0.0651	0.221 0.0588	0.231 0.0621	0.228 0.0623	0.006 0.002

				LABORAI	rory					
	Z	۲	8	υ	۵	ш	Ľ.	IJ	Mean	SD
Sample ID	-	ى ا	7.3	7.1	8.1	4	1.3	Ξ		
Thickness mm	49.3	50.0	48.0	50.0	51.7	47.4	49.9	50.3		
Temp °C R m²C/W	13.0	27.13 1.57	25.2 1.377	25.73 1.471	26.35 1.474	25.1 1.481	24.84 1.481	24.05 1.481		
k W/m°C	0.0321	0.0318	0.0349	0.0340	0.0351	0.0320	0.0337	0.0339		
R m²C/W } at 24°C k W/m ℃ }	1.476 0.0334	1.587 0.0315	1.383 0.0347	1.480 0.0338	1.486 0.0348	1.487 0.0319	1.485 0.0336	1.481 0.0340		
Sample ID	æ	9	7.4	7.2	8.2	3	1.4	1.2		
Thickness mm	50.7	49.5	49.0	50.0	52.3	48.9	50.0	50.0	49.8	1.16
Temp B m2C/W	13.0	26.08 1.59	25.2 1 429	26.09 1.488	25.55 1.485	25.1 1.476	24.88 1.466	24.05 1.459	1.488	0.050
k W/m°C	0.0329	0.0311	0.0341	0.0336	0.0352	0.0332	0.0341	0.0343	0.0335	0.0012
R m²C/W } at 24⁰C k W/m ⁰C }	1.482 0.0342	1.601 0.0309	1.434 0.0340	1.499 0.0334	1.493 0.0350	1.482 0.0330	1.470 0.0340	1.459 0.0343	1.487 0.0335	0.048 0.0012
Sample ID	1+8	5+6	73+74	7.1 + 7.2	8.1 + 8.2	3+4	1.3 + 1.4	1.1 + 1.2		
Thickness mm	9.99 1.05	99.5 26.70	97.0 25.0	100.0	104.0	97.8 25.4	99.2 24.06	100.3 24.05	69.7	1.95
R m²C/W k W/m°C	3.07 0.0325	2.47 2.47 0.0403	2.775 0.0349	2.959 0.0338	2.946 0.0353	2.950 0.0331	2.932 0.0338	2.963 0.0339	2.883 0.0347	0.173 0.0023
R m²C/W } at 24°C k W/m°C }	2.968 0.0336	2.484 0.0400	2.786 0.0348	2.981 0.0336	2.965 0.0351	2.961 0.0330	2.942 0.0337	2.964 0.0338	2.882 0.0347	0.161 0.0021

TABLE 4 -- Results from all laboratories, 50 mm and 100 mm glass fibre, 50 kg/m $^3$ 

samples. Excess plate spacing of up to 2-3mm will tend to add linearly to the observed resistance, about 18% per mm for the 15mm sample, 2.7% per mm and 1.3% per mm for the 50mm and 100mm samples. The decision on plate spacing is therefore crucial especially in thin samples. Only one laboratory (E) made any distinction between sample thickness and plate spacing.

The resistance for the 14mm boards was consistently higher in the Laboratory Z results than in others. The mean difference was 0.012  $m^2$ °C/W, about 5.1%. Applying Students 't' test to this result showed the difference to be "highly significant". It is attributed to an imperfection in the sample holding frame, the two faces of which can slightly warp differentially. The effect in these tests was to add a small cavity averaging 0.46mm wide. For cavities less than ~10mm convection is suppressed and at less than 3-4mm even radiation is swamped by gas conduction. Thus provided that the cavity is also large compared with mean free path, the heat transfer is dominated by gas conductivity of some 0.025 W/m.°C. Thus a 0.46mm cavity is expected to add a resistance of about 0.018  $m^2$ °C/W to the observed result, see Figure 4.

The resistance for these l4mm boards is estimated as  $0.255 \pm 0.003$  m<sup>2°</sup>C/W. This estimate is the mean of the 5 participant results plus the average Laboratory Z result corrected as above.

Results for 50mm and 100mm boards can be investigated together. The 100mm board resistances should be almost equal to the sum of the two 50mm boards included, different only by the "thickness effect". From Pelanne [6] and Hollingsworth [7] we should expect the combined 100mm samples to have some 0.01 m<sup>2</sup>°C/W (0.3%) less resistance than the two 50mm pieces. In this survey we found the deficit to average 0.00236 m<sup>2</sup>°C/W but the standard deviation on this was 0.013 m<sup>2</sup>°C/W. Thus this survey is not able to contribute usefully to "thickness effect" data, which at 0.3% has been ignored.

A comparison of the glass fibre board results in Figure 5, shows overall mean and 90% confidence limits for the set, and the individual results from each laboratory. Laboratory A reported a systematic problem, obtaining high results for low-R samples and vice-versa.

The inter-board variance is quite small. The Laboratory Z -premeasurements - see Figure 3 - show the maximum deviation from the mean to be little over 2%, includi-ng the laboratory variability. Inspection of Table 4 and Figure 4 does not suggest much variation with board, most of the variation being associated with the laboratory.

The mean thermal resistance for the 50mm glass fibre boards is estimated as  $1.470 \pm 0.006 \text{ m}^2 \,^\circ \text{C/W}$ . This estimate is the average of the 21 results (50mm and 100mm) reported by the seven laboratories with smallest variability. The effect of laboratory exclusion is shown below:-

	Mean	Standard Error
All laboratories included	1 479	
Most variable laboratory analysis	1,472	0.014
The meet model is a line of the second of th	1.4/0	0.006
iwo most variable laboratories excluded	1.481	0.002

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			BOAR	)						
Source	1	2	3	4	5	6	best estimate <sup>b</sup> of mean value			
	Measured R Value, m²ºC/W									
Laboratory Z - as reported	0.238	0.234	0.239	0.239	0.234					
- corrected for mounting gap	0.220	0.216	0.221	0.221	0.216					
Other Participants			0.223	0.220	0.233	0.231				
			•••••		_		0.225			
		%	difference fr	om mean						
Laboratory Z	-2.2	-4.0%	-1.8%	-1.8%	-4.0%					
Others	 -1.8%		-0.9%	-2.2%	+3.6%	+2.7%				

### Table 5 -- Comparison from individual boards

				BO	ARD				
Source	1	2	3	4	5	6	7	8	best estimate <sup>b</sup> of mean value
			Me	asured R-v	alue m²°C	w			
Laboratory Z	1.476	1.460	1.480	1.478	1.418	1.508	1.495	1.482	
Other Participants	1.485 1.470 1.481 1.459		1.482	1.487	1.587	1.601	1.480 1.499 1.383 1.433	1.481 1.459	1.470
			%	difference	from mea	n			
Laboratory Z	+0.4%	-0.7%	+0.7%	+0.5%	-3.5%	+2.6%	+1.7%	+0.8	
Others	+1.0% 0.0% +0.7% -0.7%		+0.8%	+1.2%	+8.0%	+8.9%	+0.7% +2.0% -5.9% -2.5%	0.7% -0.7%	

#### (a) Mineral fibre board

#### (b) Glass Fibre Board

• see "Discussion"

The mean thermal conductivity for this material is therefore 0.0340  $\pm$  0.0002 W/m°C. The "typical" data of ASHRAE [1] Fundamentals Chapter 20, shows the conductivity of fibre glass of 8-9 micron diameter, at 24°C and 50 kg/m<sup>3</sup> density, as 0.0345 W/m°C - only 1.45% different. (The ASHRAE data was based on 25mm samples, and a small further "thickness effect" correction applies). See Figure 1.

This survey can be subdivided into laboratories using proprietary equipment "A" [8], and custom-made equipment. Comparing these two sets does not show either statistically significant or apparent differences.

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If the results from each custom-made equipment is individually compared with the average for proprietary equipment results, it differs in the worst case by less than 1.4 standard deviations and in all other cases by less than 0.75 standard deviations. The standard deviations amongst the results from custom-made equipment were slightly smaller than the proprietary results.

#### CONCLUSIONS

A Round-Robin comparison of thermal laboratories has been conducted using typical industrial board insulants 14-100mm thick, thermal resistance 0.22 -  $3.0 \text{ m}^2 \text{°C/W}$ . Eight laboratories using four measurement methods were included; three from New Zealand, four from Australia, one from United States of America.

Each laboratory measured different samples, but the intersample standard deviation was small - about 2% for the combined sample and laboratory deviation in the coordinating laboratory measurements.

Results from one laboratory using a comparative procedure to BS 5335 had more variation than other laboratories, with individual results reaching a maximum 15% from the mean.

Results from the seven laboratories using GHB and HFM methods were all individually within 5.9% of the best estimate of mean value for each material. Excluding the seventh ranked laboratory would leave all results within 4.0% for the mineral board samples, and 2.0% of the mean for fibre glass samples.

The mean thermal resistance for fibre glass found in this survey was within 1.5% of the "typical" data in the ASHRAE Handbook of Fundamentals for the same conditions.

Results for 100mm samples from the seven laboratories using GHB and HFM methods were individually within 1.5% of the sum of their respective 50mm measurements. The mean difference was consistent with previous data on "thickness effect" difference, but not of sufficient certainty to confirm or modify it.

No significant differences could be established between different methods (except BS 5335), between Australasian and United States laboratories, or between users of proprietary equipment and custom-made equipment types.

## ACKNOWLEDGEMENTS

The authors acknowledge with appreciation the support of all participating laboratories.

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David J. McCaa<sup>1</sup> and David R. Smith<sup>2</sup>, et al.<sup>3</sup>

INTERLABORATORY COMPARISON OF THE APPARENT THERMAL CONDUC-TIVITY OF A FIBROUS BATT AND FOUR LOOSE-FILL INSULATIONS<sup>4</sup>

REFERENCE: McCaa, D. J., Smith, D. R., et al., "Interlaboratory Comparison of the Apparent Thermal Conductivity of a Fibrous Batt and Four Loose-Fill Insulations," <u>Insulation Materials: Testing and</u> <u>Applications, 2nd Volume, ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

**ABSTRACT:** An interlaboratory comparison of measurements of the apparent thermal conductivities of four loose-fill insulations was conducted according to ASTM Standard Practice C 687 in order to prepare a revised statement of precision and bias. The apparatus used included one guarded hot plate, one thin-heater apparatus, and eight heat-flow meters. All specimens were tested at a mean temperature of 24 °C (75 °F). Test results for fibrous glass blanket were used to characterize the basic imprecision (numerical measure of precision) of the test instruments. The 20 imprecisions of measurement of apparent thermal conductivity were 2.8% for fibrous glass blanket, 5.8% for cellulose, 9.4% for unbonded glass fiber, 10.5% for mineral wool, and 5.0% for perlite.

**KEY WORDS:** apparent thermal conductivity, bias, blanket, cellulose, fibrous glass, insulation, loose fill, mineral wool, perlite, precision, round robin, thermal resistance.

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<sup>4</sup> Official contribution of the National Institute of Standards and Technology; not subject to copyright in the United States.

### INTRODUCTION

Loose-fill building insulation materials are important thermal insulations for reducing the cost of thermal energy in residential, commercial, and industrial buildings. Loosefill insulation can be easily installed in new construction, and can be retrofitted into existing structures. Performance standards play an important role in the effective use of loose-fill insulation.

Within North America the most popular means of determining the thermal performance of loose-fill insulation materials is that specified by ASTM C 687, Standard Practice for Determination of the Thermal Resistance of Loose-Fill Building Insulation [1]. This practice permits the thermal resistance of loose-fill insulation to be measured by any of five different apparatus: the guarded hot plate (C 177) [2], the heat-flow meter (C 518) [3], either of the two hot-box test methods (C 236 or C 976) [4,5], or the thin-heater apparatus (C 1114) [6]. The heat-flow meter is the recommended apparatus for saving time and operating costs in measuring thermal resistance.

In 1986 a round-robin testing program was organized and conducted under the auspices of ASTM Subcommittee C 16.30 (Thermal Measurement) of Committee C 16 (Thermal Insulation), in order to obtain test data needed to prepare a statement of the precision and bias of Standard C 687. There were also two other secondary purposes: (1) to investigate the effect of using only the mass of the test specimen within the metered area to determine the test density, rather than the mass of the entire specimen, which gives the box density; and (2) to see whether thermal test results for loose-fill specimens blown at one density and then compressed to achieve the (higher) design density are significantly different from test results for specimens blown directly at the design density.

In that round robin, reported by Adams and Hust [7], eleven different participants within the USA and Canada reported the apparent thermal conductivity,  $\lambda$  (which is directly related to thermal resistance, R), of five materials. One laboratory used a guarded hot plate, another used a thin-heater (thin screen) apparatus, and the remaining nine used ten different heat-flow meters. They measured  $\lambda$  as a function of density for five different materials: specimens of glass fiber blanket ranging in density from 11.5 to 16.2 kg/m<sup>3</sup>; bonded glass fiber, from 5.7 to 12.3 kg/m<sup>3</sup>; rock/slag mineral wool, from 26 to 39 kg/m<sup>3</sup>; and unbonded glass fiber, from 5.8 to 13 kg/m<sup>3</sup>. All measurements used a mean specimen temperature of 24°C (75°F).

The 1986 round-robin test results for the glass fiber blanket, basically the same material as NIST SRM 1451 [8], exhibited a mean deviation from the certification values of only 0.25%. For this reason Adams and Hust concluded that the bias of the mean was negligible. They arrived at an overall imprecision of 3.0% ( $2\sigma$ ;  $\sigma$ =1.5%) for the measurement capability of the participants; the imprecision (the <u>numerical measure</u> of precision [9]) was defined as two standard deviations of the measured values from the reference values.

However, the reported values of  $\lambda$  for the loose-fill materials exhibited much larger  $2\sigma$  imprecisions: 10% for the rock/slag wool, 21% for the cellulose, 14% for the bonded glass fiber, and 16% for the unbonded glass fiber. To summarize: for  $\lambda$  of loose-fill materials, for which Standard Practice C 687 is specifically designed, the imprecisions of the test results of the eleven participants ranged from 10 to 21%.

Additionally, Adams and Hust found that a "compression effect" existed; that is,  $\lambda$  decreased more rapidly with increasing density, when given specimens were compressed, than it did when different specimens were blown over the same range of density. This effect was more pronounced at the lower densities.

They also reported that the box densities for the four loose-fill materials were less than the test densities by about 4 to 6%, or 5% on the average. The imprecision  $(2\sigma)$  of the ratio of box density to test density was greater than 5%, however, indicating that any single box density could be greater than the test density. They concluded that the 5% difference between average box densities and average test densities was statistically significant.

In light of the high precision of the results obtained for the fibrous glass blanket, which involves minimal preparation before measurement of thermal resistance, they suggested that the reason for the disagreeably large imprecisions of the loose-fill results may have been due to inadequate standardization of the technique for preparing specimens.

The results from the 1986 loose-fill round robin led the task group responsible for C 687 to rewrite that standard. In particular, the task group described more carefully the methods to be followed in preparing specimens for measurement of thermal resistance. It was generally agreed that differences in specimen preparation among the eleven participants of the 1986 round robin had led to the large spread in the reported values of thermal resistance. As the revised version of C 687 was being ballotted, this (1990) interlaboratory comparison was initiated late in 1989, in order to provide precision and bias data for the revised C 687 document.

#### SCOPE

The present interlaboratory comparison was designed to study four loose-fill insulation materials that are readily available, and one blanket insulation, which was to be used to compare testing apparatus. Letters of invitation were sent to all participants in the 1986 round robin and to all members of ASTM Subcommittee C16.30 on Thermal Measurements; an announcement of the round robin was also placed in ASTM Standardization News. The nine laboratories that participated in this study are listed below. Test materials were shipped to participants in the summer of 1990, and all data were received by November of 1990.

### Objectives

The primary objective of this study was to provide the data needed for developing a statement of precision and bias for ASTM Standard Practice C 687. Two secondary objectives were (a) to provide a further interlaboratory comparison of measurements on a blanket specimen by use of heat-flow meter apparatus; and (b) to include specimens of poured insulation, because C 687 does describe the testing of this type of insulation, although the 1986 round robin had not included it.

### Participants

The nine laboratories that took part in the round robin were, in alphabetic order,

CertainTeed Corporation, Blue Bell, Pennsylvania, 19422 Fiberglas Canada, Inc., Sarnia, Ontario, Canada N7V1X4 Holometrix, Inc., Cambridge, Massachusetts, 02139

- \*Knauf Fiberglass, Inc., Shelbyville, Indiana, 46176 Manville Sales Corp., Denver, Colorado, 80127
- NAHB Research Foundation, Upper Marlboro, Maryland, 20772
- National Institute of Standards and Technology (NIST), Gaithersburg, Maryland, 20899
- Oak Ridge National Laboratory, Oak Ridge, Tennessee, 37831
- Owens-Corning Fiberglas, Granville, Ohio, 43023.

With the exception of the laboratory denoted by an asterisk (\*), eight of the participants took part in the previous (1986) round robin. Three of the eleven participants in the previous effort did not take part in this one. Thus eight laboratories took part in both round robins.

# INSTRUCTIONS TO PARTICIPANTS

In response to the results of the 1986 round robin, the revised (1990) round robin required the participants to determine the test density in the metering area of their thermal test apparatus.

Participants were informed that three specimens of each material were to be tested by each laboratory, except for the fibrous glass blanket, of which there would be only one spec-Thermal tests were to be conducted at mean imen tested. specimen temperatures of 75°F (24°C).

A further question which arose from the results of the 1986 round robin concerned the effect of the time needed to achieve steady-state on the testing of cellulose loose fill. Participants in this (1990) interlaboratory comparison were instructed not to measure  $\lambda$  of the cellulose loose fill until at least 24 hours had passed from the time they inserted specimens into their test apparatus.

A complete list of instructions given to the participants is found in the Appendix.

#### APPARATUS

The sizes and configurations of the ten different apparatus reported by the nine participants are given in Table 1. The apparatus used included one guarded hot plate (GHP; C 177), one thin-heater apparatus (THA; C 1114) and eight heatflow-meter apparatus (HFM; C 518). The thin-heater apparatus (THA) was used by a laboratory represented elsewhere in the list by another entry for a heat-flow meter.

The blowing machines used by each laboratory to blow the fibrous loose-fill (cellulose, mineral wool, and unbonded glass) material into forms for testing are listed in Table 2. The perlite was simply poured.

Apparatus	Apparatus	Plate	Metered
Code	Type	Area, mm	Area, mm
1114 appli	es to the this	n-heater appar	atus (THA).

TABLE 1.	Test A	pparatus	Used by	Parti	cipant	.s.	
ASTM C 177	appli	es to the	guarded	hot p	late	(GHP)	;
C 518 app]	lies to	the heat	z-flow me	eter (	HFM);	and	С
111 <b>4</b> appli	les to	the thin-	heater aj	pparat	us (Tl	HA).	

			-		
1	HFM	610x610	254x254		
2	HFM	610 <b>x</b> 610	254x254		
3	HFM	610x610	254x254		
4	HFM	610x610	254x254		
5	HFM	610 <b>x</b> 610	254x254		
6	GHP	1016 (diam.)	406 (diam.)		
7	HFM	914x914	457x457		
8	HFM	610x610	254x254		
9	HFM	610x610	254x254		
10	THA	<b>914x15</b> 00	305x305		

## TABLE 2. DETAILS OF BLOWING MACHINES AND SETTINGS USED BY PARTICIPANTS TO PREPARE LOOSE-FILL SPECIMENS. Laboratory 7 reported use of a "smooth-bore" hose. Laboratory 8 did not describe hose. All others reported use of corrugated hoses.

	Concis Icpereed	<u></u>	agaeea ne				
Lab. Code	Blowing Machine	Material	Nominal Hose diam. (in)	Nomin. Hose length (ft)	Gear	Nomin Gate Setg. (in)	Feeder (rpm)
	Unisul [10]	CELL	2	100	3	6	76.7
1	(Volumatic II)	GLASS	3	150	3	12	76.7
		MWOOL	3	150	3	6	76.7
	Krendel Hopper/ Tornado Blower	CELL	2	100			
2	Unisul	GLASS	3	150	4	10	
	(Volumatic III)	MWOOL	3	150	1	8	
	Shelter Shield	CELL	2	100			
3	Unisul	GLASS	3	150	3	12	58
	(Volumatic II)	MWOOL	3	150	3	6	58
4	Unisul	GLASS	3	150	4	10	
5	Star-Roots (Dresser)	GLASS		150	FIXED	60%	95
	Unisul	CELL	2	100	3	6	76.7
6	(Volumatic II)	GLASS	3	150	3	12	76.7
		MWOOL	3	150	3	6	76.7
		CELL		150	3	12	49
7	Universal 5E	GLASS		150	3	12	49
		MWOOL		150	3	6	49
		CELL		100	2	4	
8	Unisul Inc.	GLASS_		100	3	10	
		MWOOL		100	3	10	
9	Unisul	CELL		100	3	12	1000
10	Krendel Hopper/ Tornado Blower	CELL	2	100			
	Unisul, (Volumatic III)	GLASS MWOOL	3 3	150 150	4 1	10 7,8	

		-			
APPARATUS	BATT	CELL	GLASS	MWOOL	PERL
1	x	x	x	x	·
2	х	х	х	x	х
3	х	х	х	х	х
4	х		х		
5	х		х		
6	х	х	х	х	
7	х	x	х	х	х
8	х	х	х	х	х
9	х	x			х
10	X	х	x	x	
Totals:	10	8	9	7	5

TABLE 3. Materials Tested by Each Participating Laboratory

### SPECIMENS (MATERIALS AND LABELS)

### Test Materials

The five test materials used in this study were selected as representative of commonly used and widely available loose fill insulations. These materials were cellulose (CELL), unbonded fibrous glass wool (GLASS), mineral (rock or slag) wool (MWOOL), and pourable perlite (PERL); a six-inch thick fibrous glass blanket (BATT) was also included. One material tested in the 1986 round robin, bonded fibrous glass wool, was not tested in this test program; perlite, tested here, was not tested in the previous program. Thus in this round robin there were four materials tested which were similar to those tested in the previous round robin, and which can be used to compare the bias and precision.

Each loose-fill material was selected from a single lot by the respective manufacturer, and then a quantity sufficient for testing was shipped to each participant. The fibrous glass blanket was prepared by one of the participants and then nine separate specimens were shipped, one to each laboratory. One laboratory measured the same specimen on two different apparatus.

The materials tested by each participant are listed in Table 3. The identities of the participants are represented by randomly chosen integers, one for each apparatus.

Only four laboratories (Nos. 2, 3, 7 and 8) measured all five materials. The fibrous glass blanket was measured on all apparatus, but perlite was measured on only five apparatus; the remaining materials were measured on at least seven different apparatus.

#### TEST RESULTS

The participants submitted conductivity data both in traditional units (conductivity in Btu·in/[(hr·ft<sup>2.</sup>°F]; density in lb<sub>m</sub>/ft<sub>3</sub>), and in SI units. All data are exhibited here in SI units (density in kg/m<sup>3</sup>,  $\lambda$  in mW/(m·K), and temperature in °C). The conversion factors used were: (a) 1.00 Btu·in/(hr·ft<sup>2.</sup>°F) = 144.13 mW/(m·K), and (b) 1.00 pcf = 16.02 kg/m<sup>3</sup>. The converted data for  $\lambda$  are listed in Tables 4 through 8 and plotted in Figures 1 through 5.

The mathematical correlation we chose to represent  $\lambda$  as a (nonlinear) function of density was of the form

$$\lambda = \mathbf{A}_0 + \mathbf{A}_1 \cdot \boldsymbol{\rho} + \mathbf{A}_2 / \boldsymbol{\rho}, \qquad (1)$$

where  $\rho$  is density in kg/m<sup>3</sup>. This correlation is based on a radiative-flux model and can be supported for use with lowdensity, fibrous insulations [11, 12]. The first, constant, term on the right side models the conductive transport through the fill-gas (air), which is kept at constant temp-erature and nearly constant pressure throughout these measurements. Although some participating laboratories are situated near sea level and others are located significantly above sea level, the conductivity of air is insensitive to the associated differences in barometric pressure among the various laboratories [13]. For these reasons, and for ease of fitting the data to this nonlinear relation, the value of this constant term was fixed at 0.180 Btu in/(hr ft<sup>2.o</sup>F)  $(25.94 \text{ mW}/(\text{m}\cdot\text{K}))$ . The second term represents conduction through the solid fibers, with the conductivity increasing directly proportionally to the density of the loose-fill insulation. The last term models the tendency for radiative transport of thermal flux to decrease as the density increases, due to scattering and absorption of the thermal radiation by the particles or fibers. There are more exact correlations [14, 15, 16, 17, 18, 19], to model these transport processes, but this simple model is adequate for the task at hand, as will be seen from the standard deviations of the data from the fitted curves.

In each of Figs. 1 through 5, the curve represents the correlation of the form of Eq. (1) found for the specific material listed in the caption. The coefficients of the correlations are listed in Table 9. Also shown for comparison in each of these five figures is a simpler, linear fit, found by the method of linear least squares, of the form

$$\lambda = B_0 + B_1 \cdot \rho.$$
 (2)

The coefficients of this linear fit for each material are also listed in Table 9.

TABLE 4. Thickness, Mean Temperature, Density ( $\rho$ ) and Apparent Thermal Conductivity ( $\lambda$ ) of Fibrous Glass Blanket as Measured on Eleven Different Apparatus. Column headed "COND. (Exp.)" denotes experimental values of conductivity; values of "COND. (Calc.)" are calculated from Eq. (1) with coefficients listed at bottom of table (see first row of Table 9). "DEV." indicates deviation of experimental values from calculated fit: DEV. = COND<sub>exp</sub>-COMD<sub>calc</sub>. "DEV. (Pct.)" (or %dev), is 100\*DEV./COND<sub>calc</sub>. Column headed "DEV. SQRD." lists squares of values of %dev. Standard deviation ( $\sigma$ , STD. DEV.) is computed from  $\sigma^2 = \Sigma$  (%dev)<sup>2</sup>/(N-2).

Pt.	APPA-	THICK-	MEAN	DENSITY	COND.	COND.	DEV.	DEV.	DEV.
Nr.	RATUS	NESS	TEMP.		(mW/	(m.K) )			SQRD.
		(Cm)	(°C)	(kg/m*)	(Exp.)	(Calc.)		(Pct.)	(Pct.)
1	1	15.24	23.86	13.18	36.75	38.05	-1.30	-3.41	11.597
2		15.24	23.85	13.18	36.90	38.05	-1.15	-3.01	9.067
3		15.24	23,83	13.18	36.90	38.05	-1.15	-3.01	9.067
4	2	15.24	23.96	11.48	38.21	38.57	-0.36	-0.93	0.859
5		15.24	23.96	11.48	38.23	38.57	-0.34	-0.87	0.766
6		15.24	23.96	11.48	38.19	38.57	~0.3B	-0.98	0.958
7	3	15.24	23.97	11.57	38.34	38.53	-0.19	-0.50	0.247
8		15.24	24.00	11.57	38.48	38.53	~0.05	-0.13	0.018
9		15.24	23.92	11.57	38.34	38.53	-0.19	-0.50	0.247
10	4	15.24	23.96	14.69	38.45	37.80	0.65	1.72	2.974
11		15.24	23.88	14.69	38,63	37.80	0.83	2.20	4.843
12		15.24	23.86	14.69	38.44	37.80	0.64	1.70	2.883
13	5	15.24	24.11	14.25	37.62	37.85	-0.23	-0.62	0.380
14		15.24	24.11	14.25	37.57	37.85	-0.28	-0.75	0.560
15		15.24	24.11	14.25	37.57	37.85	-0.28	-0.75	0.560
16	6	15.24	23.90	11.20	38.79	38.69	0.10	0.27	0.073
17		15.24	23.89	11.20	38.84	38.69	0.15	0.40	0,159
18		15.24	23.88	11.20	38.77	38.69	0.08	0.22	0.047
19	7	15.24	24.17	12.50	38.48	38.22	0.26	0.69	0.470
20		15.24	24.17	12.50	38.63	38.22	0.41	1.08	1.162
21		15.24	24.17	12.50	38.63	38.22	0.41	1.08	1.162
22	8	15.24	23.89	10.89	38,68	38.83	-0.15	-0.39	0.149
23		15.24	23.89	10.89	38.93	38,83	0.10	0.26	0.067
24		15.24	23.94	10.89	38,93	38,83	0.10	0.26	0.067
25	9	15.24	23.94	11.66	38,63	38,50	0.13	0.35	0.120
26		15.24	23.94	11.66	39.06	38.50	0.56	1.46	2.140
27		15.24	23.89	11,66	38.92	38.50	0.42	1.10	1.208
28	10	15.18	23.89	11.09	39,44	38,74	0.70	1.82	3.310
29		15.17	23.89	11.10	38,77	38.73	0.04	0.10	0.010
30		15.17	23.89	11.10	39.21	38.73	0.48	1.24	1.531
			(Fixe	ed) A0	= 25,94		AVG.	AVG.	STD.
				<b>A</b> 1	= 0.348	3	DEV.	%DEV.	DEV.
				<b>A</b> 2	= 99.1		0.00	0.00	1.42
									=== <b></b>

COND. (Calc.) = A0 + A1\*DENS + A2/DENS

TABLE 5. Thickness, Mean Temperature, Density and Apparent Thermal Conductivity of Cellulose Loose Fill as Measured on Eight Different Apparatus. Column headings and calculations are as described in caption of Table 1.

Pt.	APPA-	THICK-	MEAN	DENSITY	COND.	COND.	DEV.	DEV.	DEV.
Nr.	RATUS	NESS	TEMP.		( mW/	(m.K) )			SQRD.
		(cm)	(°C)	(kg/m³)	(Exp.)	(Calc.)		(Pct.)	(Pct.)
1	1	15.23	23.94	28.66	40.07	39.59	0.48	1.21	1.475
2		15.23	23.86	26.99	40.50	39.63	0.87	2.21	4.874
3		15.25	23.87	27.49	40.07	39.61	0.46	1.16	1.355
4	2	13.60	23.99	29.90	39.30	39.59	-0.29	-0.74	0.549
5		14.39	24.03	29.79	39.38	39.59	-0.21	-0.54	0.287
6		14.68	24.02	28.04	39.39	39.60	-0.21	-0.52	0.272
7	3	15.24	23.94	31.40	37.76	39.63	-1.87	-4.72	22.235
8		15.24	23.89	33.64	38.34	39.73	-1.39	-3.51	12.309
9		15.24	23.90	33.80	37.04	39.74	-2.70	-6.80	46.278
10	6	18.01	23.89	28.70	39.84	39.59	0.25	0.63	0.402
11		17.65	23.89	28.60	39.02	39.59	-0.57	-1.44	2.070
12		17.83	23.88	28.80	39.64	39.59	0.05	0.13	0.017
13	7	15.48	24.17	31.86	40.79	39.65	1.14	2.89	8.334
14		15.14	24.17	31.86	40.64	39.65	0.99	2.51	6.293
15		15.48	24.17	31.86	40.64	39.65	0.99	2.51	6.293
16	8	16.89	23.94	32.84	41.05	39.69	1.36	3.43	11.745
17		16.89	23.89	29.16	38.86	39.59	-0.73	-1.84	3.380
18		16.89	23.94	34.60	40.51	39.80	0.71	1.79	3.220
19	9	14.48	24.00	35.08	39.06	39.83	-0.77	-1.93	3.739
20		14.48	24.00	33,96	40.93	39.75	1.18	2.96	8.757
21		14.48	24.06	34,76	39,92	39.81	0.11	0.28	0.081
22	10	16.48	23.89	28.70	37.52	39.59	-2.07	-5.23	27.310
23		15.00	23.89	29.14	40.18	39.59	0.59	1.50	2.237
24		16.75	23.89	29.80	41.20	39.59	1.61	4.06	16.494
			(Fix	ed) A0 =	25,94		AVG.	AVG.	STD.
				A1 =	0.235		DEV.	%DEV.	DEV.
				A2 =	198.32		0.00	0.00	2.94
				••••					=====
		COND.	(Calc	.) = A0 +	A1*DENS	+ A2/DE	NS		

TAE	<b>LE</b>	6.	Thic	ckness	, Mean	Temper	rature,	, Dens	lity a	and	Appa	rent	The	rmal
Cor	nduc	tivi	ty o	f Unbo	nded F	ibrous	Glass	Loose	Pill	as	Meas	ured	on	Nine
Dif	fer	ent 1	Appa:	ratus.	Colum	n headi	ings an	nd calo	culat	ions	are	as d	esci	ibed
in	cap	tion	of	Table	1.		•							

Pt.	APPA-	THICK-	MEAN I	DENSITY	COND.	COND.	DEV.	DEV.	DEV.
Nr.	RATUS	NESS	TEMP.		( mW/	(m.K))			SQRD.
		(cm)	(°C)	(kg/m <sup>3</sup> )	(Exp.)	(Calc.)		(Pct.)	(Pct.)
1	1	17.42	23.85	7.10	66.16	67.46	-1.30	-1.93	3.727
2		17.06	23.86	6.78	70.19	69.63	0.56	0,80	0.644
3		17.33	23.92	6.79	69.76	69.56	0.20	0.29	0.082
4	2	16.99	24.34	8.72	57.08	58.81	-1.73	-2.94	8.662
5		16.33	24.36	8.05	63.31	61.99	1.32	2.13	4.521
6		15.62	24.30	7.71	62.72	63.80	-1.08	-1.70	2.884
7	3	15.24	23.84	7.87	58.66	62.93	-4.27	-6.79	46.094
8		15.24	23.88	7.90	60.97	62.77	-1.80	-2.87	8.250
9		15.24	23.88	7.83	60.39	63.15	-2.76	-4.37	19.065
10	4	17.32	23.87	9.22	58.56	56.72	1.84	3.25	10.548
11		17.65	23.73	9.21	58.83	56.76	2.07	3.65	13.331
12		17.98	23.93	9.30	57.41	56.40	1.01	1.79	3.192
13	5	16.26	24.00	7.54	71.50	64.77	6.73	10.40	108.077
14		15.75	24.00	7.71	70.00	63.80	6.20	9.71	94.321
15		16.05	24.00	7.32	70.60	66.08	4.52	6.85	46.872
16	6	16.38	23.90	7.17	66.49	67.01	-0.52	-0.78	0.608
17		16.74	23.90	7.13	68.17	67.27	0.90	1.34	1.796
18		16.69	23.90	7.12	67.65	67.33	0.32	0.47	0.222
19	7	19.30	24.17	7.75	63.56	63.58	-0.02	-0.04	0.001
20		17.18	24.11	7.88	62.55	62.88	-0.33	-0.52	0.274
21		17.73	24,11	8.09	61,26	61.79	-0.53	-0.85	0.731
22	8	16.89	23.89	6.57	67.32	71.16	-3,84	-5.40	29.175
23		16.89	23.89	8.65	53,60	59.12	-5.52	-9.34	87.227
24		16.89	23.89	8.49	55,63	59.85	-4.22	-7.05	49.717
25	10	18.40	23.89	8.38	60.43	60.37	0.06	0.11	0.011
26		17.42	23.89	9.08	58,66	57.28	1.38	2.40	5.783
27		17.88	23.89	9,06	58.21	57.36	0.85	1.47	2.173
			(Fixe	d) A0 =	25.94		AVG.	AVG.	STD.
				A1 =	-0.32		DEV.	%DEV.	DEV.
				A2 =	310,9		0.00	0,00	4.68
									====

COND. (Calc.) = A0 + A1\*DENS + A2/DENS

Pt.	APPA-	THICK-	MEAN	DENSITY	COND.	COND.	DEV.	DEV.	DEV.
Nr.	RATUS	NESS	TEMP.	_	( mW/(m.K) )				SQRD.
		(cm)	(°C)	(kg/m³)	(Exp.)	(Calc.)		(Pct.)	(Pct.)
1	1	14.91	23.93	34.39	52.18	49.43	2.75	5.57	30.984
2		14.91	23.91	30.79	56.50	49.40	7.10	14.37	206.454
3		14.41	23.94	33.43	52.90	49.39	3.51	7.10	50.369
4	2	15.20	23.99	33.95	45.08	49.41	-4.33	-8.76	76.822
5		14.10	23.98	33.55	45.81	49.40	-3.59	-7.26	52.750
6		14.40	23.97	31.79	47.29	49.38	-2.09	-4.23	17.915
7	3	15.24	23.93	49.79	52.75	51.64	1.11	2.15	4.634
8		15.24	23.92	47.95	53.04	51.26	1.78	3.47	12.062
9		15.24	23.88	50.64	51,60	51.82	-0.22	-0.43	0.181
10	6	15.75	23.90	35.80	48.73	49.51	-0.78	-1.57	2.479
11		16.38	23.90	31.00	50.42	49.40	1.02	2.07	4.305
12		16.33	23.88	30.70	49.44	49.40	0.04	0.07	0.005
13	7	20.57	24.17	36.59	47.13	49.57	-2.44	-4.92	24.218
14		18.53	24.17	34.23	49.00	49.42	-0.42	-0.85	0.728
15		17.50	24.11	33.23	48.86	49.39	-0.53	-1.07	1.150
16	8	16.89	23.94	27.87	48.31	49.62	-1.31	-2.65	7.005
17	_	16.89	23.94	27.39	49.61	49.69	-0.08	-0.15	0.023
18		16.89	23.94	25.63	51.63	49.99	1.64	3.28	10.740
19	10	17.58	23.89	34.57	47.49	49.44	-1.95	-3.94	15.511
20		17.97	23.89	32.89	48.54	49.38	-0.84	-1.71	2.916
21		18.09	23.89	33.88	49.10	49.41	-0.31	-0.62	0.389
			(Fixe	ed) A0 =	25.940		AVG.	AVG.	STD.
				A1 =	0.364		DEV.	%DEV.	DEV.
				A2 =	377.40		0.00	-0.00	5.24

TABLE 7. Thickness, Mean Temperature, Density and Apparent Thermal Conductivity of Fibrous Mineral Wool Loose Fill as Measured on Seven Different Apparatus. Column headings and calculations are as described in caption of Table 1.

COND. (Calc.) = A0 + A1\*DENS + A2/DENS

TABLE 8. Thickness, Mean Temperature, Density and Apparent Thermal Conductivity of Perlite as Measured on Five Different Apparatus. Column headings and calculations are as described in caption of Table 1.

....

Pt. Nr.	APPA- RATUS	THICK- NESS	MEAN I TEMP.	DENSITY	COND.	COND.	DEV.	DEV.	DEV.
		(cm)	(°C) (	(kg/m <sup>3</sup> )	(Exn.)	(Calc.)		(Pet )	(Pct)
		(•)		(	(2000)	(curci)		(2001)	(1001)
1	2	10.20	24.00	90.08	44.51	46.15	-1.64	-3.56	12.645
2		10.21	24.01	86,98	45.03	45.99	-0.96	-2.09	4.359
3		10.32	24.06	95.58	45.86	46.48	-0.62	-1.33	1.780
4	3	15.24	23.96	84.11	44.39	45.86	-1.47	-3.20	10.262
5		15.24	23.98	97.88	45.83	46.63	-0.80	-1.72	2.960
6		15.24	23.95	89.07	44.82	46.10	-1.28	-2.77	7.670
7	7	16.51	24.17	75.45	46.27	45.59	0.68	1.50	2.239
8		16.38	24.17	75.29	45.98	45.58	0.40	0.87	0.752
9		16.61	24.17	75.77	46.12	45.59	0.53	1.15	1.331
10	8	16.95	23.94	72.09	45.08	45.54	-0.46	-1.02	1.035
11		17.02	23.94	82.98	46.09	45.81	0.28	0.61	0.367
12		17.02	23.94	85.55	46.86	45.92	0.94	2.04	4.167
13	9	15.01	23.89	93.72	47.85	46.36	1.49	3.21	10.286
14		15.01	23.89	95.64	47.56	46.48	1.08	2.31	5.359
15		15.01	23.89	94.68	48.28	46.42	1.86	4.00	16.006
			(Fixed	I) A0 =	25.94		AVG.	AVG.	STD.
			• • • • • • • •	A1 =	0.1396	;	DEV.	%DEV.	DEV.
				A2 =	687.4		0.00	-0.00	2.50
					=				

COND. (Calc.) = A0 + A1\*DENS + A2/DENS



Figure 1. Apparent thermal conductivity,  $\lambda$ , of fibrous glass batt (blanket) as a function of density, at a mean specimen temperature of 24°C (75°F), as measured on 11 different apparatus. The curve is a least-squares fit of Eq. (1), and the line is a least-squares fit of Eq. (2), shown for comparison. The coefficients for both fits are given in Table 9. The standard deviation ( $\sigma$ ) of the residuals between the data and the curve is 1.3%; for the line  $\sigma$  is 1.5%.


Figure 2. Apparent thermal conductivity,  $\lambda$ , of cellulose loose fill as a function of density, at a mean specimen temperature of 24°C (75°F), as measured on 8 different apparatus. The curve is a least-squares fit of Eq. (1); the line is a least-squares fit of Eq. (2), shown for comparison. The coefficients for both fits are given in Table 9. The standard deviation ( $\sigma$ ) of the residuals between the data and the curve is 2.9%; for the line  $\sigma$  is also 2.9%.



Figure 3. Apparent thermal conductivity,  $\lambda$ , of unbonded fibrous glass loose fill as a function of density, at a mean specimen temperature of 24°C (75°F), as measured on 9 different apparatus. The curve is a least-squares fit of Eq. (1), and the line is a least-squares fit of Eq. (2), shown for comparison. The coefficients for both fits are given in Table 9. The standard deviation ( $\sigma$ ) of the residuals between the data and the curve is 4.7%; for the line  $\sigma$  is 4.8%.



Figure 4. Apparent thermal conductivity,  $\lambda$ , of loose-fill mineral wool as a function of density, at a mean specimen temperature of 24°C (75°F), as measured on 7 different apparatus. The curve is a least-squares fit of Eq. (1), and the line is a least-squares fit of Eq. (2), shown for comparison. The coefficients for both fits are given in Table 9. The standard deviation ( $\sigma$ ) of the residuals between the data and the curve is 5.1%; for the line  $\sigma$  is 5.5%.



Figure 5. Apparent thermal conductivity,  $\lambda$ , of perlite as a function of density, at a mean specimen temperature of 24°C (75°F), as measured on 5 different apparatus. The curve is a least-squares fit of Eq. (1), and the line is a least-squares fit of Eq. (2), shown for comparison. The coefficients for both fits are given in Table 9. The standard deviation ( $\sigma$ ) of the residuals between the data and the curve is 2.5%; for the line  $\sigma$  is also 2.5%.

Mater- ial	Coefficients for Eq. 1: $\lambda = \mathbf{A}_0 + \mathbf{A}_1 \cdot \rho + \mathbf{A}_2 / \rho$			Straight-line Coefficien $\lambda = B_0 + B_1 \cdot \rho$	
	A <sub>0</sub>	A	A2	Bo	B <sub>1</sub>
Batt	25.94	0.348	99.1	41.90	-0.2877
Cell	25.94	0.235	198.3	39.77	3.683·10 <sup>-3</sup>
Glass	25.94	-0,320	310.9	104.57	-5.241
M Wool	25.94	0.364	377.4	46.20	0.1030
Perl	25.94	0.1396	687.4	42.13	0.04519

TABLE 9. Coefficients of the Correlations of Apparent Thermal Conductivity,  $\lambda$ , with Density,  $\rho$ , for Each Material Tested in the Round Robin.

The reader is cautioned that the purpose of this work was not primarily to measure the apparent conductivities of the insulations used, but to compare the reproducibility of the C 687 test method among different laboratories. The data obtained in this study were acquired at a single temperature  $(75^{\circ}F)$  and over a limited range of densities; thus the correlations (1) and (2) and the associated numerical coefficients are not necessarily reliable guides to the values of apparent thermal conductivity of other insulation specimens of the same type.

The standard deviations  $\sigma$  of the residuals between the data and the fitted curves (Eqs. (1) and (2)) are listed in Table 10 as imprecisions, measured by  $2\sigma$ . For comparison, the  $2\sigma$  imprecisions of the test results from the 1986 round robin are also listed.

For all five materials tested in this round robin, there is no appreciable difference between the standard deviations obtained by use of either Eq. 1 (curve) or Eq. 2 (line) for the data from this (1990) round robin. The imprecisions for cellulose and for perlite are slightly less for the straightline fit, but for glass fiber and mineral wool the imprecisions are slightly less for the curve. The imprecision for the fibrous glass blanket is essentially the same for both round robins. Except for mineral wool, the imprecisions for the loose-fill materials tested in this round robin are appreciably reduced from those obtained in the 1986 round robin. The imprecision for cellulose is less than one third of that obtained in the previous round robin. The imprecision for the unbonded glass fiber is reduced to almost one half the previous value. The range of imprecisions  $(2\sigma)$  for the four materials tested in common to both round robins was (a) 1986: 10 to 21%, and (b) 1990: 5 to 11%. Eight of the present nine participants also took part in the 1986 round robin, out of a total of eleven. Thus the comparison of imprecisions between the two round robins is especially meaningful, reflecting improvement in technique.

Material	Imprecision, 20 (Percent)				
	1986 Round Robin	<b>1990 Rou</b> Eq. (1) (curve)	nd Robin Eq. (2) (line)		
Glass blanket	3	2.8	3.0		
Cellulose	21	5.9	5.8		
Bonded glass fiber	14				
Unbonded glass fiber	16	9.4	9.5		
Mineral Wool	10	10.5	11.0		
Perlite		5.0	5.0		
Range (percent)	3 to 21	2.8	8 to 11		

TABLE	10.	COMPARISON (	OF	IMPRECISIONS	OF	TEST	RESULTS	FROM
		1986 AND 199	90	ROUND ROBINS.				

#### SUMMARY

In the analysis of the previous (1986) round robin<sup>1</sup>, the 3% imprecision in the measurements of  $\lambda$  for fibrous glass blanket showed that the bias of the mean was negligible when measuring  $\lambda$  with heat-flow-meter apparatus. However, the range of 10 to 21% in the imprecision of  $\lambda$  for loose-fill insulation showed that the techniques of preparing loose-fill specimens for thermal testing needed to be improved.

The test results obtained in this (1990) effort for  $\lambda$  of four different loose-fill materials similar to those used in the previous round robin indicate that the imprecision of the measurements by the nine participants can be characterized by an imprecision ranging from 5 to 11 percent, appreciably better than the range of 10 to 21 percent for the imprecision of results obtained in the earlier round robin. The observation that eight laboratories took part in both round robins shows that appreciable improvement in the practices of the participants has taken place. Standard Practice C 687 has been revised in an attempt to more clearly spell out the procedures to be used in such measurements.

It must be noted that these measures of imprecision apply, strictly speaking, only to the measurements reported here for the participating laboratories. Other workers desiring to measure  $\lambda$  of loose-fill insulation with the same imprecision, by means of either the heat flow meter or the guarded hot plate, must ensure that the same care in experimental technique is used in preparing the specimens.

#### ACKNOWLEDGMENTS

The authors thank the participants for their prompt response and diligent effort in performing the measurements upon which this work is based. We also thank the following individuals representing manufacturers who supplied the materials for this interlaboratory comparison. They were, in alphabetic order of manufacturer, D. McCaa of CertainTeed Corporation, for unbonded fibrous glass loose fill; Whitney Waugh of Chemrock Corporation, for perlite loose fill; John Mumaw of Owens-Corning Fiberglas, for fibrous glass blanket; Duvon McGuire of Regal Industries Inc., for cellulose loose fill; and Jim Watson of US Gypsum Interior Systems, Inc., for mineral wool. Without their support this study would have required a much longer time to complete. The very helpful and constructive comments of three anonymous reviewers are also appreciated.

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# APPENDIX

COPIES OF FORMS SENT TO PARTICIPANTS

IN THE 1990 LOOSE-FILL ROUND ROBIN

ASTM C- 687 SURVEY FORM

TEST	APPARATUS:	Instrument Type :	
		Plate Area:	
		Metering Area:	
		Specimen Orientation:	Horizontal Vertical
		Heat Flux Direction:	Up Down Horizontal

MATERIALS TO BE TESTED: (Please indicate those you will test)

6 Inch Fiberglass Batt	
Perlite	_
Cellulose	All Materials
Rockwool	
Insulsafe	

PNEUMATIC EQUIPMENT:

Type of Blowing Machine(s)\_\_\_\_\_

		<u>Asth</u>	<u>C-68</u>	7 ROUND	ROBIN D	ATA SE	<b>FRT</b> S	L
Laborato	)rY					Da	te:_	
Material	·		<u>.</u>			_		
Blowing	Machi	ne Ma	nufac	turer			_	
Hose Typ	e			Hose	e Lengt	h		
Gear Set	ting			Gate	setti	ng		
Date Spe	cimen	Blow	n	Peed	ler RPM			
	47 (0000) F.M.		THE	RMAL DATA				
Thes	T	24	Delta T	Tmeen	Mete Area	ring Wt.	W	hole Box Wt.
		_		1				
k-valu	e	Test	ty	Test Thicknes	s Re	sistiv	ity	Time at Steady State
Date of Method o Conditio	Therma f Cone ning (	al Ter dition condit	st ning: ions	Labor :	Date M atory	ateria  Temper % Rela Durati	l Re _Cab atur tive on	ceived inet e Humidity
ASTM The	rmal '	fest )	letho	i Followe		-177		
Test Dur 1. Date 2. 3.	ation /Time	: IN	<u></u>		_Date/' 	Time O	 UT	
Sample T Pin Gage	hickn Thici	ess: kness	+ '	Fest Thic	kness	Set	tled	Thickness
			+			+		

		Page 2
Was Plate C	ontact Maintained During Test	?
Apparatus:	Metering Area	
nyyara cas .	Total Plate Area	_
	Specimen Orientation	
	Heat Flow Direction	
	Dimensions of Sample Box	
	Date of last Calibration	
	R-Value of Standard	
Estimate of	Accuracy of Measurement	
Method of D	etermining Settled Density	
Settle	d Density (if applicable)	
Observation:	s and/or Remarks :	
	<u> </u>	
<b>-</b>		
	<u>.</u>	

Send Completed Data Sheets to: Dr. David R. Smith N.I.S.T. Thermophysical Properties 325 Broadway Boulder, CO 80303

Phone: (303) 497-5374

Erv Bales

ASTM/DOE HOT BOX ROUND ROBIN

REFERENCE: Bales, E., "ASTM/DOE Hot Box Round Robin," Insulation Materials: Testing and Applications. 2nd Volume, ASTM STP 1116, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: This paper presents the results of inviting the operators of all guarded and calibrated hot boxes in North America to participate in a round robin, an interlaboratory test program according to ASTM E 691, "Conducting an Interlaboratory Test Program to Determine the Precision of a Test Method", to establish the precision and bias of the corresponding ASTM standard test methods: ASTM C 236, "Steady-State Thermal Performance of Building Assemblies by Means of a Guarded Hot Box", and C 976, "Thermal Performance of Building Assemblies by Means of a Calibrated Hot Box". The data and data analysis are reported from twenty-one laboratories that completed a series of tests on 4-inch thick specimens of cellular polystyrene thermal insulation. The data analysis resulted in two recommended statements of precision and bias.

Ninety-five per cent of the time measurements from two different laboratories on the same specimen would differ by up to +/-7.8% at a mean temperature of 75 F for the guarded hot box, and by up to +/-5.6% for the calibrated hot box. Based on guarded hot plate data for the specimen material, the reference value is 15.9 hr-F-ft sq/Btu. The mean of the thermal resistance reported for guarded hot boxes was 15.77 hr-F-ft sq/Btu, which differs by -1.2% from the reference value. The mean of the thermal resistance reported for calibrated hot boxes was 16.32 hr-F-ft sq/Btu, which differs by +2.6% from the reference value.

**KEYWORDS:** bias, hot box, precision, polystyrene, round robin, thermal insulation

The national importance of using energy efficiently in buildings requires that the thermal performance of typical,

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composite building sections be accurately specified. The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) publishes in the Handbook of Fundamentals a set of data on the thermal resistance of building materials, but there is no comparable set of data for built-up sections.(1) ASHRAE does provide steady-state, one-dimensional analytical models and equations. These are used for design guidance and are not precise enough to make the needed evaluations of competing energy conservation techniques in building construction. More sophisticated, extensive computer models, such as DOE-2, require accurate quantitative data on wall systems for the models use and Apparatus to test composite sections exist verification. and systems data are becoming available. ASHRAE has funded a project to gather and evaluate the data.

### Test Apparatus

The testing procedures for the original hot box are given in the American Society for Testing and Materials (ASTM) Standard Test Method, C 236, (2). The apparatus has a heated metering box placed over the center area of a test The complete specimen is surrounded by a larger specimen. guard box fitted against the periphery on the warm surface side, the metering box. The temperature in the guard box is controlled to match that in the metering box. Therefore, at steady state, all energy supplied to the metering box passes through the test specimen. Another box fits against the complete specimen on the cold surface opposite the quard box. The temperature in the guarded box simulates the interior temperature of a building . Temperature outside a building is simulated in the large box on the opposite side of the specimen. Thermal conductance of the specimen is the net rate of energy supplied to the metering box divided by the surface area and temperature difference between the specimen surfaces.

However, testing full-scale building sections with the guarded box apparatus becomes impractical because of the necessary large guard area. K. R. Solvason, at the National Research Council, Canada, designed the first Calibrated Hot Box to meter the heat flow into representative sections of building walls exposed to controlled temperature. It was built to operate without forced air circulation over the warm side of the test wall in order to simulate a wall in practice. It operated with the cold-side temperature either constant or varying according to some predetermined cycle.(3) Papers have been presented on the design of six calibrated hot boxes, (4,5,6,7,8,9,10 and 11). An ASTM Standard Test Method, C 976, is based on this experience.(12)

### Precision and Bias Statements

The ASTM Committee on Standards requires that a statement of precision and bias be included in all test method standards. Conditions of the requirements are published in "Form and Style for ASTM Standards",(13), the "Blue Book". Precision indicates to the user of a test how closely the test results will agree within a statistical sample, if the testing is conducted correctly. It indicates the reproducibility of the measurements of a single specimen in a number of similar apparatuses. Bias is the difference between the mean of a statistical sample and the true or accepted value. Note that accuracy pertains to how close a particular measurement is to the true value. As random error is reduced ,i.e., as the test method is improved, the precision will improve. Bias is a consequence of systematic error.(14,15)

A task force was established within ASTM Committee C16, Thermal Insulation, to generate the necessary data for precision and bias statements for inclusion in the two hot box standard test methods, C 236 and C 976.(2,12) A series of ASTM Standard Practices on how to conduct an interlaboratory test program and how to report the results as precision and bias statements were referred to in conducting this ASTM/DOE Hot Box Round Robin, (15,16,17)

## Data

Three different sets of data were measured during the Hot Box Round Robin. The first set was the data to characterize the specially manufactured lot of specimen material. A second set was data used to establish the reference value of the thermal resistance for the 4 inch thick specimen material. The round robin data of thermal resistance as a function of temperature were the third set. A comprehensive analysis of the data is presented in reference 18.

### Characterization Data

The specimen material used was expanded polystyrene, ASTM C 578, "Standard Specification for Preformed Cellular Polystyrene Thermal Insulation".(19) The density and apparent thermal conductivity of samples from the total inventory of material was measured and used to characterize the specimens. The conductivity was measured at 75 F deg with a heat flow meter apparatus, ASTM C 518, "Standard Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus".(20)

The manufacture's 65 data points were analyzed using a linear regression. The data indicated that the lot of material manufactured for the round robin had an average density of 1.32 lbs/cu ft +/- 0.14 lbs/cu ft before the surfaces were painted. The resultant equation is, R =0.557 D + 3.542, for thermal resistance as a function of density for the one inch thick, unpainted material; a thermal resistance of 4.27 hr-sq ft-deg F/Btu-in at 75 deg F for the average density. Bias Data In order to determine the bias of the round robin data it is necessary to know the absolute value of the thermal The National Institute of Standards and resistance. Technology (NIST), a participant, using its one-meter guarded hot plate apparatus, ASTM C-177, (21), tested a sample of its round robin, 4 inch specimen material. In addition several other Round Robin participants and Oak Ridge National Laboratory tested the 4 inch specimen material. All the data from the five laboratories are shown in table 1 as function of mean temperature. A regression analysis for all the data in table 1, resulted in the following linear equation. The data are plotted in figure 1.

> $k = 0.213 + 5.03 \times 10-4 t$ (correlation coefficient of +0.9948)

Therefore, the reference thermal resistance for a 4 inch thick, painted specimen of 1.56 lb/cu ft density at a mean temperature of 75 F is: R = 15.94 hr-ft sq ft-deg F/Btu.

### Interlaboratory Data

A hot box requires a large cross section of specimen material. It is not practical to have one sample passed among laboratories. Therefore, a large lot of material was manufactured under controlled conditions for use in the round robin. It was cut into 4x7 ft and 4x14 ft boards as required by the particular laboratory. The boards were cut by the participants to fit their hot box specimen frame. The joints were fasten with an adhesive; the complete specimen was fitted into the frame an seal around the edges. The boards were painted with two coats of adhesive and two coats of latex exterior paint to prevent leakage between hot box chambers and to protect the surfaces. A manual on specimen handling, construction, mounting and instrumentation techniques was supplied to each participating laboratory. (18)

Twenty-one laboratories submitted data with a total of 76 data sets. The principal data are specimen mean temperature, thermal resistance and density. The R-value for all data points are plotted in figure 2. Statistical analysis according to ASTM procedures indicated that there are seven outliers which are identified in the figure. These seven data points were discarded from the data base for the final analysis. A regression equation is included in figure 2 with the outliers excluded. Figure 3 shows the data for the guarded hot box only. A linear regression equation is plotted and given in equation form. The calibrated hot box data is similarly presented in figure 4.









## Table 1

## DATA USED TO ESTABLISH BIAS

## Specimen: Expanded Polystyrene, ASTM C 578 4 inches thick, painted surfaces

<u>Apparatus</u>	<u>Apparent Conductivity</u>	<u>Mean</u>	<u>Density</u>
	Btu-in/hr-sq ft-deg F	<u>Temp</u> degF	lb/cu ft
Laboratory A			
GHP	0.251	70.3	1.72
GHP	0.262	95.3	1.72
HFM	0.254	75.3	1.72
Laboratory B			
GHP	0.262	99.5	1.60
GHP	0.249	75.5	1.60
GHP	0.242	59.6	1.60
Laboratory C			
HFM	0.249	75.2	1.72
HFM	0.247	75.2	1.72
HFM	0.247	75.2	1.72
HFM	0.244	75.0	1.72
Laboratory D	(NIST)		
GHP	0.264	100.1	1.56
GHP	0.251	75.0	1.56
GHP	0.230	31.9	1.56
GHP	0.231	32.1	1.56
Laboratory E	(ORNL)		
Thin Screen	0.259	91.9	1.55
Thin Screen	0.272	116.4	1.55
Thin Screen	0.278	127.9	1.55
Thin Screen	0.287	144.8	1.55
Thin Screen	0.256	85.0	1.55
Thin Screen	0.265	102.8	1.55
Thin Screen	0.282	135.9	1.55
Thin Screen	0.267	105.5	1.55

GHP - Guarded Hot Plate, ASTM C 177 (21) HFM - Heat Flow Meter, ASTM C 518 (22) Thin Screen - "Standard Test Method for Steady-State Thermal Transmission Properties by Means of the Thin-Heater Apparatus," ASTM C 1114 (23)

### Data Analysis

The data on mean temperature, specimen thermal resistance, specimen density, and hot box type were statistically analyzed using ASTM guidelines.(15,16,17) The initial analysis of the data indicated that seven observations are outliers. An outlier is an observation that deviates markedly from the other observations in a data set.(16) Linear regression equations were fitted to the data with and without the outliers. The outliers shift the linear equation and change the slope. These seven data points were discarded because of the statistically undue influence on the resultant statistics.

The analysis of the data set including both types of box leads to results that are different from those obtained when each of the guarded and calibrated box sets are analyzed separately. The object of the round robin was to develop individual precision and bias statements for each apparatus. Therefore, the individual data sets are used as the basis for recommended precision and bias statements for the individual apparatus.

Regression Equation for data from the Guarded Hot Box

There are 44 sets of data for the guarded hot box. The data and the linear regression equation are shown in Figure 3. The statistical analysis provided the following results:

<u>Mean Temp</u> deg F	<u>Resistance</u> Hr-Ft <sup>2</sup> -F/Btu	<u>Two Sigma</u> Hr-Ft <sup>2</sup> -F/Btu	<u>95% Confidence</u> Hr-Ft <sup>2</sup> -F/Btu
40	16.75	1.22	15.5,18.0
75	15.77	1.23	14.5,17.0
110	14.79	1.27	13.5,16.1

Regression Equation for data from the Calibrated Hot Box

There are 25 data sets for the calibrated hot box. The data and the linear regression equation are shown in Figure 4. The statistical analysis provided the following results:

<u>Mean Temp</u>	<u>Resistance</u>	<u>Two Sigma</u>	<u>95% Confidence</u>
deg F	hr-ft <sup>2</sup> -F/Btu	hr-ft <sup>2</sup> -F/Btu	hr-ft <sup>∠</sup> -F/Btu
40	17.44	0.92	16.5,18.4
75	16.32	0.92	15.4,17.2
110	15.20	0.98	14.2,16.2

## Conclusions and Recommendations

The round robin satisfied the objective of providing the background necessary for statements of precision and bias. It was found that the calibrated hot box is more precise that the guarded hot box. Based on the reference value of thermal resistance, 15.94 hr-sq ft-F/Btu at 75 F, the calibrated hot box has a positive bias of +2.4% and the guarded hot box has a negative bias of -1.1% The bias for both types of box is small, indicating adequate standard testing practices. The precision could be improved with further refinement of the operation of the individual hot boxes.

A comparison should be made of the techniques being used to account for the thermal flanking around specimens in the calibrated hot box. The flanking energy is on the order of 5-10% of the total being supplied to the box. Each laboratory has developed its own analysis, frequently very complex finite numerical techniques.

This round robin included only North American facilities. It is recommended that a world wide round robin be conducted using specimen of another material.

The following Precision and Bias Statements were recommended to C-16.30.

for the Guarded Hot Box:

Background: A round robin for guarded and calibrated hot boxes was conducted with 21 laboratories participating; 15 had guarded hot boxes. Data were reported for 4 inch thick, homogeneous specimens of expanded polystyrene board (ASTM C-578). Each laboratory received material from a special manufacturer's lot that was controlled to maintain a uniform density. At a mean temperature of 75 F, the average R value was 15.77 hr-F-ft sq/Btu for the 44 data sets measured with a guarded hot box . The regression equation for the data was:

R = 17.867 - 0.028 t (t in deg F)

over the range of 40 F to 110 F. The specimen density ranged from 1.26 to 1.49 lbs/cu ft. The final report is available from ASTM.(18)

<u>Precision</u>: At a specimen thermal resistance predicted by the above equation on the basis of test error alone, the difference in absolute value of two test results obtained in different laboratories on the same specimen materials will be expected to exceed the reproducibility interval only 5% of the time according to the following table:

mean	thermal	reproducibility	<u>change in R</u>
<u>cemp</u> deg F	hr-ft <sup>2</sup> -F/Btu	<u>interval</u>	hr-ft <sup>2</sup> -F/Btu
40	16.75	+/-7.3%	+/-1.22
75	15.77	+/-7.8%	+/-1.23
110	14.79	+/-8.6%	+/-1.27

That is, measurements from two different laboratories correctly conducted on the same specimen would differ by up to +/-7.8% at a mean temperature of 75 F ninety-five percent of the time.

<u>Bias</u>: Based on guarded hot plate data (ASTM C-177) from the National Institute of Standards and Technology and supported by measurements from other laboratories, the reference value at 75 F mean temperature is 15.94 hr-F-ft sq/BTU. The mean thermal resistance of 15.77 hr-F-ft sq/ Btu as measured with a guarded hot box differs by -1.1% from the reference value.

for the Calibrated Hot Box:

<u>Background</u>: A round robin for guarded and calibrated hot boxes was conducted with 21 laboratories participating; 6 had calibrated hot boxes. Data was reported for 4 inch thick, homogeneous specimens of expanded polystyrene board (ASTM C 578). Each laboratory received material from a special manufacturer's lot that was controlled to maintain a uniform density. At a mean temperature of 75 F, the average R value was 16.32 hr-F-ft sq/Btu for the 25 data sets measured with the calibrated hot box . The regression equation for the data was:

R = 18.717 - 0.032 t (t in F)

over the range of 40 F to 110 F. The mean specimen density ranged from 1.26 to 1.49 lbs/cu ft. The final report is available from ASTM.(18)

<u>Precision</u>: At a specimen thermal resistance predicted from the above equation on the basis of test error alone, the difference in absolute value of two test results obtained in different laboratories on the same specimen materials will be expected to exceed the reproducibility interval only 5% of the time according to the following table:

mean	thermal	reproducibility	<u>change in R</u>
deg F	hr-ft <sup>2</sup> -F/Btu	Incerval	hr-ft <sup>2</sup> -F/Btu
40	17.44	+/-5.3%	+/-0.92
75	16.32	+/-5.6%	+/-0.92
110	15.20	+/-6.4%	+/-0.97

That is, measurements from two different laboratories correctly conducted on the same specimen would differ by up to +/-5.6% at a mean temperature of 75 F ninety-five percent of the time.

<u>Bias</u>: Based on guarded hot plate data (ASTM C-177) from the National Institute of Standards and Technology supported by measurements from other laboratories, the reference value at 75 F mean temperature is 15.94 hr-F-ft sq/BTU. The mean thermal resistance of 16.32 hr-F-ft sq/ Btu as measured in the calibrated hot box differs by +2.4% from the reference value.

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THERMAL PROPERTIES OF SELECTED MATERIALS FROM STEADY-STATE AND TRANSIENT TESTS

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ABSTRACT: The Unguarded Thin-Heater Apparatus (UTHA, ASTM C 1114) was used to determine the thermal conductivity (k), specific heat (C), and thermal diffusivity ( $\alpha$ ) of selected building materials from 24 to 50°C. Steady-state and transient measurements yielded data on four types of material: gypsum wall board containing 0, 15, and 30 wt% wax; calcium silicate insulations with densities ( $\rho$ ) of 307, 444, and 605 kg/m<sup>3</sup>; three wood products: southern yellow pine flooring (575 kg/m<sup>3</sup>), Douglas fir plywood (501 kg/m<sup>3</sup>), and white spruce flooring (452 kg/m<sup>3</sup>); and two cellular plastic foams: extruded polystyrene (30 kg/m<sup>3</sup>) blown with HCFC-142b and polyisocyanurate rigid board  $(30.2 \text{ kg/m}^3)$  blown with CFC-11. The extruded polystyrene was measured several times after production (25 days, 45 days, 74 days, 131 days, and 227 days). The UTHA is an absolute technique that yields k with an uncertainty of less than  $\pm 2$ % as determined by modeling, by determinate error analyses, and by use of Standard Reference Materials SRM-1450b and SRM-1451.

In the transient mode of operation, a step-change in heat flux was applied to specimens that were initially isothermal or that had a steady, imposed temperature gradient. Analysis of the temperature response of the thin heater for short times predicts a temperature rise that is a linear function of the square-root of time. This behavior was observed for materials with densities above 300 kg/m<sup>3</sup> and the slope of this relation provided experimental values for the product  $k \cdot \rho \cdot C$ . The overall error in determining the product  $k \cdot \rho \cdot C$ is less than l%. Values of C were calculated from experimental values of k,  $\rho$ , and the product of  $k \cdot \rho \cdot C$ .

However, for the 30 kg/m<sup>3</sup> foams, the thermal mass of the thin heater delayed the onset of the linear temperature rise with the square-root of time. This effect negated the use of the short-time transient analysis for low density materials. A finite difference program that described this observation provided an alternate method to obtain values for the product  $k \cdot \rho \cdot C$  from the experimental data.

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KEYWORDS: thermal conductivity, thermal diffusivity, specific heat, thin-heater apparatus, gypsum containing wax, wood products, calcium silicate, plastic foams, extruded polystyrene, polyisocyanurate

#### INTRODUCTION

This paper describes the use of the ORNL Unguarded Thin-Heater Apparatus (UTHA) in the steady-state and transient modes of operation to determine thermophysical properties of building materials from 24 to 50°C (75 to 120°F). The thermophysical properties determined were thermal conductivity (k), specific heat (C), and thermal diffusivity ( $\alpha$ ). These properties were determined for gypsum wall board with and without wax, calcium silicate insulations, three wood products, and two cellular plastic foams.

The need for transient analysis of buildings has increased interest in measuring the nonsteady-state thermal behavior of building materials. Transient analysis is possible with the ORNL UTHA because the heat capacity of the screen heater is small relative to that of commonly used measurement systems and can be small relative to that of many test specimens of building materials.

The use of the UTHA in the steady-state and transient modes of operation was initially prompted by the need to test gypsum board specimens containing 0, 15, or 30 weight percent wax ( $C_{18}H_{38}$ , n-octadecane) in the temperature range 20 to 50°C. These tests were part of an effort to develop a gypsum wall board with enhanced thermal storage capacity.

These transient tests showed the potential value of applying this technique to other building materials to develop a thermophysicalproperty data base.

### EQUIPMENT

The UTHA is an absolute, longitudinal heat flow technique with less than 2% uncertainty for steady-state thermal resistance tests at mean temperatures from 20 to 50°C. This technique has been described extensively in the literature (1-9).

Transient tests can be performed in the UTHA as a two-sided test by applying a step-change in heat flux to an initially isothermal specimen, or to a specimen with a steady-state temperature gradient. The resulting temperature-time behavior of the heater can be analyzed to obtain specimen properties (3,7).

A Hewlett-Packard (HP) 9121 computer was used to control the test and data was logged by a HP 3497A data acquisition system (10). The average heater temperatures were determined from the readings of five thermocouples located in the central area of the heater. The emf of each thermocouple was measured to 1  $\mu$ V and converted to temperature by use of a calibration table (11). Data were recorded every 12 seconds from 10 minutes prior to changing the power until 60 minutes after the power was changed, then every 120 seconds for the next 300 minutes, and then every 300 seconds until steady-state was achieved. The software allows 1000 points to be recorded. For each specimen, the first transient test was conducted on an isothermal specimen with the initial temperature difference between the plates and the screen less than 0.05 K for plate temperatures between 25 and 50°C. At the conclusion of the first transient test, a steady-state measurement was made of the specimen k. The second transient test was initiated with a temperature gradient on the specimen and was followed by a steady-state measurement of the specimen k. This procedure was repeated to obtain three or four heating transient tests on most specimens. Finally, a cooling transient test was conducted by a step-change in the heat flux to zero.

### ANALYSIS OF TRANSIENT TESTS

#### <u>Method 1</u>

A solution of the heat conduction equation is given by Carslaw and Jaeger (12) for the case of a semi-infinite solid that is initially at constant temperature, T°, and which is subjected to a constant heat flux, F, at the surface x = 0 for time  $(t) \ge t_o$ . The temperature-time relationship at x = 0 (the heater) is:

$$T = T^{\circ} + \frac{2F}{k} \left[ \frac{\alpha}{\pi} \cdot (t - t_o) \right]^{V_k} \qquad (1)$$

For a short time interval this equation for a semi-infinite material is applicable to a material of finite thickness (L), such as the specimens in the UTHA. The short time interval for the denser materials of interest is about 1800 s. In the current work, the temperature versus time and the heat flux are measured, so Eq. 1 allows the quantity  $k\rho C$  to be determined from the slope of a leastsquares fit of the temperature versus  $(t-t_o)^{\frac{1}{2}}$  data.

Equation 1 can be arranged as follows:

$$T = T^{\circ} + \frac{2F}{(\pi k \rho C)^{\frac{1}{2}}} (t - t_o)^{\frac{1}{2}} = A + B (t - t_o)^{\frac{1}{2}} .$$
 (2)

Thus, a linear-least-squares fit of the heater temperature as a function of  $(t-t_o)^{\frac{1}{2}}$  can be performed.

The principle of superposition allows the Eq. 2 to be extended to a material which has an initial temperature gradient. In the determination of  $k\rho C$  using Eq. 2 with a temperature gradient, F equals the change in heat flux applied at time  $t_o$ . For a two-sided test, the change in heat flux is F/2.

Figure 1 shows an early test where the power step was initiated (arbitrarily) at 877 s. The x-axis is the square root of time after the power step, i.e. 1600 s into the power step corresponds to 40 s<sup> $\frac{1}{2}$ </sup>. The y-axis is the heater temperature. Figure 1 shows that the experimentally determined heater temperature is a linear function of (time)<sup> $\frac{1}{2}$ </sup> after the heat flux was changed at 877 s for a transient test (T1R) on a gypsum board. A least-squares fit of 151 points between 877 and 2689 seconds yielded

$$T = 20.96 + 0.06446 (t - 877)^{\frac{1}{2}}$$
(3)

...

with an average percent deviation of less than 0.06%. The T° value of 20.96°C is within 0.01°C of the average cold plate temperature value of 20.97°C. The slope term and the heat flux change of 41.281 W/m<sup>2</sup> yield a  $k\rho$ C value of 130560 W<sup>2</sup>·s/m<sup>4</sup> K<sup>2</sup>, which yields a C of 1055 J/kgK for this specimen with a k of 0.178 W/m·K and  $\rho$  of 695.1 kg/m<sup>3</sup>. This procedure for data analysis was called Method 1.1 and used the complete data set for times from 200 to 1800 seconds after the heat flux change. Method 1.2 used only 10 data points selected in uniform increments from the complete data set for the same time period. Method 1.2 used less data but yielded coefficients within 1% of those of Method 1.1.



FIG. 1 -- Temperature of the thin heater as a function of  $\sqrt{\text{time}}$  after initiating the power step (1600 seconds into power step is 40 s<sup>k</sup>) for run TlR on a gypsum board (695 kg/m<sup>3</sup>).

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Several indeterminate errors arise in the analysis of the transient tests. The development of Eq. 2 assumes that the heat flux is constant. During the early stage of a transient, however, the thin heater of the experimental apparatus undergoes a significant temperature change, since this heater has a small thermal mass. This was illustrated in the results as an offset of approximately 0.03°C between the known initial temperature and that resulting from the least-squares analysis (see Fig. 1). This effect has been discussed by Domingos and Voelker (13).

Figure 1 shows that deviations of the temperature rise of the heater from a straight line begins to occur near 1800 s. Comparisons of the complete solution to the short-time solution at 1800 s show the short-time solution is 1 mK too low (7, 8). To minimize this error, the limit of applicability of Eq. 1, the short-time solution, was taken as 1800 s. In general, the results of the transient experiments show very good internal agreement for the values of  $k\rho C$  for dense materials. Thus, overall error in  $k\rho C$  is less than  $\pm 1$ %.

### <u>Method 2</u>

This method of analyzing transient data was used after tests on low density  $(30 \text{ kg/m}^3)$  extruded polystyrene yielded a temperature response of the thin heater that was delayed in time. This time lag was associated with the thermal mass of the heater. To reduce the effect of this time lag, the step change in heat flux was reduced.

$$F(reduced) = F - F_1 = F - \frac{dT}{dt} \left[ \frac{x}{2} \cdot \rho \cdot C \right]$$
  
= F - 303.79 \cdot  $\frac{dT}{dt}$  (4)

where dT/dt is the rate of temperature change of the heater,  $\rho$  is the heater density, 2115 kg/m<sup>3</sup>, C is the heater specific heat, 435 J/kg K, and x/2 is half the heater thickness, 330 x 10<sup>-6</sup> m.

#### <u>Method 3</u>

This method of analyzing the transient data evolved from Method 2 and is described in detail in Ref. 8. It is based on a finite difference model of the transient behavior of an infinite slab of thickness L and a thin heater of thickness 0.5  $t_{sc}$ . The boundary of the slab at x = L is maintained at constant temperature T(0). The boundary at x = zero is in intimate contact with the thin heater. The heater has a finite thermal mass, 0.5  $t_{sc} \cdot \rho_{sc} \cdot C_{sc}$ , per unit area and high thermal conductivity relative to the slab. The slab may initially be isothermal or have a temperature profile. The model yields the heater temperature versus time for a given step change in the power to the heater.

The model involves dimensionless position, X = x/L, Fourier Number (or dimensionless time), N<sub>Fo</sub> =  $\alpha t/L^2$ , dimensionless temperature, and thermal mass ratio, TMR = 0.5 t<sub>sc</sub>· $\rho_{sc}$ · $C_{sc}/L \cdot \rho \cdot C$ . The zero-heaterthermal-mass case, i.e. TMR equal zero, corresponds to Method 1. As the thermal mass ratio increases, a nonlinear regime in the relationship between temperature and the square root of time occurs at short times. The length of this regime increases as TMR increases. In all cases, the nonlinear regime is followed by a linear region. The linear region is then followed by curvature in the time-temperature relationship, as was the case for TMR equal zero. For TMR less than or equal to 1, however, the linear region occurs for dimensionless temperatures between 0.3 and 0.5.

The slopes of the dimensionless heater temperatures versus the square root of Fourier number,  $N_{\rm Fo}$ , are a function of the TMR and  $N_{\rm Fo}$ . In the linear region between dimensionless temperatures of 0.3 and 0.5, the slope is solely a function of TMR and can be expressed as a polynomial in TMR:

$$\frac{S(TMR)}{S(0)} = 1 + a TMR + b (TMR)^2 + \dots$$
 (5)

S(TMR) is measured in a test. S(o) is given in Eq 2. Since C appears on both sides of Eq. 5 an iterative technique is used to solve for C.

#### SPECIMENS

Two-sided and one-sided steady-state tests and two-sided transient tests were conducted on the specimens described in this section. The heater was positioned at the mid-plane of each test specimen, and the entire heater was covered by the specimen or by perimeter materials of similar properties. Table 1 lists the materials tested, the test specimen density and thickness, and the values of k, C, and  $\alpha$  at 24°C estimated from published data (14-23) as described below and in Ref. 7 and 8. This paper does not describe results on the gypsum with and without wax test specimens, since this has been published elsewhere (7,8).

The calcium silicate board test specimens were duplicate slabs, each nominally 0.9 m x 1.5 m x 0.038 m, of three densities (307, 444, and 605 kg/m<sup>3</sup>) that were conditioned in the Building Materials laboratory prior to testing (24). Calcium silicate insulation is a reacted hydrous calcium silicate material made using uncalcined diatomaceous earth and lime blended with reinforcing fibers (25). ASTM Standards (26, 27) and literature (16) give k as a function of density for calcium silicate and yielded the estimates of k given in Table 1. Specific heat values of 836 - 878 J/kg·K and 1045 - 1170 J/kg·K are listed for hydrous calcium silicate (16), and yield an average C of about 1000 J/kg·K. Table 1 shows that the estimated thermal diffusivity decreases as the specimen density increases.

The southern yellow pine flooring and white spruce flooring test specimens were made by gluing individual tongue-in-groove boards together and planing their surfaces to produce  $0.9 \text{ m} \times 1.5 \text{ m}$  slabs of uniform thickness. The Douglas fir plywood test specimen was made from  $1.2 \text{ m} \times 2.5 \text{ m}$  plywood boardstock by cutting a  $0.9 \text{ m} \times 1.5 \text{ m}$  slabs and sanding one surface to produce a uniform thickness. Tests were conducted with heat flowing perpendicular to the board face, as would be the case for a flooring application (30). Test specimens were

conditioned in the Building Materials laboratory for several weeks prior to testing, and the average moisture content did not change significantly as a result of the thermal tests. Some moisture redistribution from warm to cold regions was noted. The estimated properties given in Table 1 were obtained from an assessment of the thermal conductivity and specific heat of wood and wood products by TenWolde, McNatt, and Krahn (31).

Thermal test specimens,  $0.9 \text{ m} \times 1.5 \text{ m}$ , were prepared from  $1.2 \text{ m} \times 2.4 \text{ m}$  boardstock of extruded polystyrene and polyisocyanurate plastic cellular foam. Each polystyrene test specimen was nominally 0.051 m thick, had a density of  $30 \text{ kg/m}^3$ , and was tested with the as-produced surfaces (32). The polystyrene was foamed with HCFC-142b. The polyisocyanurate test specimen was prepared by planing a rigid foam boardstock blown with CFC-11. The density of the test specimen was  $30.2 \text{ kg/m}^3$  and each specimen was nominally 0.032 m thick (33). Table 1 contains typical thermal properties given by Strzepek (34), Sparks (35), and Luikov, et al (36). The estimated  $\alpha$  for both increase with time and the value for extruded polystyrene is about 60% greater than that of the polyisocyanurate foam.

### THERMAL CONDUCTIVITY RESULTS

This section describes the k-values as a function of temperature that were determined for the building materials from steady-state tests and those determined before and at the end of the transient tests. The k-values found for the test specimens GOW, G15W, and G3OW were independent of temperature and found to be: GOW,  $0.1786 \text{ W/m} \cdot \text{K}$ from 24 to 45°C, G15W (liquid wax),  $0.1919 \text{ W/m} \cdot \text{K}$  from 29 to 34°C; and G3OW (liquid wax),  $0.2325 \text{ W/m} \cdot \text{K}$  from 29 to 32°C. The k-values increase with wax content and the average k-value for GOW is 25% greater than the estimated value given in Table 1. This value and the data given in Ref. 16 are described to 2% by

$$k = 0.1076 + 9.8344 \cdot 10^{-5} \rho$$
 (6)

...

This equation predicts k-values that are about 17% greater than those given in the ASHRAE Handbook (17).

Table 2 contains equations that describe the measured k-values as a function of temperature for calcium silicate test specimens: CS1, CS2, and CS3. The k-values increase with temperature and density. The data on each specimen are described by a linear equation in temperature and the average deviation of the data from the equation 0.26% or less. The k at 24°C and published values (16, 26, 28, and 29) are described by an equation with an average deviation of less than 2%

$$k = 0.03620 + 9.2316 \times 10^{-5} \rho - 9.505 \times 10^{-9} \rho^2 , \qquad (7)$$

Equation 7 yields k-values at  $24^{\circ}$ C that are 23 to  $36^{\circ}$  less than those given in Table 1.

Equations for the k results for the wood products are given in Table 3. The k-values increase with temperature for the three products and were fit by a least-squares method to a linear function

Code	Material	pensity kg/m <sup>3</sup>	lotat Thickness, m	k W/m·K	C J∕kg∙K	α m <sup>2</sup> /s <sup>-</sup> 10 <sup>7</sup>
10	Gypsum board	695.1	0.1002	0.142	1079	1.893
15W	Gypsum board - 14.1 wt/O wax	816.9	0.1011	0.161	1262	1.562
304	Gypsum Board - 30 wt/o wax	1000	0.1006	0.194	1465	1.324
SI	Calcium silicate 1	307.4	0.077	0.081	1000	2.635
S2	Calcium silicate 2	6-277	0.077	0.093	1000	2.095
23	Calcium sílicate 3	605.1	0.077	0.121	1000	2.000
ISF	White spruce flooring <sup>a</sup>	451.5	0.097	0.127	1627	1.729
FP	Douglas fir plywood <sup>a</sup>	500.9	0.073	0.115	1554	1.477
YPF	Southern yellow pine flooring <sup>a</sup>	575.2	0.098	0.151 0.020	1548	1.696
Ň	Polystyrene (HCFC-142b) <sup>b</sup> After 100 days	30.0	0.101	0.020 0.0288	1150 1150	5.797 8.348
IR	Polyisocyanurate (CFC-11) <sup>b</sup> After 1 <u>y</u> ear	30.2	0.064	0.017 0.024	1525 1525	3.691 5.211

Table 1 -- Estimated properties for building materials for steady-state and transient tests in the UTHA

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TABLE	2	 Equations	for	the	thermal	cone	ductivity	val	ues	for	three
		calcium si	ilica	ate s	specimens	s as	measured	in	the		
		ORNL	Ungu	arde	d Thin-H	eate	r Apparat	us			

CS1, 307 kg/m <sup>3</sup>	$k = 6.3842 \times 10^{-2} + 6.3009 \times 10^{-5} t, \pm 0.268^{a}$
CS2, 444 kg/m <sup>3</sup>	$k = 7.2640 \times 10^{-2} + 7.9893 \times 10^{-5} t, \pm 0.077$ %
CS3, 605 kg/m <sup>3</sup>	$k = 8.7021 \times 10^{-2} + 7.9729 \times 10^{-5} t, \pm 0.20$ %

<sup>a</sup>Average deviation.

TABLE 3 -- Equations for the thermal conductivity values for three wood products

WSF	$k = 0.09920 + 1.5537 \times 10^{-4} t, \pm 0.288^{a}$
DFP	$k = 0.09786 + 1.6039 \times 10^{-4} t, \pm 0.188^{a}$
SYPF	$K = 0.1234 + 1.16153 \times 10^{-4} t, \pm 0.118^{a}$

<sup>a</sup>Average deviation

of temperature. The resulting equations show that an average deviation of the results from the equations is less than 0.3%. The measured k-values at 24°C are within 3% of the estimated k-values for 0% moisture content: WSF (2.9%), DFP (2.2%), and SYPF (2.5%) (8). However, the measured k-values at 24°C are significantly lower than the estimated k-values for the measured moisture content of the test specimens: WSF (-19%), DFP (-11.7%), and SYPF (-15.9%). This result suggests that the equations used to estimate the k-values (31) should be modified.

Equations for the k-values as a function of temperature and time after manufacture for extruded polystyrene foamed with HCFC 142b and the results 398 days after manufacture on the polyisocyanurate foamed with CFC-11 are given in Table 4. The average percent deviation is less than  $\pm 0.15$ % for the five data sets for the extruded polystyrene and is  $\pm 0.3$ % for the polyisocyanurate. Figure 2 shows that the kvalues increase with temperature and increase with time after manufacture for the extruded polystyrene and the polyisocyanurate.

The equations given in Table 4 for extruded polystyrene were used to obtain the values of k at  $24^{\circ}$ C, and these values show the initial k ( $24^{\circ}$ C) value, 0.02492, is about 25% greater than the initial value estimated in Table 1, but the 100 day value, 0.0281, is within 3% of the estimated 100 day value. The k at  $24^{\circ}$ C for the polyisocyanurate is within the predicted range.

In a study on the aging of thin specimens planed from polyisocyanurate roof insulation boards (37), we have noted that ln 100 k is a linear function of  $(time)^{\frac{1}{2}}/thickness$ . The first three extruded polystyrene data points are described by

$$ln 100 k = 2.7896 + 10.81 \times 10^{-8} t^{\frac{1}{2}}/h$$
 (8)

where h is 5.08 cm, and t is time in seconds. The coefficient for the variable,  $10.81 \times 10^{-8} \text{ cm}^2/\text{sec}$ , is an apparent diffusion coefficient for air components in this polystyrene foam.

TABLE 4 -- Equations for the thermal conductivity as a function of temperature and time after production for extruded polystyrene foamed with HCFC 142b and polyisocyanurate foamed with CFC-11

PS (HCFC-142b): 25 days	$k = 0.02302 + 1.1653 \times 10^{-4} t, \pm 0.158^{a}$
PS (HCFC-142b): 45 days <sup>b</sup>	$k = 0.02368 + 1.2618 \times 10^{-4} t$ , ±0.014%
PS (HCFC-142b): 74 days	$k = 0.02480 + 1.1841 \times 10^{-4} t$ , ±0.081%
PS (HCFC-142b): 131 days <sup>b</sup>	$k = 0.02567 + 1.1834 \times 10^{-4} t$ , $\pm 0.085$ %
PS (HCFC-142b): 227 days <sup>b</sup>	$k = 0.02602 + 1.2181 \times 10^{-4} t$ , $\pm 0.012$ %
PIR (CFC-11): 398 days <sup>b</sup>	$k = 0.01794 + 9.9032 \times 10^{-5} t, \pm 0.29$ %

Average deviation

Transient tests were conducted at these times.



FIG. 2 -- Thermal conductivity as a function of temperature and time after manufacture for extruded polystyrene (HCFC-142b) and polyisocyanurate (CFC-11).

#### TRANSIENT DATA ANALYSIS RESULTS

This section presents the results of analyzing the transient temperature response of the thin heater. Table 5 lists the results obtained by analyzing the transient data for the GOW, G15W, and G3OW specimens using Method 1.1. Each specimen was subjected to positive and negative step changes in heat flux. The slope was determined from data that spanned the first 1.8 ks of each transient.

The five C values for GOW increase with temperature and are described to 1% by

$$C(J/kg \cdot K) = 159.5 + 3.0070 T (K)$$
 (9)

and are about 2.5% below the value given in Table 1. Three of the five C values for G15W are very large and indicate that a phase change was occurring during the transient. The other two C values average 1285 J/kg·K, which is 2% above the value in Table 1. Four of the eight values for G30W are large and are indicative of a phase change. The other four C values average 1527 J/kg·K, which is 4% above the value in Table 1. These four C values apply for gypsum containing liquid wax between 28 and 38°C. The large apparent specific heat values were derived using the short-time solution. This equation is for determining the specific heat only, and does not apply for cases where a phase change is occurring. An analysis that includes the latent heat of fusion is needed to treat these cases.

Table 6 contains  $\alpha$  values from 20 to 40°C that were calculated from the experimental values of  $\rho$ , k, and C for the three specimens. The  $\alpha$  values for the gypsum without wax decrease with increasing temperature due to C increasing with temperature. The  $\alpha$  values for the composites decrease with increasing wax content and the  $\alpha$  (solid wax) values are 3 to 5% larger than the  $\alpha$  (liquid wax) values. The experimental  $\alpha$  values (Table 6) are 15 to 25% larger than the  $\alpha$  values estimated from the literature (Table 1).

Table 5 lists the results obtained by analyzing the 14 transient tests for the CS-1, CS-2, and CS-3 specimens using Method 2. Each specimen was subjected to positive and negative step changes in heat flux.

The slope was determined using the data that spanned the range 200 to 1800 seconds of each transient. The heat flux correction for the screen thermal mass was less than 2% for all 14 cases. The specific heat (C) was calculated using the k-values calculated from the equations given in Table 2.

The 14 values of C for the calcium silicate test specimens are plotted in Fig. 3 and increase with temperature. Each of the transient tests with negative step changes in heat flux yielded low C values. We do not understand why these are low, but if these are excluded, the remaining 11 values are described with an average deviation of 1.0% by

$$C = 951.5 + 2.556 t$$
 (10)
Equation 10 yields a C (24°C) of 1012.6 J/kg·K, which is 1.3% above the estimated value for C given in Table 1. Table 7 contains thermal diffusivity values calculated from the experimental values of  $\rho$  and C and the k-values calculated from the equations in Table 2. From 20 to 50°C the increase in C of 7.6% is over twice the average increase in k, so the average decrease of  $\alpha$  is 5.5%.

Material (TMR)	Temperature (°C)	C-M3 (J/kg K)	C-Ml or M2 (J/kg K)
GOW (0.0075)	23.3	1089	1048
	29.3	1121	1056
	34.8	1136	1078
	31.0	1115	1092
	23.6	1111	1055
G15W (0.0055)	22.8	3021	2461
	23.9	2923	2098
	29.0	1428	1425
	36.1	1301	1296
	35.2	1295	1274
G30W (0.0038)	23.9	7155	4116
	30.1	9336	5891
	29.2	1549	1534
	24.5	9270	6012
	25.0	4507	1942
	28.6	1503	1528
	34.3	1502	1520
	33.6	1491	1526
CS-1 (0.0231)	25.8	1100	1003
	28.3	1065	1005
	33.1	1050	1026
	41.0	1101	1050
	50.8	1123	1069
	43.9	1061	1021
CS-2 (0.0164)	26.8	1008	1027
	36.2	1074	1052
	50.5	1120	1078
	44.3	1024	1008
CS-3 (0.0121)	26.6	1041	1028
	34.0	1070	1052
	45.5	1094	1075
	40.4	1004	1000

TABLE 5 -- Specific heat values obtained by Methods 1 or 2, and 3

WSF (0.00777)	23.7	1672	1656
	28.8	1704	1668
	39.5	1748	1712
	37.3	1720	1723
DFP (0.0105)	23.3	1438	1400
	31.8	1487	1411
	44.2	1533	1421
	35.8	1492	1412
SYPF (0.0065)	23.4	1544	1576
	27.6	1596	1642
	36.3	1600	1676
	34.9	1601	1669
		•	
PIR+CFC-11	26.7	1218	1530
(0.237)	40.0	1327	1640
	43.6	1300	1680^
			*
	27.2	1359	1250 <u>°</u>
PS@45 d (0.132)	35.2	1397	1309
	49.7	1552	1441
	46.5	1485	1381*
	26.9	1275	1157.
PS@127 d (0.144)	34.4	1323	1226
	49.0	1349	1239
	46.8	1364	1283*
	27.1	1308	1168.
PS@227 d (0.145)	34.2	1270	1165
	48.7	1350	1245*
	46.6	1353	1266*

\*C-M2. \*\* Based on slope for T equal 0.3 to 0.4.

	GOW	<u>G15W</u>	<u>G30W</u>
Percent Wax Density kg/m <sup>3</sup>	0 695 1	14.1 816 9	30 1000 0
C, J/kg·K	a 0 1780	1285	1527
		-	0.2325
	$\alpha$ , m <sup>2</sup> /s x	107	
t, °C 20	2.460	(1.919) <sup>c</sup>	(1.565)
25	2.425	1.919	1.565
30	2.391	1.828	1.523
40	2.264	(1.828)	(1.523)
•			

TABLE 6 -- Calculated thermal diffusivity for three gypsum-wax

<sup>a</sup>Eq. 8 <sup>b</sup>k of liquid-wax gypsum composite <sup>c</sup>Extrapolated.



FIG. 3 -- Specific heat as a function of temperature for three calcium silicate insulations.

		(a x 10	', m²/s)	
		CS-1	CS - 2	CS - 3
Density, kg,	/m <sup>3</sup>	307.4	443.95	605.1
t, °C	20	2.145	1.712	1.441
	30 50	2.107 2.031	1.629 1.596	1.419 1.377

TABLE 7 -- Calculated thermal diffusivity for three calcium silicate specimens

The  $\alpha$  values decrease with increasing density. The  $\alpha$  values are 19 to 28% lower than the estimated  $\alpha$  values given in Table 1.

Table 5 contains the results obtained by analyzing the 12 transient tests conducted on the wood products using Method 2. Each specimen was subjected to three positive and one negative step change in heat flux. The slope was determined using data that spanned the range 200 to 1800 seconds for each transient. The heat flux correction for the screen thermal mass was -1.03% for the WSF specimens, -1.09% for the DFP specimens, and -0.85% for the SYPF specimens. The specific heat was calculated using the k-values calculated from the equations in Table 3.

The C values for each wood specimen are plotted in Fig. 4, and span the range 1400 J/kg·K to 1725 J/kg·K. Each wood product has a different temperature dependence, with C increasing from DFP to SYPF to WSF.

The WSF specimen, yielded a C (24°C) of 1658 J/kg·K, which is 1.9% above the estimated C given in Table 1. The SYPF has a lower moisture content than the WSF specimen and has a lower C value. The SYPF, has a C (24°C) of 1604 J/kg·K, which is 3.6% above the estimated C given in Table 1. The DFP specimen showed the lowest C values and a C (24°C) of 1402 J/kg·K, which is 9.8% lower than the estimated C given in Table 1. Table 8 contains  $\alpha$  values for the three wood product test specimens. The  $\alpha$  values for the WSF and SYPF are about 20% lower and the DFP is 2% lower than the estimated  $\alpha$  in Table 1.

Table 5 contains the results obtained by analyzing three transient tests conducted on the polyisocyanurate (CFC-11) and 12 transient tests conducted on the extruded polystyrene (HCFC-142b). This analysis used Method 2. Each specimen was subjected to positive and negative step changes in heat flux. The heat flux corrections for the heater thermal mass were large. The slope was determined using data that spanned the range 200 to 1800 seconds for the polyisocyanurate and 100 to 900 seconds for the extruded polystyrene.

The C values obtained from each test of the cellular plastic foams is plotted in Fig. 5 and span the range  $1140 \text{ J/kg} \cdot \text{K}$  to  $1680 \text{ J/kg} \cdot \text{K}$ . The C values for the polyisocyanurate are highest and yield a C (24) of 1496 J/kg \cdot K, and this is 1.9% above the estimated C given in Table 1. The C values for the extruded polystyrene specimen decrease as aging time increases.



FIG. 4 -- Specific heat as a function of temperature for three wood products. Filled symbols are estimated values from Table 1.

TABLE 8 -- Calculated thermal diffusivity for three wood product specimens  $\alpha \ge 10^7$ , m<sup>2</sup>/s

		WSF	SYPF	DFP
 Density, kg,	/m <sup>3</sup>	451.5	575.2	500.9
t°, C	20	1.3773	1.3876	1.4430
	24	1.3735	1.3756	1.4485
	30	1.3676	1.3578	1.4567
-	50	1.3482	1.2982	1.4841

The PS (227 d) has a C (24) of  $1147 \text{ J/kg} \cdot \text{K}$ , which is 0.25% less than the estimate of C given in Table 1. It is noteworthy that C (45 days) is greater than C (227 days) and this may be due to inward diffusion of air and outward diffusion of HCFC-142b, with the higher C being associated with the HCFC blowing agent.

Table 9 contains thermal diffusivity values calculated from the experimental  $\rho$  and C values and the k-values calculated from the Equations in Table 5.

The experimental  $\alpha$  values for PIR (CFC-11) are within the limits defined by the estimated values in Table 1. The  $\alpha$  values for the PS (HCFC-142b) started out greater than the initial  $\alpha$  estimated in Table 1 and exceed the  $\alpha$  (beyond 100 days) value by 2.7%.



FIG. 5 -- Specific heat as a function of temperature for cellular plastic foams. Filled symbols are estimated values from Table 1.

	PIR (CFC-11)		PS (HCFC-142b)	
Density, kg/m <sup>3</sup>	30.2	30.0	30.0	30.0
Age, days	398	45	131	227
t, °C 20	4.516	6.891	8.181	8.415
24	4.495	6.888	8.170	8.421
30	4.463	6.875	8.206	8.447
50	4.356	6.833	8.311	8.529

TABLE 9 -- Calculated thermal diffusivity for cellular plastic foams  $\alpha~x~10^7~m^2/sec$ 

Because of the high thermal mass ratios for these materials, Method 2 was used to find their specific heats. Method 2 has no fundamental basis and large differences in C resulted, i.e. PIR (CFC-11), -21% or -335 J/kg K; PS (45 d), +7.7% or 103 J/kg K; PS (131 d), +8.3% or 102 J/kg K; and PS (227 d), +9.1% or 109 J/kg K. The thermal mass ratios for these materials were approximately 0.14 for the PS and 0.24 for the PIR. If the C values for the plastic foams from Method 3 were used to compute the  $\alpha$  values, then the  $\alpha$  (PIR) would increase by about 25% and the  $\alpha$  (PS) would decrease by about 10%. Table 10 summarizes the materials with the highest TMR studied.

Material	TMR	Methods	Delta C $^{\star}$	<pre>% Difference</pre>
GOW	0.0075	3 & 1	49	4.6
G15W	0.0055, ***	3&1	13	$1.0^{**}$
G30W	0.0038**	3 & 1	-16**	-1.0**
CS-I	0.0231	3&1	54	5.3
CS-II	0.0164	3&1	15	1.4
CS-III	0.0121	3&1	14	1.3
WSF	0.0077	3&1	21	1.3
DFP	0.0105	3 & 1	77	5.4
SYP	0.0065	3&1	- 56	-3.4
PIR-CFC	0.237	3&2	-335	-20.7
PS@45	0.132	3 & 2	103	7.7
PS@130	0.144	3 & 2	102	8.3
PS@230	0.145	3 & 2	109	9.1

TABLE 10 -- Summary of differences between calculation methods

"C for M3 minus C for M1 or M2.

Gypsum plus liquid wax only.

#### CONCLUSIONS

(1) Steady-state and transient modes of operation of the UTHA yielded thermophysical properties (k, C, and  $\alpha$ ) of four types of building materials (gypsum containing wax, calcium silicate insulations, wood products, and cellular plastic foams) from 24 to 50°C. (2) A procedure was developed to operate the UTHA in a transient mode by applying a step change in heat flux to specimens that were initially isothermal or that had a steady imposed temperature gradient. The procedure used a computer to control the test and to record the temperature-time behavior of the thin-heater. (3) Experimental values for the product  $k\rho C$  were determined from the temperature change of the thin-heater that is a linear function of the square root of time after the step change in heat flux. The error in the product kpC was less than 1% for building materials with  $(k\rho C)^{-\frac{1}{2}}$ values below 10  $m^2 K/W s$  (densities above 300 kg/m<sup>3</sup>, and low values of TMR). However, for the cellular plastic foams with  $(k\rho C)^{-\frac{1}{2}}$  values above 30 m<sup>2</sup>K/W  $\sqrt{s}$  (densities near 30 kg/m<sup>3</sup> and large TMR values), alternative analyses were required due to the large relative thermal mass of the heater and larger differences were found for  $k\rho C$ . (4) For low TMR materials (gypsum, calcium silicate, and two wood specimens) the measured  $\alpha$ -values were as much as 30% below estimated  $\alpha$ -values. Applying this test method would improve the data base for building materials.

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**Performance Factors II** 

William A. Lotz

# UNDERGROUND CELLULAR GLASS PIPE INSULATION FAILURES IN QUEBEC AND LOUISIANA

**REFERENCE:** Lotz, W. A., "Underground Cellular Glass Pipe Insulation Failures in Quebec and Louisiana," <u>Insulation</u> <u>Materials: Testing and Applications. 2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

**ABSTRACT:** The author has participated as an expert witness on two major failures of the subject insulation on steam/condensate piping. One installation was in Quebec and one installation was in central Louisiana. The two systems were very different in design and installation. In both cases the water table was occasionally above the pipes. The soil in Louisiana was clay. There were no expansion nor anchoring provisions. The waterproofing consisted of multi-layers of cutback asphalt mastic and glass mesh reinforcing. The solution was to relocate the piping above ground. The soil in Quebec was clay except that the piping was in sand fill or a sand trench. There were many drainage ditches on the site - some of which were well below the level of the pipes. Dewatering pumps were installed in the trench in an attempt to save the insulation. The waterproofing system was designed and installed in strict accordance with the manufacturer's directions and under the supervision of the manufacturer's technical service representative.

**KEYWORDS:** cellular glass, underground pipe insulation, steam pipe insulation, failed insulation.

The failure of insulation on insulated steam underground pipes has been a common occurance since the first high pressure steam pipe was buried. In the 1960's the Building Research Advisory Board did an extensive field study of failed underground insulated steam and high temp. hot water pipes at US military bases in all types of climates and soils. The results of this study was several reports, underground hot pipe system criteria and a boiling test for pipe insulation. Today, many underground insulation specifiers and buyers require certification that the insulation passes the boiling test.

William A. Lotz, P.E. is a Consulting Engineer, Acton Ridge Road, West Newfield, Maine 04095 The following is the story of two applications where the insulation did not survive boiling "in-situ".

#### QUEBEC PROJECT

In 1989 in a small town paper mill complex in Quebec, several miles of steam and condensate pipes were buried. The system consisted 9 manholes and 15 expansion loops. The pipe sizes ran from 2 1/2" to 12" schedule 40 steel pipe (6 - 30 cm) The pipe insulation was single layer cellular glass, 3" thick (7 1/2 cm) on the steam pipes and 2" (5 cm) thick on the condensate pipes. The steam temperature averaged 400 F  $\pm$  35 F (240 C  $\pm$  22 C) and the condensate was 260 - 285 F (127 - 140 C). The pipes were buried from 1 ft. (.3 m) to 13 ft. (4 m) deep. The local soil is mostly clay. However, the piping trench was backfilled with sand, gravel, and in some areas, rounded stones (under the pipes). The property bordered the Nicolet River.

The cellular glass was waterproofed with an 1/8" (3 mm) thick proprietary jacket consisting of asphalt, glass fabric, aluminum foil, and plastic film. This jacket was supplied by the cellular glass manufacturer. The jacket comes with separate 4" wide (10 cm) strips to seal the butt joints. All of the jacket joints were heat sealed using a propane torch.

The insulation manufacturer provided installation details for the trench, anchors, guides, wall penetration, expansion loops, and supports which were reasonably carefully followed. The manufacturer also supplied an technical expert to review the engineers design (actually an exact copy of the manufacturer's recommendations) and the contractor's installation. The only difference between the manufacturer's recommended specifications and the project specifications was that the project engineering was in French.

A local consulting engineering firm translated the U.S. manufacturer's design details and specifications into French. The engineer's drawings included a plan view of the site plus elevations of the pipe vs. grade. All of the design details were reproductions of the insulation manufacturer's recommendations.

In November 1989, just prior to the start up of the new steam system, a paper mill engineer noticed water pouring into a manhole from the space between the pipe and the insulation. Excavation revealed a high water table (above the pipe). The plant engineer quickly had a trench drainage system installed consisting of a flexible perforated pipe below the insulated pipes connected to a system of 5 "sump pumps". The manufacturer's technical expert went to the site and agreed that there were problems. Assuming the cause was contractor's workmanship, the expert and the contractor each sealed a section of waterproofing lap in the shop. When the two test section laps were cut open, they were equally full of voids.

It soon became obvious that the waterproofing on the cellular glass had failed and the insulation was wet.

In June 1990, we opened the trench in 6 locations along the 5500 foot (1650 meters) trench. We made the following observations:

#### <u>Hole #1</u>

Soil - sandy, lumps of clay, dry, depth to pipe 10 ft. (3.3 m) The insulation was dry and looked "normal".

#### <u>Hole #2</u>

Soil - sandy, crushed rock, wet, "steamy", depth to pipe 3 1/2 ft. (1 meter). Some of the cellular glass insulation had fallen off the pipe. In one location, the crushed rock had cut into the water-proofing membrane. The membrane on the bottom side of the insulation was blistered and sagging away from the cellular glass.

#### <u>Hole #3</u>

Soil - sandy, damp, depth to pipe 6 ft. (1.8 meters). The cellular glass insulation cells were saturated with water. The insulation had many cracks and fractures. The pipe was rusty. The insulation had crushed 1/2" (12 mm) on the top of the pipe and hence was sagging 1/2" (12 mm) below the pipe. This condition existed on both the steam and condensate pipes. The waterproof membrane on the steam pipe insulation was sagging below the cellular glass.

# <u>Hole #4</u>

Soil - sandy, much water, depth to pipe 5 ft. (1.5 m). When the backhoe finished excavating this hole, there was  $\pm 1000$  gallons (28,000 liters) of 130 F (54 C) hot water in the hole. The water table stabilized at 2 ft. (.6 m) below the pipes. Steam came out of a hole (resulting from the backhoe) in the waterproof membrane on the cellular glass. The insulation was crushed on top and sagging. The cellular glass cells were very wet.

#### <u>Hole #5</u>

Soil - sandy, warm and damp, depth to pipe 7 ft. (2.1 m). The cellular glass cells were saturated with water in the exterior half of the pipe insulation.

#### <u>Hole #6</u>

Soil - sand in trench, clay in native ground, water table 2 ft. (.6 m) under pipe depth to pipe 2 ft. (.6 m). The cells of the glass insulation were very wet. The insulation was sagging on the pipe and one butt joint had opened 3/4" (19 mm) on the steam pipe.

In August 1990, two more excavations were made with the following observations:

#### <u>Hole #1A</u>

Soil - sandy, crushed rock below sand with drainage pipe, depth to 6 ft. (1.8 m). The cellular glass was dry but cracked into many pieces.

# <u>Hole #2A</u>

Soil - sandy, dry, depth to pipe 3 ft. (.9 m). The cellular glass was severely cracked and saturated.

The manufacturer suggested that the insulation system be excavated and repaired with the same materials and procedures. I disagreed. Under ideal conditions, in the shop, neither the most experienced contractor's insulator <u>nor</u> the manufacturer's technical expert were able to seal a joint in the waterproofing membrane over the cellular glass pipe insulation without voids and fishmouths (ie. potential leak holes). If they couldn't get a good seal under ideal conditions then working in the mud in the trench would be even less likely to obtain a good seal. I have recommended the entire underground piping system be abandoned and replaced by a new prefabricated underground insulated conduit system that is pressure testable and dryable.

At this site, the native soil is clay and the natural water table is near the surface. When the paper mill dug the piping trenches and backfilled them with sand, they created a wonderful trough for the water to collect in. This system of insulation works well in dry soil. When you combine high pressure steam, cellular glass pipe insulation with an asphaltic membrane and a high water table, you are courting failure. It has been my experience (dating back to the early 1950's) with underground cellular glass pipe insulation that it is essentially impossible to obtain a leak free asphaltic coating in field applications.

#### LOUISIANA PROJECT

This project was located in central Louisiana in clay soil not far from the Red River. The system consisted of 300 F (149 C) steam, condensate, water and chilled water pipes buried 3 - 6 ft. (1-2 m)below grade. The water table was almost always above the pipes. The steam and condensate pipes were insulated with 1 1/2" thick (38 mm) cellular glass and the chilled water pipes were insulated with 1" thick (25 mm) urethane pipe insulation. Both insulations had the same waterproofing - 3 hand applied layers of cutback asphalt mastic reinforced with 2 layers of glass mesh. The insulation was secured to the pipe with stainless wires on 6"centers (150 mm). The consulting engineer and the insulation contractor both expected each other to design the "details" - anchors, expansion loops, etc. for the insulation. The net result is that there was no provision for expansion between the pipe and the insulation.

Within weeks of the start up of the school complex, the chilled water temperature at the far end of the underground system was 140 F (60 C) rather than the design 45 F (7 C). Ground temperatures just below grade (above the buried steam pipes) was 188 F (87 C). The city water was near the boiling point.

When we opened up the trench to inspect the insulation, we observed:

1. In some locations, the cellular glass and urethane were intact but very wet.

2. In other locations, the cellular glass and urethane pipe insulation were pulverized and the only items recognizable were the stainless wires that had held the insulation in place. When the groundwater penetrated the waterproofing membrane, it flashed into steam and reduced the insulation and membrane into small particles which intermingled with the clay mud. Because of the close proximity of all of the pipes in the trench, the urethane on the chilled water pipes was also destroyed.

3. There was one hole that we excavated that was in dry soil and all of the insulation looked in good condition.

The underground pipes on this project were abandoned and all of the pipes were routed above ground.

In summary, in ASTM STP 922, Govan and Demetroulis stated in "Design Criteria for Underground Insulated Piping Systems" the following:

"In two of these field investigations, closed-cell cellular glass (Foamglas) was inspected. Foamglas, because of its frangibility and susceptibility to destruction where potential boiling action is present, is limited in its use by the Tri-Service group to applications below 122 C (250 F)." Tri-Service refers to U.S. Government agencies.

I agree with this conclusion, however, I suggest one step further. It is my recommendation that cellular glass not be used to insulate underground pipes above 250 F and not be used at all in a damp or wet soil condition. Again, this recommendation comes from my experience (that spans 40 years) with many, many applications of cellular glass underground. Having insulated many underground pipes myself, I am especially aware of the difficulty attaining a perfect water seal on cellular glass while working in a muddy trench. If the seal is not perfect, it will leak in a wet soil and if the pipe temperature is above 250 F it will destory the insulation. Cellular glass cannot withstand the mechancial and chemical action of steam under pressure.



PHOTO #1 - Steam pipe in Canada. Note water stains on cellular glass.



PHOTO #2 - Cellular glass sags away from pipe due to crushing of the insulation on the top of the pipe.



PHOTO #3 - Louisiana saturated cellular glass.



PHOTO #4 - Cellular glass on Louisiana steam pipe has "disappeared" into the mud.

# E. Eugene Stansbury

EFFECTS OF AERATION ON CORROSIVENESS OF WET RESIDENTIAL BUILDING THERMAL INSULATION ON LOW CARBON STEEL

REFERENCE: Stansbury, E. E., "Effects of Aeration on Corrosiveness of Wet Residential Building Thermal Insulation on Low Carbon Steel," <u>Insulation Materials</u>: <u>Testing and Applications</u>, <u>2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: The corrosion behavior of low carbon steel in wet residential building thermal insulation and filtered insulation leachates were investigated with variable access of air to the corroding interface. Aqueous environments were characterized in terms of solubility of leachable species, pH and specific resistivity. Several electrochemical methods were used to evaluate the general corrosion behavior and to estimate the corrosion rate. Direct corrosion rates were determined by weight-loss and comparisons made to rates in wet cotton using distilled water. Although the pH of the environments varied from 4 to 9.5, a major variable is access of oxygen to the metal interface. This was confirmed by studying the effect of depth of embedding and by partial embedding of steel specimens in wet insulation. Limiting oxygen concentrations in leachates were established by air and nitrogen sparging which changed the corrosion rate by factors greater than ten. An additional variable is the effect of composition of the leachate on the adherence and porosity of the corrosion product deposit. Electrochemical polarization measurements provided information on the corrosion mechanisms.

KEYWORDS: wet insulation, steel corrosion, aeration, deaeration, corrosion mechanism

# INTRODUCTION

Residential building thermal insulations are broadly classed as glass fiber, cellulosic, mineral fiber or foam types. Cellulosic insulations consist of disintegrated paper products, predominantly

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newspapers, to which selected additives have been made to meet performance standards and the foam insulations are organic-based products. These insulations contain materials that are leachable by water resulting in wet insulations that can cause damaging corrosion to metals used in building construction. These include steel, copper and aluminum pipes, wiring, boxes and fasteners. The subject was covered at the Cellulose I Conference [1] in 1978 and reviewed in an Assessment of the Corrosiveness of Cellulosic Insulation Materials by Weil and Graviano in 1979 [2]. The latter covered specifications, test methods, data and properties of cellulosic insulation related to corrosiveness. ASTM Standards contain sections for evaluating the corrosiveness of wet insulations in contact with steel, copper and aluminum. ASTM C665 [3] and C764 [4] pertain to mineral-fiber insulation and ASTM C739 [5] to cellulosic-fiber insulation. These Standards specify exposure of thin sheets of steel, copper and aluminum to the insulation under controlled humidity, temperature and time conditions. Severity of attack is based on surface appearance under C665 and by penetration of the sheet under C739. C951 [6], a now discontinued standard on urea formaldehyde-based insulation, required reporting a pitting factor based on visual inspection and a corrosion rate based on weight-loss.

The procedures under the above standards are time consuming and qualitative, and the results are frequently difficult to reproduce. Also there is sufficient difference in the procedures to question their value for comparison of the corrosiveness of the three types of insulations. These uncertainties led Weil, Sheppard and co-workers [7,8] to initiate development of electrochemical methods for evaluating the corrosiveness of insulations which could be conducted in a short time and would be applicable to a wide range of insulations and metallic materials that they may contact.

Several electrochemical techniques were investigated in detail by Stansbury [9,10] who included studies of the mechanisms of corrosion of low carbon steel in wet insulations and in filtered insulation leachates. Corrosion rates calculated from electrochemical measurements were assessed by comparison to corrosion rates determined directly by weight-loss. Poor correlation was obtained between the calculated and directly measured rates. Reasons included: wide variation in the nature of corrosion products formed in the different wet insulations; the accompanying wide variation in the access of dissolved oxygen to the metal interface; and the influence of these factors on the time required for establishing steady state corrosion which precluded a viable short time test. A major conclusion of the investigation was that the corrosion product deposits, and the location of the metal in the wet insulation, lead to control of diffusion of oxygen to the corroding metal interface as the corrosion rate determining variable. Further, under these conditions, the models and calculations used in converting electrochemical measurements to corrosion rates are questionable.

The present paper covers a number of investigations conducted to show that dissolved oxygen is generally the corrosion rate determining variable of metals, low carbon steel in particular, in wet insulations. The evidence is presented in terms of weight-loss measurements, surface appearances including penetration of thin sheets and electrochemical measurements which are significant to the interpretation of corrosion mechanisms and helpful in estimating corrosion rates. There was no evidence of microbiologically induced corrosion in this investigations and mechanisms relating to this mode are not considered.

## THE WET INSULATION ENVIRONMENT

Cellulosic, glass fiber and foam insulations contain water leachable chemical species. The cellulosic insulations contain 10-25% additives primarily to meet fire resistance standards; these include ammonium sulfate, borax, boric acid, calcium sulfate, aluminum sulfate and several natural mineral products of complex composition. Since these materials vary widely in solubility in water, extending from <0.1 to >75 grams per 100 grams of water, leachates within or filtered from cellulosic insulations are aqueous solutions with concentrations of electrolytes extending from very dilute and unsaturated to highly concentrated and saturated. The amount of leachable material available from fiber glass and foam insulations is much less and hence the leachates are generally more dilute in soluble species. In this

_	Leachate	рН	Specific resistivity, ohm-cm
13	sources of cellulosic insulation	4.4-8.2	49-255
3	sources of glass fiber insulation	9.1-9.6	896-2300
4	sources of foam insulation	5.6-7.5	5370-13160

TABLE 1 -- Properties of leachates

investigation, leachates were prepared by contacting 100 grams of insulation with 1000 mL of distilled water, holding stagnant for three days at  $73\pm3$ °F and filtering. The leachates were characterized in terms of pH and specific resistivity, the latter being an approximate indicator of the ionic concentration. Representative ranges of values are listed in Table 1. It should be noted that for a cellulosic insulation containing 20% additive, preparation of a filtered leachate at the ratio of 100 grams of insulation per 1000 mL water yields only a 2% solution. Considering that the solubility of ammonium sulfate is about 75 grams per 100 grams of water, wet insulation in contact with a steel surface will result in a local leachate more concentrated than that which is obtained by filtering. The insulation-to-water ratio used however in preparing the filtered solutions was necessary to provide enough leachate for investigating its properties and for conducting electrochemical and weight-loss measurements. As a consequence, local conditions at the metal surface may have a different affect on the corrosion process than the more dilute filtered leachate.

In support of understanding the corrosion processes occurring in wet insulations, measurements were conducted in saturated and 5% saturated solutions of several additives that may be used in the manufacture of cellulosic insulations. These included reagent and commercial grades of borax, ammonium sulfate, aluminum sulfate,

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calcium sulfate and boric acid. The pH of these solutions spanned a greater range than observed in wet insulations. Aluminum sulfate hydrolizes to give solutions of pH = 2.5 to 3; the borax solutions have pH >9.5. Brief reference is made to the corrosiveness of these solutions to low carbon steel in the present paper.

# OVERVIEW COMMENTS ON RELEVANT CORROSION MECHANISMS

Over the pH range of the wet insulation leachates, the dominant cause of corrosion of steel is dissolved oxygen. Only near the lower limit of pH≈4 does the acidity (H<sup>+</sup> ions) become of equal significance. Because of the low solubility of oxygen in water (~8 ppm) and even lower solubility in the presence of dissolved chemical species, the rate of corrosion is controlled by the rate of diffusion of oxygen to the metal interface. Corrosion product layers or presence of other diffusion limiting conditions such as wet insulation, can restrict diffusion of oxygen and hence further affect the corrosion rate. Over the pH range of the wet insulations, corrosion products do form on steel with a range of adherence and porosity, and hence have a range of influences on the diffusion of oxygen to the metal interface.

The corrosion process is shown schematically in Figure 1 under conditions of formation of a corrosion product deposit on iron. The



Figure 1. -- Uniform corrosion with solid corrosion product deposit.

corrosion is supported by dissolved oxygen and hydrogen ions. Iron atoms pass into solution as ions according to the anodic reaction  $Fe \rightarrow Fe^{m^+} + me$  resulting in corrosion. The released electrons are consumed at other positions on the surface by the cathodic reactions,  $2H^+ + 2e \rightarrow H_2$  and  $0_2 + 2H_20 + 4e \rightarrow 40H^-$ . Since these reactions involve transport of electrons, their rates can be expressed as electrical currents. For a freely corroding metal these rates must be equal, i.e., electrons released by the corrosion reaction must be consumed by the cathodic reactions. Both of the cathodic reactions increase the pH at the surface, directly by release of  $OH^-$  ions from the oxygen reaction and by loss of  $H^+$  ions from the hydrogen reaction. In the pH range of the wet insulation environments, hydroxides, hydrated oxides and oxides of iron form. Compounds related to the black oxide, Fe<sub>3</sub>O<sub>4</sub>, form at the metal interface but are converted by the oxygen to the red oxide, Fe<sub>2</sub>O<sub>3</sub> as an outer layer. It is evident that the corrosion rate can be controlled by the diffusion of dissolved oxygen to or through the corrosion product layer if diffusion is the slow step in the overall process.

When the steel surface is embedded in wet insulation, three additional factors influence the corrosion rate. First, the necessity for diffusion of oxygen through the wet insulation. Second, since corrosion products are restricted from diffusing from the immediate vicinity of the surface, the pH increases more than would occur in the absence of the insulation. Both of these factors tend to decrease the



Figure 2 -- Non-uniform corrosion at metal-insulation-air interface.

corrosion rate. A third variable is the distance of any local metal surface from a wet insulation/air interface as a source of dissolved oxygen. The worst-case situation exists when the metal surface extends from the wet insulation allowing short diffusion paths for oxygen at the metal-air-wet insulation interface. This condition is illustrated schematically in Figure 2 in which the arrows represent the relative in-flux of air as the source of dissolved oxygen. Leachate can extend onto the metal surface at the interface allowing transport of oxygen through a thin film of liquid which accelerates the corrosion as shown by the penetration profile in Figure 2. In contrast, deeply embedded surfaces may show very low corrosion rates due to restricted access of dissolved oxygen.

In an extensive investigation of the corrosiveness of residential building thermal insulations [10], electrochemical measurements were made for low carbon steel in wet insulation and insulation leachates. The electrochemical processes occurring on the surface of a metal produce an electrical potential which can be measured relative to a standard reference such as the calomel electrode. The corroding surface exhibits a mixed potential which depends on the potentials of the individual anodic and cathodic reactions and the areas over which

these reactions occur. A potentiostat can be used to shift this mixed potential which is accompanied by current flowing to or from the corroding surface. Current flowing from the surface is associated with an increase in the anodic corrosion rate and decrease in the cathodic rate and conversely for current flowing to the surface. Thus the external current progressively becomes a measure of the individual reactions. By an analysis of the potential-current density curves near the open circuit mixed potential, information is obtained on the processes responsible for the corrosion and, within limits, an estimate can be made of the corrosion rate. A limiting factor however in estimating corrosion rates is the conception of realistic models for the interface corrosion process allowing calculations of rates in reasonable agreement with those determined by weight-loss. This correlation is particularly complicated when the corrosion is associated with formation of corrosion products and overlying deposits which lead to control of access of oxygen to the metal surface by diffusion. These are the conditions that exist at the metal surface in wet insulation and limit quantitative evaluation of corrosion rates by electrochemical measurements. However qualitatively useful information can be obtained.

Although the present report is concerned with the effect of environmental variables in wet insulation on corrosion rates of low carbon steel, reference will be made to potentiodynamic polarization measurements which provided insight into the corrosion mechanisms. Specifically, in the present research, significant information was obtained on the effects of pH, aeration and tendencies for uniform versus localized (pitting, crevice) corrosion. Reference can be made to Figure 3 for a schematic representation of variations in the shapes that polarization curves exhibit and for some of the terminology used in their description. References [9] and [10] contain additional discussion of both experimental procedure for and interpretation of polarization measurements.

Figure 3a is representative of the anodic polarization curve for a metal and cathodic polarization curves for hydrogen ion reduction and dissolved oxygen reduction. The current density at any potential along the anodic curve is directly proportional to the corrosion rate of the metal when existing at the particular potential. The intersection of the metal and hydrogen reduction curves define the conditions  $(E_1=E_{\rm corr}$  and  $I_1=I_{\rm corr}$  ) which are spontaneously established at steady state corrosion in a completely deaerated environment. The intersection  $(E_2, I_2)$  of the metal and oxygen reduction curves define the corrosion conditions for a highly aerated environment. In the latter case, the intersection is in the passive potential range of the anodic curve where a protective film results in low corrosion rates. The positions of these cathodic curves depend on the pH, dissolved oxygen content and diffusion rates of species to and from the corroding surface. At higher pH, the hydrogen curve is depressed giving smaller  $E_1, I_1$  and hence a smaller corrosion rate. As the amount of dissolved oxygen decreases, the cathodic oxygen curve shifts progressively as represented by the dashed curve. At the position shown, the intersection with the anodic curve is now at a higher current density than I. and this amount of aeration has increased the corrosion rate. Thus the availability of dissolved oxygen governs the position of the oxygen polarization curve and may have a dominant influence on the corrosion rate. Also, environmental factors such as chlorides can shift the





(c)

(d)

Figure 3. -- Schematic representation of individual and experimental polarization curves.

- (a) Individual polarization curves for anodic metal dissolution (corrosion) and cathodic curves for oxygen and hydrogen ion reduction.
- (b) Solid curve represents experimentally determined curve when intersection occurs in passive potential range. Dashed sections would not be revealed experimentally. Aerated environment.
- (c) Solid curve represents experimentally determined curve when intersection occurs in active potential range. Dashed sections not revealed experimentally. Deaerated environment.
- (d) Solid curve represents experimental curve when pitting is initiated at  $E_B$  on up-scan and stops at  $E_P$  on down-scan. Dashed curve is followed when pitting does not occur.

anodic metal curve to higher current densities resulting in intersections corresponding to higher corrosion rates. In fact, the insulation leachate environments generally result in curves showing no or very little tendency to produce a passive section of the polarization curve. These intersection potentials,  $E_{corr}$ , can be measured experimen-

tally as the potential between the corroding metal and a reference electrode.

If a potentiostat is used to establish a range of potentials and the associated current between the specimen and auxiliary electrode measured, the solid curve in Figure 3b is obtained for the aerated environment and the solid curve in Figure 3c in the deaerated environment. These solid curves, which would result from experimental measurement, are the difference in the current density of the anodic and cathodic curves at each potential. At  $E_{corr}$ , there is no difference in the values of the current density of the two curves and the solid curve moves to and rapidly returns from small values of current thus identifying the corrosion potential. Extrapolation (shown as dashed lines) of the anodic and cathodic branches of the polarization curve until they intersect at the potential,  $E_{corr}$ , the potentiostat forces the cathodic reaction to dominate; above  $E_{corr}$  the anodic reaction dominates. The dashed sections of these curves show only the positions of the polarization curves for the individual reactions; these sections are not determined in a single potential scan.

Figure 3d is representative of an experimentally determined polarization curve for a metal/environment system susceptible to localized corrosion, either pitting or crevice corrosion. On the upscan, pitting is initiated at  $E_{\rm g}$  and the initiation and propagation of local corrosion results in a large increase in current density. If the potential is reversed, a loop forms and the localized corrosion finally stops propagating at the potential  $E_p$ . If the metal/environment system is not conducive to breakdown to localized corrosion, the measured polarization curve continues to increase as shown by the dashed extension and, on potential reversal, the curve returns at lower values of current density than on the up-scan due to increased protection by the thickening corrosion product. The practical significance of knowing these characteristics of the potential/current density relationship is estimation of conditions conducive to pitting. If an environment causes the corrosion potential to exist above  $E_{0}$  then failure by pitting corrosion is predicted. If the corrosion potential is below  $E_{P}$ , pitting is not predicted. Figure 3-d is representative of a deaerated environment,  $E_{\rm corr}$  is <<E, and localized corrosion will not occur. If aeration (dissolved oxygen) or other strongly cathodic reactants are present to increase  $E_{\rm corr}$  into the range above  $E_{\rm p}$  localized corrosion may occur. Also, if the return curve shows a large positive loop on the return scan such that  $E_p$  is below the initial  ${\rm E}_{\rm corr},$  then localized corrosion can be expected in time on exposure to the environment. Having determined the polarization curves, measurements of polarization potentials of freely corroding systems allows prediction of eventual uniform or localized corrosion.

Although broader aspects of the present investigation of the corrosiveness of wet insulations made extensive use of potentiodynamic polarization measurements, the corrosion rates based on electrochemical measurements which are reported here were obtained using a simple direct reading instrument (e.g. the Corrater 9030 by Rohrback Cosasco Systems). The instrument utilizes a probe to which two cylindrical specimens are attached. The specimens are prepared as identically as possible so as to have the same corrosion behavior. Under internal microprocessor control, a potential difference of +20 mV and then -20 mV is applied between the specimens and the resulting current averaged. The instrument then performs what is referred to as a polarization resistance analysis which is based on an assumed model of the corrosion process. The calculated corrosion rate is displayed digitally. The accuracy of the method therefore is subject to the same limitations as described using the potentiostat. Specifically, errors are introduced when corrosion product or other deposits introduce diffusion limiting steps in the corrosion process. In the present research, the direct reading instrument, although not providing quantitative information, was useful in confirming the effects of the several variables on the corrosiveness of wet insulations evaluated by weight-loss and visual observation.

#### PROCEDURE

Potentiostatic polarization measurements were conducted on AISI 1010 low carbon steel rods 3.18 mm in dia. insulated such that a 10 mm length would be exposed to the environment. Measurements using the direct reading corrosion rate meter were made on dual specimens 5.14 mm in diameter and 29.16 mm long attached to a probe and inserted into narrow 400 mL beakers containing insulation leachate or wet insulation. Weight-loss specimens were low carbon AISI 1010 steel 50 mm x 50 mm x 0.53 or 0.076 mm. All specimens were uniformly polished with 600 grit abrasive paper, washed with acetone and immediately used. Tests were conducted at  $73\pm3^{\circ}F$ .

Most leachates were prepared by contacting 100 grams of insulation with 1000 mL of distilled water, holding for three days at  $73\pm3$ °F and filtering. Smaller ratios of water to insulation were used for direct immersion of steel specimens in wet insulation. Specimens were exposed at various depths in both leachate and wet insulation for times up to three weeks at  $73\pm3$ °F.

Partial exposures were made by placing the 50 mm x 50 mm x 0.076 mm steel specimens vertically in wet insulation to a depth of 37.5 mm in crystallizing dishes 50 mm high and 100 mm in diameter. Specimens were exposed for up to three weeks at  $75\pm3$ °F and subsequently examined for penetration along the metal-air-wet insulation interface and for the distribution in thickness using ball-pointed micrometers.

#### RESULTS AND DISCUSSION

The pH of the thirteen cellulosic insulations leachates, listed in Table 2, ranged from 4.44 to 8.20, from 9.1 to 9.62 for three fiber glass insulation leachates and from 5.58 to 7.48 for the four foam insulation leachates. The pH values of the cellulosic insulation leachates are consistent with measurements made on saturated and 5% saturated single additives that may be used in producing these products. For example, borax solutions have pH $\approx$ 9.5, aluminum sulfate hydrolyzes to pH values in the range 2.5 to 3 and ammonium sulfate hydrolyzes to the range 4 to 5. The pH range of the leachates reflect mixtures of these additives.

ENVIRONMENT <sup>a</sup>	рН	CORROSION	RATE, mpy
		Corrater	Weight loss <sup>©</sup>
		(48 hrs)	(14 days)
	Cellulosic insula	tions	
Cellulosic l	7.42	3.2	0.45
Cellulosic 2	7.07	8.0	2.1
Cellulosic 3	7.58	3.6	0.28
Cellulosic 4	6.48	6.0	1.62
Cellulosic 5	6.37	10.0	n/d
Cellulosic 6	5.98	10.0	n/d
Cellulosic 7	4.40	7.0	n/d
Cellulosic 8	6.05	3.0	n/d
CelLulosic 9	4.75	5.0	n/d
Cellulosic 10	7.67	1.9	n ⁄d
Cellulosic 11	4.44	5.0	n/d
Cellulosic 12	8.20	0.2	n/d
Cellulosic 13	8.08	0.4	n/d
	Glass fiber insul	ations	
Fiber Glass 1	9.1	0.6	2.0
Fiber Glass 2	9.62	0.1	1.1
Fiber Glass 3	9.24	2.8	1.4
	Foam insulatio	ns	
Foam 1	6.70	3.8	n/d
Foam 2	5.58	3.5	n/d
Foam 3	5.60	3.8	n/d
Foam 4	7.48	3.1	n/d
	Water		•
Domestic	6.80	3.4	2.1
Distilled			1.8

Table 2 -- Summary of results on insulation leachates

<sup>a</sup> All values for AISI 1010 low carbon steel at  $73\pm3$ °F.

<sup>b</sup> Average of 4-8 determinations.

<sup>c</sup> Filtered leachates after 3 days exposure.

Included in the table are the corrosion rates calculated by the direct reading corrosion rate instrument (Corrater) based on its internal conversion of electrochemical measurements to corrosion rate in mils per year (mpy). The values were taken at 48 hours from continuously recorded output from the instrument. The corrosion rates are in the range 1.9 to 10 mpy except for the lower rates in leachates from Cellulosic Insulations 12 and 13, and for two of the fiber glass insulations. These four environments have the higher pH values.

Table 2 includes several corrosion rates determined by weightloss based on two weeks immersion in filtered leachates. Since considerable variation was observed in the weight-loss, each value in the table is an average of four to eight specimens suspended individually in 700 mL of leachate. The corrosion rates, with two exceptions, are consistently lower than those determined electrochemically. Also the ratio of the rate indicated electrochemically to the rate measured by weight-loss varies from 2 to 12. The erratic variation in this ratio, for reasons discussed earlier, does not allow establishing a useful single correlation factor to convert between the two measurement methods.

At pH values greater than about 5, the dominant driving force for corrosion in types of environments characterized by the insulation leachates is dissolved oxygen from air and the corrosion products are hydrated iron oxides as illustrated in Figures 1 and 2. The rate of corrosion however depends on the availability of dissolved oxygen at the metal interface which is almost always controlled by diffusion of oxygen through a liquid boundary layer and more commonly by diffusion through corrosion product layers. Wet insulation in contact with the surface provides additional diffusion resistance. Below pH = 5, acidity makes a progressively larger contribution to the rate. The variations in the corrosion rates in Table 2 are attributed largely to effects on the oxygen diffusion rate due to differences in the adherence and porosity of the corrosion products as influenced by the specific anions in the leachate. This influence was partially substantiated by observations over a three-week period of specimens placed in the insulation leachates and solutions of additive chemicals. The corrosion products varied significantly in color and adherence which was consistent with a variable effect on the corrosion rate.

In stagnant leachates the diffusion rate decreases with time due to corrosion product build-up. For example, in Cellulosic Insulation 1 leachate, the rate indicated by electrochemical measurements decreased from 6.4 mpy at one hour exposure to 3.2 mpy at 48 hours. From weightloss measurements in the same environment, the average rate based on two days exposure was 1.5 mpy and 0.45 mpy based on 14 days exposure. These time-dependencies of the corrosion rate explain part of the difference in rates determined electrochemically and by weight-loss since the two types of measurements were conducted using different time profiles.

<u>Environment<sup>a</sup></u>	Sparge	Corr. Rate, mpy
Corrater probe in Source 2	none	8.0
cellulosic insulation		
leachate. 100 g/L dist. water.		
11	air	22.0
Corrater probe in Source 1	none	3.0
fiber glass insulation		
leachate. 100 g/L dist. water.		
11	air	25.0
н	nitrogen	0.6
Corrosion rate from weight loss in	none	0.46
Source 1 cellulosic insulation		
leachate. 100 g/L dist. water.		
H	air	3.3
Corrater probe in domestic water.	none	3.4
Corrosion rate from weight loss in	none	2.07
domestic water.		
Corrosion rate from weight loss in	none	1.79
distilled water.		
н	air	9.1
n	nitrogen	0.1
<sup>a</sup> All values for AISI 1010 low carbo	n stool at 73+3°F	

Table 3 -- Effect of sparging on corrosion rate in leachates

alues for AISI IUIU IOW Carbon steel at 73±3 F.

Examples of effects of large changes in availability of oxygen on corrosion rate are listed in Table 3 showing the influence of air sparging to increase the dissolved oxygen at the metal interface and the influence of nitrogen sparging to remove oxygen from the bulk leachate environment. Rates in distilled water are included for comparison. In distilled water, the low carbon steel in the air sparced environment becomes covered with a somewhat flocculent Fe<sub>2</sub>O<sub>3</sub> deposit overlying a black  $Fe_3O_4$  layer. In the deaerated, nitrogen sparged, environment a thin adherent black  $Fe_3O_4$  forms restricting the rate of corrosion. The corrosion rates under stagnant versus nitrogen sparged conditions are greater by factors of 5 and 7.3 in the leachates and 18 in the distilled water. In contrast, the rates with air sparging versus stagnant are greater by factors of 8.3 in the fiber glass leachate and 5.4 in the distilled water. Further examination of the values in Tables 2 and 3 indicate that the corrosion rates in some of the leachates are greater and others smaller than in water. This indicates that in some insulations, chemical species can be leached that may have an inhibiting effect on the corrosion of low carbon steel.

Environment	Corrat	er, mpy	Weight loss, mpy
Source 1 cellulosic insulation	2.9	0.2 <sup>a</sup>	
100 g/1000 ml dist. water		F	
Source 2 cellulosic insulation	3.8	<b>0.6</b> <sup>0</sup>	n/d
100 g/1000 mL dist. water	<u> </u>	<b>0 F</b> <sup>C</sup>	- <b>/</b> I
Source 2 cellulosic insulation	6.5	0.5°	n/a
Source 1 cellulosic insulation			0.27
Immersed 3 in., 14 days			
100 g/400 mL dist. water			
Sterile cotton	4.5	1.3 <sup>d</sup>	n/d
Immersed 2 in., 14 days			
<u>   100 q/400 mL dist water                                    </u>			
<sup>a</sup> 0.2 mpy at >48 hrs., <sup>b</sup> 0.2 mp	oy at >60	hrs., °0.3	3 mpy at >60 hrs.,
° 0.2 mpy at 120 hrs.			

Table 4 -- Corrosion rates with deep immersion

The effects of total immersion on corrosion rates of low carbon steel in wet insulations at  $73\pm3$  F are shown in Table 4. With one exception, the rates are those indicated by the Corrater which are higher than actual rates determined by weight-loss. The values do indicate a large decrease in rate from 1 to 48 hours exposure, the values at the latter time being much lower than those indicated in Table 2 in stagnant leachate. The single weight-loss value of 0.27 mpy in wet Source 1 cellulosic insulation compares to 0.45 mpy in stagnant leachate. These decreased rates result from the decreased access of oxygen to support the corrosion. Confinement of corrosion products and local increase in pH also contribute to a decrease in the corrosion rate.

Effects of partial immersion of 50 mm x 50 mm x 0.076 mm low carbon steel specimens in wet insulations and wet cotton are shown in Table 5. These specimens were immersed with 12.5 mm extending above

Environment <sup>a</sup>	nt <sup>a</sup> pH Maximum mpy penetration at air interface (24 days)		mpy based on total immersion in leachate (14 days)
Cellulosic			
Source 1	7.42	6.1	0.45
Source 4	6.48	6.1	1.62
Source 11	4.44	>22.8(1 hole)	.n/d
Source 13	8.08	8.7`́	.n/d
Glass fiber			,
Source 1	9.10	7.0	2.0
Source 2	9.62	9.8	1.1
Source 3	9.24	>22.8(2 holes)	1.4
Distilled water		. ,	
Sterile cotton	5.22 <sup>b</sup>	9.1	1.8

Table 5 -- Corrosion rates with partial immersion

<sup>a</sup> Specimens were 50 mm x 50 mm x .076 mm immersed in wet insulation to a depth of 37.5 mm. Containers were 50 mm deep and 100 mm in diameter. Distilled water was added to maintain liquid level visible at the insulation surface. <sup>b</sup> pH of the liquid at the end of exposure.

the wet insulation into the air. The specimens were exposed for three weeks at  $73\pm3$  F, cleaned and examined for complete penetration along the vicinity of the interface. Otherwise, a survey was made to determine the minimum thickness which is reported as the maximum penetration rate in Table 5. Two of the environments had caused complete penetration of the 0.076 mm thick steel specimens. The maximum penetration was always along the metal-insulation interface and the corrosion rates were very large compared to those under stagnant (Table 2) and deep immersion (Table 4) conditions. Complete penetration occurred for the more acid Source 11 cellulosic insulation and for the high pH Source 3 glass fiber insulation. The acidity of this cellulosic insulation contributes to its higher corrosion rate.

The complete penetration of the specimen in the Source 3 glass fiber insulation is attributed to the localized failure of passive films that tend to form on steel surfaces in the leachates of pH>9. In borax solutions with pH=9.5, passive films quickly form and the steel remains bright with a negligible corrosion rate for months. In contrast, the alkaline pH leachates exhibited borderline tendencies for passive film formation. The specimen partially immersed in the Source 3 glass fiber insulation for three weeks had the majority of the area passivated. Dark patches occurred along the air interface; and the twc holes, each larger than one mm, were in these areas. Thus, the passive film is unstable and undergoes localized breakdown with rapid local corrosion rates aggravated by the large cathode (passive) to anode (active) area ratio. The corrosion rate deeper into the wet insulation was very small.

The behavior of the Source 3 glass fiber insulation relative to the cellulosic insulations was substantiated by comparing the polarization curves for the low carbon steel in leachates from the insulations. Figure 4a shows the cyclic polarization curve in the Source 1 cellulosic insulation leachate. On the down-scan, the current densi-



Figure 4 -- Cyclic polarization curves for low carbon steel in (a) Source 1 cellulosic and (b) Source 3 glass fiber insulation leachates

ties are lower than on the up-scan. In contrast, the cyclic curve for Source 3 glass fiber leachate in Figure 4b exhibits a down-scan at higher current density than on up-scan. After the cycle, the specimen in the cellulosic insulation leachate was uniformly corroded. The specimen in the glass fiber leachate was severely pitted with bright passivated regions between the pits. Although none of the leachates resulted in polarization curves of the usual shapes shown in Figure 3 with clearly developed passive potential regions on up-scan, the curves in Figure 4 are consistent with the behavior represented by Figure 3d.

# CONCLUSIONS

Exposure of low carbon steel to wet insulations and insulation leachates results in corrosion sensitive to the availability of dissolved oxygen from air. Governing variables are (1) the adherence and porosity of corrosion product deposits which differ due to the variety of leachable chemical species in insulation products and (2) the depth of immersion in wet insulation establishing a barrier to the diffusion of oxygen. The worst-case situation exists at the metal-air-wet insulation interface on partial immersion. Access of oxygen along this interface accelerates corrosive attack and may lead to corrosion rates an order of magnitude greater than deep in the insulation. This behavior suggests that a generally applicable comparative evaluation of the corrosiveness of building thermal insulations could be made by partial immersion of test specimens in wet insulation including wet sterile cotton, prepared in accord with ASTM C665 [3], as a reference. With 0.076 mm thick specimens, complete penetration at the interface could be taken as a pass/fail criterion with the distribution of changes of thickness as a more quantitative means of comparative evaluation.

# ACKNOWLEDGEMENTS

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TENSILE TESTING OF EIFS LAMINAS

**REFERENCE:** Flanders, S. N., Lampo, R. G., and Davies, A. G. Jr., "Tensile Testing of EIFS Laminas," <u>Insulation Materials: Testing</u> and <u>Applications. 2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

ABSTRACT: Information about tensile properties of exterior insulation finish system (EIFS) laminas has been unavailable to the engineering profession. Knowledge of the tensile properties of the reinforced laminas over the exterior surface of the insulation is key to 1) comparing EIFS performance and 2) establishing rules for spacing expansion and control joints in EIFSs for different thicknesses in different climates. This is a report that 1) establishes a procedure for tensile tests that may become an ASTM standard and 2) presents first-cut results on a variety of EIFS laminas, presented generically and anonymously. The paper discusses the results in detail and discusses what remains to be done to establish design guidelines for expansion and control joints in EIFS applications.

KEYWORDS: insulation, exterior insulation and finish system, EIFS lamina, tensile strength, tensile modulus, expansion joints, cracking

#### INTRODUCTION

An exterior insulation and finish system (EIFS) typically comprises (Fig. 1): 1) a layer of polystyrene insulation board, mechanically fastened or adhered to the building wall, or both, and 2) a laminated layer (lamina) on the exterior surface of the

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insulation board that incorporates a reinforcing mesh embedded in a base coat material and a final finish coat, giving a stucco-like finish. An improperly installed EIFS is subject to cracking of the lamina (Fig. 2). Despite this potential problem, neither designers nor manufacturers have a basis, beyond field experience, for determining placement of EIFS expansion or control joints. Cracking is most likely at locations that precipitate stress concentrations, such as the corners of windows, building joints, and changes in the substrate. However, large expanses of wall may also be crack prone due to thermal movement.



FIG. 1 -- Characteristic EIFS cross section.

### EIFS Classification

The Exterior Insulation Manufacturers Association classifies EIFSs into two broad categories: 1) polymer-based (PB) and 2) polymer-modifed (PM) systems. The following description of the two systems is paraphrased from Lampo and Trovillion [1].

<u>PB Systems</u> -- In general, PB systems have resinous lamina that are less than 6.4 mm thick. PB lamina are usually applied over molded, expanded polystyrene insulation board, which is adhered and sometimes also mechanically fastened to the building substrate. The base coat material may be a polymer cement or all polymer. The standard reinforcement is typically a polymer-coated glass fiber mesh, which is embedded in the base coat. A heavier grade of polymer-coated glass fiber mesh may be included in the lamina, typically where greater impact resistance is needed. <u>PM Systems</u> -- In general, PM systems are greater than 6.4 mm thick. PM lamina are typically applied over extruded polystyrene insulation board, which is mechanically fastened to the building substrate. The base coat is generally cementitious, modified with polymers. Most systems have a polymer-coated glass fiber mesh, which is mechanically attached to the insulation board, prior to application of the base coat. This mesh is heavier than the standard mesh for PB systems and serves partly as a lath for the base coat. One system tested uses a steel mesh reinforcing of poultry netting.

#### Cracking Mechanisms

The causes for thermally induced cracking problems are not well understood. The most common'sites for cracking in an EIFS, as with other building materials and systems, are at sharp-cornered penetrations, such as windows and doors. Such corners create stress concentrations from which cracks may propagate. The solution to such cracking usually entails substituting curved corners for sharp ones, incorporating control or expansion joints, or adding reinforcing. Removal of the stress concentration usually is more effective than increasing the strength of the assemblage.

Cracks may occur with a change in substrate. An EIFS placed continuously over a building expansion joint will usually initiate cracking, although the building within the EIFS now experiences much smaller temperature swings. Thermal movement is usually the source of force that induces cracking in configurations like those described above. Problems may occur under at least two scenarios:

1. When the insulation with its high thermal coefficient of expansion (TCOE) expands, it induces tensile stresses in the surface lamina which has a lower TCOE and a higher tensile modulus (E). When the lamina does not expand thermally as rapidly as the insulation, then substrate tensile stresses may exceed the cracking stress  $(F_{crack})$  of the lamina and cause cracking at intervals.

2. When the lamina shrinks around the envelope or pulls aways from fixed features such as windows.

Cracking in both scenarios can be alleviated with placement of expansion joints that prevent the forces in the lamina from reaching  $F_{crack}$ . Therefore, a stiff EIFS lamina will need more frequent expansion joints than a flexible one of equal cracking strength.

Most PM system manufacturers specify a maximum square region of wall surface before a control joint is required. This requirement is based largely on industry experience with products in the field. PB manufacturers currently do not have a control joint spacing requirement that is based on surface area. Problems on actual installations of both PM and PB systems indicate that not enough is currently understood about their mechanical properties to significantly reduce the probability of cracking for either type of system.



#### FIG. 2 -- Cracking in an improperly installed EIFS.

#### Purpose of This Study

Currently, the TCOE, E, and F values of EIFS laminas are unavailable to engineers to assist in designing expansion joints in EIFSs or for comparing different systems. This paper focuses on the experimental determination of the tensile properties, E and F crack, of EIFS laminas. From such experiments it is possible to determine the number of samples required to attain desired confidence levels for measuring those properties. The experiments also make possible comparisons of 23 different types of EIFS.

This study does not include determination of TCOE for the systems tested, nor does it examine the effects on cracking of different EIFS expansion joint strategies, re-entrant corners at doors and windows or other system geometries.

As part of a program within the U.S. Army Corps of Engineers to study EIFS impact resistance and in cooperation with ASTM, eight manufacturers supplied four panels each of 23 different examples of an EIFS. These represent a range of system types and characteristics. These specimens were also used for the tensile testing of lamina reported here, and were organized in a manner that was blind to which manufacturer supplied which specimens. The key to who supplied which sample resided in one person who was able to return results pertaining to a given manufacturer to that manufacturer, so that each would know its standing among the others.

#### EIFS LAMINA TENSILE TESTING

#### Summary

In this study, EIFS laminas, with the number and character of reinforcing meshes indicated in Table 1, were tensile tested. In most cases, five replicates of each EIFS was tested.

TABLE 1 -- Types of EIFS lamina systems tested.

Number of Standard Mesh Layers <sup>1</sup>	Number of Heavy-Duty Mesh Layers <sup>2</sup>	Number of Systems Tested
Polymer-Based (PB):		
Polymer-Cement Base	. Coat	
1		6
1	1	5
2		1
	2	1
Polymer Base Coat		
1		2
1	1	1
2		1
Polymer-Modified (F	MA ·	
Standard	<u>117</u> .	4
Thick Base Coat		1
Wire Mesh Instead a	f Class Fiber	1
wile resu instead o	or Grass fiber	T

<sup>1</sup> As defined by the manufacturer. Standard mesh  $\approx 0.14 \text{ kg/m}^2$ . <sup>2</sup> As defined by the manufacturer. Heavy duty mesh  $\approx 0.48-0.68 \text{ kg/m}^2$ .

The testing procedure incorporated elements of ASTM Practice E 4 and ASTM Method E 83. Samples were prepared from full-scale panels contributed by several major EIFS manufacturers. The insulation was removed from the lamina which became a specimen. The specimen was bonded to plexiglass tabs and tested in a universal testing machine. Data for E and F were obtained from specimen cross-sectional areas and test load-extension relationships.

#### <u>Apparatus</u>

The universal testing machine (Fig. 3) was equipped with a digital extensiometer. An analog plotter and a personal computer, equipped with digitizing hardware and a commercial data collection software package, recorded load and extension data at a rate of one data set per ten seconds.



FIG. 3 -- Testing machine and typical EIFS specimen.

#### Sample Preparation

A 0.46- x 1.2-m strip was sawed from a 1.2- x 2.4-m panel of each system. Each strip was further disassembled by sawing through the insulation to separate most of it and panel substrate from the lamina. The lamina with about 2.5 cm of insulation attached was then cut into 7 to 10 strips, measuring approximately 6.35 x 40.6 cm. The remaining insulation was then removed by hand sanding to leave the EIFS lamina as a test specimen.

Of the candidate specimens from each panel, five that were free of fasteners or other irregularities were chosen for testing. To facilitate testing, 5.1- x 6.3- x 0.34-cm plexiglass tabs with the ends beveled to 45 degrees were bonded to the ends of each specimen with a two-part "five-minute" epoxy. The test specimens were then conditioned at  $23^{\circ}$ C and 50% relative humidity for at least 24 h. Prior to testing, each specimen was photographed. The specimen width was measured at three locations with dial calipers. The specimen thickness was measured at the same three locations with a 0.34-cm-diameter flat anvil micrometer. These measurements were made with a precision of 0.0254 mm. The averages of these values were used to compute an average cross-sectional area of the specimen.

#### Test Procedure

The specimen was placed with the plexiglass tabs in the grips of the testing machine. The digital extensometer was placed at the center 100 mm of the approximately 300-mm long specimen. The specimen was loaded at a constant crosshead speed of 0.003 mm/s. This corresponds to an approximate strain rate of  $10^{-5} \text{ s}^{-1}$ . The specimen was visually monitored throughout the test for the formation of cracks. Loading continued until a drop in load of at least 0.5 kN from the maximum load occurred. During loading, load and extension data were plotted and recorded to floppy disk. Individual tests typically lasted for 30 minutes and resulted in about 180 data points each. After testing, all specimens were again photographed.

#### <u>Calculations</u>

The following physical properties were calculated for each test:

1. Crack strength (F ), calculated by dividing the force, at which the specimen itself was first seen to crack, by the average cross-sectional area.

2. Ultimate strength (F ), calculated by dividing the maximum force by the average cross-sectional area.

3. Primary tensile modulus (E), calculated by dividing the change in stress during the most prolonged period of linear extension by the change in strain for that period of extension.

This report focuses on F the system, because cracks cause an EIFS to lose its weather-proof integrity. E is chosen because that value dominates the majority of the system's extension before F the crack.

#### <u>Results</u>

Results show that, in most cases, the lamina specimens' stressstrain relationships were linear for much of the range. Figure 4 illustrates typical load-extension plots of five replicates. Many systems had consistent slopes and a reasonably consistent ultimate load. Figure 5 illustrates the chaotic nature of some specimens. Otherwise, one can see characteristic shapes that are repeated, but the behavior is nonlinear.



FIG. 4 -- Replicate load-extension plots typical of many EIFSs tested.



FIG. 5 -- Nonlinear load-extension plots of an EIFS that was tested.

The problem of choosing an appropriate value for E is more difficult for results like those shown in Fig. 5. Fortunately, most plots displayed the linear response depicted in Fig. 4. The following summary of tensile testing results is by category for each type of system, type of base coat and type of mesh shown in Table 1.

<u>PB Systems</u> -- Table 2 shows averaged tensile testing results for each type of PB EIFS lamina, presented in arbitrary order. The the coefficients of variation (CV = estimate of standard deviation / expected value) are based on 5 replicates, except for the example that is marked as an exception, which had 18 replicates. These 18 replicates provided the estimate of standard deviation s that determined how many replicates of other systems might achieve the desired precision for these tests (Eq 1 below). Polymer lamina systems had, on the whole, a higher F than the polymer cement systems. Likewise the E values were somewhat higher for polymer systems.

TABLE 2 Results of tensile testing of PB EIFS lamin
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Layers of Layers of Mean Coeff. of	f Mean	Coeff. of
Standard Heavy-Duty F Variation	n E	Variation
Mesh Mesh (Mapa) (% Mean)	) (G Pa)	(% Mean)

Polymer-Cement Base Coat

1 1 1 1 1 1 1 1 1 1 2	1 1 1 1 2	2.04 2.98 3.85 1.36 6.16 8.51 4.85 3.64 2.42 8.15 6.35 5.38	15 26 9.5 15 12 12 11 82 16 20 17 29 4 6	0.315 0.494 0.388 0.258 0.583 1.40 0.802 1.92 1.55 1.28 1.09 0.545	8.4 26 5.3 26 29 6.3 0.5 36 5.9 12 7.5 6.0
<u>Polymer Ba</u>	se <u>Coat</u>	10.4		1.50	
1 1		6.42 9.17	8.2 6.2	0.478 0.462	9.2 7.3
1	1	16.1	7.0	1.78	1.6
2		9.42	45	0.752	12

<sup>1</sup> 18 replicates for this type of lamina.

Among all PB systems **E** correlated fairly well with the number and types of reinforcing meshes. Among polymer systems, **F** correlates somewhat with the reinforcement mesh. Among polymer cement systems, the **F** value does not correlate strongly with the type of mesh. Only two layers of heavy-duty mesh are markedly stronger than other possible combinations.

The testing technique appears to be compatible with PB lamina. In 12 out of 16 cases, CV for F is 20% or less. In a different 12 out of 16 cases, the CV for  $\mathbf{E}$  is 20% or less. Ironically, the system with the highest CV for  $\mathbf{F}_{crack}$  of 82% had the lowest CV for  $\mathbf{E}$ of 0.5%.

Based on these experiments one may expect any given PB system to have coefficients of variation ranging from 5 to 82 % for F when testing PB EIFS systems, with a mean value of 21 %. One may expect the average CV for E to be 11 % with a maximum CV of 36 % and a minimum CV of 0.5 %.

<u>PM Systems</u> -- Table 3 indicates somewhat higher CVs for testing PM systems than for PB. Only 2 of 6 systems resulted in an CV less than 20% for  $\mathbf{F}$ . Only 1 of 6 systems resulted in an CV less than 20% for  $\mathbf{E}$ . In fact, some systems bowed while in tension, a shape that may be unrepresentative of the behavior of the laminas in use.

PM systems, as represented here, had a somewhat lower F than the standard PB systems. These systems were, in most cases, much stiffer.

Mean F (M <sup>c</sup> Fack (M <sup>c</sup> Fa)	Coeff. of Variation (% Mean)	Mean E (G Pa)	Coeff. of Variation (% Mean)
Standard			
1.10 1.51 2.85 2.22	19 25- 22 9.3	9.22 5.82 9.13 0.139	175 23 17 93
<u>Thick Base</u>	Coat		
1.85	34	13.0	72
<u>Wire Mesh I</u>	nstead of Glass	<u>Fiber</u>	
0.914	24	12.3	152

TABLE 3 -- Results of tensile testing of PM EIFS laminas.

Based on these experiments one may expect any given PM system to have CVs ranging from 9 to 33  $for F_{crack}$  when testing PB EIFS systems, with an average CV of 20 %. One may expect an average CV for E to be 77 % with a maximum of 175 % and a minimum of 5 %.

#### Discussion of Results

<u>Applicability of the Test</u> -- The range of CVs reflects in some cases on the variabilities among different specimens of the same lamina system. However, PM systems are not well adapted to this test because the reinforcing mesh occurs on one side of the lamina. This results in bending of the sample of the specimen under load, as the stress attempts to follow the mesh. The bending may cause earlier cracking than for a confined lamina. PB systems normally have the reinforcing mesh at the center of the lamina. Consequently, they are not as prone to distortion under load.

TABLE 4 -- Number of replicates required to calculate the mean value of the property shown within 5% of its true mean value 95 times out of 100 for PB EIFS laminas.

Layers of Standard Mesh	Layers of Heavy-Dut Mesh	Number of y Systems	Median F E <sup>crack</sup>	Maximum F E <sup>crack</sup>	Minimum F E <sup>crack</sup>
Polymer-Ce	ment Base	Coat			
1		6	71 82	210 215	27 9
1	1	5	88 13	2052 394	36 1
2		1	260 18	 	• • •
	2	1	7 11	• • • • • • •	•••
<u>Polymer Ba</u>	se Coat				
1		2	17 22	21 26	12 17
1	1	1	16 1		•••
2		1	627 35	· · · · · · ·	

<u>Number of Replicates Required</u> -- Initial testing of a limited number of EIFS types indicated that only five replicates of a given sample would be required to determine the mean tensile modulus and the mean yield strength of the reinforced coating to be within 15% of the true mean value 95 times in 100, according to the following formula [2]:

 $n = \frac{t^2 s^2}{d^2}$ (1)
where  $t = \frac{d^2}{t}$ t value for n degrees of freedom and probability  $\alpha$ s = estimate of standard deviation for n degrees of freedom
d = desired precision of the estimate.

The 15% level of precision would be appropriate for comparing properties among EIFS systems, without testing too many samples. To know the true mean value within a 5% level of precision would require nine times as many samples to obtain triple the precision, in this case 45 samples.

Any given manufacturer might want to know the mean value of their products' physical properties to within 5% of the true mean. This would require substitution of s for each system into Eq 1 and t = 2.776 for four degrees of freedom from five samples and for  $\alpha = 0.05$ . Table 4 demonstrates that only a few samples would be required for some manufacturers of PB EIFS laminas, but as many as several thousand would be required for others. Again, at the 95% confidence level, one-ninth the number of replicates would give one-third precision at ± 15%, and one-fourth the number of replicates would give half the precision at ± 10%. Table 5 shows that PM systems would require more than 100 replicates at the same confidence level.

Number of Systems	f Median for F (for E)	Maximum for F (for E)	Minimum for F for E)
<u>Standard</u>			
4	136 (1372)	194 (9447)	27 (9)
<u>Thick Bas</u>	<u>se Coat</u>		
1	346 (1615)	• • • • • •	• • • •
<u>Wire Mesh</u>	<u>Instead</u> of	<u>Glass Fib</u>	er
1	182 (7132)	· · · · · · ·	· · · · · · ·

TABLE 5 -- Number of replicates required to calculate the mean value of the property shown within 5% of its true mean value 95 times out of 100 for PM EIFS laminas. The amount of variability within one manufacturer-supplied specimen that produces these figures suggests the need for further study of the variability that would be obtained by different applicators using the same manufacturer's product.

Failure at Tab -- One aim in specimen preparation and testing techniques was to promote failure in regions of the specimen away from the grips. Failure at the grips raises the question of whether stress concentrations caused by the test procedure itself caused failure. Two methods of avoiding failure at the grips are to provide a narrower region in the center, or to provide thicker tabs for the grips to attach to. The specimens in these tests had a built-up tab. Of all systems tested, 18% consistently failed near where the tab began, 27% failed well away from the tab, and the rest failed at mixed locations either near or far from the tab. In the tests reported here, all results were included.

#### CONCLUSIONS

#### Validity of Test Method

ASTM Test Possible -- The experience in performing tensile testing demonstrated that the techniques used could form the basis for an ASTM test to determine the following EIFS properties: F , and E. The test method warrants closer evaluation in the case of PM systems, because of their tendency to distort under load, which may happen in conditions of use, but with a different curvature. A confined test would prevent this, but might make observation of crack formation difficult.

These tests have indicated the number of tests that may be needed to obtain mean values to a desired confidence interval for each type of system tested. For some systems this reflects the variability of the product; for others the test may require refinement to adapt to the types of specimens tested.

A standard test method would provide a comparison between systems in the open literature that engineers could use for specifying EIFS performance. Such a test would also aid manufacturers of any given EIFS in developing application and design guidelines for expansion joints that avoid cracks in the lamina.

<u>Expansion Joints</u> -- These tests show that PM systems have a significantly higher E and, on the whole, a lower F than PB systems. This means that expansion and control joints are more critical for PM systems.

#### Recommended Further Study

<u>Cracking Mechanisms</u> -- A better understanding of cracking mechanisms is needed. In what locations and under what temperature regimes do cracks occur? What are the roles of 1) discontinuous insulation, 2) insulation thickness, or 3) the insulation attachment method?

<u>Modeling of EIFS Stresses</u> -- Adequate computer models of the thermal expansion interactions of EIFS lamina-insulation-substrate systems are needed. For some lamina configurations, like window and door corners, three-dimensional models may be required. These would facilitate testing the effects of alternative lamina configurations and formulations of lamina E and  $F_{crack}$  values.

<u>Physical Properties</u> -- The remaining physical data required are for the thermal coefficient of expansion, TCOE, for EIFS laminas. How E,  $\mathbf{F}_{crack}$ , and  $\mathbf{F}_{ult}$  change as a function of temperature is also unknown. Only with all these pieces in place can one make definitive statements about appropriate placement and design of EIFS expansion joints.

<u>EIFS Variability</u> -- EIFS properties are sensitive to applicator technique. Base coat thickness and mesh overlap are among variables that affect system properties. The different systems varied widely in the consistency of results obtained in testing specimens from a single, manufacturer-supplied panel. It would be important to know further the variability that would be attributable to different applicators who are nominally adhering to manufacturer-recommended application methods. Given this information, it would be easier to recommend design values for  $\mathbf{F}_{crack}$ ,  $\mathbf{F}_{ult}$ , and  $\mathbf{E}$  that incorporate a safety factor and can be used with a high degree of reliability. In the absence of such a value, an ASTM consensus should be reached that relates the standard deviation obtained in laboratory tests to a recommended design value for these properties.

<u>Alternative Tests</u> -- An adaptation of ASTM Standard Test C 580 may be more effective for PM systems than the test discussed here. That test is designed for homogeneous materials, such as mortars and grouts. The presence of the reinforcing mesh would have to be accounted for.

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# FIELD PERFORMANCE, HYDROLYSIS AND DURABILITY OF UREA FORMALDEHYDE FOAM INSULATION IN AUSTRALIAN DWELLINGS

**REFERENCE:** Brown, S. K. "Field Performance. Hydrolysis and Durability of Urea Formaldehyde Foam Insulation in Australian Dwellings," <u>Insulation Materials: Testing and Applications, 2nd Volume</u>, <u>ASTM STP 1116</u>, R. S. Graves and D. C. Wysocki, Eds., American Society for Testing and Materials, Philadelphia, 1991.

**ABSTRACT**: Durability and field performances of urea formaldehyde foam insulation (UFFI) in Australian dwellings and the physical and chemical properties important to these performance have been investigated by a combination of field inspections and product sampling, a postal survey of building problems and occupant complaints, formaldehyde gas concentration measurements and laboratory assessment of product behaviour. Products gathered in the field exhibited large differences in physical and chemical properties and stability. Ill effects have been reported to Health Departments by occupants in 0.2% of installations and were typical of those known to result from exposure to formaldehyde gas, showed strong association to building occupancy and may have been associated with poorly formulated and unstable products. Laboratory assessment of commercial products showed that all products hydrolysed, lost strength and emitted more formaldehyde gas under high humidity, some exhibiting linear shrinkage of more than 30% after only short periods of exposure to high humidity.

**KEYWORDS**: urea formaldehyde foam insulation (UFFI), formaldehyde, durability, hydrolysis, physical properties, health, indoor air, survey.

## INTRODUCTION

Sharp increases in energy costs in the 1970s resulted in greater use of thermal insulation in dwellings and its installation in walls as well as ceilings. Urea formaldehyde foam insulation (UFFI) found extensive use as a post-construction (*in situ*) wall insulation in many countries, as it could be produced on site as a fully expanded foam which was readily pumped into wall cavities. UFFI also found use in some countries as a ceiling insulation, both as a site-produced foam and as factory-produced, pre-dried batts.

UFFI is produced by mixing urea formaldehyde (UF) resin, foaming/hardening agent solution, and compressed gas (usually air) in an appropriately designed foaming gun. The freshly produced foam usually contains approximately 75% by weight of water which must be allowed to escape after manufacture, leaving a dry, rigid, open-celled foam. Freshly produced UFFI also contains a small proportion of free formaldehyde to assist curing of UF resin and this is emitted from the foam as it dries.

After UFFI has dried and free formaldehyde has escaped, a further (slower) release of formaldehyde may occur by hydrolysis of the polymeric structure, the rate of which will depend on the climate to which the foam is exposed and its hydrolytic stability, factors which are poorly understood. This hydrolysis may also lead to physical degradation of the product [1,2].

UFFI has been used as building insulation in Europe and Great Britain since the 1960s, although predominantly as retrofit insulation for walls and often with restriction to cavitybrick walls [3]. Its use in other countries such as Australia, USA and Canada expanded

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rapidly in the 1970s in response to increased energy costs. However, by the late 1970s there was increasing concern over the release of formaldehyde gas from UFFI and the effect this may have on occupant health. Many cases where occupants experienced ill effects after installation of UFFI were reported by the media. More significantly, laboratory tests had indicated that formaldehyde gas could induce nasal cancer in some animals [4].

Authorities in many countries responded to these problems in various ways. Some produced Standards to specify the properties of and the installation requirements for UFFI. These typically dealt with installation of UFFI in wall cavities since doubts existed about the physical stability of the product when exposed to temperatures and humidities experienced in ceiling cavities. However, UFFI was still used extensively in ceilings in some countries. In other countries, UFFI was banned from further use. Health authorities in Australia [5] recommended that the formaldehyde gas concentration in the air of dwellings after installation of UFFI be limited to 0.1 ppm and that Standards be developed for the products. The work reported here was carried out to assist in this Standards development and to gain better understanding of factors affecting product performance.

#### EXPERIMENTAL METHOD

#### **Field Surveys**

Two types of field surveys were performed.

<u>Product survey</u>: The physical and chemical properties (see Laboratory Assessment) of 22 UFFI samples taken from buildings were determined, generally in response to some contact by building owners which was not necessarily related to health complaints.

<u>Postal survey</u>: A postal survey was carried out in 1985 of occupants of UFFIinsulated dwellings from two populations:

(a) 155 dwellings in Canberra (cool-temperate climate) most of which were singlelevel brick-veneer buildings with UFFI installed in wall cavities 5 to 7 years prior to the survey and selected as representative of installations of that period; and

(b) 204 dwellings with UFFI installed for periods of 2–14 years (average  $\pm$  SD was 6  $\pm$  2 years), gathered from six of Australia's State or Territory Government health authorities from whom assistance had been sought, generally after experience of ill health or apprehension of such.

The same survey form was distributed to both populations and is fully described elsewhere [6]. It gathered information on building related problems (installation damage, electrical faults, rain penetration, timber decay) and health complaints from occupants (ill effects and their nature, duration and association with occupancy of buildings). Also, in the case of population (b) measurements of formaldehyde gas concentrations in the dwellings at the time of assistance were supplied by Government authorities.

#### Laboratory Assessment

<u>Products</u>: Samples of nine UFFI products were requested from three Australian foam producers in 1986. These were supplied either as factory-produced batts with aluminium foil skins (two products) or as blocks of *in situ* foam typical of that produced on building sites with portable equipment (seven products). Products 1 and 2 were from a producer claiming to use a phenol-modified UF resin with greater hydrolytic stability. Products 3, 4, 5 and 7 used typical UF resins from two different manufacturers. Products 6 and 8/9 were modifications to products 5 and 7, respectively, in which the producer had included additives claimed to improve product stability and reduce formaldehyde emission. Products 1–4 were supplied as recently produced dry foams, while products 5–9 were supplied as wet foams in sealed plastic bags from which they were removed within 3 days of foaming. All products were aged in a ventilated building for several months before testing commenced.

<u>Test methods</u>: Test specimens were cut from blocks of dry foam at least 10 mm below specimen surfaces to avoid anomalous behaviour from acid build-up at these locations. Unless specified otherwise, specimens were conditioned at 23°C and 50% RH for at least 1 week before testing. The following test methods were used to characterise the initial properties of foam products and to monitor their change with aging under several temperature/humidity conditions.

(a) Moisture adsorption: A specimen of foam of approximate dimensions  $100 \times 100 \times 25$  mm, and which had equilibrated under laboratory or environmental aging conditions, was weighed and then dried at 40°C to constant weight (±0.01 g) in a laboratory oven (approximately 24 hours) and the mass loss determined:

#### moisture adsorption = mass loss/foam mass after drying.

(b) Dry density: The specimen from field or laboratory samples of foam was oven-dried as described in (a); each dimension of the dried specimen was measured in triplicate to the nearest 0.5 mm; average dimensions were calculated and specimen volume was determined; the specimen was weighed to the nearest 0.01 g and density determined in  $kg/m^3$ .

(c) Acid value and free formaldehyde content: A 50 mm cube of foam was sampled and then pulverised in 400 ml of distilled water for 60 s in a high-speed blender and the foam residue was filtered out using Whatman 41 filters; the filtrate was collected and titrated to a neutral end point with 0.1 M sodium hydroxide (NaOH) solution; the procedure was repeated using distilled water alone and the sample titre volume corrected by subtracting blank titre volume:

acid value = volume (ml) of 0.1 M NaOH/mass (g) of foam.

Forty millilitres of 1 M sodium sulfite (Na<sub>2</sub>SO<sub>3</sub>) was added to the neutralised filtrate:

 $CH_2(OH)_2 + Na_2SO_3 \rightarrow NaCH_2(OH)SO_3 + NaOH$ 

and the liberated NaOH titrated with 0.05 M  $H_2SO_4$ ; the procedure was repeated using distilled water alone and the sample titre volume corrected for the blank:

free formaldehyde content = 0.3 x volume (ml) of  $0.05 \text{ M} \text{ H}_2\text{SO}_4/\text{mass}$  (g) of foam.

(d) Linear shrinkage after environmental aging exposure: A  $100 \times 100 \times 25$  mm foam specimen was dried at  $40^{\circ}$ C (as described above) and both long dimensions were immediately measured in triplicate to the nearest 0.5 mm and averaged; dimensions were remeasured immediately after environmental aging exposure; linear shrinkages were determined for each dimension and then averaged:

linear shrinkage (%) = 100 x decrease in dimension/original dimension.

(e) Compressive strength: The compressive strength at 10% deformation ( $S_{10}$ , kPa) was determined with a 100 x 100 x 25 mm foam specimen according to ASTM Method C165-77 "Measuring compressive properties of thermal insulations".

(f) Formaldehyde offgassing: A 40 mm cube of foam was dried at 40°C for 24 hours and suspended above a saturated sodium chloride solution (providing 75% RH) at 25°C in a sealed jar for 7 days; the formaldehyde collected by the solution was determined using acetylacetone reagent:

formaldehyde offgassing = mass (mg) of HCHO collected/mass (g) of dry foam.

This method is used by the Bureau for Quality Control of Cavity Fillings [7] who specify a limit for formaldehyde offgassing of 10 mg/g of foam.

(g) Formaldehyde gas concentrations: Measured in environmental aging exposure containers (see below) using a commercially available fuel cell detector [8].

(h) Scanning electron microscopy (SEM): 10 mm diameter plugs of dry foam were fractured at room temperature and then coated with gold at  $10^{-3}$  torr in a magnetron head

sputter coater. The fracture surfaces were examined in a Cambridge 250 Mark III scanning electron microscope at an accelerating voltage of 10 keV and at magnifications from 40 x to 4000 x.

(i) Infrared spectrum: Approximately 0.2 mg of dry foam was milled with 300 mg potassium bromide and pressed into a disc for analysis with a Perkin-Elmer Model 1430 Ratio Recording Infrared Spectrophotometer.

Environmental aging exposure: Dry set foam specimens were exposed to several temperature/humidity environments in a procedure developed from ASTM Methods D2126-75 "Standard Test Method for Response of Rigid Cellular Plastics to Thermal and Humid Aging" and E104-51 "Standard Recommended Practice for Maintaining Constant Relative Humidity by Means of Aqueous Solutions". Specimens with dimensions 100 x 100 x 25 mm (weighing 1.8 to 2.5 g)were predried at 40°C for 24 hours and their dimensions and masses were recorded. They were supported flat on stainless steel mesh platforms 90 mm high inside sealed polypropylene containers 230 mm high and 175 mm diameter. Saturated salt solutions, prepared as slurries of 200-300 g of LR grade salts in 100 mL of distilled water, generated specific humidities in the containers which were held in controlled temperature ovens. The exposure conditions used and the salts that produced them are presented in Table 1.

	Table 1 — Temperatu	re and humidity	conditions en	mployed in	environmental	exposure
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Designation for condition	Temperature (°C)	Relative humidity (%)	Absolute humidity <sup>a</sup>	Salt
60/96	60	96	14.4	Potassium sulfate, K <sub>2</sub> SO <sub>4</sub>
60/85	60	81	12.8	Potassium chloride, KCl
60/75	60	75	11.3	Sodium chloride, NaCl
60/50	60	50	7.5	Sodium bromide, NaBr
60/30	60	30	4.5	Magnesium chloride, MgCl <sub>2</sub> 6H <sub>2</sub> O
50/96	50	96	8.3	Potassium sulfate, $K_2SO_4$
50/90	50	90	7.8	Sodium tartarate, Na <sub>2</sub> C <sub>4</sub> N <sub>4</sub> O <sub>6</sub> .2H <sub>2</sub> O
45/96	45	96	6.2	Potassium sulfate, $K_2^2 SO_4$
35/96	35	97	3.5	Potassium sulfate, $K_2 SO_4$
35/30	35	32	1.2	Magnesium chloride, MgCl <sub>2</sub> .6H <sub>2</sub> O

<sup>a</sup> 100 x mass of water/mass of air.

Test specimens were exposed for 6 months and were periodically examined for formaldehyde gas concentrations within the containers and for dimension and weight changes by removing specimens for 1–2 minutes. After 6 months aging, specimens were dried and moisture adsorption, total shrinkage and final compressive strength were determined.

Further details of laboratory assessment of UFFI products are provided elsewhere [9].

## **RESULTS AND DISCUSSION**

## **Field Surveys**

<u>Product survey</u>: Field survey results of product properties are summarised in Table 2. No information was available on climatic conditions when these products were installed. Sample numbers 1 to 12 were installed by a single contractor in the floor/ceiling cavities of a group of flats and exhibit similar properties. However, variation in the properties of the other products is considerable, with density and compressive strength showing good correlation (r = 0.92) to each other (Fig. 1) for both these products and others gathered for laboratory assessment. In one sample (number 22), the acidity of the foam was extremely high and this product had collapsed and crumbled into a very poor condition 8 years after installation.

Sample number	Product type <sup>b</sup>	Years installed	Dry density (kg/m <sup>3</sup> )	Acid value <sup>c</sup>	Compressive strength, S <sub>10</sub> (kPa)	Formaldehyde offgassing <sup>e</sup> (mg/g)
1–12	I	5	14-16	0.1-0.2		0.7–1.1
13	Ι	8	12	0.20	9.3	_
14	Ι	4	4.6	0.43	4.4	0.6
15	Ι	4	11	0.46	16	0.9
16	Ι	4	13	0.93		_
17	Ĩ	5	11	0.35	-	_
18	Ĩ	5	7.2	0.90	8.1	-
19	Ĩ	-	27		50	-
20	B	7	28	0.23	-	_
$\overline{21}$	B	10	16	0.50	26	_
22	B	8	16	6.3	nmd	_

Table 2 — Properties of UFFI sampled from buildings<sup>a</sup>

<sup>a</sup> Averaged from 2–3 measurements per sample.

<sup>b</sup> I = in situ foam; B = manufactured foam batt.

<sup>c</sup> Refer to method (c) of "Laboratory Assessment"; note that core samples were used.

<sup>d</sup> This product had collapsed and crumbled and was too weak to test (nm = non-measurable).

e Refer to method (f) of "Laboratory Assessment".



FIG. 1 — Variation in compressive strength ( $S_{10}$ ) and dry density for single test samples and field samples of UFFI.

<u>Postal survey</u>: Survey response rates were 77% (119 responses) for population (a) (to be referred to as the wall insulation population) and 74% (151 responses) for population (b) (to be referred to as the health complaint population). UFFI had been installed in these dwellings for 5–7 years for population (a) and 2–14 years (average 6 years) for population (b). Table 3 summarises the incidence of building and odour problems experienced by these populations.

Odour was the only major problem encountered by the wall insulation population, being experienced by approximately 20% of respondents, although odour persistence was less than 4 weeks for 85% of respondents and greater than 6 months for 7%, and was generally

Problem description	Health complaint population (% responses)	Wall insulation population (% responses)
Lining damage (bulging, cracking, blistering of paint)	14	8
Difficulty opening doors or windows	15	3
Electrical faults	2	8
Delayed drying of foam	12	ŏ
Dampness of building interior	10	2
Rain penetration into building	7	2
Decay of timber	1	0
Odour during or after installation	59	19
Shrinkage of UFFI <sup>a</sup>	27	6
Dust evolution from UFFI when disturbed <sup>a</sup>	24	3
Formaldehyde odour	55	23
Ill-health experience	58	12

Table 3 — Incidence of problems with *in situ* UFFI reported in health complaint survey

<sup>a</sup> Problems identified from respondents' comments and not specifically surveyed

considered mild to barely discernible. Twelve per cent of the wall insulation survey population reported ill effects, generally irritation of eyes, nose and throat, but over onethird of these had not detected odours. Also these ill effects often persisted for several months, in contrast to odour persistence, and in some cases may have resulted from causes other than formaldehyde gas as suggested by respondent remarks (e.g. possible hayfever, contact lenses, skin irritation only).

In contrast, the health complaint population exhibited a higher incidence of building related problems, especially those related to the wet condition of installed foam (incidences of staining of linings, dampness and delayed drying were higher to statistical significance at the 2% level), suggesting a higher incidence of poor foam formulation and production. Also, the incidence of odour during and after installation was approximately 60% of responses which is significantly higher for this population, further implicating installation as a factor in these complaints. The odour persisted for less than 4 weeks for one-half of these cases but persisted for longer than 6 months for 28% of the cases, again in marked contrast to the experience of the wall insulation population (4% of odour cases persisted > 6 months).

As a perspective to these incidences, it should be noted that the number of health complaint cases gathered for this survey represents a very small proportion of UFFI installations in Australia. UFFI has been used in Australia in the absence of product standards to insulate 46 000 ceilings (probably plasterboard) and 26 000 walls (predominantly brick-veneer and double-brick construction) [10]. It is estimated from these figures that the incidence of health complaint to authorities is approximately 0.2%, which is comparable to the incidence reported in the USA, UK and Netherlands but less than that reported in Canada [11–14]. A more widespread study could provide a better estimate of the health complaint incidence in Australia but has not been undertaken. It is also found that the distribution of health complaint respondents according to area installed (wall or ceiling) is the same as the distribution of national usage of the product. This indicates that the building element insulated may not be a factor in the occurrence of complaints.

The ill effects reported by the health complaint population were typical of the effects caused by formaldehyde and occurred to high incidences (Table 4) and generally persisted for several months or more (Table 5). Many of these are non-specific effects that could occur in any general population. Hence two survey questions dealt with the association between respondent presence in or absence from UFFI-insulated buildings and the occurrence of ill effects. For those 80 respondents reporting ill effects, 33 exhibited this association and four did not, while the remainder did not answer the questions. This is considered strong evidence of an association between exposure to UFFI and the ill effects encountered.

Ill effect	Frequency <sup>a</sup> (%)	Ill effect	Frequency <sup>a</sup> (%)
Eye irritation	52	Nausea	18
Nose irritation	55	Dizziness	16
Throat irritation	45	Asthma	15
Skin irritation	18	Chest pain	7
Headache	40	Nose bleeding	7
Cough	26	Vomiting	4
Drvness of mouth	27	Cramps	6
Shortness of breath	18	Diarrhoea	3
Sleep disturbance	22	Numbness	3
Fatigue	16	Loss of voice	2

Table 4 — Ill effects and their frequencies reported by health complaint population

<sup>a</sup> Per cent of total number of cases reporting ill health.

Table 5 — Duration of ill health reported by health complaint population

Duration of ill health	Number of reports				
<1 week	2				
2–4 weeks	6				
1–2 months	8				
3–6 months	13				
>6 months	42				
Unknown	9				

State Government Health Departments measured formaldehyde gas concentrations (by impinger sampling/colormetric analysis) in 98 of the health complaint population houses, generally in living areas under normal building conditions and from six months to several years after UFFI was installed. Formaldehyde concentrations ranged from 0.0 to 0.4 ppm and exceeded 0.1 ppm (the indoor air quality goal set for formaldehyde [5]) in approximately 10% of the houses but these did not appear to be related to climatic differences, building element insulated or the experience of ill effects by occupants. However, the measurement conditions may not have been appropriate for assessing occupant ill effects. Several studies [15–17] have demonstrated that formaldehyde concentrations in indoor air vary considerably throughout a one-year period or even within hours, probably in response to changes in ambient climate, building ventilation and air infiltration through building cavities. Many respondents noted that odour and ill effects were intermittent and since the formaldehyde gas concentrations were measured under normal building conditions, it is probable that they are not representative of maximum concentrations.

### Laboratory Assessment

Physical and chemical properties of the UFFI products supplied from Australian manufacturers are summarised in Table 6. Similar to products gathered from buildings, a considerable variation in properties is exhibited. Also, dry density and compressive strength show strong correlation (Fig. 1) as discussed earlier.

When exposed in sealed containers to the environmental conditions described in Table 1, all products adsorbed moisture to an extent depending on relative humidity but with little influence from product properties or test temperature. Moisture adsorption was small up to approximately 80% RH but increased sharply at higher humidities reaching more than 40% w/w for some products (Fig. 2).

Product number	Dry density <sup>a</sup> (kg/m <sup>3</sup> )	Acid value <sup>a</sup>	Compressive strength <sup>a</sup> , S <sub>10</sub> (kPa)	Formaldehyde offgassing <sup>b</sup> (mg/g)		
1	13.5±0.2	0.56±0.09	29.9±0.6	1.6		
2	12.8±0.8	$2.2\pm0.9$	15.8±1.7	1.7		
3	12.9±0.7	$0.08\pm0.02$	20.0±0.5	1.3		
4	12.8±1.0	0.16±0.06	23.3±2.9	1.9		
5	9.0±3.0	4.5±4.0	3.3±0.8	2.9		
6	9.9±1.6	2.9±3.1	5.4±0.2	2.1		
7	9.9±0.6	0.36±0.15	6.4±1.0	2.2		
8	7.2±0.9	$2.3\pm2.2$		1.6		
9	7.2±0.3	~2.3	_			

Table 6 — Physical and chemical properties of dry set foams

<sup>a</sup> Averaged from 5 measurements.

<sup>b</sup> Averaged from 2–3 measurements.



FIG. 2 — Moisture adsorption of UFFI products after six months of exposure to different relative humidities at temperatures of 35°C to 60°C (see Table 1).

During exposure to these environments over a 6-month period, only products 1, 2, 5 and 9 exhibited linear shrinkages greater than 4% and only at conditions of 96% RH for all temperatures tested. Typical behaviour for products 2, 5 and 9 are shown in Figs 3 and 4; product 1 behaved almost identically to product 2. All products exhibited rapid shrinkage with first exposure to these environments, this shrinkage being more rapid and of greater magnitude for higher exposure temperatures. At high temperatures, most shrinkage occurred within only a day, showing that these products are highly sensitive to such environments and even short exposure to these conditions can cause deterioration. It is notable that all these products have high acid values, well above the limit of 0.5 being considered in Australian product standards. In contrast, products 3, 4 and 7 have acid values below this limit and exhibited lowest shrinkages (below 4% linear shrinkage under exposure conditions, and below 6% when redried after the 6 month exposure period) even under the most severe environmental condition.



FIG. 3 — Linear shrinkage of product 2 with period of exposure to different temperature/humidity conditions.



FIG. 4 — Linear shrinkage of products 5 and 9 with period of exposure to different temperature/humidity conditions.

The compressive strengths  $(S_{10})$  of products were determined after 6 months exposure to the environments (and preconditioning to 23°C and 50% RH for 1 week). However, since the unexposed products varied somewhat in compressive strengths and densities with a direct relationship between these two factors (Fig. 1), it was considered that relative changes in  $S_{10}$ /density would allow assessment of strength stability with environmental exposure. Normalised values of  $S_{10}$ /density are presented in Table 7. These show similar behaviour in strength stability for the different products. Most products lost strength after exposure to 96% RH, the extent of loss increasing markedly with test temperature. Most retained strength at 30% RH, even at 60°C, some products becoming stronger, probably as a result of post-curing of the UF resin. At intermediate humidities, products exhibited different strength stabilities but with no correlation to product acidity. For most products, the maximum reduction in strength was to approximately 25% of original. Consequently, stronger foam products are expected to retain greater strengths after exposure. From experience of handling foams, it was estimated that residual foam strengths should be at least  $S_{10} = 5$  kPa. Assuming the above strength reduction after exposure, new foam should have a minimum strength of  $S_{10} = 20$  kPa which corresponds to a foam density of greater than 12 kg/m<sup>3</sup> (Fig. 1). Since UFFI exhibits little variation in thermal conductivity with changes in foam density [18], this is being considered as the minimum density for UFFI products in Australian standards.

Product number	Normalised $S_{10}$ /DD values after exposure to conditions										
	Unexposed	60/96	60/85	60/75	60/50	60/30	50/96	50/90	45/96	35/96	35/30
1	1.00	0.20	1.01	1.01	1.09	1.19	0.21	0.96	0.63	0.90	1.03
2	1.00	0.16 0.28	1.04	1.04	0.97	1.18	0.12	0.47	0.29	0.34	0.99
3	1.00	0.35	_	_		1.09	0.57	0.93	_	0.80	1.17
4	1.00	0.18	0.51	_	_	0.81	0.44	0.36	0.56	0.79	0.87
5	1.00	0.16	-	0.45	-	1.31	0.63	0.97	0.05 0.15	0.95	0.85
6	1.00	0.21	_	_	_	1.17		0.61	_	1.01	1.26
7	1.00	0.36	1.04	_	-	0.92	0.77	0.84	_	0.81	1.27
8	1.00	0.21	_		—	0.68		0.36	_	-	0.78
9	1.00	0.22	-	-	-	0.86	0.17	—	-	0.38	1.00

Table 7 — Normalised values of S<sub>10</sub>/dry density (DD) for specimens exposed to different temperature/humidity conditions for 6 months

Scanning electron microscopic (SEM) examination of products after exposure to  $60^{\circ}$ C and 96% RH showed changes in the cellular morphology of foams which were consistent with physical behaviour. Product 2, a high acid value foam which shrank 39% by length (77% by volume) was found to have completely lost its cellular structure which had collapsed (Fig. 5). Product 5, also a high acid value foam which shrank 24% by length (56% by volume), appeared to have lost its cell walls leaving an open network of cell ribs which had partly collapsed (Fig. 6), consistent with the high shrinkage and large loss of strength found for this product. Product 4, a low acid value foam exhibiting low shrinkage (1.3% by length, 3.8% by volume) showed no change in cellular morphology with exposure (Fig. 7). However, this product still lost considerable strength with exposure to these conditions (Table 7) indicating that some weakening of the cellular structure occurred that was not evident by SEM examination. Infra-red spectra of this product showed a relative increase of absorbances at 1640 cm<sup>-1</sup>, 1245 cm<sup>-1</sup> and 1130 cm<sup>-1</sup> compared to that at



FIG. 5 — SEM micrographs of product 2 before (top) and after (bottom) exposure to  $60^{\circ}$ C and 96% RH for 3 months.

1380 cm<sup>-1</sup> (and a substantial narrowing of peaks) after exposure to  $60^{\circ}$ C and 96% RH. These changes are consistent with formation of CN and NH end groups during hydrolytic breakdown of the polymeric structure [18,19], which may have caused the above loss of strength without shrinkage or apparent change to cellular morphology.

Formaldehyde gas concentrations were measured in environmental aging exposure chambers and exhibited the behaviour shown in Fig. 8. Formaldehyde release properties of products were assessed according to chamber concentrations after 20 days exposure. These varied significantly between products but showed no correlation to foam acidity. However, the concentrations were significantly influenced by environmental conditions. When corrected for specimen weight, formaldehyde release increased steadily with increasing absolute humidity (Fig. 9), this appearing to be the main factor controlling release rather than product properties or temperature.



FIG. 6 — SEM micrographs of product 5 before (top) and after (bottom) exposure to  $60^{\circ}$ C and 96% RH for 3 months.



FIG. 7 — SEM micrographs of product 4 before (top) and after (bottom) exposure to 60°C and 96% RH for 3 months.

# CONCLUSIONS

Building problems and occupant health complaints have occurred with Australian houses insulated with UFFI, and although evidence suggests their rate of incidence is low, further investigation would allow more definite conclusions. Poor performance by UFFI (whether judged by occupant complaints or assessment of product stability) appears to be associated with several factors such as poor foam production and control, low foam density (resulting in poor product strength after environmental aging), high foam acidity (resulting in cellular collapse and excessive shrinkage), and sensitivity of all products to high humidities (resulting in high moisture adsorption, loss of strength and higher formaldehyde release). Better understanding of these factors and product standards to control them are required for improved product performance.



FIG. 8 — Formaldehyde gas concentration in test chamber for product 1 exposed to different temperature/humidity environments.



FIG. 9 — Formaldehyde gas concentration after 20 days [HCHO]<sup>20</sup> per unit specimen weight for exposure to different absolute humidites.

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