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APPLICATION

Part II of Fluid Meters

Sixth Edition 1971

Interim Supplement 19.5 on Instruments and Apparatus

Report of ASME Research Committee on Fluid Meters

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Instruments and Apparatus

APPLICATION

Part II of Fluid Meters

Sixth Edition 1971

Report of ASME
Research Committee
on Fluid Meters

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
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*Pages 1-148 are "Part One: Theory and Mode of Operation" of the Sixth Edition of *Fluid Meters: Their Theory and Application* and appear only in the complete version of the report.

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EXPLANATORY NOTE

This publication is issued in accordance with an agreement made by the Research Committee on Fluid Meters and the Performance Test Code Committee in 1964. The basis for this agreement was that, in the past, Chapters Two through Five of Part Five on Instruments and Apparatus dealt with various meters and methods of measuring quantities of fluids. Practically all of the material on these chapters was taken from Fluid Meters, and most of the writers of these chapters were members of the Research Committee on Fluid Meters. Chapter One of Part Five on Weighers and Weighing was an exception.

This resulted in duplication of committee membership and activity. It was the decision of the two committees that combining the material into one publication in such a way that the sections dealing with specifications and instructions could be published separately would reduce the work of the committees and the number of separate publications.

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FOREWORD

WHEN the Research Committee on Fluid Meters was organized in 1916, one of its stated objectives was "the preparation of a textbook on the theory and use of fluid meters sufficient as a standard reference." In carrying out this objective the first edition of Part 1 of this report was published in 1924 and received immediate approval and wide usage by the users of fluid meters and by educators. As originally planned by the committee, the report was to be issued in three parts, and Part 1, "Theory and Application," was the first one published. It was to be followed by Part 2, "Description of Meters," and Part 3, "Installation." After its publication, Part 1 was so well received that the number printed sold so rapidly that the second and third editions of this part were needed before time could be found to prepare the other two parts of the report. The second edition of Part 1 was considerably different from the first; however, it followed about the same format and arrangement while the third edition was very little different from the second. These were published in 1927 and 1930, respectively.

Part 2 of the report was published in 1931 and contained a complete description of the physical characteristics of the meters then being manufactured. However, it was found that the material in this part became obsolete so rapidly that it was decided not to try to keep it up but to tell anyone interested in these descriptions that they should be secured from the manufacturers, since their literature must necessarily be up to date.

Part 3, published in 1933, gave instructions for correct installation of meters and discussed the effect of incorrect installations. However, Part 3 was abandoned also because the committee decided the material in it should be an integral part of the complete report of the committee.

The fourth edition of Part 1 was prepared in 1937 and was a completely new draft of this part of the report. It was altered because there had been considerable criticism of the fact that the material presented was difficult to put to practical use. The changed format and additional material presented apparently corrected this condition, since this edition went through many printings.

The fifth edition, issued in 1959, followed the same general format as the fourth, and included material gained in the long interval between the two editions.

Another publication by the committee is a manual "Flowmeter Computation Handbook," which was issued in 1961. The procedures in it can be adapted to computer programming.

The format of the sixth edition differs slightly from that of the fourth and fifth editions. Each chapter is complete in itself, so that altering one chapter will not affect preceding or following chapters. Also, somewhat like the third edition and Part 3, the material on installation and application will be both a part of the complete report and a separate publication.

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PREFACE

FIRST Edition, 1924. After six years of effort on the part of the Research Committee on Fluid Meters of The American Society of Mechanical Engineers, it now presents its first progress report entitled "Fluid Meters, Part 1."

This report takes the form of a reference book on fluid meters of all kinds. It contains not only such practical instruction and information, including formulas, constants, and the like, as may be needed by the actual or prospective user but also more general information—the physical principles of design and operation—which may be useful to students, designing engineers, and inventors.

Part 1 treats the general types of fluid meter as well as the principles and methods involved and gives information which may, in many cases, be applicable to various commercial meters. In this part, instruments of individual makers are not discussed in detail but are referred to only incidentally or for illustrative purposes. The general physical principles are in the body of the text, while the derivation of formulas and the refinements of the theory involved has been placed in the appendixes of the report.

Fluid meters are of great and rapidly increasing importance, but, hitherto, the information available on this group of instruments has been incomplete. The material forming Chapters 1, 2, 3, 4, 5, 6, and Appendix C, were recently rewritten. This material contains a new presentation of the subject based on a mathematical analysis which is more specifically applicable to fluid flow than Bernoulli's theorem. This analysis did not include certain additional experimental data now covered in the report, but these data required no serious modification of the text.

The most important modern advance in experimental aerodynamics and hydraulics is the application to them of dimensional analysis. This is absolutely indispensable to an understanding of the behavior of moving fluids, for the phenomena of fluid motion are so complicated as to defy analysis by any other known method. The use of this method is especially valuable, in that it makes possible the reconciliation of data obtained from experiments with the venturi tube and the thin disk orifice which were formerly thought to be irreconcilable. These data are now shown to be mutually confirmatory.

The personnel of the committee, which prepared this report, was as follows: Messrs. R. J. S. Pigott, *Chairman*, J. M. Spitzglass, *Secretary*, H. Bacharach, E. G. Bailey, M. M. Borden, G. S. Coffin, C. A. Dawley, L. M. Goldsmith, F. G. Hechler, Horace Judd, Leo Loeb, P. S. Lyon, H. H. Mapelsden, H. N. Packard, C. G. Richardson, and T. R. Weymouth. Dr. Edgar Buckingham served as a member of the committee until 1922 and made valuable contributions to the early drafts of the report.

SECOND and Third Editions, 1927 and 1930. Continued demand for the report necessitated the publication of a second edition in 1927, and a third in 1930. Before publishing the second edition, the reorganized Fluid Meters Committee carefully reviewed and revised practically all of the chapters of the report, exclusive of the Pitot Tube and Flow Nozzle sections. Subsequent experimental work has provided data for the revision of these two sections which are incorporated in the third edition together with certain minor corrections throughout the report.

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Work on Parts 2 and 3 of the report is progressing. The former is designed to present a brief description with illustrations of the various commercial meters on the market. Part 3 will discuss the proper methods of installation and care which meters must receive to function satisfactorily.

FOURTH Edition, 1937. A consistent demand for this report exhausted the third edition of 1600 copies in approximately one year and a half and made the printing of a fourth edition necessary. Thirteen years have elapsed since the first edition was published, and, while the three different editions have had a flattering reception, the committee feels that the fourth edition should be an improvement over the others as regards convenience in practical use. Part 2, "Description of Meters," was published in 1931 and Part 3, "Selection and Installation," was issued in 1933.

The majority of the criticisms of the several editions of Part 1 have had to do with the difficulty of putting the theoretical equations into practical form for every day use. In preparing this fourth edition, therefore, a radical change has been made in the arrangement and in certain parts of the text. This part has been subdivided into three sections—A, B, and C—of which the following is a brief description. In Section A, the classification and nomenclature of fluid meters, as used throughout the reports of this committee, are given, together with the definitions of special terms and other general information. In Section B, the theory of fluid measurements is presented. In most cases, the assumptions made and the steps taken to develop practical working equations from the theoretical relations are set forth. Section C contains figures and tables for use in solving practical fluid measurement problems, and their proper use is illustrated with examples.

The personnel of the subcommittee that prepared this edition was as follows: Messrs. H. S. Bean, *Chairman*, W. W. Frymoyer, Louis Gess, W. S. Pardoe, Ed S. Smith, Jr., R. E. Sprenkle, E. C. M. Stahl, and T. R. Weymouth. This subcommittee acknowledges with appreciation the assistance rendered in the preparation of this edition by Mr. M. A. Goetz and Mr. T. H. Smith of Mr. Stahl's staff.

FIFTH Edition, 1959. In the twenty-one years which have elapsed since this report was last revised there has been a great deal of growth in the science of fluid mechanics and a manyfold growth in the interest in fluid metering. This increase in interest and knowledge has made necessary many changes in the material covered by this report and made available many more accurate determinations of the constants and other material not appearing in the fourth edition. Realizing this and wishing to incorporate the material which had been published in Part 3 of the committee's report, it was decided that instead of trying to revise the fourth edition that the fifth edition should be a completely new publication, and so the material incorporated in it has been completely rewritten.

In the writing of the material all of the knowledge of modern fluid mechanics which was available to the committee has been used in an attempt to explain the phenomena which are used in the science of fluid metering. While no attempt has been made to go into the fundamentals of fluid mechanics, the results of the study of these fundamentals are available because of the use of more rational formulas and explanations and the use of less empiricism. In addition, the experience of many years in the solution of fluid metering problems by the members of the committee has made the material of the greatest practical use. It is believed that this edition will carry on the great tradition established by the preceding editions.

The arrangement of this edition is similar to that of the fourth. Also, the method of presenting the material is the same, with two exceptions: First, to conform with the universal teaching practice of engineering schools in this country,

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the gravitational system of units has been used in the development of equations and the presentation of physical data. Second, the coefficients of differential head meters are presented on the basis of the pipe Reynolds number in the belief this will be of greater convenience to the user, and will aid in promoting international standardization.

The personnel of the subcommittee which prepared this edition is as follows: Messrs. H. S. Bean, *Chairman,* L. K. Spink,† *Vice Chairman,* E. E. Ambrosius, R. C. Binder,* L. Gess, C. S. Hazard, V. P. Head, A. L. Jorissen,† E. J. Lindahl, I. O. Miner,* and R. E. Sprenkle, while many other members of the main committee cooperated by offering material and constructive criticism.

The success of the work is primarily the responsibility of Mr. Howard S. Bean, the Chairman of the subcommittee; and he has worked hard and long as a labor of love in preparing this material. His service has been of incalculable value in preparing the fourth and fifth editions of this report, and the committee wishes to take this opportunity to state its appreciation for these services.

SIXTH Edition, 1971. In preparing this edition the Committee has endeavored to include some mention, if only by illustration, of the meters which have come to the attention of the members during the ten years since the preparation of the fifth edition. At the same time most of the types of fluid meters and metering procedures included in preceding editions are still in use, hence it is necessary to continue to include them. Furthermore, there has been a notable increase in the interest and application of fluid meters in the fields of aeronautics and cryogenics, and this has influenced the presentation of parts of the text on differential pressure meters.

During the planning stage for this edition an agreement was made between this Committee and the ASME Standing Committee on Performance Test Codes to include in this edition all of the material in Instruments and Apparatus Supplement Part 5 on Measurement of Quantity of Materials, except Chapter 1 on Weighing Scales. To be sure, much of the material in the other four chapters of Part 5, displacement meters, velocity meters, flow measurement (i.e., differential pressure type meters) and other meters and methods had been included in varying degrees in previous editions. However, in assembling this edition much of these chapters has been included without change, supplemented with additional material as needed to make the treatment as complete as possible.

The arrangement between the two Committees provided that all of the material on application previously included in I and A Part 5-4, Flow Measurement, would be made a Part II of this edition, with some supplemental data. Thus, this Sixth Edition is composed of Parts I and II as a complete volume, and also Part II is available separately under the title "Flow Measurement Application: Part II of Fluid Meters Sixth Edition; (Replaces I and A Part 5-4)." In this way the plant engineer concerned with application does not need to be burdened with the complete volume.

Throughout the text, illustrations are used to show the application of various principles and methods of fluid metering. Some of the illustrations show proprietary equipment. The inclusion of such figures does not constitute nor imply an endorsement of the equipment by this Committee nor the Society. Furthermore, no operating data are given for such proprietary equipment.

The members of the subcommittee which prepared this edition are: Messrs. H. S. Bean, *Chairman;* H. V. Beck, *Vice Chairman;* E. E. Ambrosius, B. T. Arnberg,

*Retired from the Committee before completion of report.
†Deceased.

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R. B. Dowdell, L. P. Emerson, H. J. Evans, Louis Gess,* E. J. Lindahl, J. V. Moore, J. W. Murdock, R. M. Reimer, R. E. Sprenkle, E. F. Wehmann; and K. C. Cotton, L. A. Dodge, J. R. Jordan, A. S. McDaniel and E. L. Upp (for J. V. Moore). The last five, under Mr. Cotton's leadership had the primary responsibility for Part II. The subcommittee was aided by material and comments from other members of the main Committee and other subcommittees.

The committee wishes to acknowledge the assistance of Mr. D. R. Keyser of the Naval Engineering Center in the preparation of parts of several chapters.

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INTRODUCTION

Fluid meters can be divided into two functional groups. One measures primarily quantity; the other measures primarily rate of flow. All fluid meters, however, consist of two distinct parts, each of which has a different function to perform. The first is the primary element, which is in contact with the fluid, resulting in some form of interaction. This interaction may be that of imparting motion to the primary element; the fluid may be accelerated; or there may be an exchange of heat. The second or secondary element translates the interaction between fluid and primary element into volumes, weights or rates of flow and indicates or records the result.

For example, a weigher will have weighing tanks as its primary element and a counter for recording the number of fillings and dumpings as its secondary element. In an orifice meter, the orifice, together with the adjacent part of the pipe and the pressure connections, constitute the primary element, while the secondary element consists of a differential pressure gage together with some sort of mechanism for translating a pressure difference into a rate of flow and indicating the result, in some cases also recording it graphically and integrating with respect to time. The same sort of combination will be observed in other types of meters.

The secondary devices may obviously be varied almost without limit, but the primary elements depend for their operation on a few simple physical principles. Therefore, fluid meters may best be classified with regard solely to the nature of the primary element or to the physical principle involved, and this plan has been adopted here.

The conditions for proper installation and operation, the errors, and other characteristics of the primary element are usually altogether distinct from, and independent of, those of the secondary element, so that it is convenient to treat the two separately so far as is possible. Therefore, throughout the two parts of this edition of the report the discussions are directed particularly toward the primary elements, and secondary elements are included only to the extent necessary for an adequate description of the meter or the principle of the metering procedure.

The material on differential pressure meters occupies a considerable portion of the report. This is due, in part, to the nature of the information to be presented and, in part, to an endeavor to present the development of the theoretical equations required and the correlation of experimentally determined factors in as complete a manner as possible. Moreover, since the compiling of the preceding editions the use of various forms of flow nozzles under sonic flow conditions has become an important tool in aeronautical and aerospace programs. This has led to using a development of equations of flow from the critical or "choked" condition, and thence to the subcritical condition. This procedure of developing equations for flow nozzles (including Venturi tubes), as well as the classical procedure that starts with the incompressible and subcritical states, are presented in order that the reader may have the use of either.

In order that a user may be able to use the final working equations and factors with justifiable confidence, recommendations on construction, installation, and operation are given in considerable detail in Part II.

Numbers in brackets [ ] refer to like numbered references listed at the end of each chapter.

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Part Two

APPLICATION OF FLUID METERS–ESPECIALLY DIFFERENTIAL PRESSURE TYPES

Part One: Theory and Mode of Operation, pp.1-148, of the Sixth Edition of *Fluid Meters: Their Theory and Application* appears only in the complete version of the report.

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INTRODUCTION

Part II of this edition of *Fluid Meters* presents the recommended conditions, procedures and data for measuring the flow of fluids, particularly with the three principal differential pressure meters: the orifice, the flow nozzle, and the Venturi tube. Included here are the factors and other data that may be needed in computing the flows, using the equations developed in Part I. These data and procedures should provide a degree of accuracy suitable for most fluid measurements, whether it be a commercial transfer measurement or associated with a performance guarantee.

The accuracy of flow measurements made with orifices, flow nozzles, or Venturi tubes depends, in part, upon the values of the coefficients of discharge used in computing the flow, and these are affected by the design and quality of construction of the primary elements. Inasmuch as the coefficients presented here are the results of many thousands of tests made in many different laboratories, both in the U.S. and abroad, under many different operating conditions, it follows that certain tolerances must be applied to cover the spread of these data.

Also, other tolerances must be assigned for other variables, such as the measurements of diameters, pressures, and temperatures. The combination of all of these tolerances is then a measure of the final accuracy one can expect to achieve for a flow measurement made under a particular set of operating conditions. If this tolerance is larger than acceptable for a particular application, then the primary element together with the flow section in which it is mounted, should be individually calibrated.

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Chapter II-I

Conversion Factors, Constants and Data on Fluids and Materials

II-I-I Pursuant to statute authority, the following *exact* conversion factors between the international metric units (S.I. units) and the foot-pound units have been adopted and published by the National Bureau of Standards [1, 2]:

1 foot = 0.3048 meter
1 inch = 0.0254 meter = 2.54 centimeter
1 pound = 0.453 592 37 kilogram
1 cubic foot = 28.316 846 592 cubic decimeters
= 28.316 846 592 liters

In addition, the standard atmospheric pressure and acceleration due to gravity are [3], respectively,

$$p_o = 101\ 325\ \text{Newtons/m}^2 = 14.695\ 95\ \text{lb}_f/\text{in.}^2$$
$$g_o = 980.665\ \text{cm/sec}^2 = 32.174\ 06\ \text{ft/sec}^2$$

From the exact relations given above the following conversion factors are derived:

1. For density:
$\rho\ (\text{gram}_m/\text{cc}) \times 62.427\ 96 = \rho\ (\text{lb}_m/\text{ft}^3$ [II-1-1]
2. For absolute viscosity:
$\mu\ (\text{gram}_m/\text{sec-cm}) \times 0.067\ 197 = \mu\ (\text{lb}_m/\text{sec-ft})$ (II-I-2)
3. For kinematic viscosity:
$\nu\ (\text{cm}^2/\text{sec}) \times 0.001\ 076\ 39 = \nu\ (\text{ft}^2/\text{sec})$ (II-I-3)

II-I-2 A graphical comparison of temperature scales is given by Fig. II-I-1. There are tables of equivalents in many handbooks, and conversions of temperatures between Celsius and Fahrenheit scales can be made by the relation:

degrees C = 5/9 (degrees Fahrenheit − 32) (II-I-4).

II-I-3 Frequently pressures measured with liquid manometers must be converted into pounds per square inch. In many such cases it may be sufficient to use a factor based upon some average temperature for the manometer fluid and to take no further account of the effects of temperature. Assuming a manometer temperature of 68 F (20 C), such factors are

In. of mercury × 0.4893
= Lb_f/per sq in (II-I-5)
In. of mercury under water × 0.4532
= Lb_f/per sq in (II-I-6)
In. of water × 0.0361
= Lb_f/per sq in (II-I-7)
In. of mercury × 13.57
= In. of water at the same temperature between 45 F and 81F (II-I-8)

In other cases, the conditions may justify the use of a more exact value to be obtained by taking account of the manometer temperature. Such values may be obtained from Fig. II-I-2.

Note: The factors in Fig. II-I-2 give pressures in pounds-force per sq in. at local gravity. If, as a further refinement, it is desired to obtain pounds-force at standard gravity, it will be necessary to multiply the values by g/g_o where g is the value of the local acceleration of gravity and g_o = 32.174 ft/sec^2, the standard value of gravitational acceleration.

II-I-4 When differential pressure meters are used to meter fluids at temperatures considerably above or below ordinary room temperatures, the thermal expansion of the primary element must be taken into account. Figure II-I-3 gives the area expansion factors, F_a, to be used when the material of the primary

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element is steel, steel alloy, bronze or monel. The factor is to be used as a multiplier of a *or* d^2 in such equations as (I-5-33) and the working equations given later in Par. II-III-39.

II-I-5 Values of the velocity of approach factor, $E = 1/\sqrt{1 - \beta^4}$, used with many differential pressure meters, are given in Table II-I-1 and Figs. II-I-4(a) and (b).

Density, Viscosity and Compressibility Data

II-I-6 The density of mercury, Table II-I-2, is from data published in 1964 by tne National Physical Laboratory, England [4]. The data for Table II-I-3 on the density of dry air are from National Bureau of Standards Circular 564 [5].

II-I-7 The density of water Table II-I-4 is from equations given in the *1967 ASME Steam Tables* [6]. At temperatures and pressures intermediate beween those of the table, linear interpolation may be used. For all data on the density, specific volume, compressibility and other thermodynamic properties of steam, the *1967 ASME Steam Tables* should be used, and such data are not repeated here.

II-I-8 Values of the viscosity of water are given in Figs. II-I-5 and II-I-6, and those for the viscosity of steam in Fig. II-I-7 are from the ASME Steam Tables. The data on the viscosities of nonhydrocarbon gases, Figs. II-I-8 and II-I-9, are from Circular 564, while those for methane and other hydrocarbon gases are from a report by N.L. Carr [5, 7]. The viscosity of most gases increases slightly with increasing pressure, but the rate of increase is not the same for all gases, as shown by the upper corner insert of Fig. II-I-8.

The most common method of determining the viscosity of liquids is to observe the time in seconds required for a definite volume to pass through a small aperture or a short capillary tube. From this observed

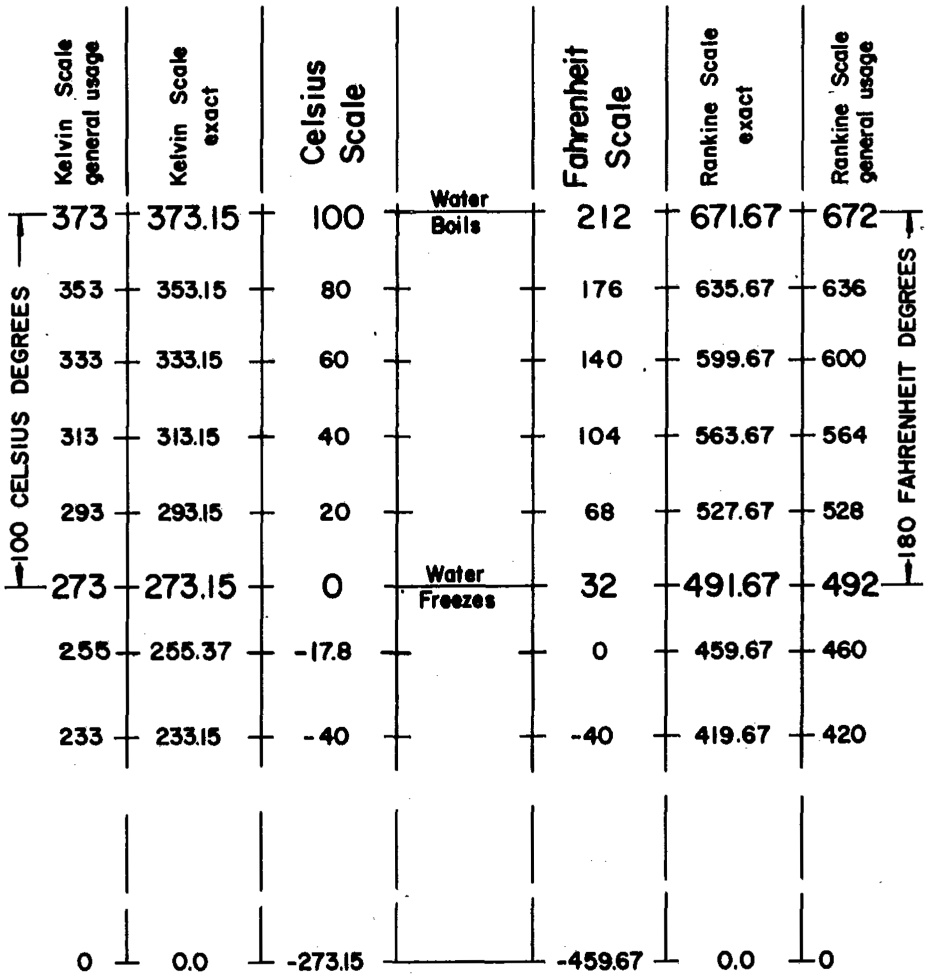

FIG. II-I-1 TEMPERATURE SCALES

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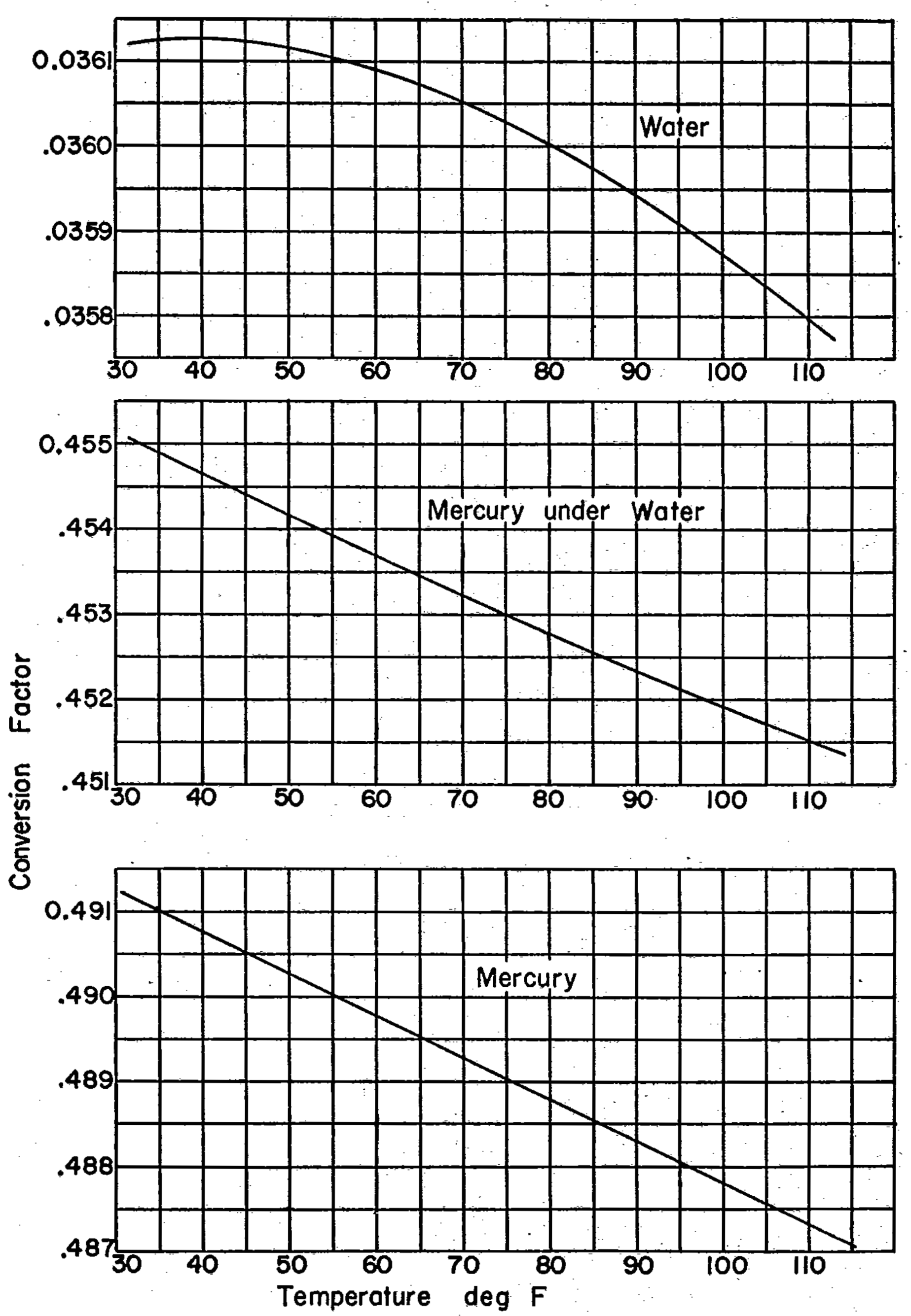

FIG. II-I-2 FACTORS FOR CONVERTING INCHES OF FLUID TO PSI AT LOCAL GRAVITY

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time, t, in seconds, the kinematic viscosity is computed by an empirical equation. The empirical equations applying to the four most commonly used viscosimeters give values in terms of *stokes*, i.e., cm^2/sec. These equations are as follows:

1. For Saybolt Universal [8, 9]:

$$\nu\ (cm^2/sec) = 0.00226t - 1.95/t \qquad \text{(II-I-9)}$$

when $32 < t < 100$

$$\nu\ (cm^2/sec) = 0.00220t - 1.35/t \qquad \text{(II-I-10)}$$

when $t > 100$

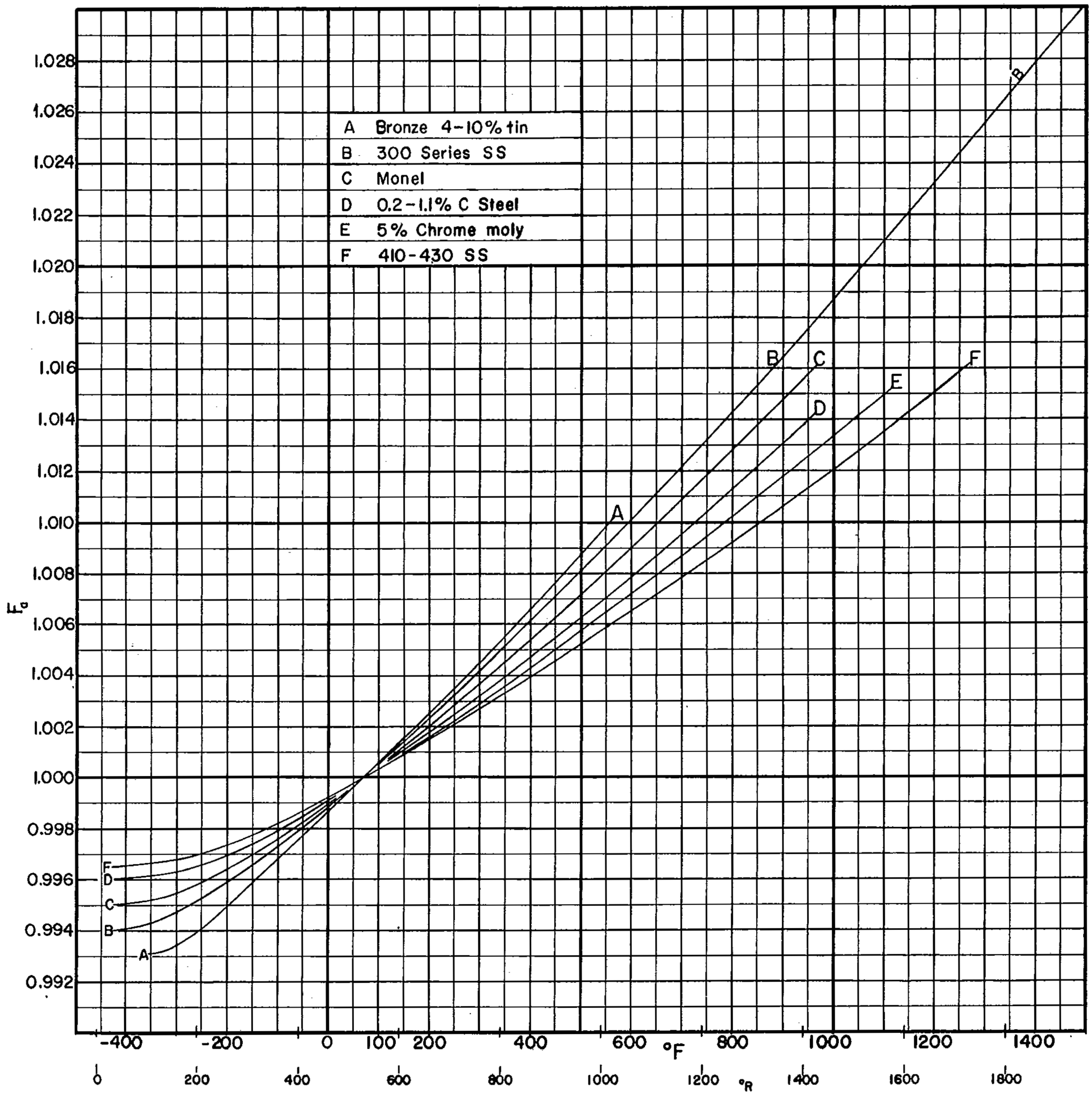

FIG. II-I-3 AREA FACTORS, F_a, FOR THE THERMAL EXPANSION OF PRIMARY ELEMENTS

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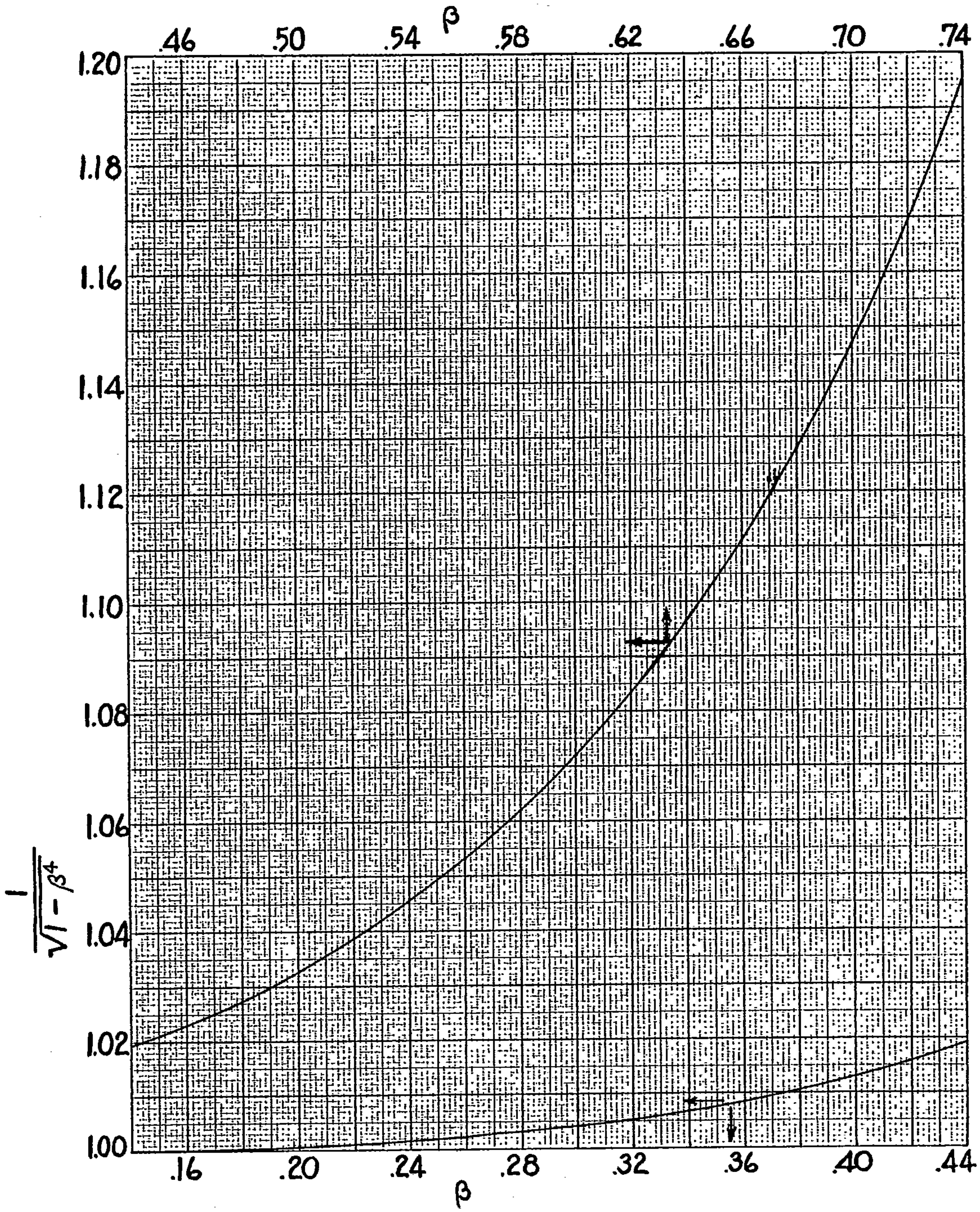

FIG. II-I-4(a) VALUES OF VELOCITY OF APPROACH FACTOR, $E = 1/\sqrt{1-\beta^4}$

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2. For Saybolt Furol [10, 11]:

$$\nu\ (\text{cm}^2/\text{sec}) = 0.0224t - 1.84/t \quad \text{(II-I-11)}$$

when $25 < t < 40$

$$\nu\ (\text{cm}^2/\text{sec}) = 0.0216t - 0.60/t \quad \text{(II-I-12)}$$

when $t > 40$

3. For Redwood Standard No. 1 [11]:

$$\nu\ (\text{cm}^2/\text{sec}) = 0.00260t - 1.79/t \quad \text{(II-I-13)}$$

when $34 < t < 100$

$$\nu\ (\text{cm}^2/\text{sec}) = 0.00247t - 0.50/t \quad \text{(II-I-14)}$$

when $t > 100$

4. For Engler [9]:

$$\nu\ (\text{cm}^2/\text{sec}) = 0.00147t - 3.74/t \quad \text{(II-I-15)}$$

Instead of using the equations, values of ν may be obtained from the curves of Fig. II-I-10.

Redwood and Engler viscosimeters are used very little in this country.

Attention is called to the fact that the values of ν obtained by any of the above equations are at best only close approximations and sometimes may be in

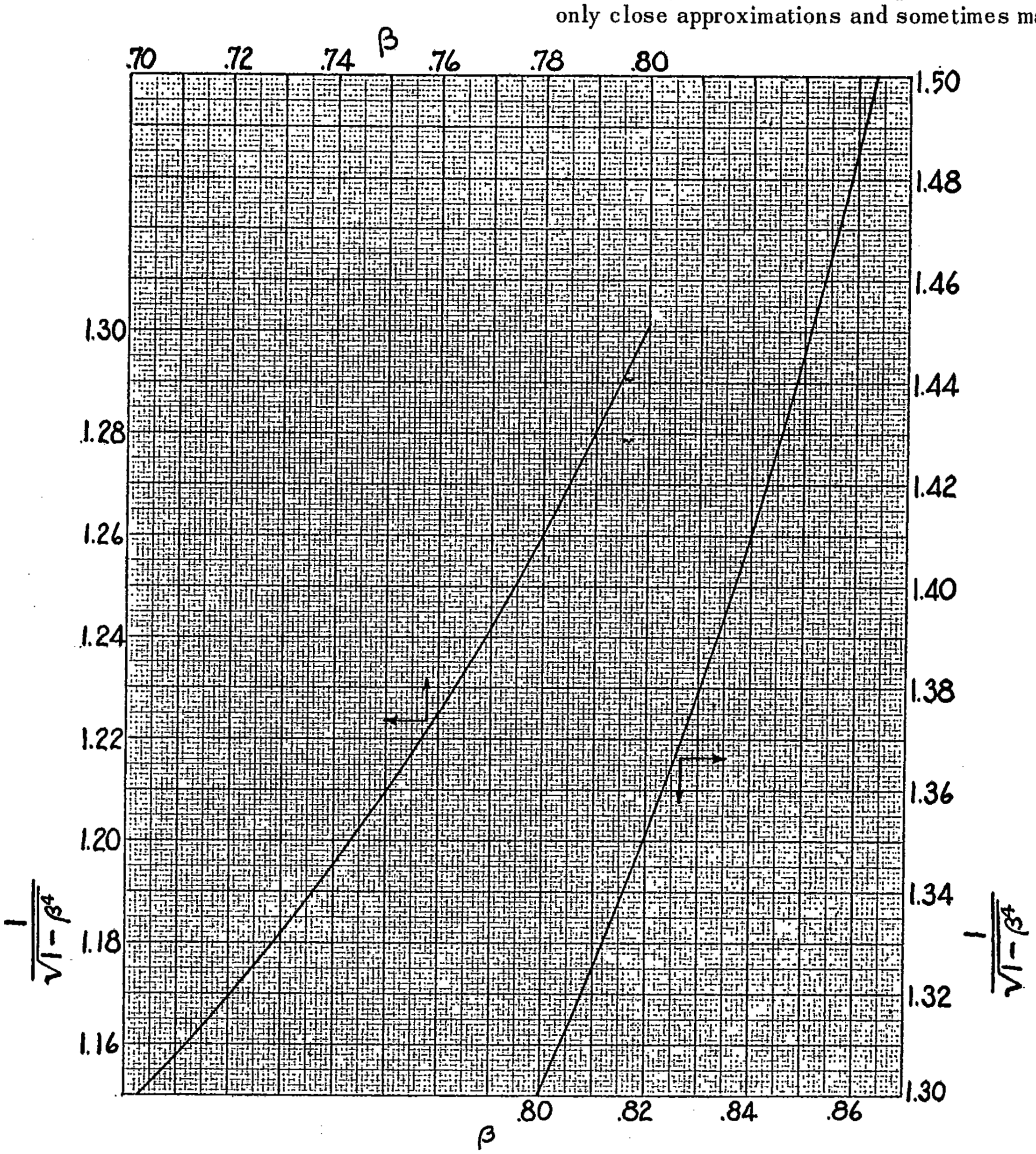

FIG. II-I-4(b) VALUES OF VELOCITY OF APPROACH FACTOR, $E = 1/\sqrt{1-\beta^4}$

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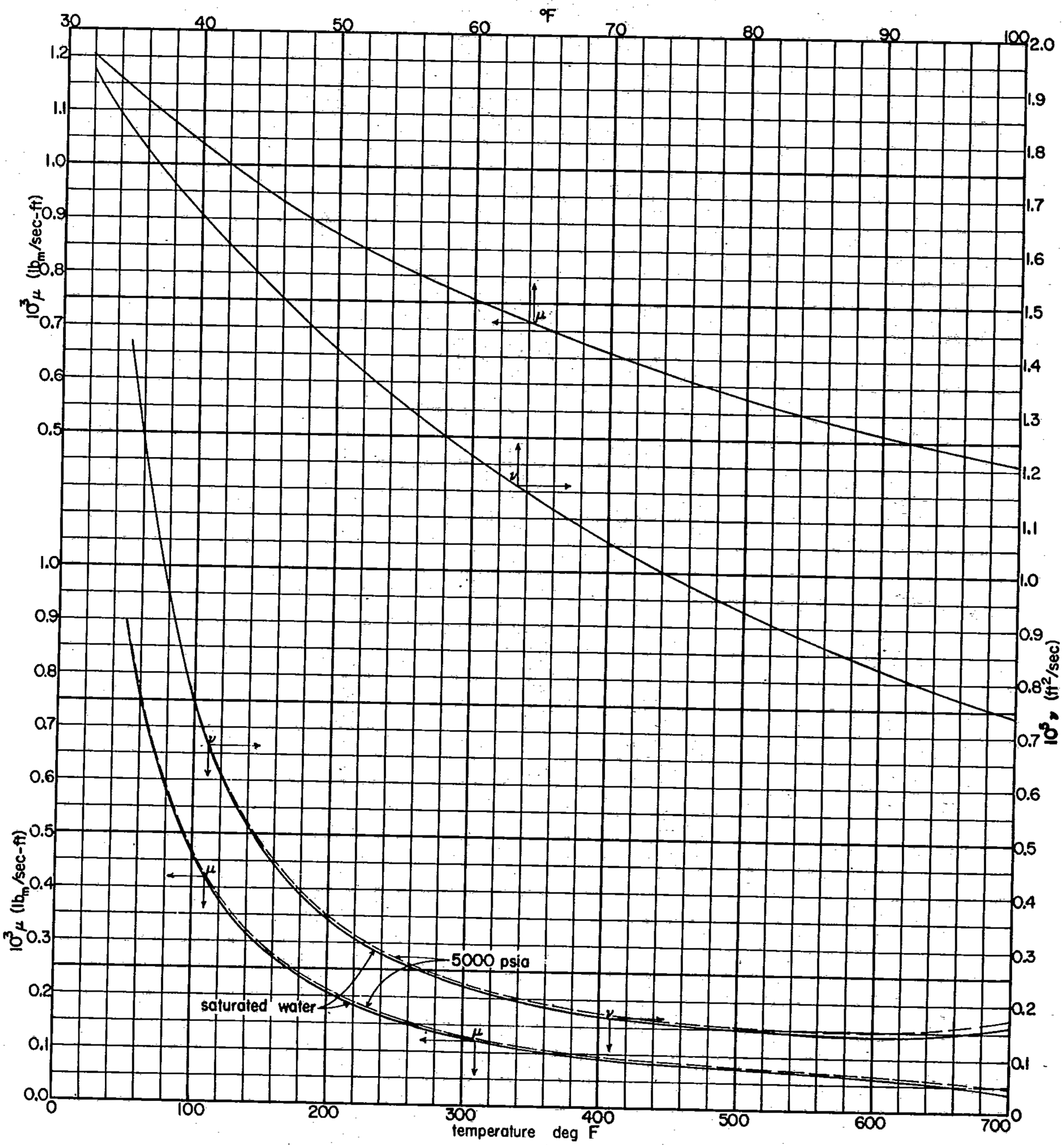

FIG. II-I-5 VISCOSITY OF WATER, μ, IN LB$_m$/SEC-FT AND KINEMATIC VISCOSITY, ν, IN FT^2/SEC

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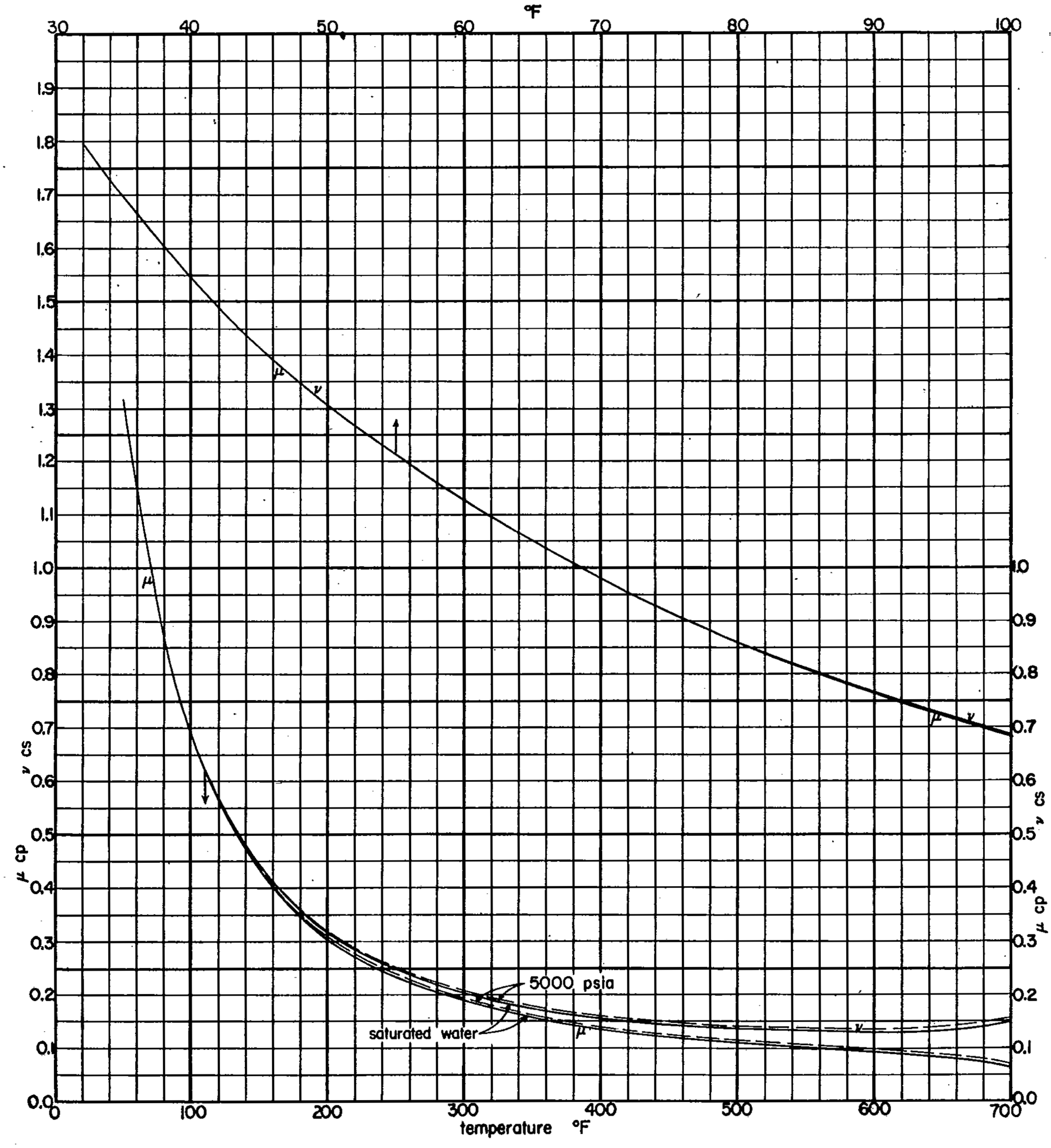

FIG. II-I-6 VISCOSITY OF WATER, μ, IN CENTIPOISE AND KINEMATIC VISCOSITY, ν, IN CENTISTOKE

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error by 2 to 3 per cent. The principal reason for this is that the manner in which the viscosimeters are used does not provide for a sufficiently accurate measurement of the temperature and, therefore, also of the volume of the liquid as it flows through the metering passage. The fact that calculations at the time breaks of 100 sec or 40 sec may not give identical results with the two equations for a particular instrument will cause no significant error. There are tables that may be used in place of the equations [8, 9]. Should it be necessary to determine more exact values of ν, reference should be made to some of the literature on viscosity determinations [12–15].

As stated in Chapter I-3, the name of the metric unit for the coefficient of absolute viscosity, or dynamic viscosity, is "poise" and that for kinematic viscosity is "stoke." The corresponding quantities in ft-lb units have no generally accepted names. In industrial practice in this country as well as in technical laboratories, the use of viscosity values in poise (or centipoise) and stokes (or centistokes) in place of the ft-lb units has become common. For this reason most of the viscosity data given here are in both systems of units.

II-I-9 The curves giving the compressibilities of air, hydrogen and carbon dioxide, Figs. II-I-11, II-I-12 and II-I-13, are from data in Circular 564 [16]. Figure II-I-14, on the compressibility of methane, is based on data correlated by Beitler, Darrow and Zimmerman, combined with data given by Din [16, 17, 18, 19].

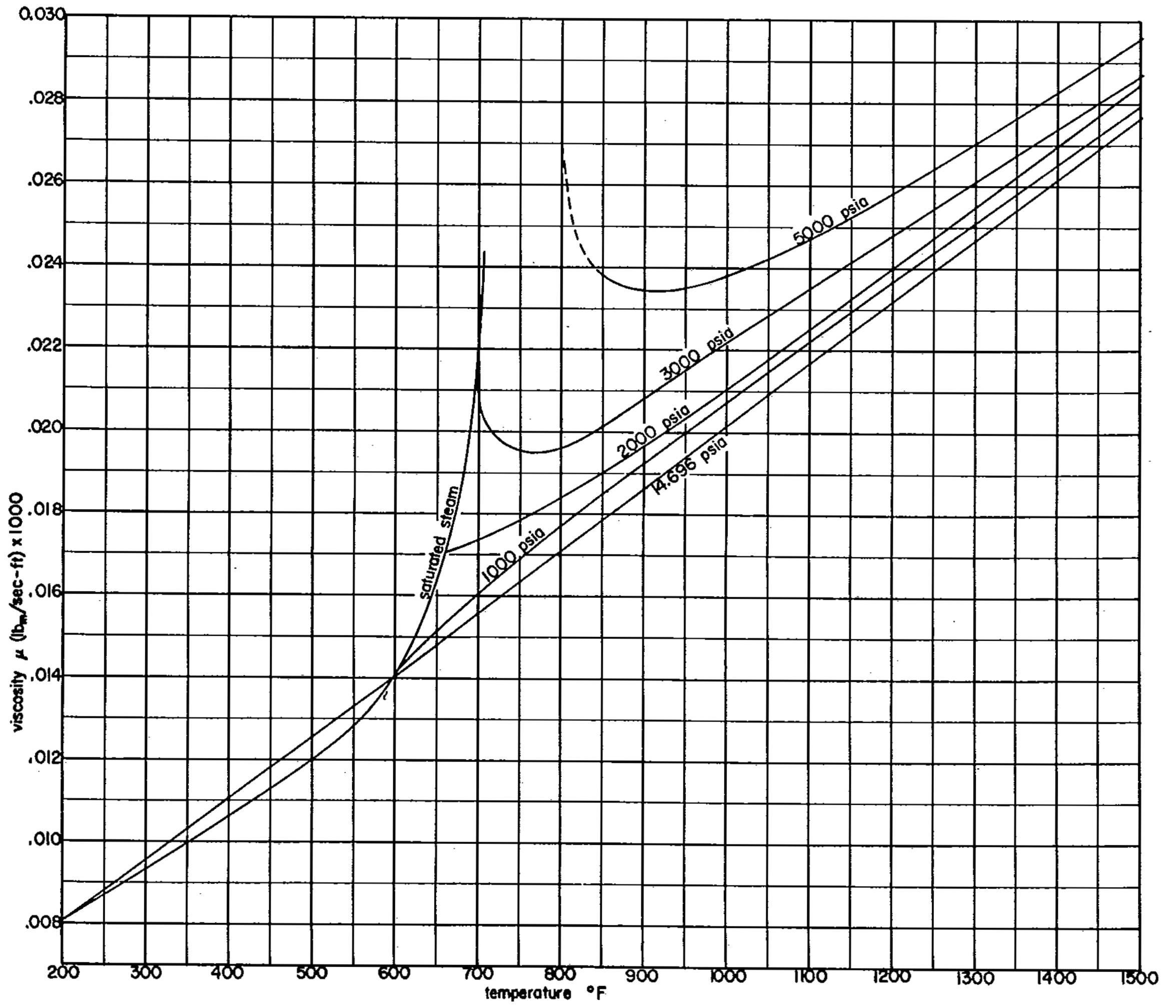

FIG. II-I-7 VISCOSITY OF STEAM, μ, IN LB_m/SEC-FT

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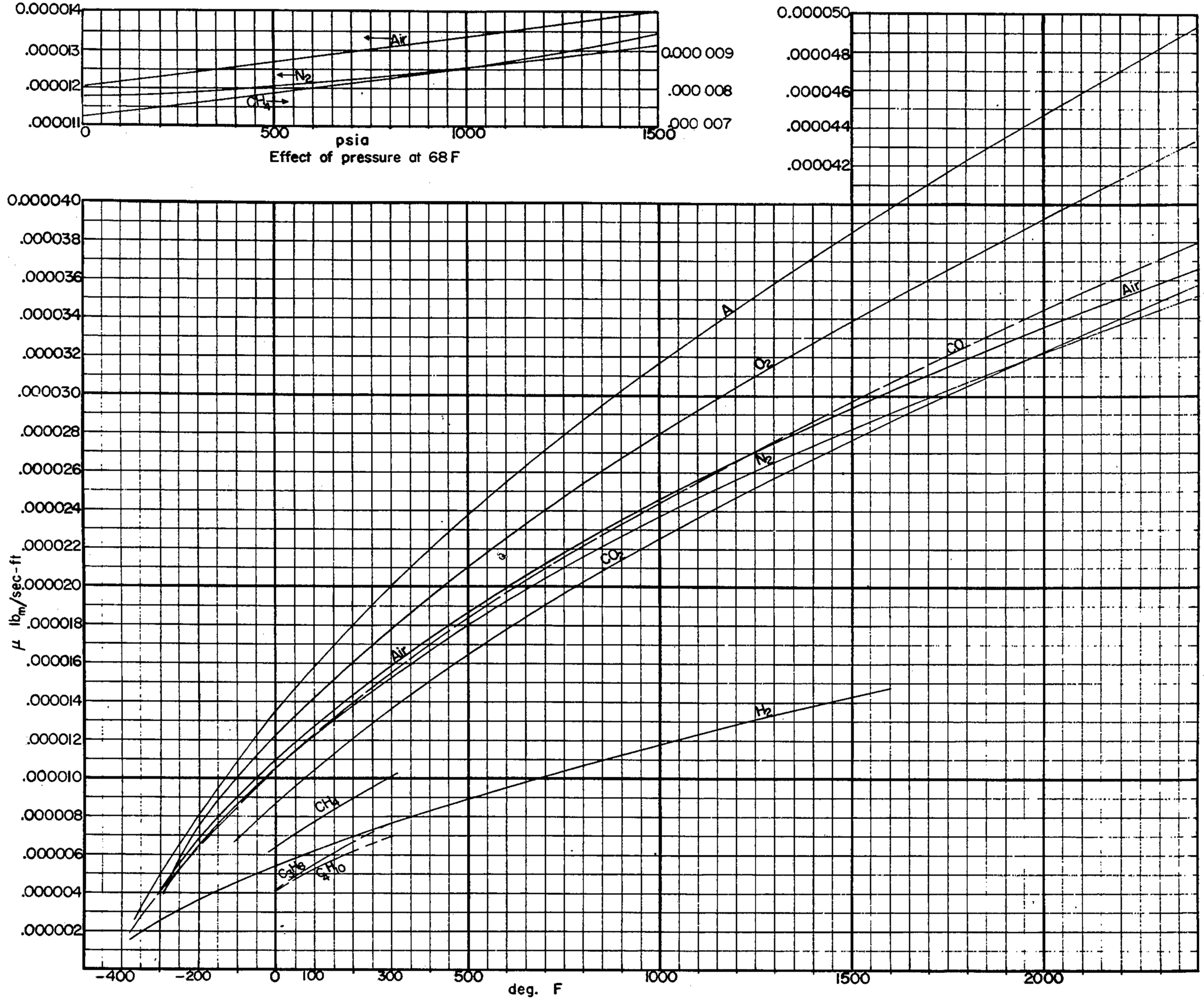

FIG. II-I-8 VISCOSITY OF GASES, μ, IN LB_m/SEC-FT

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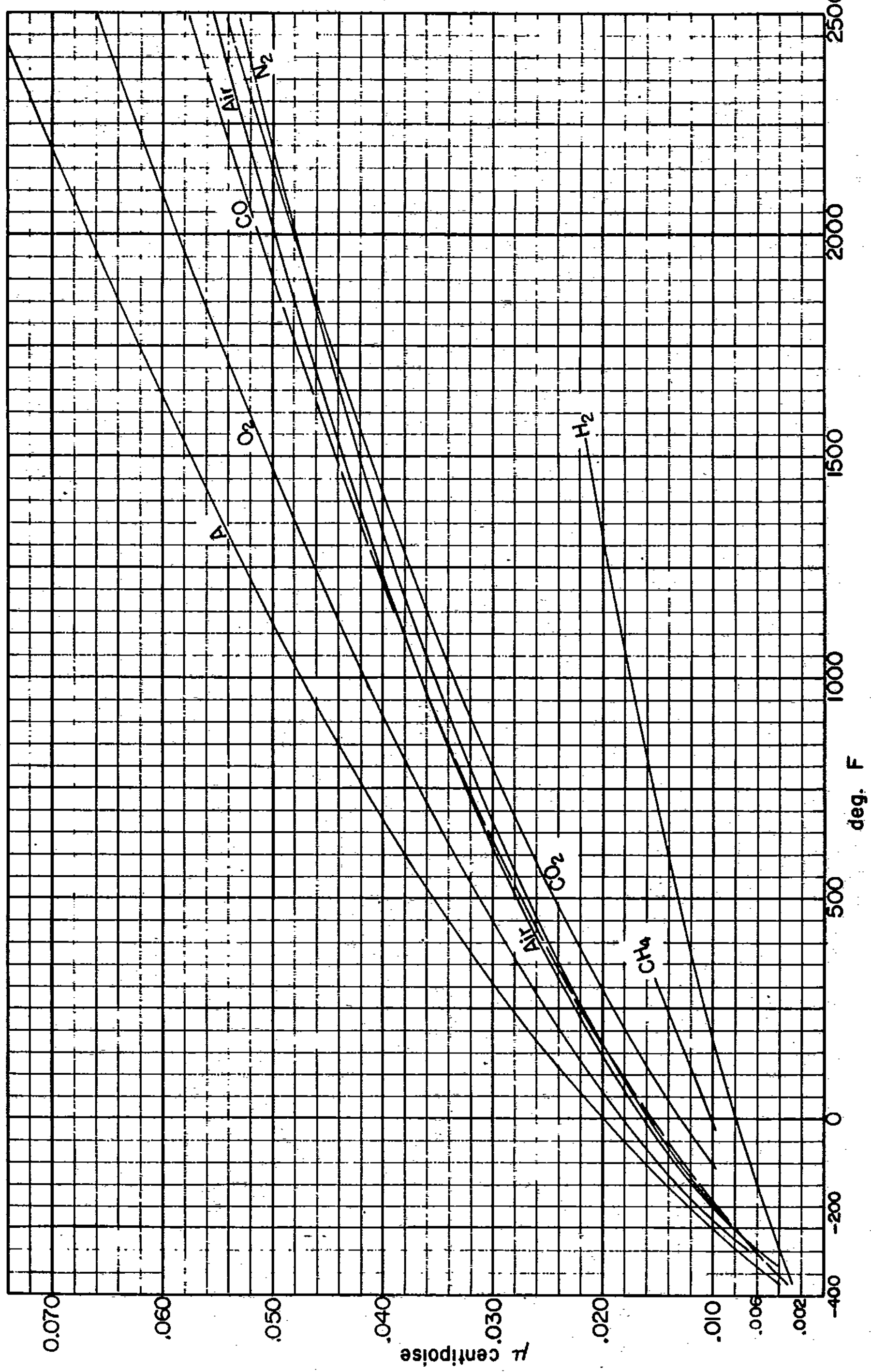

FIG. II-1-9 VISCOSITY OF GASES AT 1 ATMOSPHERE, μ IN CENTIPOISE

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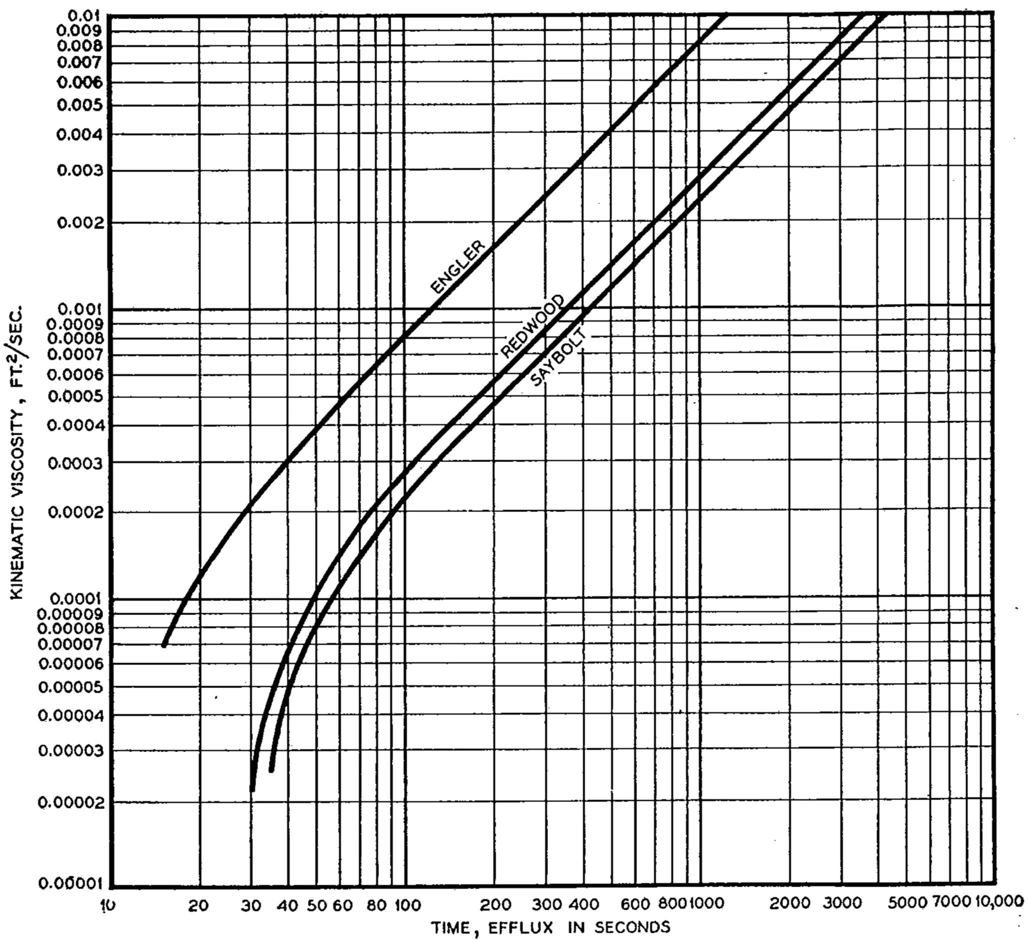

FIG. II-I-10 CHART FOR OBTAINING KINEMATIC VISCOSITY FROM FLOW TIME FOR SAYBOLT, REDWOOD AND ENGLER VISCOSIMETERS

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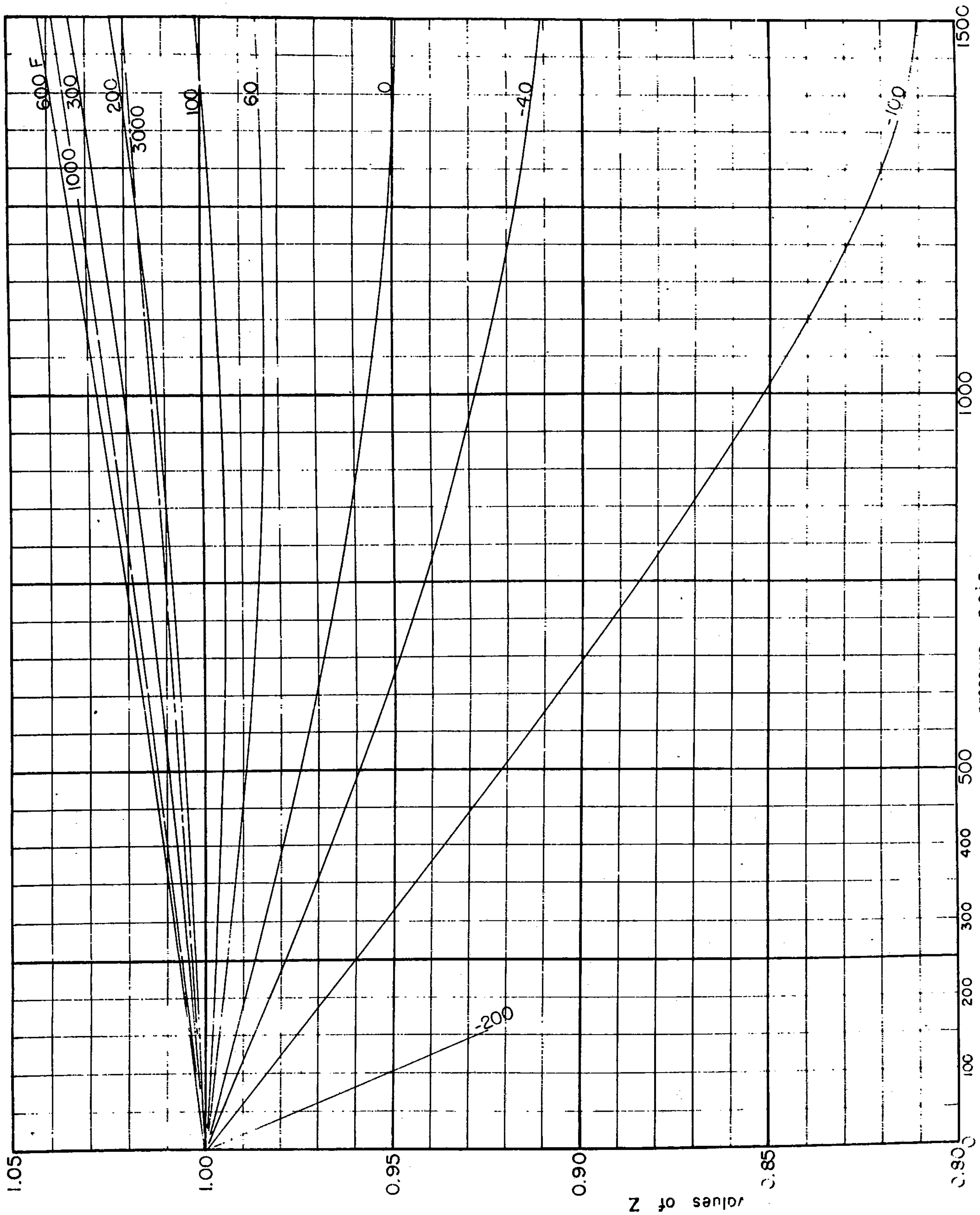

FIG. II-I-11 COMPRESSIBILITY FACTOR, Z, OF DRY AIR

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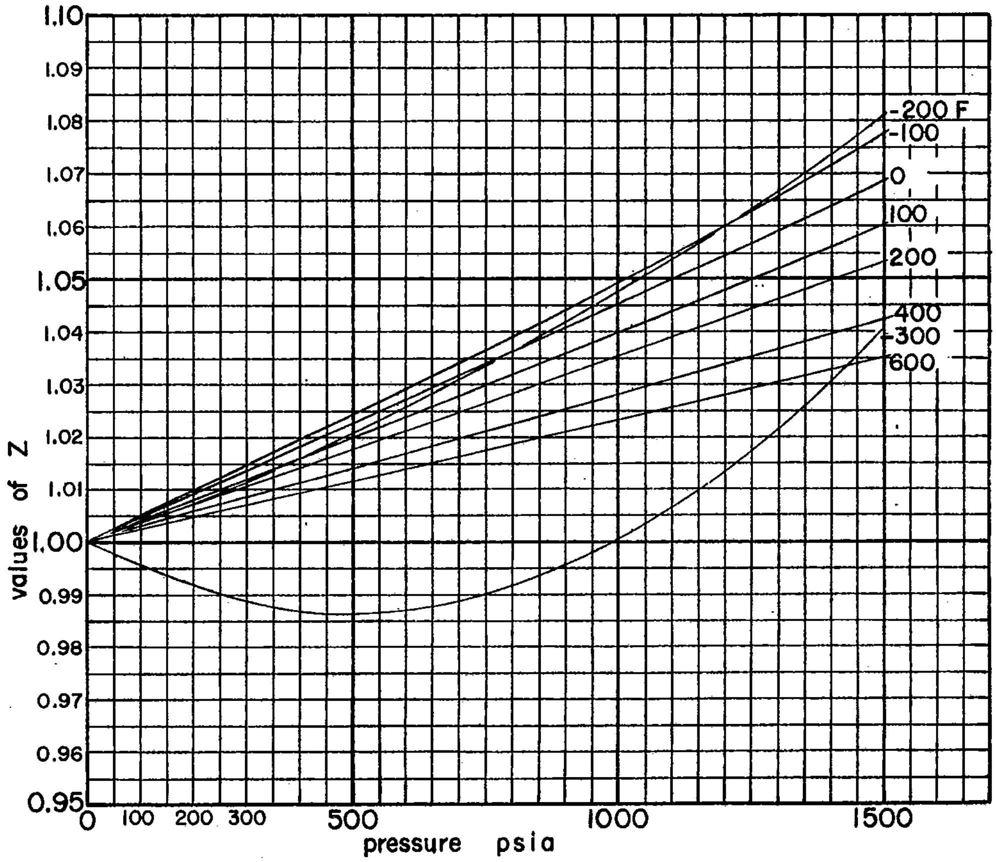

FIG. II-I-12 COMPRESSIBILITY FACTOR, *Z*, OF NORMAL HYDROGEN

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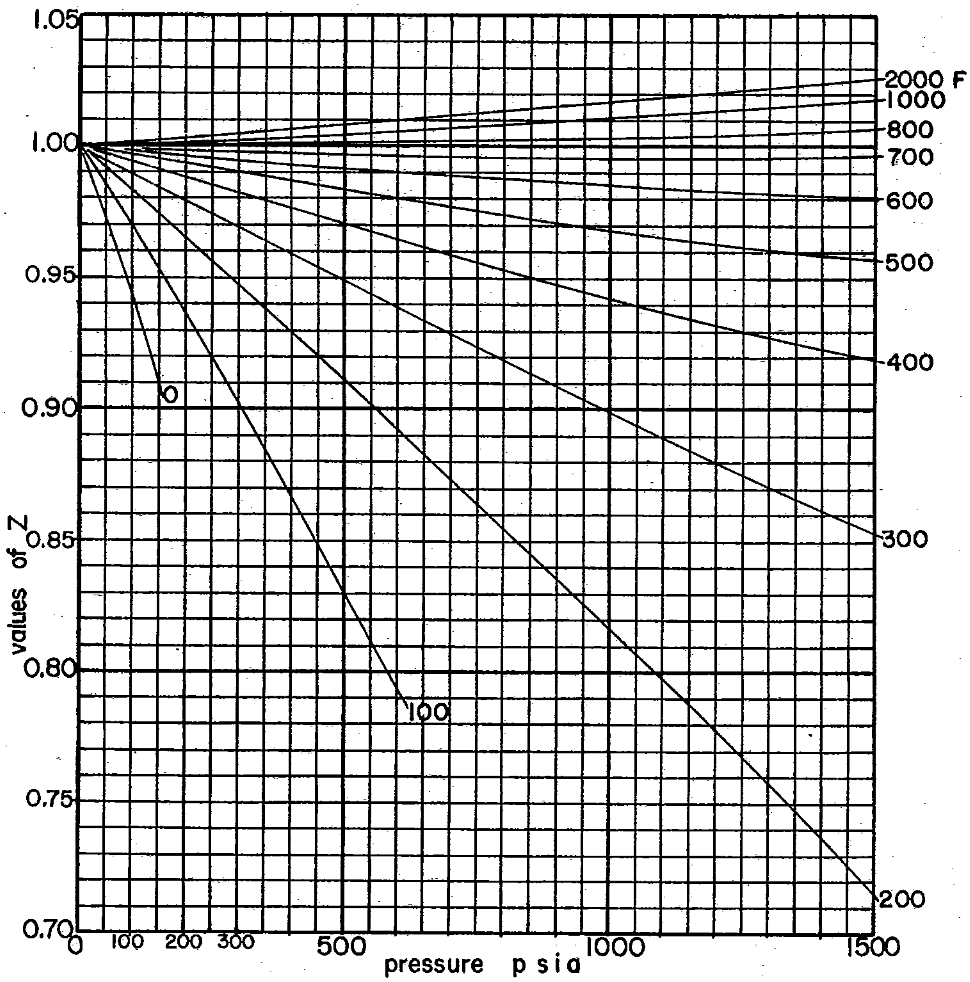

FIG. II-I-13 COMPRESSIBILITY FACTOR, Z, OF CARBON DIOXIDE

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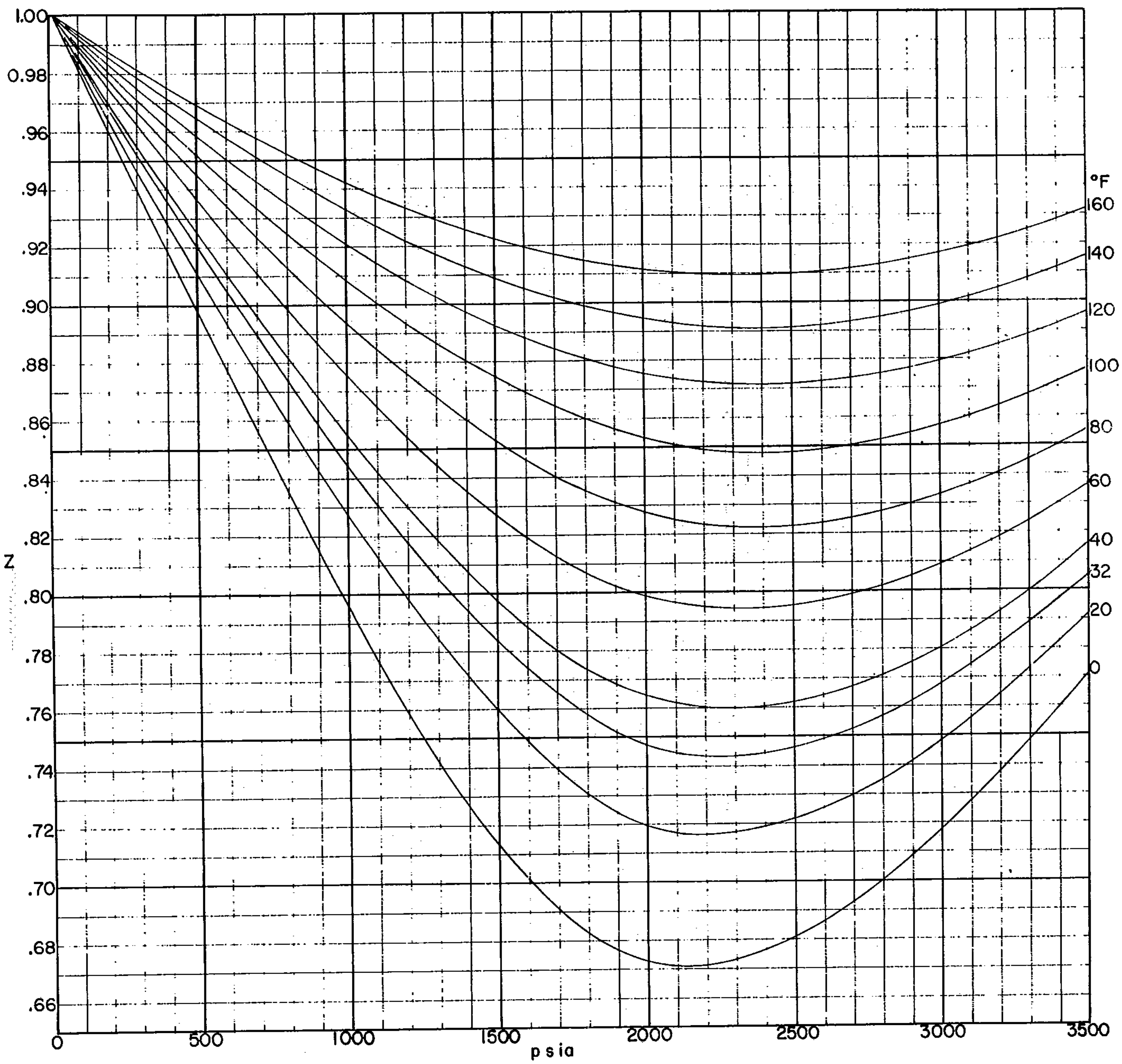

FIG. II-I-14 COMPRESSIBILITY FACTOR, *Z*, OF METHANE

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Table II-I-1. Velocity of Approach Factors

β	$\frac{1}{\sqrt{1-\beta^4}}$	β	$\frac{1}{\sqrt{1-\beta^4}}$	β	$\frac{1}{\sqrt{1-\beta^4}}$	β	$\frac{1}{\sqrt{1-\beta^4}}$
0.100	1.000 04	0.625	1.086 30	0.725	1.175 47	0.770	1.241 81
.150	1.000 25	.630	1.089 48	.726	1.176 72	.771	1.243 57
.200	1.000 80	.635	1.092 77	.727	1.177 97	.772	1.245 34
.220	1.001 17	.640	1.096 17	.728	1.179 23	.773	1.247 12
.240	1.001 66	.645	1.099 68	.729	1.180 50	.774	1.248 92
.260	1.002 29	.650	1.103 31	.730	1.181 78	.775	1.250 73
.280	1.003 08	.652	1.104 79	.731	1.183 07	.776	1.252 56
.300	1.004 07	.654	1.106 30	.732	1.184 37	.777	1.254 41
.320	1.005 28	.656	1.107 82	.733	1.185 67	.778	1.256 27
.340	1.006 74	.658	1.109 37	.734	1.186 99	.779	1.258 14
.350	1.007 58	.660	1.110 93	.735	1.188 32	.780	1.260 03
.360	1.008 50	.662	1.112 52	.736	1.189 66	.781	1.261 94
.370	1.009 50	.664	1.114 13	.737	1.191 01	.782	1.263 86
.380	1.010 59	.666	1.115 76	.738	1.192 36	.783	1.265 80
.390	1.011 77	.668	1.117 41	.739	1.193 73	.784	1.267 76
.400	1.013 05	.670	1.119 09	.740	1.195 11	.785	1.269 73
.410	1.014 43	.672	1.120 78	.741	1.196 50	.786	1.271 72
.420	1.015 93	.674	1.122 50	.742	1.197 90	.787	1.273 72
.430	1.017 54	.676	1.124 25	.743	1.199 31	.788	1.275 75
.440	1.019 28	.678	1.126 02	.744	1.200 73	.789	1.277 79
.450	1.021 15	.680	1.127 81	.745	1.202 16	.790	1.279 85
.460	1.023 16	.682	1.129 63	.746	1.203 60	.791	1.281 92
.470	1.025 32	.684	1.131 47	.747	1.205 05	.792	1.284 02
.480	1.027 64	.686	1.133 33	.748	1.206 52	.793	1.286 13
.490	1.030 13	.688	1.135 23	.749	1.207 99	.794	1.288 26
.500	1.032 79	.690	1.137 15	.750	1.209 48	.795	1.290 41
.510	1.035 64	.692	1.139 09	.751	1.210 98	.796	1.292 58
.520	1.038 69	.694	1.141 06	.752	1.212 50	.797	1.294 77
.530	1.041 95	.696	1.143 06	.753	1.214 02	.798	1.296 97
.540	1.045 43	.698	1.145 09	.754	1.215 55	.799	1.299 20
.550	1.049 15	.700	1.147 15	.755	1.217 10	.800	1.301 45
.555	1.051 10	.702	1.149 23	.756	1.218 65	.801	1.303 72
.560	1.053 12	.704	1.151 35	.757	1.220 22	.802	1.306 00
.565	1.055 20	.706	1.153 50	.758	1.221 81	.803	1.308 31
.570	1.057 36	.708	1.155 67	.759	1.223 40	.804	1.310 64
.575	1.059 58	.710	1.157 88	.760	1.225 01	.805	1.312 99
.580	1.061 88	.712	1.160 12	.761	1.226 63	.806	1.315 36
.585	1.064 26	.714	1.162 39	.762	1.228 26	.807	1.317 76
.590	1.066 71	.716	1.164 69	.763	1.229 91	.808	1.320 18
.595	1.069 24	.718	1.167 03	.764	1.231 57	.809	1.322 61
.600	1.071 86	.720	1.169 40	.765	1.233 24	.810	1.325 08
.605	1.074 56	.721	1.170 59	.766	1.234 93	.811	1.327 56
.610	1.077 36	.722	1.171 80	.767	1.236 63	.812	1.330 07
.615	1.080 24	.723	1.173 02	.768	1.238 34	.813	1.332 60
.620	1.083 22	.724	1.174 24	.769	1.240 07	.814	1.335 15

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Table II-I-2 Density of Mercury

Temperature (°F)	ρ (lb$_m$/ft^3)	Temperature (°F)	ρ (lb$_m$/ft^3)	Temperature (°F)	ρ (lb$_m$/ft^3)	Temperature (°F)	ρ (lb$_m$/ft^3)
− 5	851.888 14	35	848.456 54	75	845.047 35	115	841.656 89
− 4	.802 05	36	.371 01	76	844.962 45	116	.571 98
− 3	.715 98	37	.286 11	77	.876 92	117	.487 71
− 2	.629 92	38	.200 59	78	.792 02	118	.403 43
− 1	.543 88	39	.115 06	79	.707 12	119	.318 53
0	.457 85	40	.029 53	80	.622 21	120	.234 25
1	.371 84	41	847.944 01	81	.537 31	121	.149 97
2	.285 84	42	.858 48	82	.452 41	122	.065 07
3	.199 63	43	.772 95	83	.367 51	123	840.980 79
4	.113 90	44	.688 05	84	.282 61	124	.896 51
5	.027 95	45	.602 53	85	.197 70	125	.811 61
6	850.942 01	46	.517 00	86	.112 80	126	.727 33
7	.856 09	47	.432 10	87	.027 90	127	.643 06
8	.770 19	48	.346 57	88	843.943 00	128	.558 15
9	.684 30	49	.261 04	89	.858 10	129	.473 88
10	.598 43	50	.175 52	90	.773 19	130	.389 60
11	.512 57	51	.089 99	91	.688 29	131	.305 32
12	.426 73	52	.005 09	92	.604 01	132	.221 04
13	.340 90	53	846.919 56	93	.519 11	133	.136 77
14	.255 09	54	.834 66	94	.434 21	134	.052 49
15	.169 56	55	.749 13	95	.349 31	135	839.968 21
16	.084 04	56	.664 23	96	.264 41	136	.883 93
17	849.998 51	57	.578 71	97	.179 50	137	.799 65
18	.912 36	58	.493 80	98	.095 23	138	.715 38
19	.826 83	59	.408 28	99	.010 32	139	.631 10
20	.740 68	60	.323 38	100	842.925 42	140	.546 82
21	.655 16	61	.238 47	101	.841 14	145	.125 43
22	.569 63	62	.152 95	102	.756 24	150	838.704 67
23	.483 48	63	.068 05	103	.671 96	155	.283 90
24	.397 95	64	845.983 14	104	.587 06	160	837.863 76
25	.312 43	65	.897 62	105	.502 16	165	.443 62
26	.226 90	66	.812 72	106	.417 88	170	.023 48
27	.141 37	67	.727 19	107	.332 98	175	836.603 97
28	.055 22	68	.642 29	108	.248 70	180	.184 45
29	848.969 70	69	.557 39	109	.163 80	185	835.765 56
30	.884 17	70	.472 48	110	.079 52	190	.346 67
31	.798 64	71	.387 58	111	841.994 62	195	834.927 78
32	.713 12	72	.302 68	112	.910 34	200	.508 88
33	.627 59	73	.217 15	113	.825 44	205	.090 62
34	.542 07	74	.132 25	114	.741 16	210	833.672 97
						212	833.505 67

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Table II-I-3 Density, ρ, of Dry Air at a Pressure of 1 Atm (14.696 psia)

Temperature (°R)	Temperature (°F)	ρ (lb_m/ft^3)	Temperature (°R)	Temperature (°F)	ρ (lb_m/ft^3)
189.67	-270	0.212 663	559.67	100	0.070 890
199.67	-260	.201 515	569.67	110	.069 639
209.67	-250	.191 546	579.67	120	.068 436
			589.67	130	.067 274
219.67	-240	0.182 545	599.67	140	.066 144
229.67	-230	.174 376			
239.67	-220	.166 909	609.67	150	0.065 062
249.67	-210	.160 064	619.67	160	.064 013
259.67	-200	.156 141	629.67	170	.062 996
			639.67	180	.062 011
269.67	-190	0.147 972	649.67	190	.061 058
279.67	-180	.142 612			
289.67	-170	.137 583	659.67	200	0.060 122
299.67	-160	.132 933	669.67	210	.059 226
309.67	-150	.128 583	679.67	220	.058 354
			689.67	230	.057 507
319.67	-140	0.124 522	699.67	240	.056 683
329.67	-130	.120 623			
339.67	-120	.117 112	709.67	250	0.055 884
349.67	-110	.113 722	719.67	260	.055 101
359.67	-100	.110 541	729.67	270	.054 350
			739.67	280	.053 616
369.67	- 90	0.107 530	749.67	290	.052 897
379.67	- 80	.104 681			
389.67	- 70	.101 976	759.67	300	0.052 203
399.67	- 60	.099 401	769.67	310	.051 525
409.67	- 50	.096 963	779.67	320	.050 863
			789.67	330	.050 217
419.67	- 40	0.094 639	799.67	340	.049 596
429.67	- 30	.092 427			
439.67	- 20	.090 328	809.67	350	0.048 966
449.67	- 10	.088 294	819.67	360	.048 377
459.67	- 0	.086 365	829.67	370	.047 796
			839.67	380	.047 231
469.67	10	0.084 524	849.67	390	.046 674
479.67	20	.082 757			
489.67	30	.081 045			
491.67	32	.080 722 3			
499.67	40	.079 431			
509.67	50	0.077 865			
519.67	60	.076 355			
529.67	70	.074 918			
539.67	80	.073 522			
549.67	90	.072 182	859.67	400	0.046 125

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Table II-I-4 Density of Saturated and Compressed Liquid Water (lb_m/ft^3)

Temperature (°F)	Pressure (psia)						
	Saturated	500	1000	1500	2000	2500	3000
32	62.4140	62.5217	62.6288	62.7355	62.8415	62.9470	63.0519
33	.4167	.5240	.6308	.7370	.8426	.9477	.0522
34	.4191	.5260	.6324	.7382	.8434	.9480	.0521
35	62.4212	62.5277	62.6336	62.7390	62.8438	62.9481	63.0517
36	.4229	.5289	.6345	.7395	.8439	.9478	.0510
37	.4242	.5299	.6351	.7397	.8437	.9471	.0500
38	.4252	.5305	.6353	.7395	.8432	.9462	.0487
39	.4258	.5308	.6352	.7390	.8423	.9450	.0471
40	62.4261	62.5307	62.6348	62.7383	62.8412	62.9435	63.0453
41	.4261	.5304	.6341	.7372	.8397	.9417	.0431
42	.4257	.5297	.6330	.7358	.8380	.9396	.0407
43	.4251	.5287	.6317	.7342	.8360	.9373	.0380
44	.4241	.5274	.6301	.7322	.8337	.9347	.0350
45	62.4229	62.5258	62.6282	62.7300	62.8312	62.9318	63.0318
46	.4213	.5239	.6260	.7275	.8283	.9286	.0284
47	.4194	.5218	.6235	.7247	.8252	.9252	.0246
48	.4173	.5193	.6208	.7216	.8219	.9216	.0207
49	.4149	.5166	.6178	.7183	.8183	.9177	.0165
50	62.4122	62.5136	62.6145	62.7148	62.8144	62.9135	63.0120
51	.4092	.5104	.6110	.7109	.8103	.9091	.0073
52	.4059	.5068	.6072	.7069	.8060	.9045	.0024
53	.4024	.5031	.6031	.7026	.8014	.8996	62.9973
54	.3986	.4990	.5988	.6980	.7966	.8946	.9919
55	62.3946	62.4947	62.5943	62.6932	62.7915	62.8892	62.9864
56	.3903	.4902	.5895	.6882	.7863	.8837	.9806
57	.3858	.4854	.5845	.6829	.7808	.8780	.9746
58	.3810	.4804	.5793	.6775	.7750	.8720	.9684
59	.3760	.4752	.5738	.6718	.7691	.8658	.9620
60	62.3707	62.4697	62.5681	62.6658	62.7630	62.8595	62.9554
61	.3652	.4640	.5622	.6597	.7566	.8529	.9485
62	.3595	.4581	.5560	.6533	.7500	.8461	.9415
63	.3535	.4519	.5497	.6468	.7432	.8391	.9343
64	.3474	.4455	.5431	.6400	.7363	.8319	.9269
65	62.3410	62.4390	62.5363	62.6330	62.7291	62.8245	62.9194
66	.3344	.4322	.5293	.6258	.7217	.8170	.9116
67	.3275	.4251	.5221	.6185	.7142	.8092	.9036
68	.3205	.4179	.5147	.6109	.7064	.8013	.8955
69	.3132	.4105	.5071	.6031	.6984	.7931	.8872

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Table II-I-4 (Continued)

Temperature (°F)	Pressure (psia) Saturated	500	1000	1500	2000	2500	3000
70	62.3058	62.4029	62.4993	62.5952	62.6903	62.7848	62.8787
71	.2981	.3950	.4914	.5870	.6820	.7763	.8700
72	.2902	.3870	.4832	.5787	.6735	.7677	.8612
73	.2822	.3788	.4748	.5701	.6648	.7588	.8522
74	.2739	.3704	.4663	.5614	.6559	.7498	.8430
75	62.2654	62.3618	62.4575	62.5525	62.6469	62.7406	62.8337
76	.2568	.3530	.4486	.5435	.6377	.7313	.8242
77	.2479	.3440	.4395	.5342	.6283	.7217	.8145
78	.2389	.3349	.4302	.5248	.6188	.7121	.8047
79	.2297	.3255	.4207	.5152	.6090	.7022	.7947
80	62.2203	62.3160	62.4111	62.5055	62.5992	62.6922	62.7846
81	.2107	.3063	.4013	.4955	.5891	.6820	.7743
82	.2009	.2964	.3913	.4854	.5789	.6717	.7638
83	.1910	.2864	.3811	.4752	.5685	.6612	.7532
84	.1809	.2762	.3708	.4647	.5580	.6505	.7424
85	62.1706	62.2658	62.3603	62.4542	62.5473	62.6397	62.7315
90	.1166	.2113	.3055	.3988	.4915	.5835	.6748
95	.0585	.1529	.2467	.3397	.4320	.5236	.6145
100	61.9964	.0906	.1841	.2769	.3689	.4602	.5508
105	.9307	.0246	.1180	.2105	.3023	.3934	.4838
110	61.8612	61.9551	62.0483	62.1408	62.2325	62.3234	62.4136
115	.7884	.8821	61.9754	.0678	.1594	.2502	.3404
120	.7121	.8059	.8992	61.9916	.0832	.1740	.2641
125	.6326	.7265	.8198	.9123	.0040	.0949	.1850
130	.5500	.6440	.7375	.8301	61.9219	.0129	.1031
135	61.4643	61.5584	61.6521	61.7450	61.8369	61.9281	62.0184
140	.3757	.4700	.5640	.6570	.7492	.8406	61.9311
145	.2842	.3787	.4730	.5663	.6588	.7504	.8412
150	.1899	.2847	.3793	.4730	.5658	.6577	.7488
155	.0928	.1880	.2830	.3770	.4702	.5624	.6538
160	60.9932	61.0887	61.1841	61.2786	61.3721	61.4647	61.5565
165	.8909	60.9868	.0827	.1776	.2716	.3647	.4568
170	.7862	.8824	60.9789	.0743	.1687	.2622	.3549
175	.6789	.7756	.8726	60.9686	.0635	.1575	.2506
180	.5693	.6665	.7640	.8605	60.9560	.0506	.1442
185	60.4573	60.5549	60.6531	60.7502	60.8463	60.9414	61.0356
190	.3430	.4411	.5400	.6377	.7344	.8301	60.9249
195	.2265	.3250	.4246	.5230	.6204	.7167	.8121
200	.1076	.2068	.3070	.4062	.5042	.6012	.6972
205	59.9866	.0863	.1873	.2872	.3860	.4837	.5803

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Table II-I-4 (Continued)

Temperature (°F)	Pressure (psia)						
	Saturated	500	1000	1500	2000	2500	3000
210	59.8635	59.9636	60.0655	60.1662	60.2657	60.3641	60.4615
215	.7382	.8389	59.9416	.0430	.1433	.2425	.3406
220	.6108	.7120	.8156	59.9179	.0190	.1190	.2178
225	.4813	.5830	.6875	.7907	59.8927	59.9935	.0932
230	.3497	.4520	.5574	.6615	.7644	.8661	59.9666
235	59.2161	59.3189	59.4253	59.5304	59.6342	59.7367	59.8381
240	.0804	.1838	.2912	.3973	.5020	.6055	.7077
245	58.9428	.0467	.1551	.2622	.3679	.4723	.5755
250	.8031	58.9075	.0171	.1252	.2319	.3373	.4415
255	.6614	.7663	58.8770	58.9862	.0940	.2005	.3056
260	58.5177	58.6231	58.7350	58.8453	58.9542	59.0617	59.1678
265	.3720	.4779	.5910	.7025	.8125	58.9211	.0283
270	.2244	.3306	.4450	.5577	.6689	.7786	58.8869
275	.0747	.1814	.2970	.4110	.5234	.6343	.7437
280	57.9231	.0301	.1471	.2624	.3761	.4881	.5987
285	57.7695	57.8768	57.9952	58.1118	58.2268	58.3401	58.4519
290	.6139	.7215	.8413	57.9593	.0756	.1902	.3032
295	.4563	.5641	.6854	.8048	57.9225	.0385	.1528
300	.2966	.4046	.5275	.6484	.7675	57.8848	.0005
305	.1350	.2431	.3675	.4900	.6106	.7293	57.8463
310	56.9713	57.0795	57.2056	57.3296	57.4517	57.5719	57.6903
315	.8056	56.9137	.0415	.1672	.2908	.4126	.5324
320	.6378	.7459	56.8754	.0028	.1281	.2513	.3727
325	.4680	.5758	.7072	56.8363	56.9633	.0882	.2111
330	.2960	.4036	.5369	.6678	.7965	56.9230	.0476
335	56.1220	56.2291	56.3644	56.4972	56.6277	56.7560	56.8821
340	55.9458	.0524	.1897	.3245	.4568	.5869	.7148
345	.7674	55.8735	.0128	.1496	.2839	.4158	.5454
350	.5869	.6922	55.8337	55.9726	.1088	.2427	.3742
355	.4042	.5085	.6523	.7934	55.9317	.0675	.2009
360	55.2192	55.3225	55.4687	55.6119	55.7524	55.8902	56.0255
365	.0320	.1340	.2826	.4282	.5709	.7108	55.8482
370	54.8424	54.9430	.0942	.2422	.3872	.5293	.6688
375	.6506	.7495	54.9033	.0538	.2012	.3456	.4872
380	.4563	.5534	.7099	54.8630	.0129	.1597	.3035
385	54.2597	54.3546	54.5140	54.6698	54.8223	54.9715	55.1177
390	.0606	.1531	.3155	.4742	.6293	.7810	54.9296
395	53.8590	53.9489	.1144	.2760	.4338	.5882	.7393
400	.6548	.7418	53.9105	.0751	.2359	.3930	.5467
405	.4481	.5318	.7039	53.8717	.0354	.1954	.3518

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Table II-I-4 (Continued)

Temper-ature (°F)	Pressure (psia)						
	Saturated	500	1000	1500	2000	2500	3000
410	53.2387	53.3187	53.4944	53.6655	53.8324	53.9953	54.1545
415	.0267	.1026	.2819	.4565	.6267	.7927	53.9548
420	52.8119	52.8833	.0665	.2447	.4183	.5875	.7527
425	.5942	.6607	52.8480	.0300	.2071	.3796	.5480
430	.3737	.4348	.6262	52.8122	52.9930	.1691	.3407
435	52.1503	52.2053	52.4012	52.5913	52.7760	52.9557	53.1307
440	51.9238	51.9723	.1728	.3673	.5560	.7395	52.9181
445	.6942	.7354	51.9409	.1399	.3329	.5204	.7027
450	.4615	.4948	.7054	51.9092	.1066	.2983	.4844
455	.2255	.2501	.4661	.6749	51.8770	.0730	.2633
460	50.9862	51.0012	51.2229	51.4370	51.6441	51.8446	52.0391
465	.7434	50.7479	50.9757	.1954	.4076	.6129	51.8118
470	.4971	.4971	.7243	50.9499	.1675	.3778	.5814
475	.2472	.2472	.4686	.7003	50.9236	.1392	.3477
480	49.9935	49.9935	.2082	.4465	.6758	50.8970	.1106
485	49.7359	49.7359	49.9431	50.1884	50.4240	50.6510	50.8700
490	.4744	.4744	.6731	49.9257	.1680	.4011	.6258
495	.2087	.2087	.3978	.6582	49.9077	.1473	.3779
500	48.9387	48.9387	.1170	.3857	.6427	49.8892	.1261

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Table II-I-5 Physical Data on Some Common Commercial Gases

Gas Name	Formula	Molecular Weight (basis C^{12})	Density at 32 F and 14.696 psia (lb_m/ft^3) ρ_o	Specific Gravity (see par. I-3-23) G	Ratio Specific Heats (see Note (e)) γ	Boiling Point at 14.696 psia (°R) T_b	Critical Temperature (°R) T_c	Critical Pressure (psia) p_c	Critical Volume (ft^3/lb_m) v_c
Air		28.9644(a)	0.0807223(b)	1.00000	1.41	142.0	238.4	547	0.0517
Argon	Ar	39.948		1.3792	1.67	157.4	272.08	705.4	.0301
Acetylene	C_2H_2	26.0382	.06860	0.89897	1.24	340.7	557.1	905	.0661
Ammonia	NH_3	17.0306	.0452	.58798	1.31	431.6	731.1	1657	.0684
Benzene	C_6H_6	78.11	.0548	2.6967		635.9	1010.9	700.9	.0527
Butane-n	C_4H_{10}	58.1243	.15805	2.0068	1.09	490.8	765.3	550.7	.0704
Butane-iso	C_4H_{10}	58.1243	.15788	2.0068	1.10	470.6	734.6	529.1	.0725
Carbon dioxide	CO_2	44.00995	.12342 (b)	1.5194	1.30	350.4	547.7	1073	.0348
Carbon monoxide	CO	28.01055	.078065(b)	0.96707	1.40	143.0	241.7	510	.0515
Ethane	C_2H_6	30.0701	.07987	1.0382	1.19	332.2	549.8	708.3	.0787
Ethylene	C_2H_4	28.0542	.07391	0.96858	1.24	305.0	509.5	742.1	.0705
Ethyl alcohol	C_2H_5OH	46.07	49.2759 (liq)	1.5905	1.13	632.75	929.3	927.3	.0581
Helium	He	4.0026	.011143(c)	.13819	1.66	7.669	9.4(f)	33.0(f)	.23(f)
Hydrogen	H_2	2.0159	.0056114(b)	.069599	1.41	36.8	59.9	188	.5168
Methyl alcohol	CH_3OH	32.04	49.6942 (liq)	1.1061	1.203	608.06	923.7	1156.6	.0588
Hydrogen sulphide	H_2S	34.0799	.09050	1.1766	1.32	383.2	672.4	1306	
Methane	CH_4	16.0430	.042355	0.55389	1.31	201.0	343.2	673.1	.0993
n-Octane	C_8H_{18}	114.23	43.9257 (liq)	3.9438		715.968	1024.5	361.5	.0684
Nitrogen	N_2	28.0134	.078064 (b)	.96717	1.40	139.3	226.9	492	.0515
Oxygen	O_2	31.9988	.0892102(b)	1.1047	1.40	162.3	277.9	730	.0373
Propane	C_3H_8	44.0972	.11806	1.5225	1.33	416.0	666.	617.4	.0730
Sulphur dioxide	SO_2	64.07	.1826	2.212		473.7	774.6	1141.9	.0308
Water (steam, dry)	H_2O	18.0153		0.62198	1.30	671.7	1165.1(d)	3208.2(d)	.05078(d)

(a) U.S. Standard Atmosphere, U.S. Government Printing Office, 1962, p. 9.
(b) National Bureau of Standards Circular 564, Tables of Thermodynamic Properties of Gases, 1960. (Interpolation within these tables should be by a method of second differences [20].)
(c) U.S. Bureau of Mines Journal of Chemistry and Engineering Data, vol. 5, Jan. 1960, p. 51.
(d) *1967 ASME Steam Tables.*
(e) Values for an ideal gas and a reversible system.
(f) Provisional Thermodynamic Functions for Helium 4, R.D. McCarty, National Bureau of Standards Report 9762, 1970.

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[7] Thermodynamic Functions of Gases, F. Din; vol. 3, 1961, p. 24.

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[15] Determination of Viscosity of Exhaust-Gas Mixtures at Elevated Temperatures; J. C. Westmoreland; *NACA Tech. Note 3180*, June 1954.

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[17] Supercompressibility Factors for Natural Gas; R. H. Zimmerman and S. R. Beitler; *Trans. ASME*, vol. 74, Aug. 1952, p. 945.

[18] Evaluation fo Compressibility Factors for Natural Gas Mixtures; R. G. Darrow; MS Thesis, Ohio State University, 1953.

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Chapter II-II

General Requirements for Fluid Metering: Installation

II-II-1 The fluid to be measured may be incompressible or compressible, i.e., liquid or gaseous. In locations where there may be a choice of measuring the flow of a fluid in either the liquid or gaseous state, the measurement should be made with the liquid if at all possible. This is because in the present state-of-the-art measurements made of a liquid are more reliable.

The fluid must remain in a single phase. For example, superheated steam must remain superheated. In the case of liquids, the pressure throughout the entire metering unit must be sufficiently high to prevent evaporation or the escape of any dissolved gases. Also, the fluid stream must be free of pulsations. This applies for almost all types of meters, but particularly those of the differential-pressure type.

The fluid should not contain suspended particles, as, for example, sand. Colloidal solutions with an index of dispersion not materially different than that of a homogeneous liquid, for example, milk, may be measured. Solutions of coarse dispersion should be excluded. Fluids having high viscosities such that the Reynolds numbers are below the limits shown by the tables or curves of coefficients of discharge should be avoided if a differential-pressure-type meter is to be used.

II-II-2 Installations. The conditions under which meters such as the orifice, flow nozzle and Venturi tube are installed may have as much effect on the accuracy of the flow measurement as the degree of perfection of manufacture or the characteristics of the elements themselves. The rate of flow computed from the differential pressure produced by these elements may be in error to an unacceptable degree if the piping arrangements are such that distorted flow conditions result. Distortions of the velocity traverse, helical swirls or vortices can affect the accuracy of the flow measurement. A projecting gasket, misalignment or a burr on a pressure tap can cause considerable error. Therefore, the following instructions should be followed carefully.

Whenever possible, it is preferable to locate the primary element in a horizontal line. If the unit is in a vertical pipe, the pressure tubing to the secondary element, including reservoirs, if used, should be adjusted to the same elevation as described in Par. II-II-11 below.

II-II-3 To insure accurate flow measurement, the fluid should enter the primary element with a fully developed velocity profile, free from swirls or vortices. Such a condition is best achieved by the use of adequate lengths of straight pipe, both preceding and following the primary element. The *minimum* desirable lengths of such piping are shown in the eight diagrams of Fig. II-II-1. Each diagram shows a somewhat different arrangement of piping. That diagram which corresponds the closest to the actual piping arrangement for the meter location should be used to determine the required lengths of straight pipe on inlet and outlet. These lengths have been determined as necessary to hold errors due to piping configurations to less than ±0.5 per cent.

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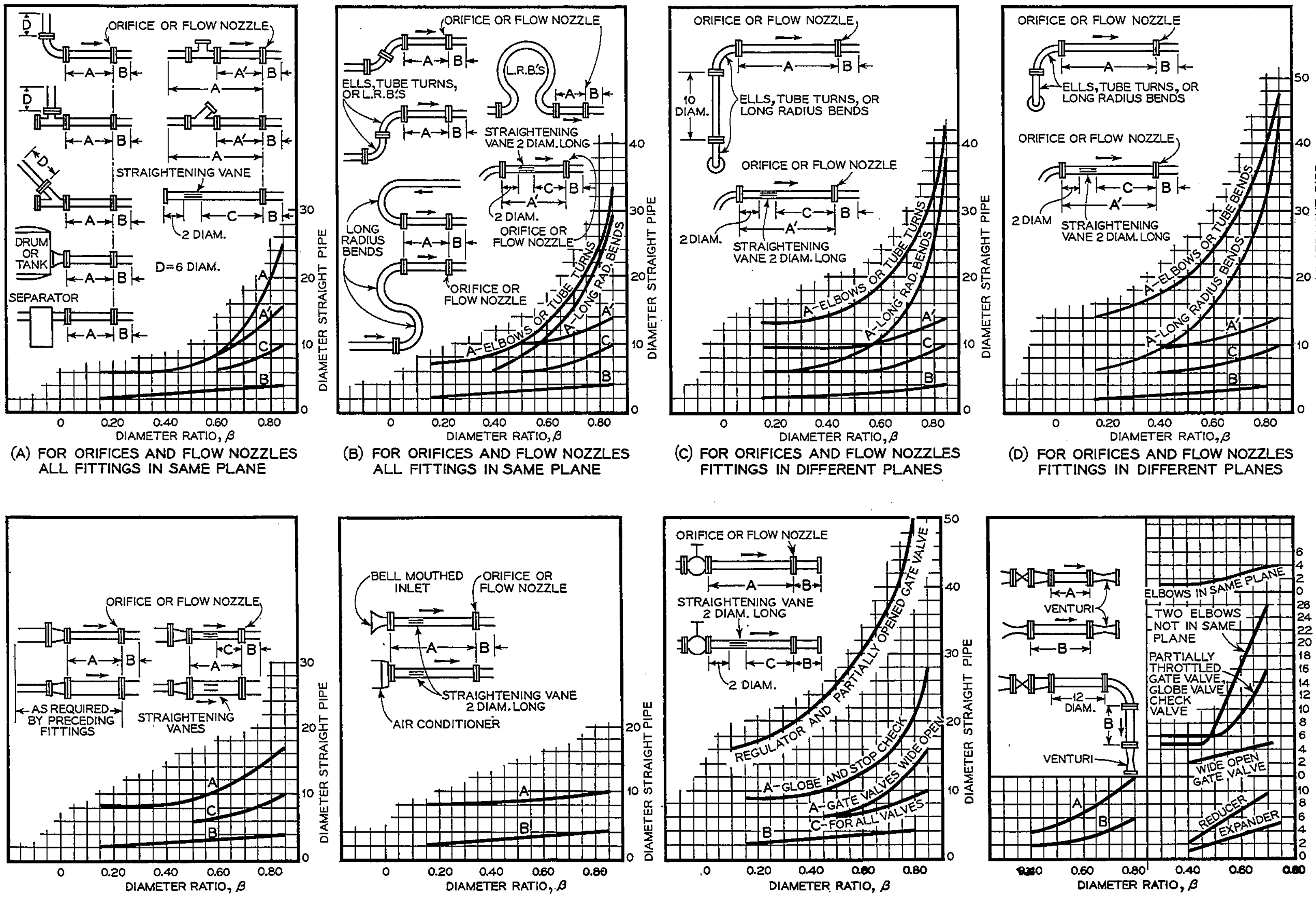

(A) FOR ORIFICES AND FLOW NOZZLES ALL FITTINGS IN SAME PLANE

(B) FOR ORIFICES AND FLOW NOZZLES ALL FITTINGS IN SAME PLANE

(C) FOR ORIFICES AND FLOW NOZZLES FITTINGS IN DIFFERENT PLANES

(D) FOR ORIFICES AND FLOW NOZZLES FITTINGS IN DIFFERENT PLANES

(E) FOR ORIFICES AND FLOW NOZZLES WITH REDUCERS AND EXPANDERS

(F) FOR ORIFICES AND FLOW NOZZLES IN ATMOSPHERIC INTAKE

(G) VALVES AND REGULATORS

(H) FOR VENTURI TUBES

FIG. II-II-1 RECOMMENDED MINIMUM LENGTHS OF PIPE PRECEDING AND FOLLOWING ORIFICES, FLOW NOZZLES AND VENTURI TUBES (ALL CONTROL VALVES, INCLUDING REGULATORS, SHOULD BE LOCATED ON OUTLET SIDE OF PRIMARY ELEMENT.)

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If it is impossible to arrange the piping so as to provide the recommended lengths of straight pipe or should there be uncertainty as to the diagram to follow, straightening vanes may be used. Diagrams (A) through (H) (Fig. II-II-1) show the minimum straight pipe that may be used with straightening vanes. The vanes should be preceded by two or more diameters of straight pipe. Greater lengths of straight pipe, both with and without vanes, than the minimums shown in Fig. II-II-1 should be provided whenever possible.

If it is impossible to provide the recommended minimum lengths, even with the use of straightening vanes, an additional tolerance of ± 0.5 per cent should be applied to the flow measurement. This additional tolerance should be treated in the same way tolerances for other elements are treated as described in Par. II-V-4.

II-II-4 The use of any type of differential pressure meter in a pipe line that includes a reciprocating pump should be avoided. However, with liquids, as explained in Par. I-3-46 (Chapter I-3), a measurement corrected for pulsation effects may be possible if a time average of the square root of the instantaneous differential pressures at the meter can be obtained by the recorder. On the other hand, with compressible fluids no reliable measurement is possible; the pulsations from a reciprocating pump must be reduced or completely suppressed by some means between the source and the primary element.

If the pump is a centrifugal or turbine type, the preferred location of the primary element is on the inlet side and as far from the pump as possible. If the meter is located on the discharge side of the pump, straightening vanes will be needed to eliminate swirl. There should be a minimum of eight pipe diameters between the vanes and primary element. If there are fittings between the pump and primary element, the vanes should be located according to the appropriate arrangement for the fittings as shown in Fig. II-II-1.

II-II-5 Cross sections of recommended designs of straightening vanes are shown in Fig. II-II-2. In the tubular and cross-plate designs, the maximum distance between tube centers or passage centers should not exceed one-fourth the pipe diameter, D; and the overall length should be at least eight times this dimension. These vanes may be constructed of thin-walled tubes, or plates, welded together. The perforated-plate design has plates held one pipe diameter apart by spacers; each has a large number of small holes, and the total area of these holes should be equal to at least 50 per cent of the cross-sectional area of the pipe. Regardless of the design of straightening vane used, they must be secured firmly in place within the pipe.

The pressure drop through the tubular and cross-plate designs is about the same as that for 20 diameters of the pipe. For the perforated-plate design, the loss is about the same as the differential pressure across a sharp-edged orifice of 0.75-diameter ratio.

II-II-6 Internal Pipe Surface and Diameter. The internal surface of the pipe immediately preceding and following an orifice or flow nozzle should be straight, free from mill scale, pits or holes, reamer scores or rifling, bumps or other irregularities. The surface roughness should not be greater than 350 microinch. The pipe should be near enough to a cylindrical shape that no diameter departs from the average diameter, D, by more than 0.33 per cent. If, to secure this degree of surface roughness and pipe roundness, boring is necessary, such boring should extend for at least $4D$ preceding and $2D$ following the inlet face of the orifice or nozzle. The bored portion should be faired into the unbored portion at an included angle of less than 30 deg. The depth of boring should be the minimum required to obtain the desired condition, and the boring should be done after any necessary welding of flanges and pressure connections has been done.

II-II-7 The internal pipe diameter, D, should be measured on four or more diameters in the plane of the inlet pressure tap hole, p_1. Check measurements should be made on three or more diameters in two additional cross sections so as to cover at least two pipe diameters from the inlet face of the orifice plate of flow nozzle, or past the weld, whichever is the greater distance. The values of all such inlet-section diameters should agree within 4 per cent when the diameter ratio, β, of the orifice or flow nozzle to be used will be 0.2 and within 0.5 per cent when the diameter ratio, β, is to be 0.75. For intermediate values of β, a linear relation can be used. The average of all diameters near the plane of the inlet pressure tap should be used in computing the diameter ratio, β, of the primary element.

Measurements of the diameter of the outlet section should be made in the plane of the outlet pressure tap to insure that the diameter of the outlet section agrees with that of the inlet section within twice the percentage spreads given above for the diameters of the inlet section.

For use with a Venturi tube, the internal surface of the pipe, attaching to the tube inlet, should be free of such imperfections as mentioned above.

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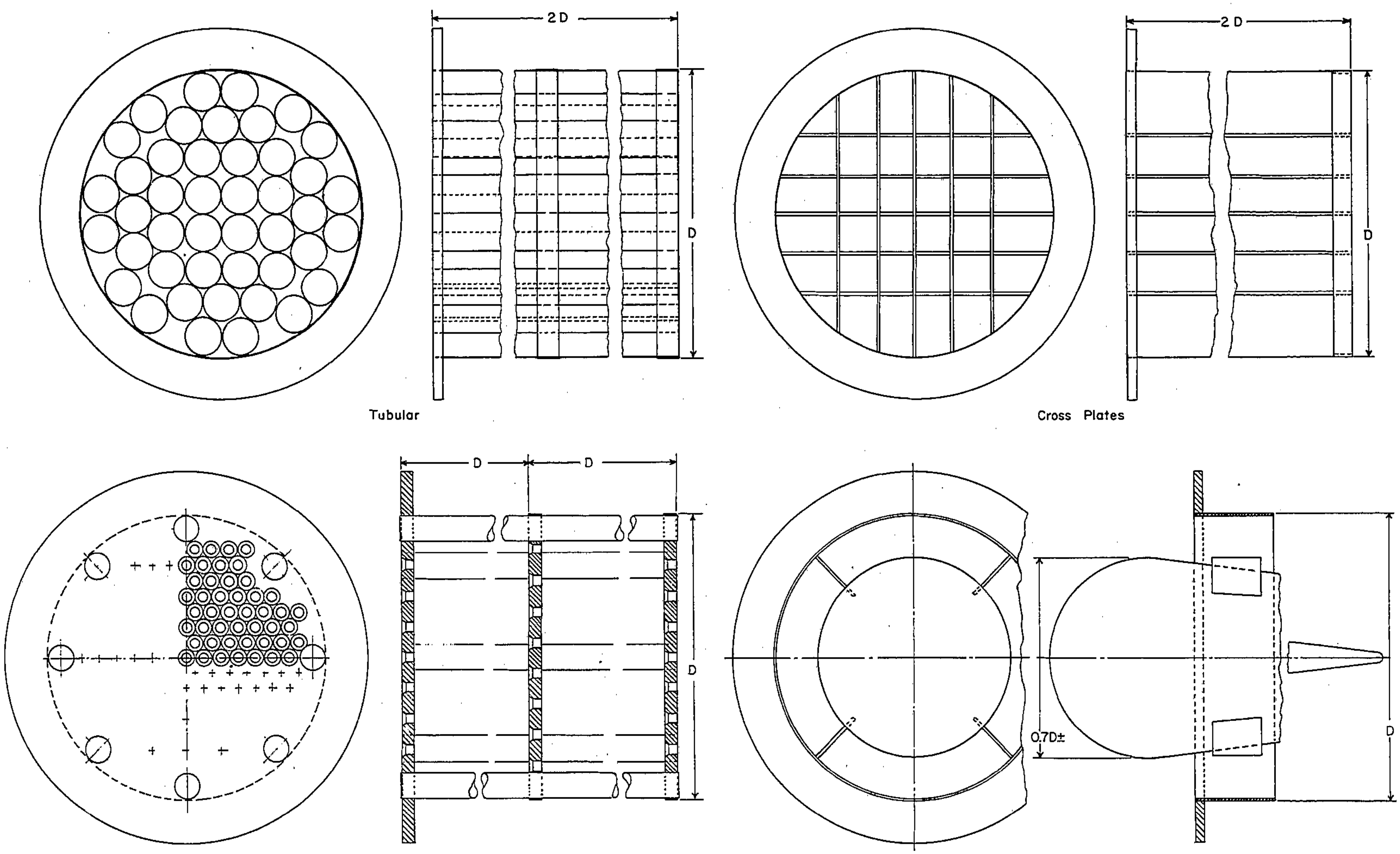

FIG. II-II-2 RECOMMENDED DESIGNS OF STRAIGHTENING VANES

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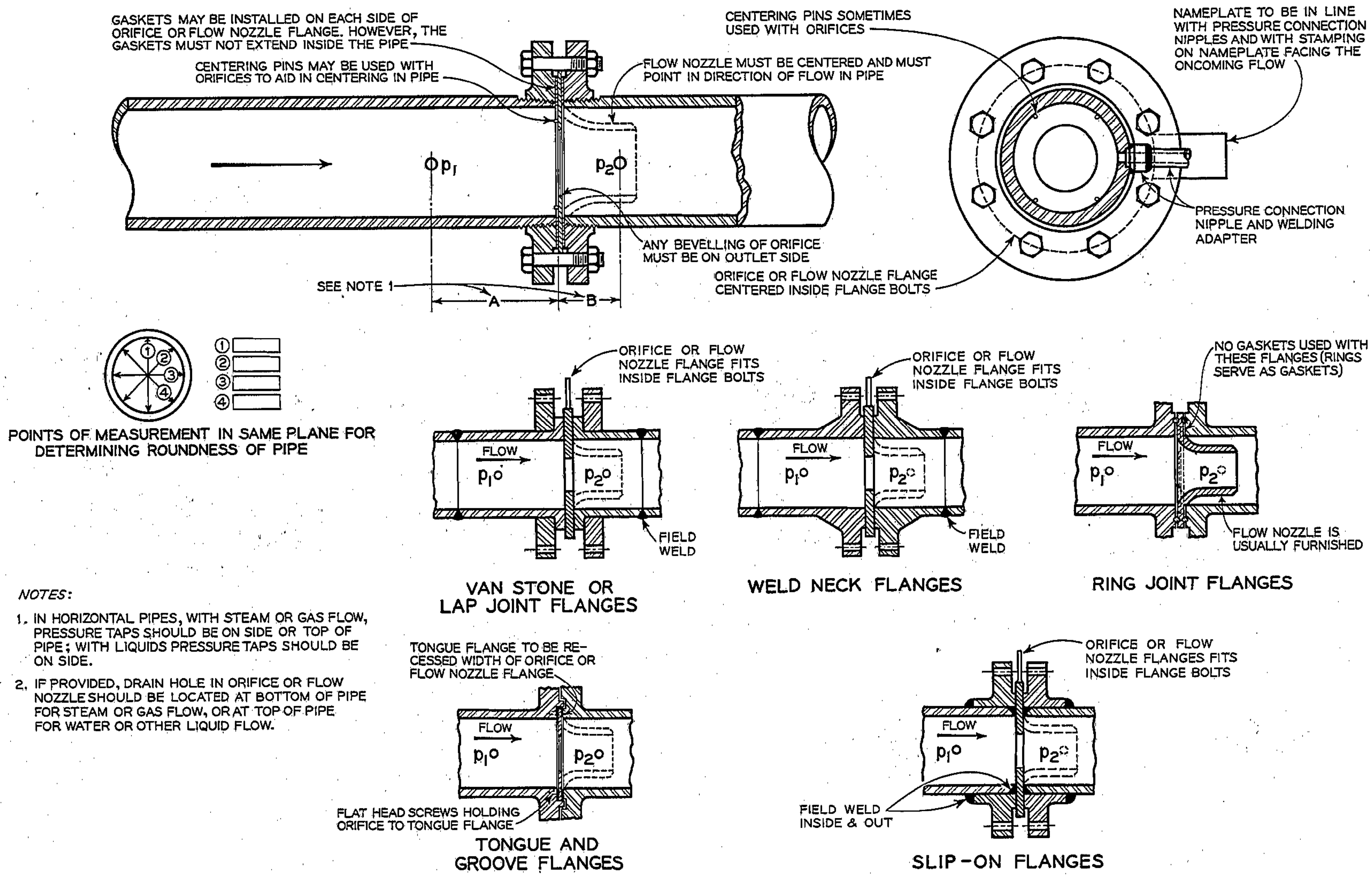

NOTES:

1. IN HORIZONTAL PIPES, WITH STEAM OR GAS FLOW, PRESSURE TAPS SHOULD BE ON SIDE OR TOP OF PIPE; WITH LIQUIDS PRESSURE TAPS SHOULD BE ON SIDE.
2. IF PROVIDED, DRAIN HOLE IN ORIFICE OR FLOW NOZZLE SHOULD BE LOCATED AT BOTTOM OF PIPE FOR STEAM OR GAS FLOW, OR AT TOP OF PIPE FOR WATER OR OTHER LIQUID FLOW.

FIG. II-II-3 METHODS OF INSTALLING ORIFICE PLATES AND FLOW NOZZLES BETWEEN FLANGES FOR LOW-PRESSURE, LOW-TEMPERATURE SERVICE

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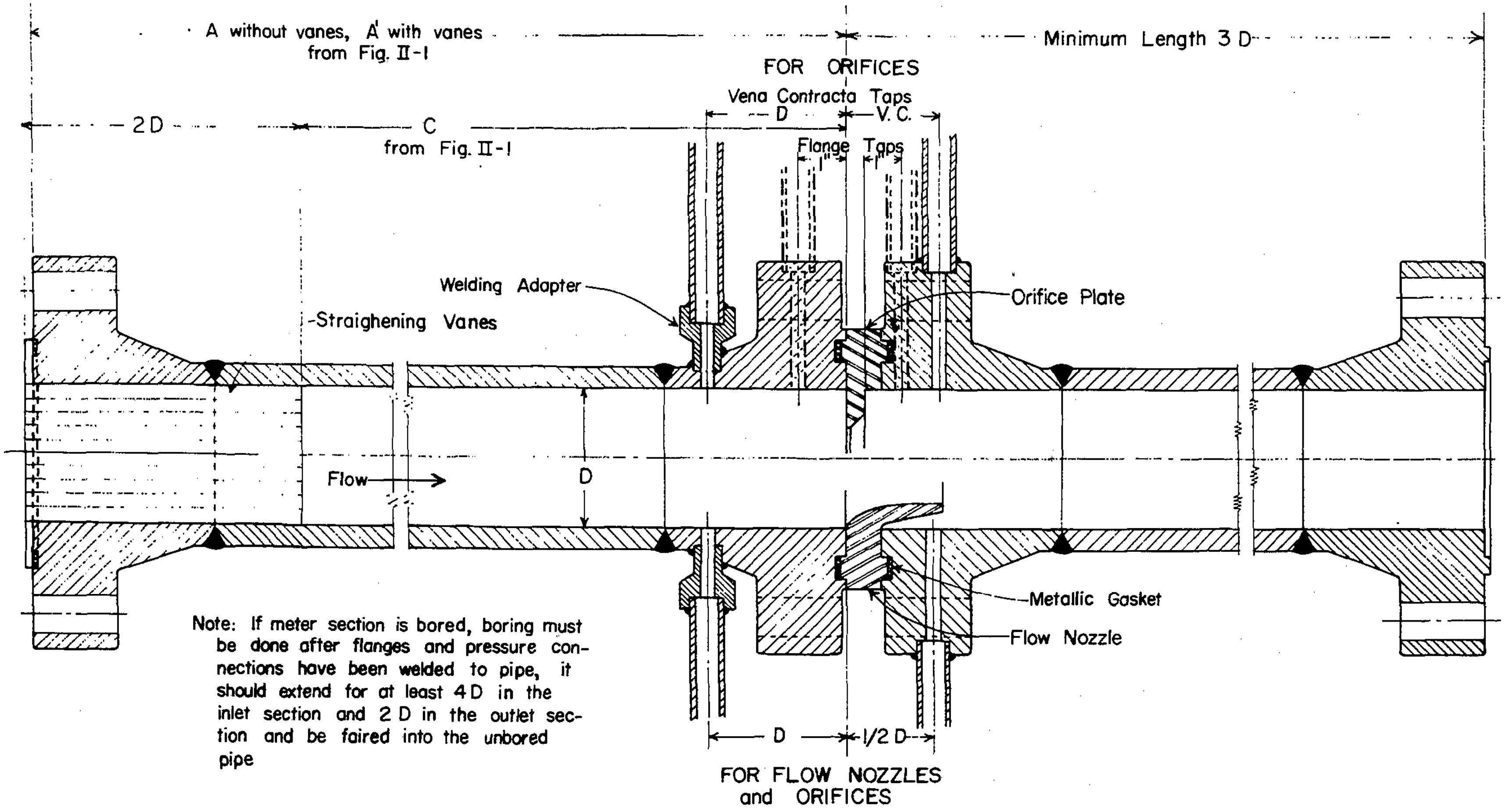

FIG. II-II-4 A RECOMMENDED DESIGN OF ORIFICE OR FLOW-NOZZLE METER SECTION FOR HIGH-TEMPERATURE, HIGH-PRESSURE SERVICE

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The average diameter of the pipe, D, where it joins the Venturi tube shall be within ±1 per cent of the diameter of the tube inlet, and the out-of-roundness should not exceed 2 per cent. Desirably, the pipe following a Venturi tube should be of the same nominal size.

II-II-8 Installation of an Orifice or Flow Nozzle. When installed between pipe-line flanges, the center of a concentric orifice or of a nozzle throat should be within ± 1/32 in. of the axis of the pipe. Several optional methods of installing and centering an orifice plate or flow nozzle for low-pressure low-temperature service are shown in Fig. II-II-3. The centering of an eccentric or segmental orifice should be such that no portion of the round hole will be closer than 1 per cent of the pipe diameter to the pipe wall. A method of centering and holding an orifice plate or flow nozzle between flanges for high-pressure high-temperature service is shown in Fig. II-II-4.

When male and female or tongue and groove flanges are used, the outside diameter of the orifice plate or the nozzle flange should be made to fit inside the female or groove flange.

For use in ring joint flanges, a flow-nozzle flange can be made thick enough so that a ring groove can be cut into each side of this flange, thereby allowing for the usual ring on each side. For an orifice plate a ring which will fit in the grooves of the flanges may be fitted to the outer circumference of the plate, or the face of the inlet flange may be recessed by an amount equal to the thickness of the orifice plate and the plate attached to the flange. This procedure may be used with a flow nozzle also, provided that it does not require an undesirably thin nozzle flange.

II-II-9 Gaskets. In all cases the inside diameter of the gasket must be made large enough and the gasket so positioned that, when in service, it will not protrude beyond the inner surface of the pipe at any point.

II-II-10 Pressure Connections. The locations of pressure tap holes used with orifices and flow nozzles are referred to the inlet face of the orifice plate or flow-nozzle flange as the datum plane, except for flange taps used with orifices. For orifices, the locations for which data are given are flange, D and $1/2D$ and vena contracta taps. For flow nozzles, data are given for pipe-wall taps at D and $1/2D$ and for D and the nozzle throat. The locations of these several pairs of pressure taps are specified in the sections dealing with each differential pressure producer in Ch. II-III.

The recommended maximum diameter, δ, of pressure tap holes through the pipe wall or flange are given in Table II-II-1. With clean fluids smaller diameters may be desirable.

Table II-II-1 Recommended Maximum Diameters of Pressure Tap Holes

Nominal Inside Pipe Diam. D	Max. Diam. δ (Fig. II-II-5)
Under 2	1/4
2,3	3/8
4 to 8	1/2
10 and over	3/4

All dimensions are given in inches.

There must be no burrs, wire edges or other irregularities on the inside of the pipe at the nipple connections or along the edge of the hole through the pipe wall. The diameter of the hole should not *decrease* within a distance of 2δ from the inner surface of the pipe but may be increased within a lesser distance.

Where the pressure hole breaks through the inner surface of the pipe there *must* be no roughness, burrs nor wire edge. The edge (corner) of the hole may be left truly square or it may be dulled (rounded) very slightly.

Connections to the pressure holes should be made by nipples, couplings or adapters (Fig. II-II-5) welded to the outside surface of the pipe. It is important that no part of any such fitting projects beyond the inner surface of the pipe.

II-II-11 When the primary element is in a horizontal pipe, for measuring steam the pressure holes and connecting nipples should be in the horizontal plane of the pipe center line. For measuring water and other liquids, the connections may be in the same horizontal plane or any other convenient position in the lower half of the pipe. For measuring gases, the usual position is in the vertical plane of the pipe center line, although, if necessary, other positions in the upper half of the pipe line may be used. Some of the connection positions are illustrated in Fig. II-II-6.

When measuring steam in a vertical pipe, an S nipple of at least 3/4-in. pipe size should be used at the lower connection and should be of such length that the top of the nipple is level with the straight nipple used at the upper connection. On the other hand, when measuring liquids in a vertical

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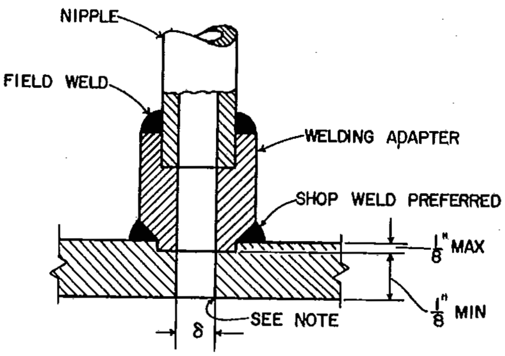
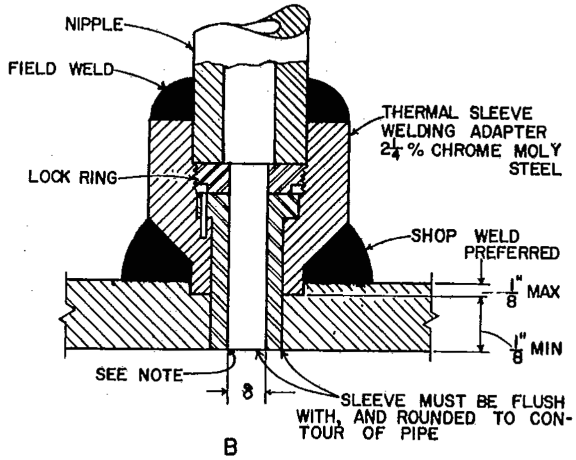
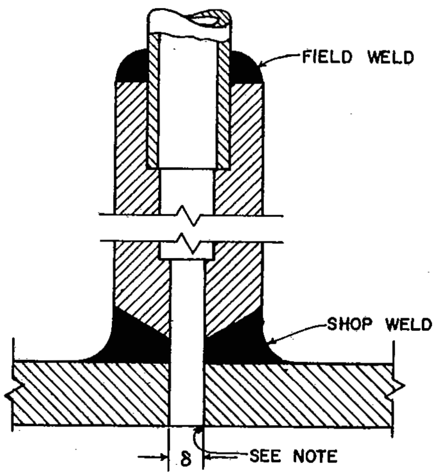


A
FOR TEMPERATURES UP TO 800 F

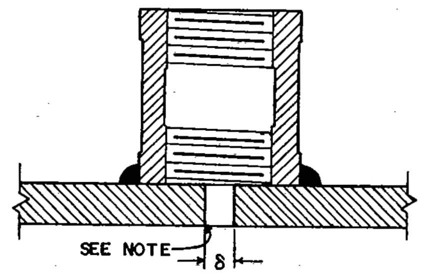


B
FOR TEMPERATURES ABOVE 800 F AND A SECONDARY ELEMENT WITH APPRECIABLE DISPLACEMENT

NOTE: EDGE OF HOLE MUST BE CLEAN AND SQUARE OR ROUNDED SLIGHTLY, FREE FROM BURRS, WIRE EDGES, OR OTHER IRREGULARITIES.



C
OPTIONAL DESIGN WHERE FULL PENETRATION WELD IS REQUIRED



D
FOR TEMPERATURES UP TO 400 F

FIG. II-II-5 METHODS OF MAKING PRESSURE CONNECTIONS TO PIPES

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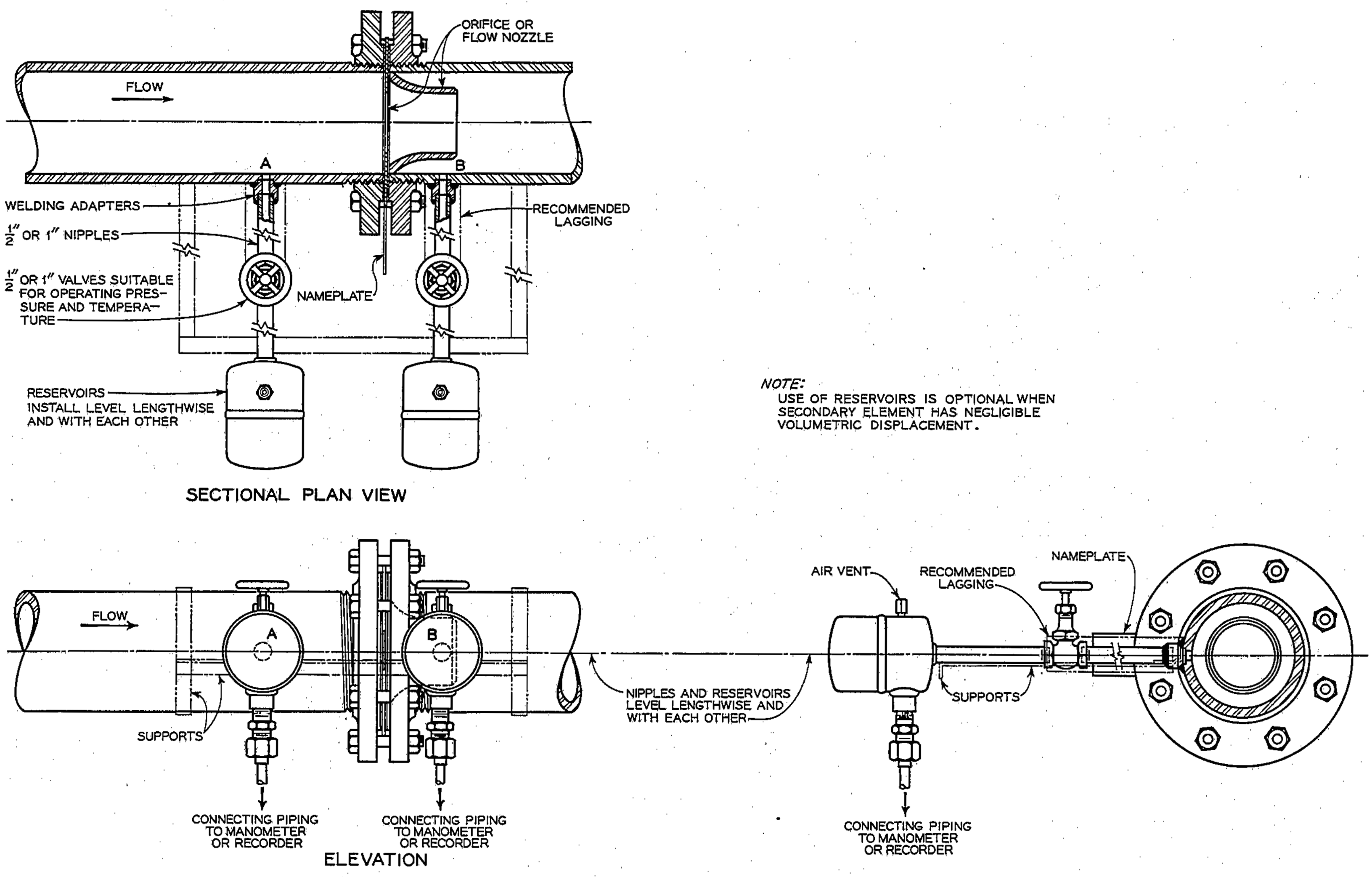

FIG. II-II-6 METHOD OF CONNECTING NIPPLES, VALVES AND RESERVOIRS TO HORIZONTAL PIPE LINES

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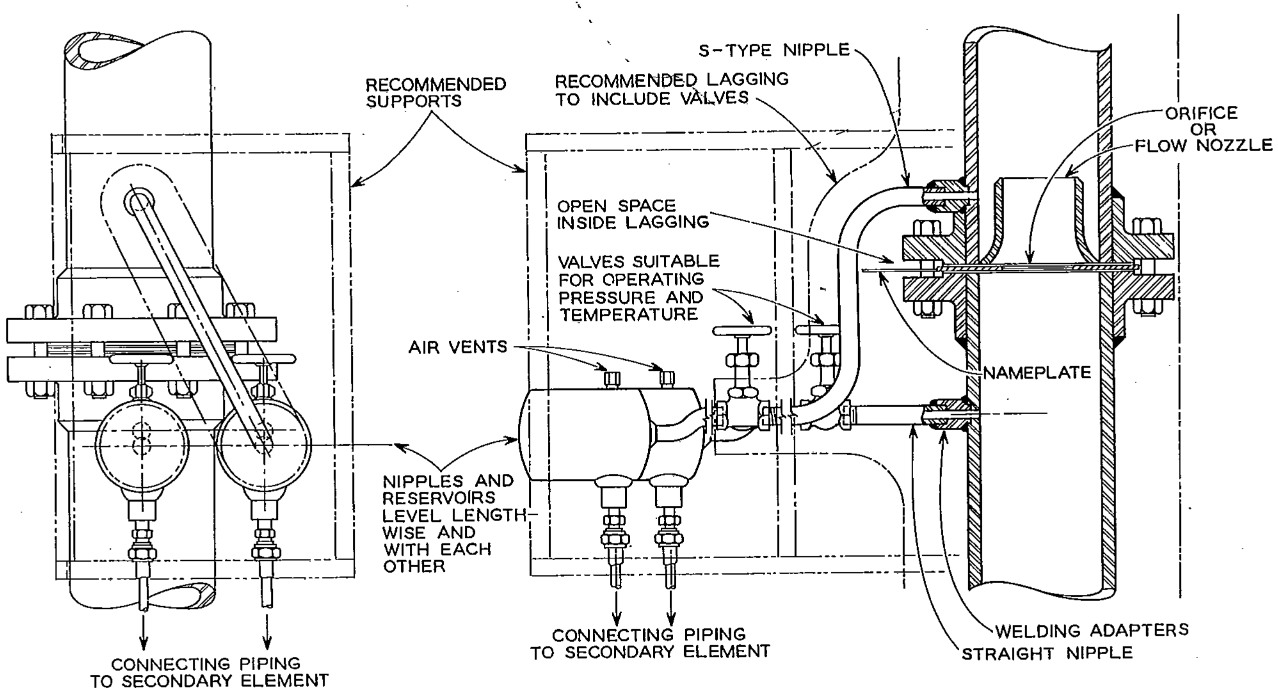

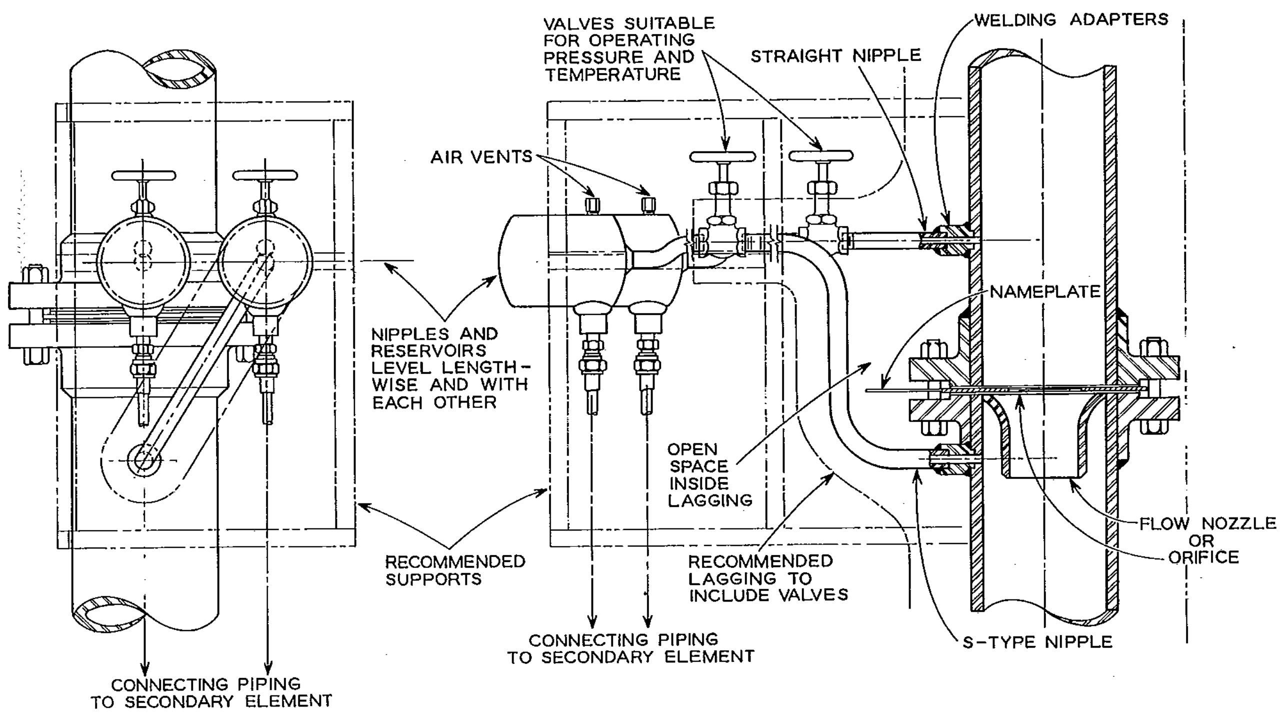

FIG. II-II-7 CONNECTING NIPPLES, VALVES AND RESERVOIRS TO VERTICAL PIPE LINES

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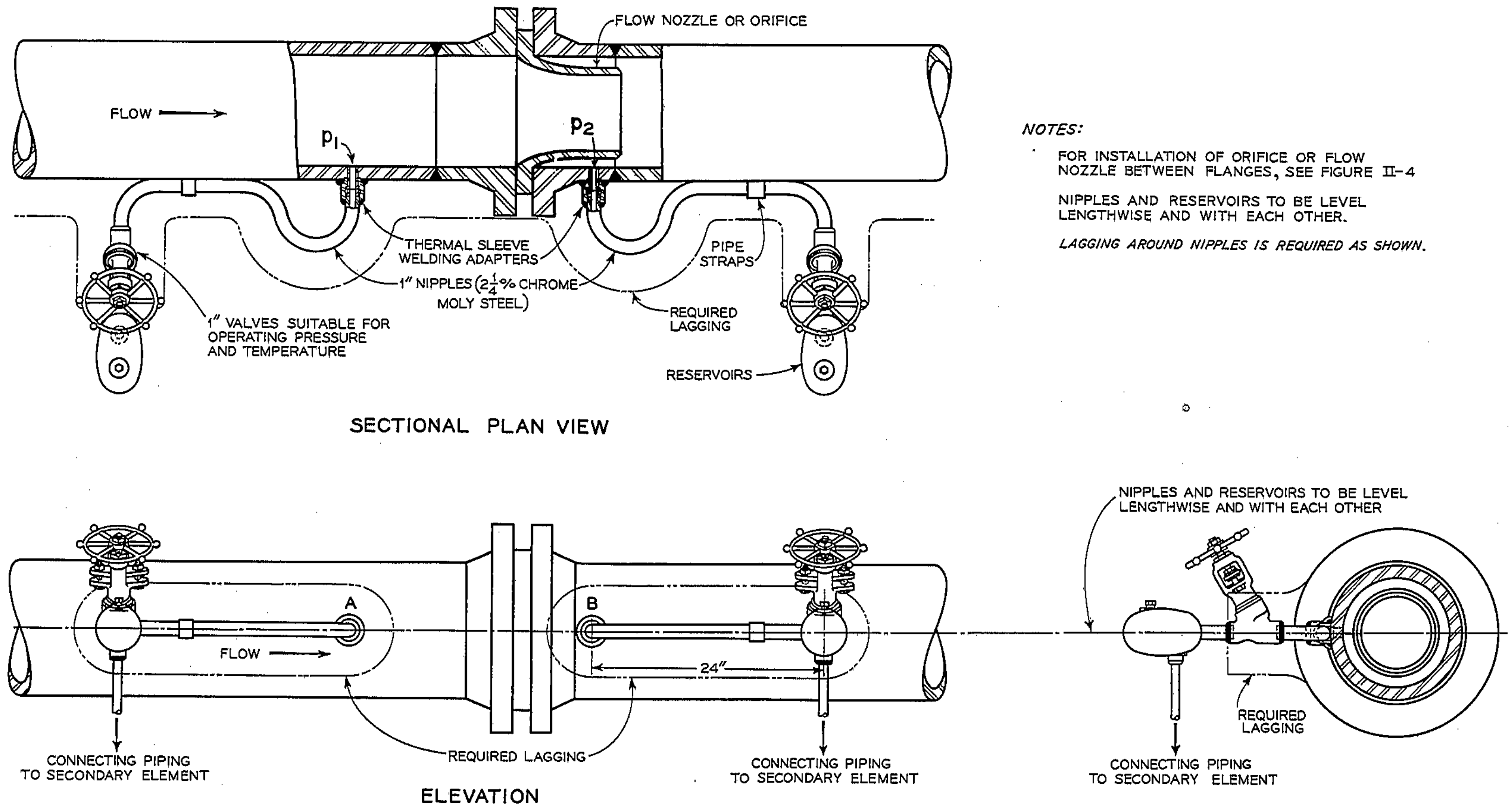

NOTES:

FOR INSTALLATION OF ORIFICE OR FLOW NOZZLE BETWEEN FLANGES, SEE FIGURE II-4

NIPPLES AND RESERVOIRS TO BE LEVEL LENGTHWISE AND WITH EACH OTHER.

LAGGING AROUND NIPPLES IS REQUIRED AS SHOWN.

FIG. II-II-8 METHOD OF CONNECTING NIPPLES, VALVES AND RESERVOIRS TO HORIZONTAL PIPE WITH STEAM AT TEMPERATURES ABOVE 850 F, ESPECIALLY IF SECONDARY ELEMENT HAS APPRECIABLE VOLUMETRIC DISPLACEMENT

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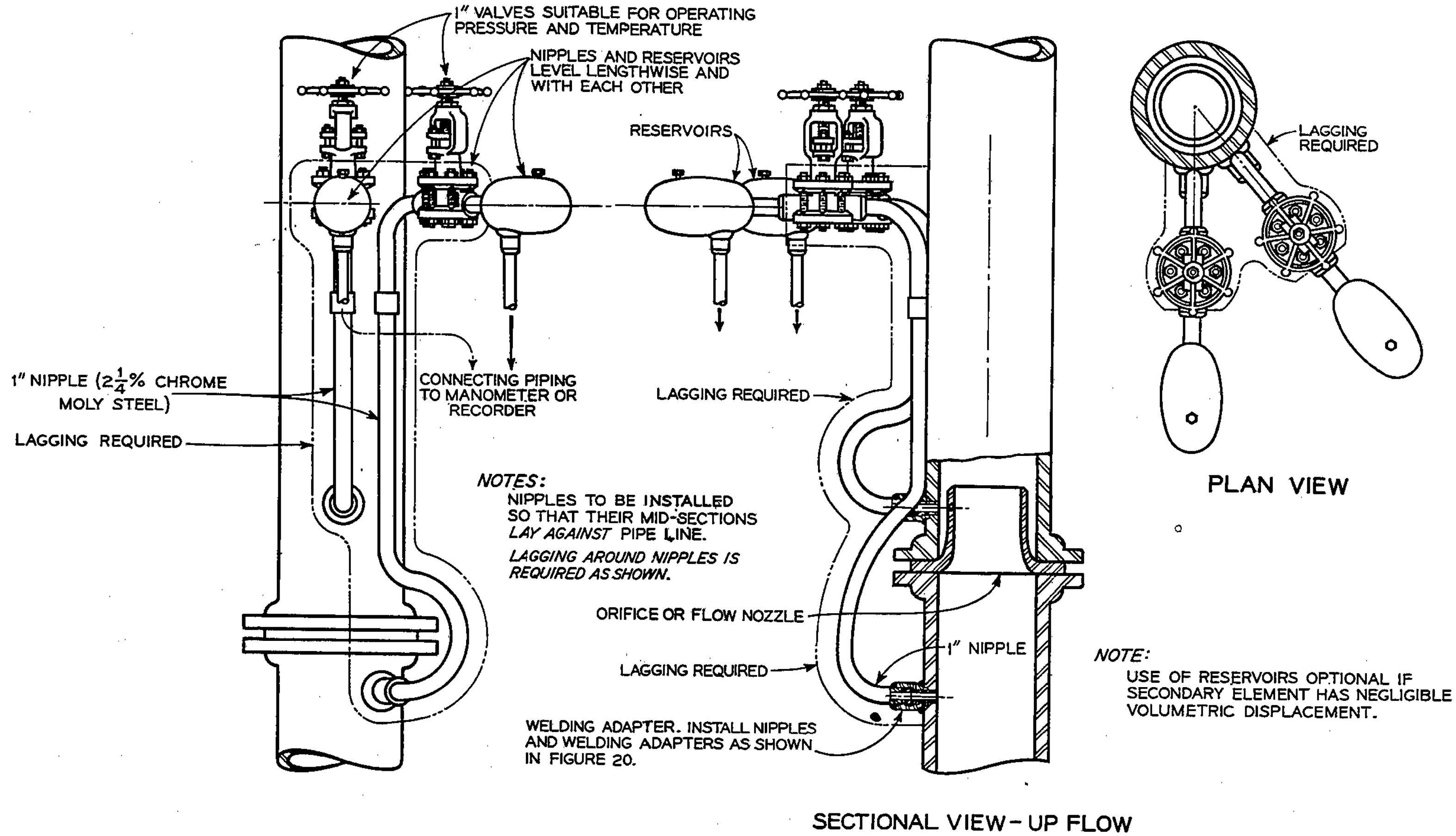

FIG. II-II-9 CONNECTING NIPPLES, VALVES AND RESERVOIRS TO VERTICAL STEAM PIPES WHEN STEAM TEMPERATURE IS ABOVE 850 F AND SECONDARY ELEMENT HAS APPRECIABLE VOLUMETRIC DISPLACEMENT

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pipe, if the temperature differs from the ambient by more than 50 F, an S nipple of at least 3/4-in. pipe size should be used at the upper connection; and its length should be such that the bottom of this nipple is level with the straight nipple used at the lower connection, as shown in Fig. II-II-7. The S nipple should be lagged in with the main flow line in order to prevent condensate forming in the nipple when measuring steam and to maintain pipe temperatures in the nipple when measuring hot liquids.

For measuring steam at temperatures higher than 850 F, special bent nipples, as shown in Figs. II-II-8 and II-II-9, should be used. In both cases the entire length of these nipples as well as the shutoff valves should be lagged in with the steam line so as to convert to steam any water returning from a manometer or other gage, before it re-enters the steam pipe, and also to keep the amount of condensate to a minimum.

II-II-12 Shutoff Valves. Shutoff valves should be provided for every pressure tap connection and should be located as close as possible to the main pipe containing the primary element. These valves must be capable of withstanding full pipe-line pressure and temperature and must be installed so as to close against the pressure in the main pipe.

II-II-13 Reservoirs. Reservoirs should be used at the ends of the inlet and outlet differential pressure connections at the primary element when measuring steam and when measuring hot water or liquids above 250 F, as shown in Figs. II-II-6 through II-II-9. These reservoirs provide water legs of equal density and elevation on both sides of a manometer or other differential pressure gages. The water volume of these reservoirs should be at least equal to the maximum displacement of the manometer or other differential pressure gage to which they are connected, and a volume two or three times this amount is preferable. The design of reservoirs and the method of connecting them should be such that they will be full of condensate at all times. Reservoirs filled partly with steam and partly with water are of little

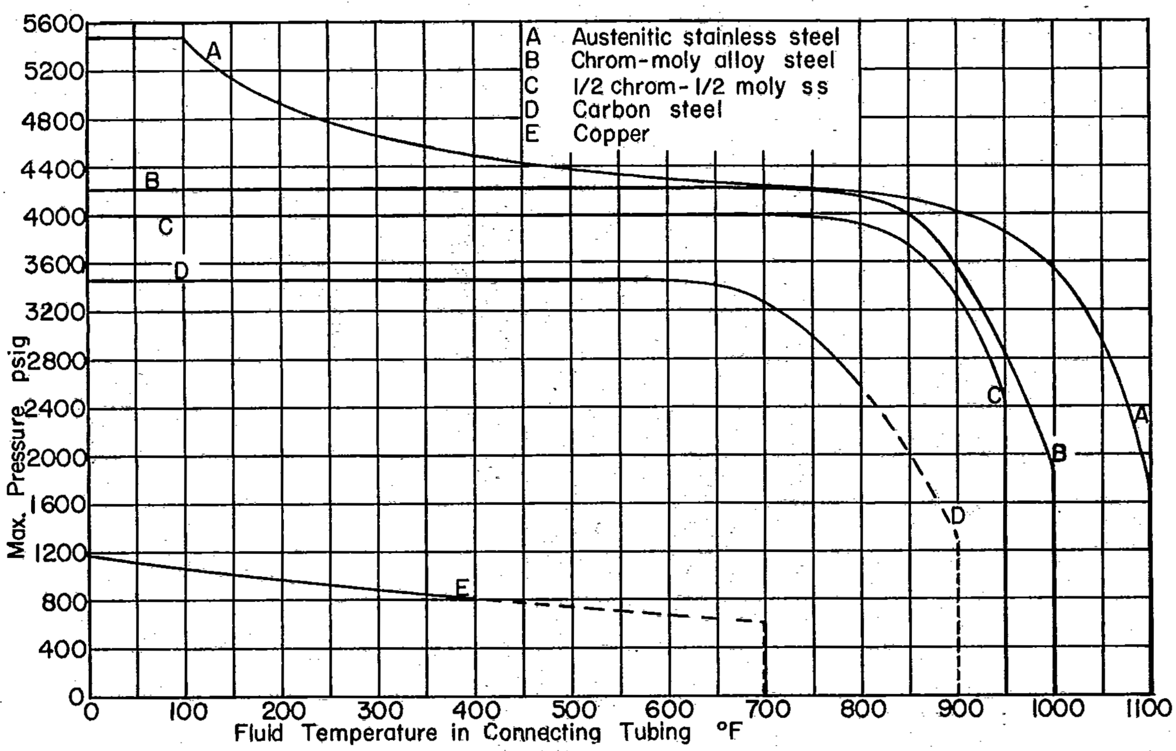

FIG. II-II-10 CHART FOR SELECTING SEAMLESS TUBING OR PIPE CONNECTING PRIMARY AND SECONDARY ELEMENTS (SELECTION MADE ACCORDING TO THE AREA WHICH CONTAINS THE POINT REPRESENTING THE OPERATING CONDITIONS. DOTTED LINES APPLY TO INTERMITTENT SERVICE. FROM ANSI STANDARD CODE FOR PRESSURE PIPING, B 31.1.0-1967.)

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value. They should be installed and supported so as to be level with each other at all times.

Reservoirs may be omitted if the differential pressure gage that is being used has zero or negligible displacement.

II-II-14 Connecting Tubing. For connecting the primary element to the secondary instruments, 1/2-in. o.d. copper tubing with steel flared fittings may be used for air, gas, steam, water, oil and most other liquids when the operating conditions are within the limits for copper tubing as shown in Fig. II-II-10. For higher pressures and temperatures, 1/2-in. carbon steel, stainless steel or chrome molybdenum steel tubing and steel flared fittings are recommended. For most gas and oil measurements, 1/2-in. steel pipe with screwed fittings may be used.

All connecting tubing should be so arranged and installed so as to have a slope of 1 in. per ft or more Some representative arrangements of connecting tubing are shown in Figs. II-II-11 through II-II-13.

II-II-15 Pressure and Temperature Instruments. Manometers and other types of differential-pressure-measuring gages, pressure gages, temperature-measuring instruments and other instruments as needed should be installed in accordance with the specific instructions furnished by the manufacturer of each instrument.

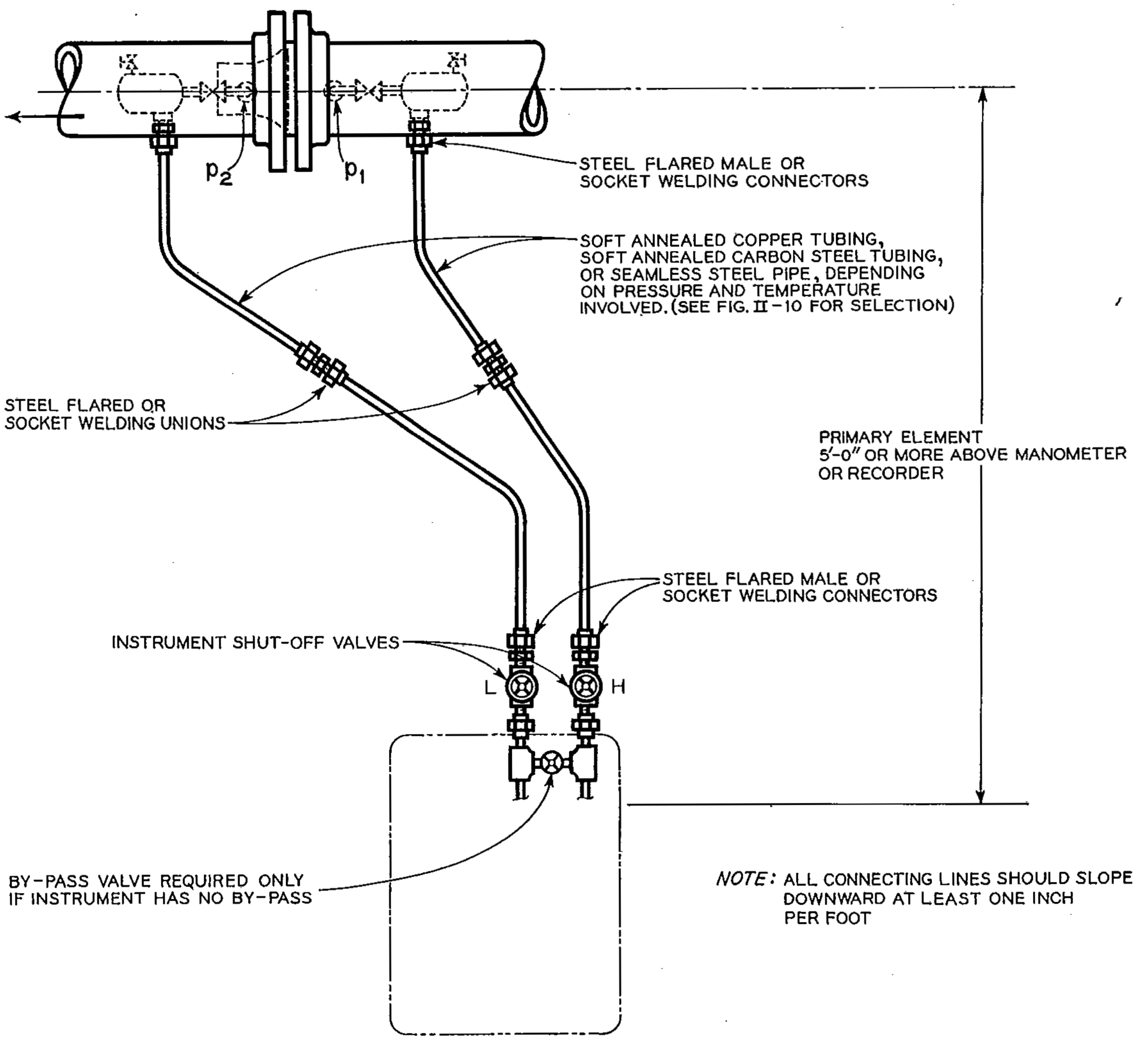

FIG. II-II-11 RECOMMENDED ARRANGEMENT OF PIPING BETWEEN PRIMARY AND SECONDARY ELEMENTS WHEN PRIMARY IS ABOVE SECONDARY AND METERED FLUID IS A LIQUID, STEAM OR CONDENSABLE GAS (SCHEMATIC)

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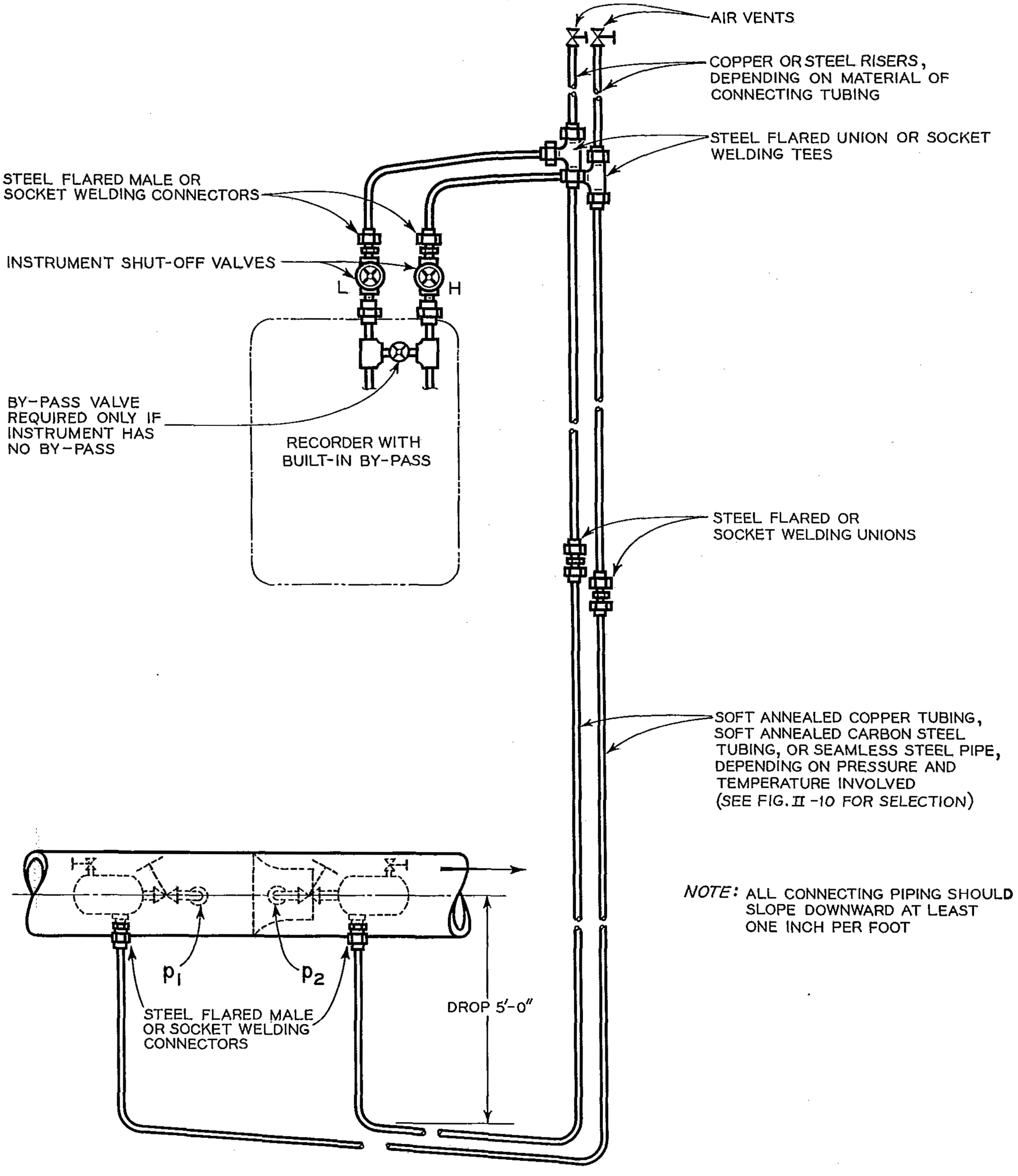

FIG. II-II-12 REPRESENTATION OF ARRANGEMENT OF PIPING BETWEEN PRIMARY AND SECONDARY ELEMENTS WHEN PRIMARY IS BELOW SECONDARY AND FLUID IS A LIQUID, STEAM OR CONDENSABLE GAS

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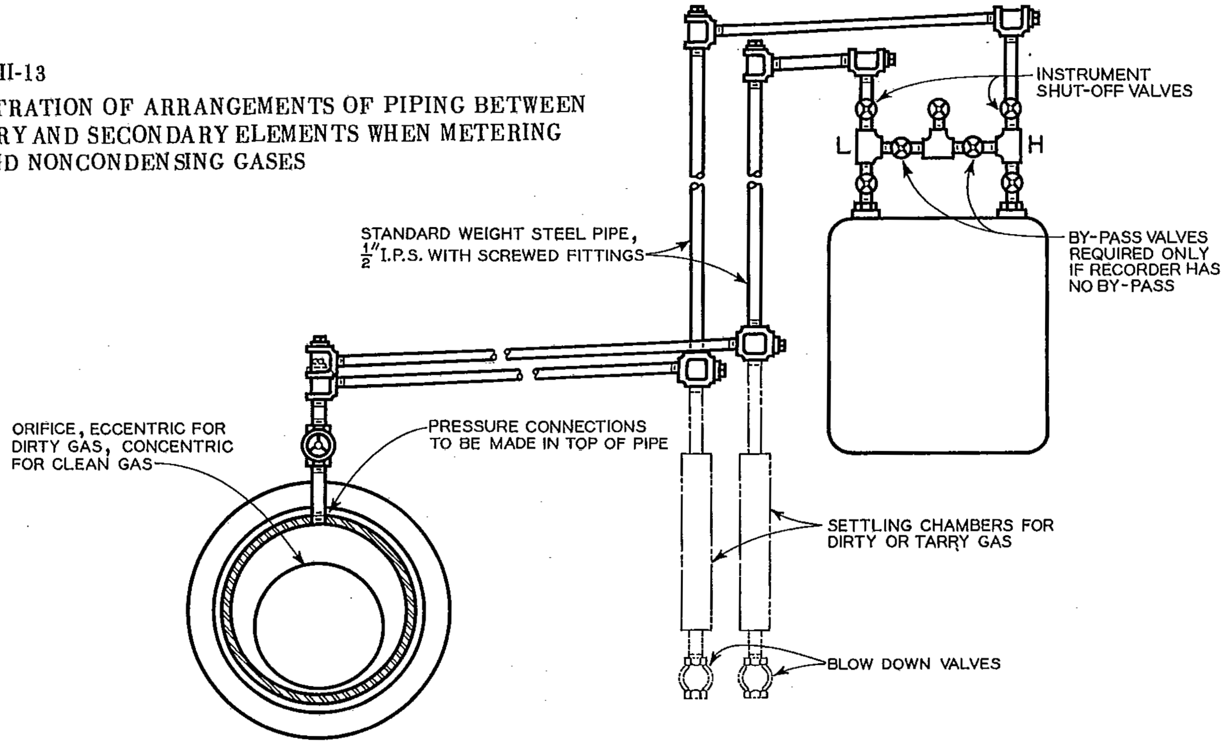
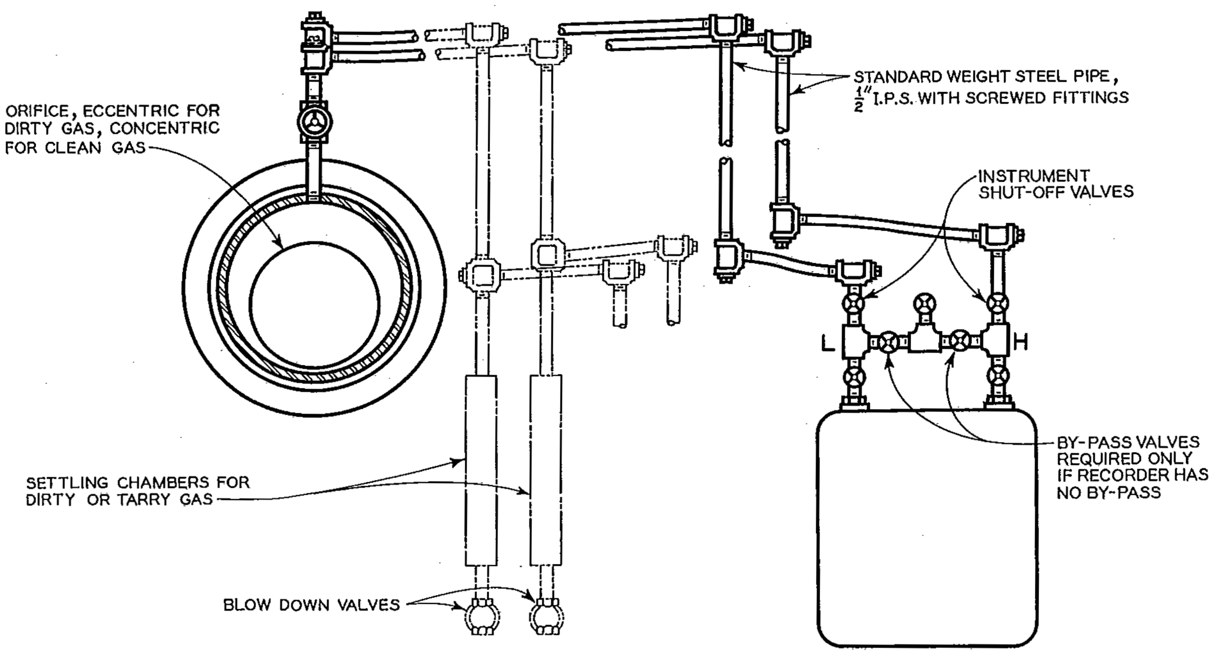
FIG. II-II-13

ILLUSTRATION OF ARRANGEMENTS OF PIPING BETWEEN PRIMARY AND SECONDARY ELEMENTS WHEN METERING AIR AND NONCONDENSING GASES



SECONDARY ELEMENT ABOVE ORIFICE OR FLOW NOZZLE

NOTE: SLOPE OF ALL CONNECTING PIPES SHOULD NOT BE LESS THAN 1 INCH PER FOOT TOWARD PRIMARY ELEMENT



SECONDARY ELEMENT BELOW ORIFICE OR FLOW NOZZLE

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PROCEDURE FOR BLOWING OUT SETTLING CHAMBERS

1. OPEN BY-PASS VALVES AT SECONDARY AND CLOSE THE VALVES IN PRESSURE LINES H AND L.
2. OPEN SETTLING CHAMBER BLOW DOWN VALVES. WHEN ALL SEDIMENT IS OUT, ALLOW CLEAN WATER, STEAM OR GAS TO BLOW THROUGH FOR 10 TO 20 SECONDS.
3. CLOSE BLOW DOWN VALVES TIGHT.
4. IF MEASURING STEAM, WAIT UNTIL CHAMBERS AND CONNECTING PIPING ARE REFILLED WITH CONDENSATE.
5. SLOWLY OPEN VALVES IN LOW PRESSURE LINE L. CLOSE BY-PASS VALVES. SLOWLY OPEN VALVES IN HIGH PRESSURE LINE H.

NOTES:

SETTLING CHAMBERS SHOULD BE AT LEAST 2'-0" TO THE SIDE OF OR ABOVE THE SECONDARY ELEMENT.

CONNECTING PIPING FROM SETTLING CHAMBERS SLOPE AT LEAST 1 INCH PER FOOT TOWARD THE SECONDARY ELEMENT.

TO ALLEVIATE ACCUMULATION OF HYDROGEN GAS IN THE PIPING OR INSTRUMENT PRESSURE CASING, INSERT A MAGNESIUM ROD IN EACH SETTLING CHAMBER.

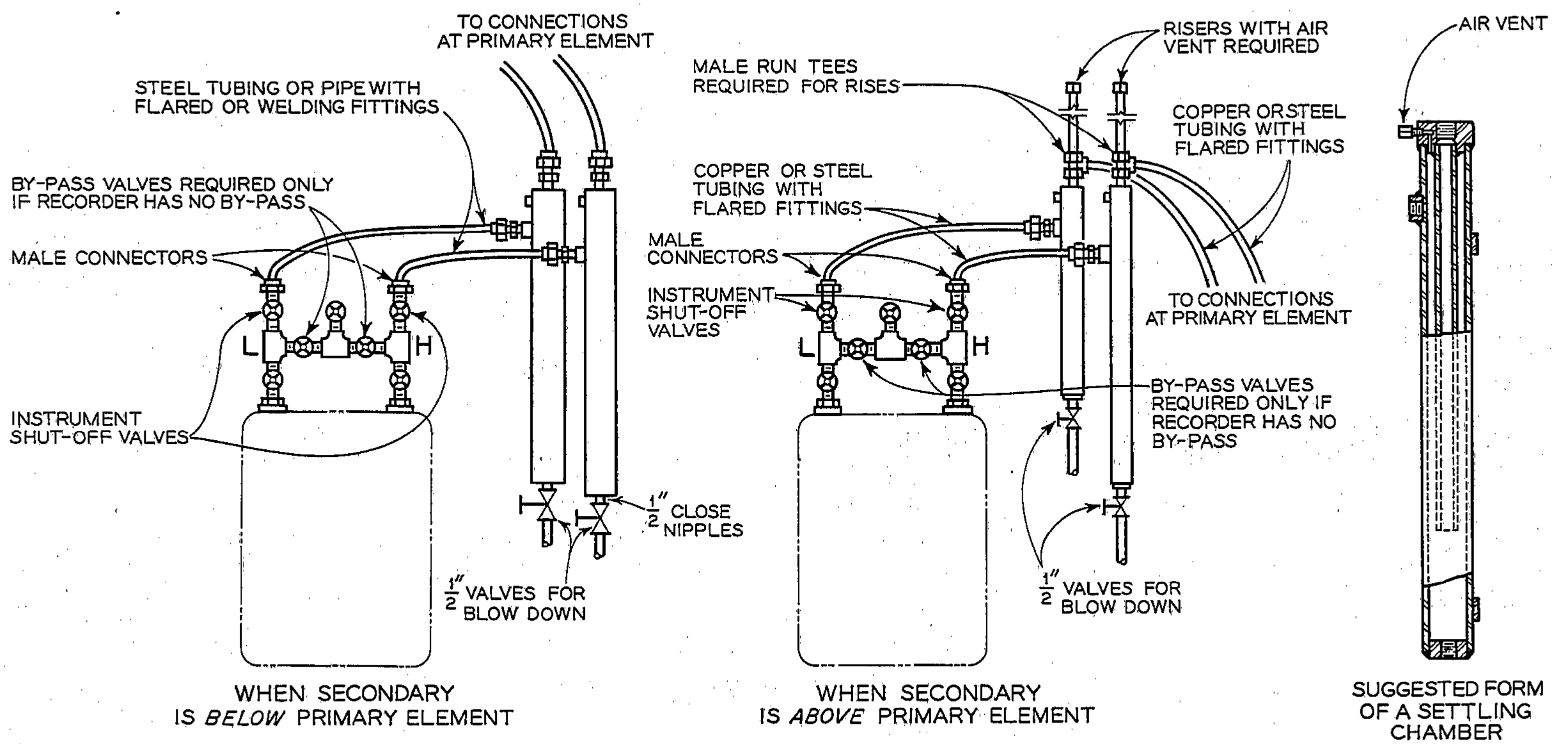

FIG. II-II-14 SUGGESTED METHODS OF CONNECTING SETTLING CHAMBERS, IF NEEDED, INTO PIPING BETWEEN PRIMARY AND SECONDARY ELEMENTS

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If needed, settling chambers or dirt traps may be installed as shown in Fig. II-II-14.

Shutoff valves to pressure instruments should be close to the instrument. For steam service, they should be suitable for the temperature of saturated steam corresponding to the actual line pressure. For use in the metering of other fluids, such valves should be suitable for the actual mainline pressure and temperature. For differential pressure gages, if a by-pass is not provided integrally with the instrument, such a valve should be incorporated between the shutoff valves and the gage itself, as illustrated in Figs. II-II-11 through II-II-13.

II-II-16 Drains. When measuring steam in a horizontal pipe, suitable drains or blowoffs should be provided on the under side of the pipe on the inlet and outlet sides of the primary element. If the pressures are measured through annular chambers, there should be drains in these chambers also. In other than horizontal installations, the pipe adjacent to the primary element should be drained at the point of minimum elevation. The valves or cocks used on these drains should be ones that will close tightly.

When measuring an incompressible fluid, vents should be located on the upper side of a horizontal pipe to eliminate any entrapped gas. In other than horizontal installations, the piping system should be vented at the highest point.

II-II-17 Calibrations. If a calibration of a particular differential pressure producer is desired or required, it should be made with the adjacent sections of pipe in which such primary element is to be used. The length of the actual piping to be used in the calibration should be at least that shown in the block of Fig. II-II-1, corresponding the closest to the actual installation arrangement. For best results, any fittings which immediately precede the inlet run should be used in the calibration.

Whenever possible, the calibration range should encompass the entire range of Reynolds numbers corresponding to the rates of flow to be encountered in use. When the calibration facilities are inadequate to attain the highest Reynolds numbers to be encountered in use, the indications of the calibration may be extrapolated graphically or analytically. However, when such an extrapolation is used, the tolerance to be applied thereto should be increased to possibly double that of the direct calibration results.

Note: The extrapolation should not be extended to a condition that would correspond to a pressure ratio, p_2/p_1, below about 0.52, as a change of flow regime may occur in this pressure-ratio region. This does not apply to sonic-flow nozzles discussed latter.

II-II-18 Other Considerations. When the temperature of the fluid is above or below the ambient temperature, so that the difference in temperature might affect the fluid properties, thermal insulation of the entire meter section may be advisable.

If the meter is to be used in a special service, e.g., an acceptance test, the primary element should be sized so as to produce as high a differential pressure as operating conditions and auxiliary equipment will permit. During the time of such use, the primary element should be clean and undamaged; the inlet edge of an orifice, square and sharp; the inlet and throat sections of a flow nozzle, clean and smooth; and the inlet and throat of a Venturi tube, free of scale or incrustations. Such conditions should be established, by inspection, if possible, both before and after use.

It should be possible to read or estimate the smallest division of the scale of a manometer, pressure gage or chart of a recording gage to a value that will give the accuracy required for the service or test.

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Chapter II-III

Primary Elements and Equations for Computing Rates of Flow

II-III-1 Symbols. The following symbols are used in describing the primary elements and in the equations given for computing rates of flow. Letters used to represent special factors in some equations are defined at the place of use, as also are special subscripts.

a	Area of an orifice, flow nozzle or Venturi throat	in.^{2}
C	Coefficient of discharge	ratio
c_p	Specific heat of a fluid at constant pressure	Btu/lb$_m$/°R
c_v	Specific heat of a fluid at constant volume	Btu/lb$_m$/°R
D	Diameter of pipe or meter tube	in.
d	Diameter of orifice, flow nozzle throat or Venturi throat	in.
E	Velocity of approach factor $= 1/\sqrt{1-\beta^4}$	number
F	Isentropic expansion function of a real gas (equation (I-5-124))	ratio
F_a	Area thermal expansion factor, from Fig. II-I-3	ratio
F_i	Isentropic expansion function of an ideal gas (equation (I-5-104))	ratio
G	Specific gravity; for a liquid the ratio of density of liquid to that of water at a defined temperature; for gases the ratio of the molecular weight of the gas to molecular weight of air	ratio
g	Acceleration due to gravity, local	ft/sec
g_c	Proportionality constant in the force-mass-acceleration equation = 32.174	number
h	Effective differential pressure	ft of fluid
h_w	Effective differential pressure	in. of water at 68 F
K	Flow coefficient $= CE$	ratio
MW	Molecular weight of a fluid	number
m	Mass rate of flow	lb$_m$/sec
p	Pressure, absolute	psia
p_t	Total or stagnation pressure	psia
q	Volume rate of flow	cu ft/sec
R	Gas constant in $pv = RT$ (here p is lb$_f$/ft^2)	ft · lb$_f$/lb$_m$ · °R

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R_D	Reynolds number based on D	ratio
R_d	Reynolds number based on d	ratio
r	Ratio of outlet to inlet static pressure $= p_2/p_1$	ratio
T	Absolute temperature	°R
V	Velocity	ft/sec
V_s	Velocity of sound (acoustic velocity)	ft/sec
v	Specific volume $= 1/\rho$	cu ft/lb$_m$
x	Ratio of differential pressure to inlet static pressure $= \Delta p/p_1$	ratio
Y	Expansion factor for a gas	ratio
Z	Compressibility factor for a real gas	ratio
β (beta)	Ratio of diameters $= d/D$	ratio
Γ (gamma)	Isentropic exponent of a real gas, a function of p_1, p_2 and T	number
γ (gamma)	Ratio of specific heats of a gas (ideal) $= c_p/c_v$	ratio
Δp (delta p)	Differential pressure $= p_1 - p_2$	psi
λ (lambda)	A Reynolds number reciprocal $= 1000/\sqrt{R_D}$ $= 1000/\sqrt{\beta R_d}$	ratio
μ (mu)	Absolute viscosity of a fluid	lb$_m$/ft · sec
ρ (rho)	Density	lb$_m$/cu ft
τ (tau)	Deflection of an orifice plate	in.
ϕ^* (phi)	Sonic-flow function of a real gas (equation (I-5-125))	number
ϕ_i^* (phi)	Sonic-flow function of an ideal gas (equation (I-5-105))	number

II-III-2 Thin-Plate Square-Edged Orifice:
Material. The orifice plate should be stainless steel or other noncorrodible material suited to the fluid to be metered at the expected operating conditions. When the temperature of the fluid will exceed 600 F, the plate material should have a coefficient of thermal expansion no greater than that of the pipe flanges between which the plate will be installed. Whenever possible, the rate of change of temperature of the entire primary assembly should be kept as low as possible to avoid distortion of the plate from thermal stress.

The recommended thicknesses of orifice plates are given in Table II-III-1. These values are based on a maximum allowable deflection of $\tau/[0.5(D-d)] \overset{=}{<} 0.05$ (Fig. II-III-1). τ is a function of D, β, Δp and the material of the plate.

Table II-III-1 Minimum Recommended Thicknesses of Orifice Plates (inches)

Diff'l Pressure (in. H_2O)	Internal Diameter of Pipe (inches): 3 and less	6	10	20	30
	$\beta < 0.5$				
< 1000	1/8	1/8	3/16	3/8	1/2*
< 200	1/8	1/8	1/8	1/4	3/8
< 100	1/8	1/8	1/8	1/4	3/8
	$\beta > 0.5$				
< 1000	1/8	1/8	3/16	3/8	1/2
< 200	1/8	1/8	1/8	3/16	3/8
< 100	1/8	1/8	1/8	3/16	1/4

*For 1/2-in. plate in 30-in. line, maximum differential = 500 in.

II-III-3 The conventional and preferred use of orifices is to have the center of the orifice on the center line of the meter tube when installed. The use of eccentric and segmental orifices is treated later.

The width of the cylindrical surface of the orifice itself, measured normal to the plane of the inlet face of the plate, should be between 0.01 D and 0.02 D or $d/8$, whichever is smaller. If the thickness of the orifice plate exceeds the minimum of this requirement, then the outlet corner of the orifice should be beveled at an angle of about 45 deg to the face of the plate sufficiently to provide the minimum face width.

The inlet edge or corner of the orifice must be square, sharp and free from burrs, nicks, wire edge or rounding.

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The inlet face of the orifice plate should be flat within 0.010 per in. of pipe diameter.

The actual diameter of the orifice hole should be carefully and accurately determined after all machine work on the plate has been completed. In doing this, particular care must be used not to damage or alter the inlet corner of the hole. Measurements should be made on at least three and preferably more diameters to determine a reliable average value of d. No diameter should differ from the average diameter by more than 0.05 per cent and preferably not more than 0.02 per cent.

II-III-4 For use in horizontal pipes, a drain hole may be provided in an orifice plate so located as to be flush with the bottom of the pipe when measuring gaseous fluids or flush with the top of the pipe when measuring liquids. If such a drain hole is provided, the diameter should be such that the hole area is

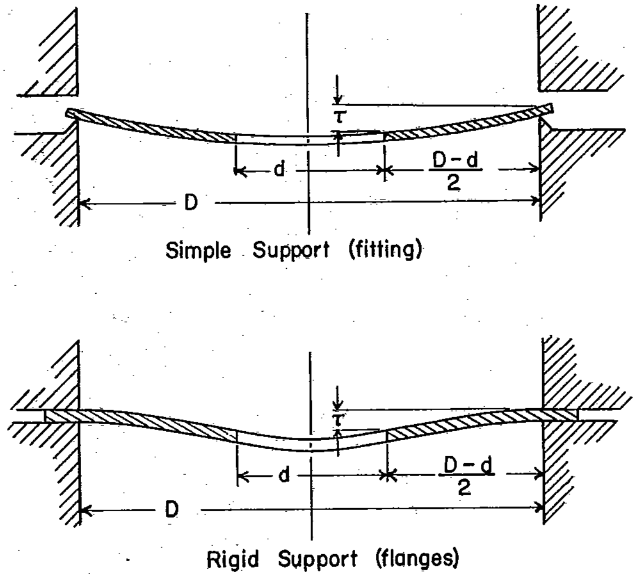

FIG. II-III-1 DEFLECTION OF AN ORIFICE PLATE BY DIFFERENTIAL PRESSURE

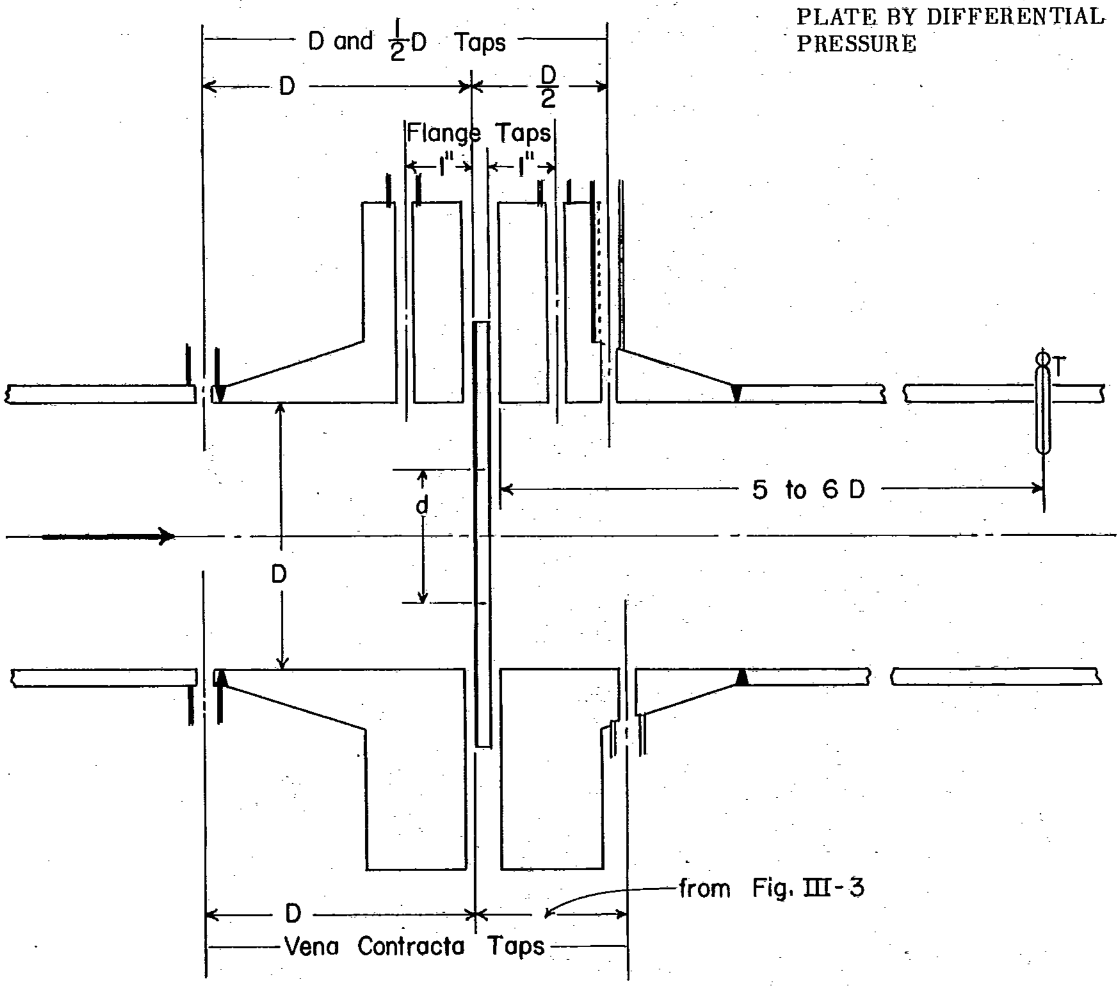

FIG. II-III-2 LOCATIONS OF PRESSURE TAPS USED WITH ORIFICES (WHEN A THERMOMETER IS REQUIRED, THE WELL FOR IT MAY BE LOCATED AS SHOWN BY T.)

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less than 0.002 of the area of the orifice. In general, drain holes are considered of little value and are not recommended.

The design of the orifice plate and its outside diameter should be such as to facilitate centering the orifice accurately in the pipe line. Any method used for centering the orifice should provide that the center of the orifice is not further than 1/32 in. from the center line of the meter tube or pipe.

II-III-5 Pressure Taps. Any one of the following three pairs of pressure taps, shown by Fig. II-III-2, may be used:

1. *Flange Taps:* The centers of the inlet and outlet pressure taps are, respectively, 1 in. from the inlet and outlet faces of the orifice plate, are subject to a tolerance of ± 1/16 in. for β up to 0.40, and vary linearly to ± 1/64 in. at $\beta = 0.75$.

2. 1 *D and* 1/2 *D Taps:* The center of the inlet pressure tap is 1 *D* preceding the inlet face of the orifice plate. The center of the outlet (downstream) pressure tap is 1/2 *D* from the *inlet* face of the orifice plate. These distances are subject to a tolerance varying linearly from ± 0.2 *D* at $\beta = 0.20$ to ± 0.05 *D* at $\beta = 0.75$.

3. *Vena Contracta Taps:* The center of the inlet tap is 1 *D* preceding the inlet face of the orifice plate. The distance of the outlet pressure tap from the *inlet* face of the orifice plate depends upon the diameter ratio, β, of the orifice to be used as shown by the heavy line of Fig. II-III-3. These distances are subject to a tolerance varying linearly from ± 0.2 D at $\beta = 0.20$ to ± 0.05 *D* at $\beta = 0.75$.

The pressure tap holes should be drilled (and preferably reamed) perpendicular to the inner surface of the meter tube or pipe in which the orifice plate is mounted. The corner of the hole with the inner surface of the pipe must be free of burrs and wire edge. It may be left square and sharp or dulled (rounded) very slightly.

II-III-6 Pressure Loss. The overall pressure loss with an orifice meter is shown by Fig. II-III-4.

II-III-7 Coefficients. Whenever possible it is desirable to calibrate an orifice in the meter-tube assembly in which it is to be used. When this is not done, the discharge coefficient to be used in computing the rate of fluid flow may be evaluated by the equation below which applies to the pressure taps used or from the value read from the appropriate table.

Values of discharge coefficients are given for flange taps in Table II-III-2, 1-*D* and 1/2-*D* taps in Table II-III-3, and vena contracta taps in Table II-III-4.

The equations and special values of the symbols by which the tables were computed are:

$$C \quad = K/E = K\sqrt{1-\beta^4}$$

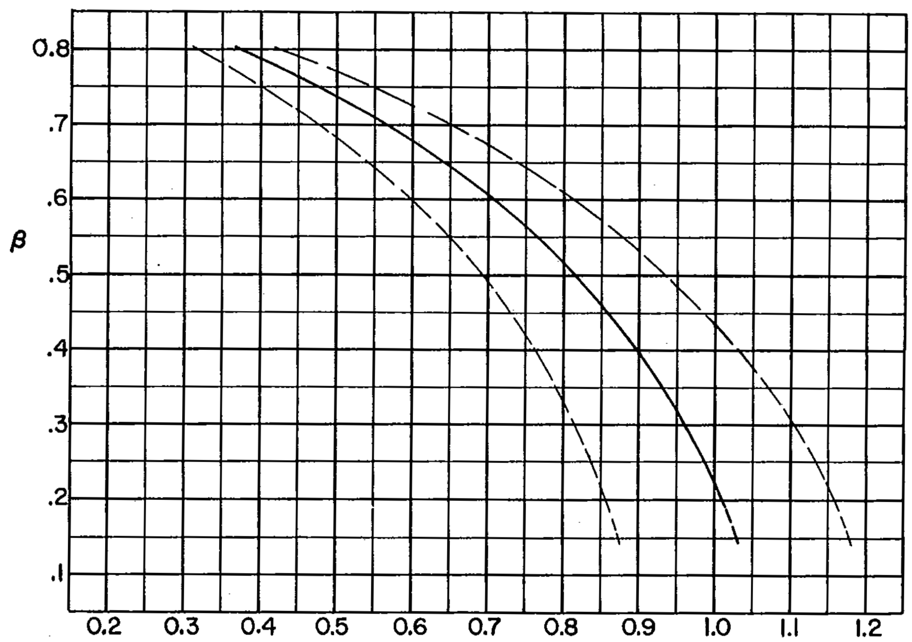

FIG. II-III-3 LOCATION OF VENA CONTRACTA OUTLET PRESSURE TAP WITH CONCENTRIC SQUARE-EDGED ORIFICES (BROKEN LINES SHOW MAXIMUM VARIATION LIMITS.)

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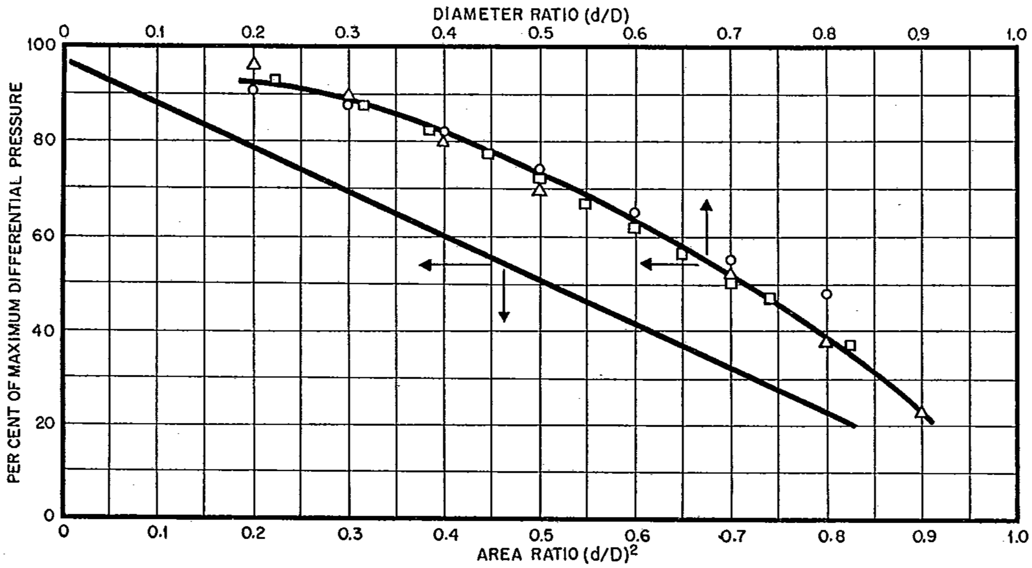

FIG. II-III-4 OVERALL PRESSURE LOSS ACROSS THIN-PLATE ORIFICES

K = Flow coefficient corresponding to any specific set of values of D, β, and R_d (or R_D)

K_o = The limiting value of K for any specific values of D and β when R_d (or R_D) becomes infinitely large

$R_D = \beta R_d$

For flange taps,

K_e = The particular value of K for any specific values of D and β when $R_d = (10^6\, d)/15$

$$K = K_o\left(1 + \frac{A}{R_d}\right)$$

$$K_o = K_e\left(\frac{10^6 d}{10^6 d + 15A}\right)$$

$$K_e = 0.5993 + \frac{0.007}{D} + \left(0.364 + \frac{0.076}{\sqrt{D}}\right)\beta^4 + 0.4\left(1.6 - \frac{1}{D}\right)^5\left[\left(0.07 + \frac{0.5}{D}\right) - \beta\right]^{5/2} - \left(0.009 + \frac{0.034}{D}\right)(0.5 - \beta)^{3/2} + \left(\frac{65}{D^2} + 3\right)(\beta - 0.7)^{5/2} \qquad \text{(II-III-1)}$$

and

$$A = d\left(830 - 5000\beta + 9000\beta^2 - 4200\beta^3 + \frac{530}{\sqrt{D}}\right) \qquad \text{(II-III-2)}$$

Note: In equation (II-III-1), each of the last three terms for some value of β reduces to the form of $x\sqrt{-1}$, i.e., to an "imaginary" number. In such cases the term is to be dropped.

For 1-D and 1/2-D taps and vena contracta taps,

$$K = K_o + b\,\lambda$$

and

$$\lambda = 1000/\sqrt{R_D} = 1000/\sqrt{\beta R_d}$$

For the 1-D and 1/2-D taps,

$$K_o = (0.6014 - 0.01352 D^{-1/4}) + (0.3760 + 0.07257 D^{-1/4})\left(\frac{0.00025}{D^2\beta^2 + 0.0025D} + \beta^4 + 1.5\beta^{16}\right) \qquad \text{(II-III-3)}$$

and

$$b = \left(0.0002 + \frac{0.0011}{D}\right) + \left(0.0038 + \frac{0.0004}{D}\right)\left[\beta^2 + (16.5 + 5D)\beta^{16}\right] \qquad \text{(II-III-4)}$$

For vena contracts taps,

$$K_o = 0.5922 + 0.4252\left(\frac{0.0006}{D^2\beta^2 + 0.01D} + \beta^4 + 1.25\beta^{16}\right) \qquad \text{(II-III-5)}$$

and

$$b = 0.00025 + 0.002325(\beta + 1.75\beta^4 + 10\beta^{12} + 2D\beta^{16}) \qquad \text{(II-III-6)}$$

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Table II-III-2 (a) Flange Taps: Discharge Coefficients, C, for Square-Edged Orifices

(2-in. Pipe, D = 2.067 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6109	.6089	.6075	.6065	.6056	.6050	.6038	.6030	.6020	.6014	.6006	.6002	.5993	.5992
.2000	.6098	.6077	.6061	.6050	.6041	.6034	.6021	.6012	.6001	.5995	.5986	.5982	.5972	.5970
.2500	.6104	.6081	.6064	.6052	.6043	.6035	.6021	.6012	.6001	.5994	.5985	.5980	.5969	.5968
.3000	.6125	.6100	.6083	.6070	.6059	.6051	.6037	.6027	.6015	.6007	.5998	.5993	.5981	.5979
.3500	.6156	.6130	.6110	.6096	.6085	.6076	.6060	.6049	.6036	.6028	.6017	.6012	.5999	.5997
.4000	.6197	.6166	.6145	.6128	.6115	.6105	.6087	.6075	.6059	.6050	.6038	.6032	.6017	.6015
.4500	.6249	.6213	.6187	.6168	.6153	.6140	.6119	.6104	.6086	.6075	.6061	.6053	.6036	.6034
.5000	.6314	.6270	.6238	.6215	.6196	.6182	.6155	.6138	.6116	.6102	.6085	.6076	.6055	.6052
.5500	.6391	.6337	.6298	.6269	.6246	.6228	.6195	.6174	.6147	.6130	.6109	.6098	.6072	.6068
.5750	.6434	.6374	.6331	.6298	.6273	.6253	.6217	.6193	.6163	.6145	.6121	.6109	.6080	.6076
.6000	.6479	.6412	.6365	.6329	.6301	.6279	.6239	.6212	.6179	.6158	.6132	.6118	.6086	.6082
.6250	.6525	.6451	.6399	.6359	.6328	.6304	.6259	.6230	.6193	.6171	.6141	.6126	.6091	.6087
.6500	.6571	.6489	.6431	.6388	.6354	.6327	.6278	.6245	.6205	.6180	.6148	.6131	.6092	.6087
.6750	.6614	.6525	.6461	.6413	.6376	.6346	.6292	.6256	.6212	.6185	.6149	.6131	.6088	.6083
.7000	.6652	.6554	.6484	.6432	.6391	.6359	.6300	.6261	.6212	.6183	.6144	.6124	.6077	.6071
.7250	.6697	.6591	.6515	.6458	.6413	.6378	.6314	.6271	.6218	.6186	.6143	.6122	.6071	.6065
.7500	.6788	.6672	.6589	.6526	.6478	.6439	.6369	.6323	.6265	.6230	.6183	.6160	.6104	.6097

Table II-III-2(b) Flange Taps: Discharge Coefficients, C, for Square-Edged Orifices

(4-in. Pipe, D=4.026 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6126	.6094	.6071	.6054	.6041	.6030	.6011	.5998	.5982	.5973	.5960	.5954	.5939	.5937
.2000	.6147	.6114	.6090	.6072	.6058	.6047	.6026	.6013	.5996	.5986	.5973	.5966	.5950	.5948
.2500	.6167	.6133	.6108	.6090	.6076	.6064	.6044	.6030	.6013	.6003	.5989	.5983	.5966	.5964
.3000	.6187	.6152	.6127	.6108	.6093	.6082	.6061	.6047	.6029	.6019	.6005	.5998	.5981	.5979
.3500		.6178	.6152	.6132	.6116	.6103	.6081	.6066	.6047	.6036	.6021	.6013	.5995	.5993
.4000		.6219	.6188	.6166	.6148	.6133	.6108	.6091	.6069	.6056	.6039	.6031	.6010	.6008
.4500		.6278	.6241	.6213	.6192	.6175	.6144	.6123	.6097	.6082	.6061	.6051	.6026	.6023
.5000			.6311	.6276	.6249	.6228	.6189	.6163	.6131	.6111	.6086	.6073	.6042	.6038
.5500			.6398	.6354	.6320	.6292	.6243	.6210	.6169	.6144	.6112	.6095	.6056	.6051
.5750			.6448	.6398	.6359	.6329	.6273	.6236	.6190	.6162	.6125	.6107	.6062	.6057
.6000			.6501	.6445	.6401	.6367	.6304	.6263	.6211	.6179	.6138	.6117	.6067	.6061
.6250				.6493	.6445	.6406	.6336	.6289	.6231	.6196	.6149	.6126	.6070	.6063
.6500				.6541	.6487	.6444	.6366	.6313	.6248	.6209	.6157	.6131	.6069	.6061
.6750				.6587	.6527	.6479	.6392	.6334	.6262	.6219	.6161	.6132	.6062	.6054
.7000				.6628	.6562	.6508	.6413	.6349	.6269	.6221	.6157	.6125	.6049	.6039
.7250					.6594	.6535	.6430	.6360	.6272	.6220	.6150	.6115	.6031	.6020
.7500					.6634	.6571	.6456	.6379	.6283	.6226	.6149	.6111	.6019	.6007

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Table II-III-2(c) Flange Taps: Discharge Coefficients, C, for Square-Edged Orifices

(8-in. Pipe, D=7.981 in.)

β \ R_d	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6166	.6137	.6114	.6096	.6064	.6042	.6015	.5999	.5977	.5967	.5941	.5937
.2000	.6184	.6155	.6132	.6114	.6081	.6060	.6032	.6016	.5994	.5983	.5957	.5954
.2500	.6190	.6162	.6140	.6122	.6091	.6070	.6044	.6028	.6007	.5997	.5971	.5968
.3000	.6197	.6170	.6148	.6131	.6101	.6080	.6055	.6039	.6019	.6009	.5984	.5981
.3500	.6217	.6189	.6167	.6149	.6117	.6096	.6070	.6054	.6032	.6022	.5996	.5993
.4000	.6261	.6228	.6203	.6183	.6147	.6123	.6093	.6074	.6050	.6038	.6009	.6006
.4500	.6334	.6294	.6263	.6238	.6193	.6164	.6126	.6104	.6074	.6059	.6023	.6019
.5000	.6443	.6390	.6350	.6318	.6259	.6220	.6172	.6142	.6104	.6084	.6037	.6032
.5500	.6586	.6518	.6464	.6421	.6344	.6292	.6228	.6190	.6138	.6112	.6051	.6043
.5750		.6592	.6531	.6482	.6393	.6334	.6260	.6216	.6157	.6127	.6056	.6047
.6000		.6674	.6603	.6547	.6446	.6378	.6294	.6243	.6175	.6142	.6061	.6050
.6250			.6680	.6616	.6501	.6424	.6328	.6270	.6193	.6155	.6062	.6051
.6500			.6759	.6687	.6556	.6469	.6361	.6295	.6208	.6165	.6061	.6048
.6750				.6757	.6610	.6513	.6391	.6317	.6220	.6171	.6053	.6039
.7000				.6823	.6660	.6551	.6415	.6333	.6224	.6170	.6039	.6023
.7250					.6705	.6584	.6433	.6343	.6222	.6162	.6017	.5999
.7500					.6749	.6617	.6451	.6351	.6219	.6153	.5993	.5973

Table II-III-2(d) Flange Taps: Discharge Coefficients, C, for Square-Edged Orifices

(16-in. Pipe, D=15.25 in.)

β \ R_d	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6244	.6213	.6158	.6122	.6076	.6049	.6012	.5994	.5950	.5944
.2000	.6249	.6220	.6167	.6131	.6087	.6060	.6025	.6007	.5965	.5960
.2500	.6236	.6210	.6161	.6129	.6089	.6065	.6032	.6016	.5977	.5973
.3000			.6157	.6128	.6091	.6068	.6039	.6024	.5989	.5984
.3500			.6167	.6138	.6101	.6079	.6050	.6035	.5999	.5995
.4000			.6202	.6168	.6127	.6102	.6068	.6052	.6011	.6006
.4500			.6269	.6226	.6172	.6140	.6098	.6076	.6025	.6018
.5000				.6314	.6241	.6197	.6139	.6110	.6039	.6031
.5500				.6433	.6332	.6272	.6191	.6150	.6054	.6042
.5750				.6504	.6386	.6315	.6221	.6173	.6060	.6046
.6000				.6581	.6444	.6362	.6252	.6197	.6065	.6049
.6250				.6664	.6505	.6410	.6283	.6220	.6068	.6049
.6500				.6749	.6568	.6459	.6314	.6241	.6067	.6045
.6750				.6836	.6630	.6506	.6341	.6259	.6061	.6036
.7000				.6919	.6687	.6548	.6363	.6270	.6047	.6020
.7250				.6999	.6740	.6585	.6378	.6275	.6026	.5995
.7500				.7077	.6791	.6619	.6390	.6276	.6001	.5966

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Table II-III-3 (a) Taps at 1 D and 1/2 D: Discharge Coefficients, C, for Square-Edged Orifices

(2-in. Pipe, D = 2.067 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6125	.6106	.6092	.6080	.6071	.6063	.6047	.6035	.6019	.6008	.5990	.5980	.5943	.5934
.2000	.6109	.6092	.6078	.6067	.6058	.6051	.6036	.6025	.6009	.5999	.5982	.5973	.5938	.5930
.2500	.6107	.6090	.6076	.6066	.6057	.6049	.6035	.6024	.6009	.5998	.5982	.5973	.5938	.5930
.3000	.6114	.6096	.6083	.6072	.6063	.6056	.6041	.6030	.6015	.6004	.5988	.5978	.5943	.5935
.3500	.6129	.6111	.6097	.6086	.6077	.6069	.6053	.6042	.6026	.6015	.5999	.5989	.5953	.5944
.4000	.6151	.6132	.6118	.6106	.6097	.6088	.6072	.6061	.6044	.6033	.6015	.6005	.5967	.5958
.4500	.6181	.6161	.6146	.6133	.6123	.6115	.6098	.6085	.6068	.6056	.6038	.6027	.5987	.5978
.5000	.6216	.6196	.6180	.6167	.6156	.6147	.6129	.6116	.6098	.6085	.6066	.6054	.6013	.6003
.5500	.6257	.6235	.6218	.6205	.6193	.6184	.6165	.6151	.6132	.6119	.6098	.6086	.6042	.6032
.5750	.6279	.6256	.6239	.6225	.6213	.6203	.6184	.6170	.6150	.6136	.6115	.6103	.6058	.6047
.6000	.6301	.6278	.6260	.6245	.6233	.6223	.6203	.6189	.6168	.6154	.6133	.6120	.6073	.6062
.6250	.6323	.6299	.6281	.6266	.6253	.6243	.6222	.6207	.6186	.6172	.6149	.6136	.6088	.6077
.6500	.6346	.6321	.6302	.6286	.6273	.6262	.6241	.6226	.6204	.6189	.6165	.6151	.6102	.6090
.6750	.6369	.6343	.6323	.6307	.6293	.6282	.6260	.6243	.6220	.6204	.6180	.6165	.6113	.6101
.7000	.6395	.6367	.6345	.6328	.6314	.6302	.6278	.6260	.6236	.6219	.6193	.6178	.6122	.6109
.7250	.6424	.6394	.6371	.6352	.6336	.6323	.6298	.6278	.6252	.6234	.6205	.6188	.6128	.6114
.7500	.6463	.6429	.6403	.6382	.6364	.6349	.6320	.6299	.6269	.6249	.6217	.6198	.6130	.6114

Table II-III-3 (b) Taps at 1 D and 1/2 D: Discharge Coefficients, C, for Square-Edged Orifices

(4-in. Pipe, D = 4.026 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6067	.6054	.6044	.6037	.6030	.6024	.6014	.6006	.5994	.5987	.5975	.5968	.5943	.5937
.2000	.6063	.6051	.6041	.6033	.6027	.6022	.6011	.6003	.5993	.5985	.5974	.5967	.5942	.5936
.2500	.6068	.6055	.6046	.6038	.6031	.6026	.6015	.6007	.5996	.5989	.5977	.5970	.5945	.5939
.3000	.6080	.6067	.6056	.6048	.6041	.6036	.6025	.6016	.6005	.5997	.5984	.5977	.5951	.5945
.3500		.6084	.6073	.6064	.6057	.6051	.6039	.6030	.6018	.6009	.5996	.5989	.5961	.5954
.4000		.6107	.6095	.6086	.6078	.6072	.6059	.6050	.6036	.6027	.6013	.6005	.5975	.5968
.4500		.6136	.6124	.6114	.6105	.6098	.6084	.6074	.6060	.6050	.6035	.6026	.5994	.5986
.5000			.6157	.6146	.6138	.6130	.6115	.6104	.6089	.6078	.6062	.6052	.6018	.6009
.5500			.6195	.6183	.6174	.6166	.6150	.6138	.6121	.6110	.6093	.6082	.6045	.6036
.5750			.6215	.6203	.6193	.6184	.6168	.6156	.6138	.6127	.6109	.6098	.6059	.6050
.6000			.6235	.6223	.6212	.6203	.6186	.6173	.6156	.6143	.6124	.6113	.6073	.6063
.6250				.6243	.6232	.6223	.6205	.6191	.6173	.6160	.6140	.6128	.6086	.6076
.6500				.6263	.6251	.6242	.6223	.6209	.6189	.6176	.6155	.6142	.6098	.6087
.6750				.6284	.6272	.6261	.6241	.6226	.6205	.6191	.6168	.6155	.6107	.6096
.7000				.6308	.6294	.6283	.6261	.6244	.6221	.6205	.6181	.6166	.6114	.6102
.7250					.6321	.6308	.6283	.6265	.6239	.6221	.6193	.6177	.6118	.6105
.7500					.6356	.6341	.6312	.6290	.6260	.6239	.6207	.6188	.6120	.6103

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Table II-III-3 (c) Taps at 1 D and 1/2 D: Discharge Coefficients, C, for Square-Edged Orifices

(8-in. Pipe, D = 7.981 in.)

β \ R_d	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6028	.6022	.6017	.6012	.6004	.5998	.5990	.5984	.5975	.5970	.5950	.5946
.2000	.6029	.6023	.6018	.6014	.6005	.5999	.5991	.5985	.5976	.5971	.5951	.5947
.2500	.6036	.6030	.6025	.6020	.6012	.6005	.5996	.5990	.5981	.5975	.5955	.5950
.3000	.6049	.6042	.6036	.6031	.6022	.6015	.6006	.5999	.5989	.5983	.5961	.5956
.3500	.6066	.6059	.6053	.6047	.6037	.6030	.6019	.6012	.6001	.5994	.5971	.5965
.4000	.6089	.6081	.6074	.6068	.6057	.6049	.6038	.6030	.6018	.6010	.5984	.5978
.4500	.6117	.6108	.6101	.6094	.6082	.6073	.6061	.6052	.6039	.6031	.6002	.5995
.5000	.6150	.6140	.6132	.6125	.6112	.6102	.6088	.6079	.6064	.6055	.6024	.6017
.5500	.6186	.6176	.6167	.6160	.6145	.6134	.6119	.6109	.6093	.6083	.6049	.6041
.5750		.6195	.6186	.6178	.6162	.6151	.6135	.6125	.6108	.6098	.6062	.6054
.6000		.6214	.6205	.6196	.6180	.6168	.6152	.6140	.6123	.6112	.6075	.6066
.6250			.6224	.6215	.6198	.6186	.6168	.6156	.6137	.6126	.6086	.6077
.6500			.6245	.6236	.6217	.6204	.6185	.6172	.6152	.6140	.6097	.6087
.6750				.6258	.6238	.6223	.6202	.6188	.6166	.6152	.6105	.6094
.7000				.6285	.6262	.6245	.6221	.6205	.6180	.6165	.6111	.6099
.7250					.6293	.6273	.6245	.6226	.6197	.6179	.6115	.6100
.7500					.6337	.6313	.6278	.6254	.6218	.6196	.6118	.6099

Table II-III-3 (d) Taps at 1 D and 1/2 D: Discharge Coefficients, C, for Square-Edged Orifices

(16-in. Pipe, D = 15.25 in.)

β \ R_d	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6015	.6012	.6005	.6000	.5993	.5988	.5980	.5976	.5959	.5956
.2000	.6018	.6015	.6008	.6002	.5995	.5990	.5982	.5978	.5961	.5957
.2500	.6026	.6022	.6015	.6009	.6001	.5996	.5987	.5982	.5965	.5960
.3000			.6026	.6020	.6011	.6005	.5996	.5990	.5971	.5966
.3500			.6041	.6034	.6024	.6018	.6008	.6002	.5980	.5975
.4000			.6061	.6053	.6042	.6035	.6024	.6017	.5993	.5987
.4500			.6085	.6077	.6065	.6057	.6044	.6037	.6010	.6003
.5000				.6104	.6091	.6082	.6068	.6060	.6031	.6024
.5500				.6135	.6121	.6111	.6096	.6087	.6054	.6046
.5750				.6152	.6137	.6126	.6110	.6100	.6066	.6058
.6000				.6169	.6153	.6142	.6125	.6114	.6077	.6069
.6250				.6188	.6170	.6158	.6139	.6128	.6088	.6079
.6500				.6208	.6189	.6175	.6155	.6142	.6098	.6088
.6750				.6233	.6210	.6195	.6171	.6157	.6107	.6095
.7000				.6265	.6238	.6220	.6191	.6174	.6114	.6099
.7250				.6310	.6276	.6253	.6217	.6196	.6120	.6102
.7500				.6376	.6332	.6302	.6255	.6227	.6128	.6104

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Table II-III-4 (a) Vena Contracta Taps: Discharge Coefficients, C, for Square-Edged Orifices

(2-in. Pipe, $D = 2.067$ in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6100	.6086	.6076	.6067	.6060	.6054	.6043	.6034	.6022	.6014	.6001	.5994	.5996	.5960
.2000	.6099	.6085	.6074	.6065	.6058	.6051	.6039	.6030	.6018	.6009	.5996	.5988	.5960	.5953
.2500	.6105	.6090	.6079	.6070	.6062	.6055	.6043	.6033	.6020	.6011	.5998	.5989	.5960	.5953
.3000	.6117	.6101	.6089	.6079	.6071	.6065	.6051	.6041	.6028	.6018	.6004	.5995	.5964	.5956
.3500	.6134	.6118	.6105	.6095	.6086	.6079	.6065	.6054	.6040	.6030	.6014	.6005	.5972	.5964
.4000	.6157	.6140	.6126	.6115	.6106	.6099	.6084	.6072	.6057	.6046	.6030	.6020	.5985	.5977
.4500	.6187	.6168	.6154	.6142	.6132	.6124	.6108	.6096	.6080	.6069	.6051	.6041	.6003	.5994
.5000	.6223	.6203	.6187	.6175	.6164	.6156	.6138	.6126	.6108	.6096	.6077	.6066	.6026	.6017
.5500	.6264	.6243	.6226	.6213	.6201	.6192	.6174	.6160	.6141	.6128	.6108	.6096	.6053	.6042
.5750	.6287	.6264	.6247	.6233	.6221	.6212	.6192	.6178	.6158	.6145	.6124	.6111	.6066	.6056
.6000	.6310	.6287	.6269	.6254	.6242	.6232	.6212	.6197	.6176	.6162	.6140	.6127	.6080	.6069
.6250	.6334	.6309	.6290	.6275	.6262	.6251	.6230	.6215	.6193	.6178	.6155	.6142	.6093	.6081
.6500	.6357	.6331	.6311	.6295	.6282	.6271	.6249	.6232	.6209	.6194	.6170	.6155	.6103	.6091
.6750	.6380	.6353	.6332	.6315	.6301	.6289	.6265	.6248	.6224	.6207	.6182	.6166	.6112	.6099
.7000	.6403	.6374	.6351	.6333	.6318	.6305	.6280	.6262	.6236	.6218	.6191	.6174	.6116	.6102
.7250	.6425	.6394	.6369	.6349	.6333	.6319	.6292	.6273	.6245	.6226	.6196	.6179	.6116	.6101
.7500	.6446	.6412	.6385	.6364	.6346	.6331	.6302	.6280	.6250	.6229	.6197	.6178	.6109	.6093

Table II-III-4 (b) Vena Contracta Taps: Discharge Coefficients, C, for Square-Edged Orifices

(4-in. Pipe, $D = 4.026$ in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6084	.6071	.6060	.6052	.6045	.6039	.6027	.6018	.6006	.5998	.5986	.5978	.5951	.5944
.2000	.6089	.6075	.6064	.6055	.6048	.6042	.6030	.6021	.6008	.6000	.5987	.5979	.5951	.5944
.2500	.6099	.6084	.6072	.6063	.6055	.6049	.6036	.6027	.6014	.6005	.5991	.5983	.5953	.5946
.3000	.6112	.6097	.6085	.6075	.6067	.6060	.6047	.6037	.6023	.6014	.5999	.5990	.5959	.5952
.3500	.6131	.6114	.6101	.6091	.6083	.6075	.6061	.6051	.6036	.6026	.6011	.6002	.5969	.5961
.4000	.6155	.6137	.6124	.6113	.6104	.6096	.6081	.6070	.6054	.6044	.6027	.6018	.5982	.5974
.4500		.6166	.6152	.6140	.6130	.6122	.6106	.6094	.6078	.6066	.6049	.6038	.6001	.5992
.5000		.6201	.6186	.6173	.6163	.6154	.6137	.6124	.6106	.6094	.6076	.6064	.6024	.6015
.5500		.6242	.6225	.6211	.6200	.6191	.6172	.6159	.6140	.6127	.6106	.6094	.6051	.6041
.5750			.6246	.6232	.6220	.6210	.6191	.6177	.6157	.6144	.6123	.6110	.6065	.6054
.6000			.6268	.6253	.6241	.6231	.6211	.6196	.6175	.6161	.6139	.6126	.6079	.6068
.6250			.6290	.6274	.6262	.6251	.6230	.6214	.6192	.6178	.6155	.6141	.6092	.6080
.6500				.6295	.6282	.6270	.6248	.6232	.6209	.6193	.6169	.6154	.6103	.6090
.6750				.6315	.6301	.6289	.6266	.6248	.6224	.6207	.6181	.6166	.6111	.6098
.7000				.6335	.6319	.6307	.6281	.6263	.6237	.6219	.6191	.6175	.6116	.6102
.7250					.6336	.6322	.6295	.6275	.6247	.6227	.6197	.6180	.6116	.6100
.7500					.6352	.6337	.6307	.6285	.6254	.6232	.6200	.6180	.6110	.6093

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Table II-III-4 (c) Vena Contracta Taps: Discharge Coefficients, C, for Square-Edged Orifices

(8-in. Pipe, D = 7.981 in.)

β \ R_d	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6055	.6047	.6040	.6034	.6022	.6014	.6002	.5994	.5981	.5973	.5946	.5940
.2000	.6061	.6052	.6045	.6039	.6027	.6018	.6006	.5997	.5984	.5976	.5948	.5941
.2500	.6071	.6061	.6054	.6047	.6035	.6025	.6012	.6003	.5989	.5981	.5952	.5945
.3000	.6083	.6074	.6066	.6059	.6045	.6036	.6022	.6012	.5998	.5989	.5950	.5951
.3500	.6101	.6090	.6082	.6074	.6060	.6050	.6035	.6025	.6010	.6001	.5968	.5960
.4000	.6123	.6112	.6103	.6095	.6080	.6069	.6054	.6043	.6027	.6017	.5982	.5973
.4500	.6151	.6139	.6130	.6121	.6106	.6094	.6077	.6066	.6048	.6038	.6001	.5992
.5000	.6185	.6173	.6162	.6153	.6136	.6124	.6106	.6094	.6075	.6064	.6024	.6014
.5500	.6225	.6211	.6200	.6191	.6172	.6158	.6139	.6126	.6106	.6094	.6051	.6041
.5750	.6246	.6232	.6220	.6210	.6191	.6177	.6157	.6144	.6122	.6110	.6065	.6054
.6000		.6253	.6241	.6231	.6211	.6196	.6175	.6161	.6139	.6126	.6079	.6067
.6250		.6275	.6262	.6251	.6230	.6215	.6193	.6178	.6155	.6141	.6091	.6080
.6500			.6283	.6272	.6249	.6233	.6210	.6194	.6169	.6155	.6103	.6090
.6750			.6304	.6292	.6268	.6250	.6226	.6209	.6183	.6167	.6111	.6098
.7000				.6311	.6285	.6266	.6240	.6221	.6193	.6176	.6116	.6102
.7250				.6330	.6302	.6281	.6252	.6232	.6201	.6183	.6117	.6101
.7500					.6318	.6295	.6262	.6240	.6206	.6185	.6112	.6095

Table II-III-4 (d) Vena Contracta Taps: Discharge Coefficients, C, Square-Edged Orifices

(16-in. Pipe, D = 15.25 in.)

β \ R_d	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.6039	.6033	.6021	.6013	.6001	.5992	.5980	.5972	.5945	.5939
.2000	.6044	.6038	.6026	.6017	.6005	.5996	.5983	.5975	.5947	.5940
.2500	.6053	.6047	.6034	.6025	.6012	.6003	.5989	.5981	.5951	.5944
.3000	.6065	.6058	.6045	.6035	.6022	.6012	.5997	.5989	.5958	.5950
.3500		.6074	.6060	.6050	.6035	.6025	.6010	.6000	.5968	.5960
.4000		.6095	.6080	.6069	.6053	.6043	.6026	.6017	.5982	.5973
.4500			.6105	.6094	.6077	.6066	.6048	.6038	.6000	.5992
.5000			.6136	.6124	.6106	.6094	.6075	.6064	.6024	.6014
.5500				.6159	.6139	.6126	.6106	.6094	.6051	.6041
.5750				.6177	.6157	.6144	.6123	.6110	.6065	.6054
.6000				.6196	.6176	.6161	.6139	.6126	.6079	.6067
.6250				.6216	.6194	.6179	.6155	.6141	.6092	.6080
.6500				.6235	.6212	.6196	.6171	.6156	.6103	.6091
.6750				.6254	.6229	.6212	.6185	.6169	.6112	.6099
.7000				.6273	.6245	.6227	.6198	.6180	.6118	.6103
.7250				.6292	.6262	.6241	.6208	.6189	.6120	.6103
.7500				.6313	.6278	.6255	.6218	.6195	.6117	.6098

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It is believed that the tolerances applicable to the coefficients above and to the right of the heavy stepped lines in Tables II-III-2, II-III-3 and II-III-4 do not exceed ± 1.0 per cent. The values below and to the left of this line are, for the most part, extrapolations outside the range of the test data and are subject to a larger tolerance (see Table II-V-1). Similar tolerance values will apply to the coefficients computed by the equations for other sizes of pipe than those given in the tables, particularly pipes larger than 16 in., and corresponding values of β and R_d or R_D [1-3].

Linear interpolation may be used within the tables; however, the use of the equations for interpolating is preferable.

Note 1: If the equations are used for pipes smaller than 2 in., the tolerances are to be doubled.

Note 2: As in previous editions, the coefficients in the tables are given to four significant figures so that, in using tabular values to compute a flow, two or more parties can obtain results agreeing within 1 or 2 in the fourth significant figure, although the actual uncertainty may be about 1 per cent, as indicated by the statement above and the tolerances given in Table II-V-1.

II-III-8 Expansion Factors. When metering compressible fluids, air, fuel gas, steam, etc., if the static pressure is measured at the inlet pressure tap, i.e., p_1, the expansion factor to be used in computing the rate of flow may be read from Fig. II-III-5 or evaluated by the equation

$$Y = 1 - (0.41 + 0.35\beta^4)\, x/\gamma \quad \text{(II-III-7)}$$

in which

x = Differential pressure ratio, $\Delta p/p_1$

γ = Ratio of specific heats of the gas, assuming it to be an ideal gas

β = Ratio of diameters, d/D

If the static pressure is measured at the outlet pressure tap, i.e., p_2, a common practice when flange taps are used, then the corresponding expansion factor, Y_2, is to be computed by the equation

$$Y_2 = \sqrt{1 + x_2} - (0.41 + 0.35\beta^4)\frac{x_2}{\gamma}\frac{1}{\sqrt{1 + x_2}} \quad \text{(II-III-8)}$$

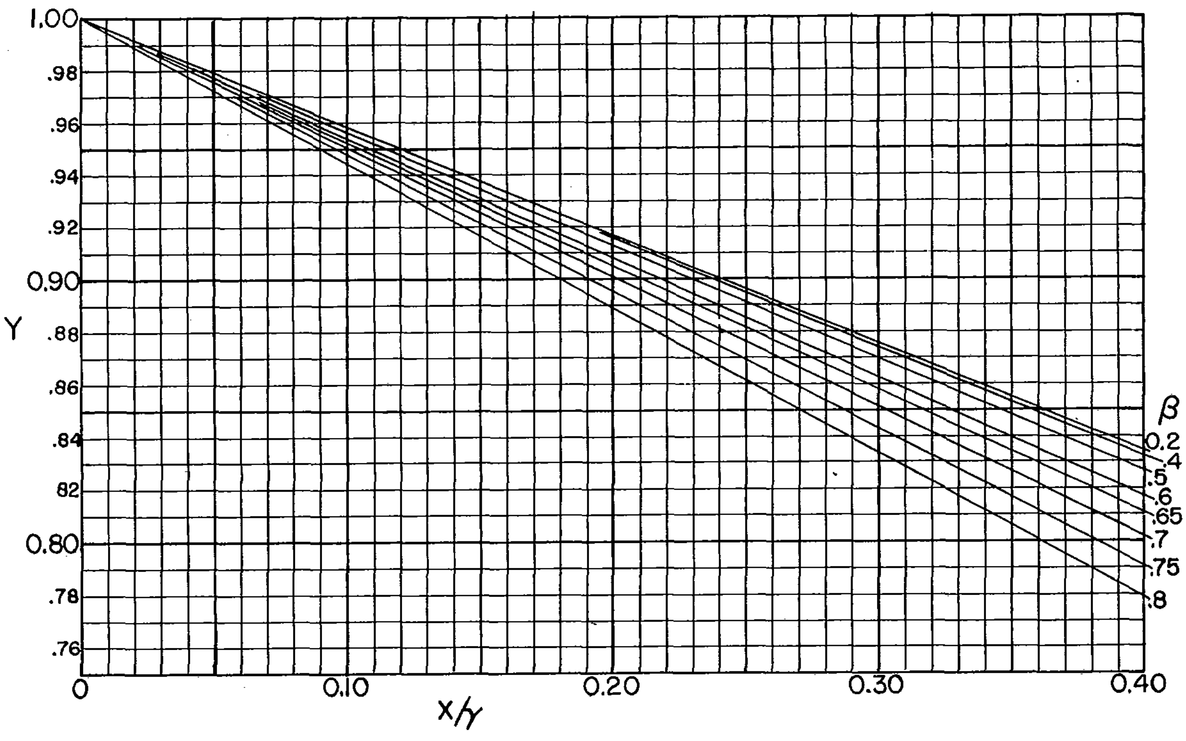

FIG. II-III-5 EXPANSION FACTORS FOR THIN-PLATE SQUARE-EDGED ORIFICES WITH FLANGE TAPS, D AND $1/2\,D$ TAPS AND VENA CONTRACTA TAPS (STATIC PRESSURE MEASURED FROM UPSTREAM PRESSURE TAP. $Y = 1 - (0.41 + 0.35\beta^4)x/\gamma$.)

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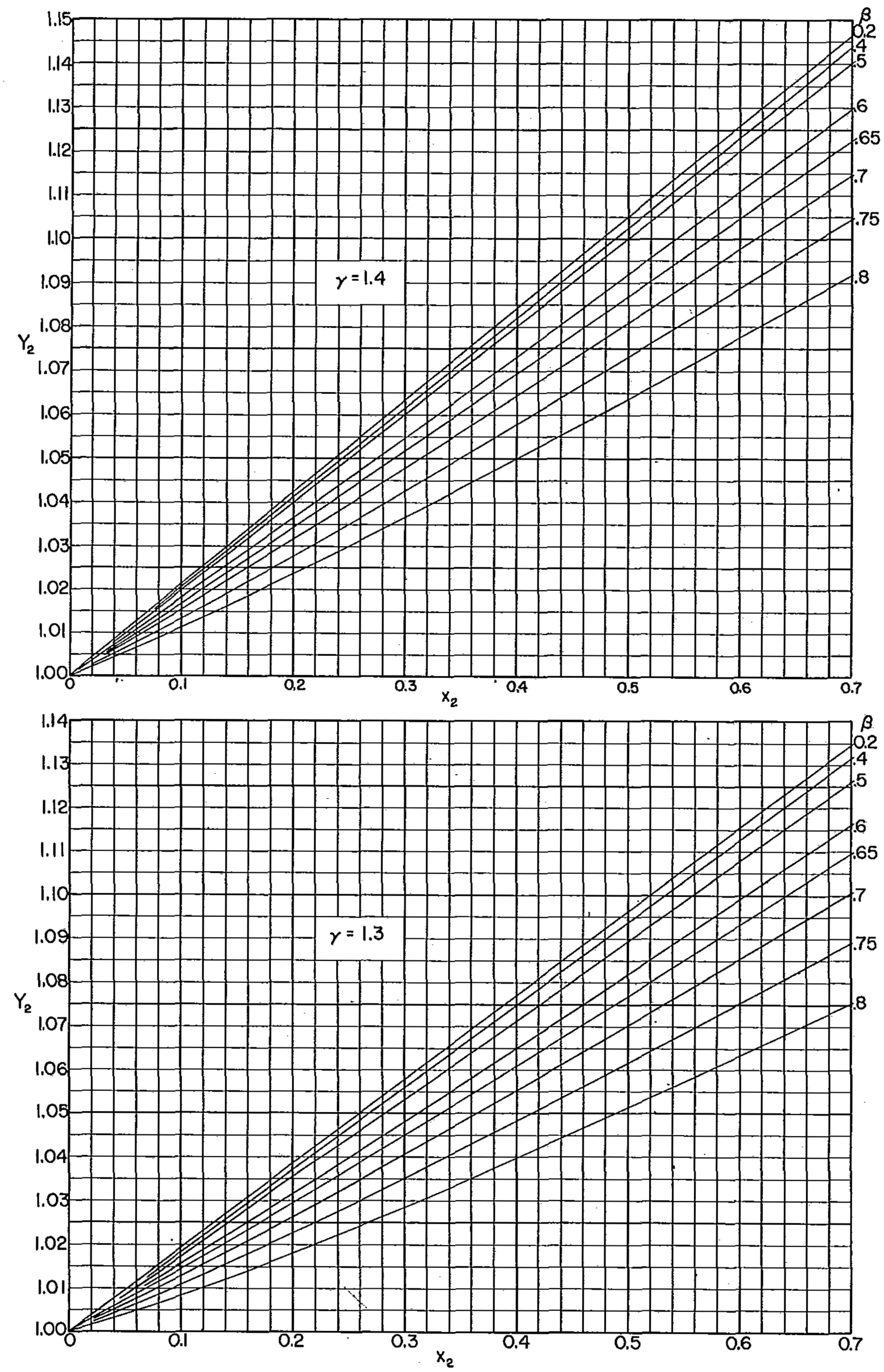

FIG. II-III-6 EXPANSION FACTORS FOR THIN-PLATE SQUARE-EDGED ORIFICES WHEN THE STATIC PRESSURE IS MEASURED FROM THE DOWNSTREAM PRESSURE TAP ($Y_2 = \sqrt{1 + x_2} - (0.41 + 0.35\beta^4)\, x_2/\gamma\sqrt{1 + x_2}$.)

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or read from Fig. II-III-6. Here $x_2 = \Delta p/p_2$, the ratio of the differential pressure to the downstream static pressure.

As explained in Chapter I-5, Par. I-5-32, these expansion factors *must* be used in conjunction with the flow coefficient, K, or the equivalent, $C/\sqrt{1-\beta^4}$.

Note: $x_2 = x/(1-x)$, and also

$$Y_2 = Y\sqrt{1+x_2} \tag{II-III-9}$$

as developed in Chapter I-5, Par. I-5-33.

Eccentric and Segmental Orifices

II-III-9 These orifice forms (Fig. II-III-7) are suitable when the fluid being metered carries a considerable amount of sediment or material in suspension.

II-III-10 Material and Installation. The material for these orifice plates should be the same as for concentric orifices. Also, the same criterion for the plate thickness as given earlier may be used. The diameter, d, or area, a, should be determined as described for concentric orifices.

As with concentric orifices, the inlet edge of the opening must be square, sharp and free of burrs, wire edge or the slightest amount of rounding.

When an eccentric orifice is installed between flanges, the circumference of the hole should lack being tangent with the pipe surface by 1 per cent of the pipe diameter. Also, the radius of the circular part of a segmental orifice should be 1 per cent less than that of the pipe and so installed that this amount of difference is uniform around the circular portion.

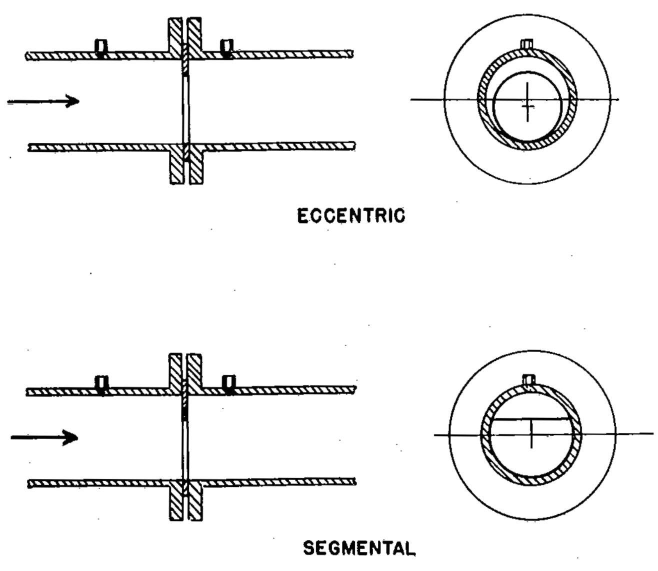

FIG. II-III-7 ECCENTRIC AND SEGMENTAL ORIFICES

II-III-11 Pressure Taps. Both flange and vena contracta pressure taps are used with these orifices, with the latter predominating. For flange taps, the locations are the same as with concentric orifices, namely, 1 in. from the adjacent face of the orifice plate. For vena contracta taps, the inlet pressure tap is one pipe diameter preceding the inlet face of the plate. The positions of the outlet pressure tap, as measured from the inlet face of the orifice plate, are given by the curves of Fig. II-III-8.

With eccentric orifices the pressure taps should, if possible, be in the side of the pipe diametrically opposite the point at which the orifice is substantially tangent to the pipe surface. However, there may be cases where the installation of the pipe does not provide room for this location of the taps, in which case the taps may be moved circumferentially up to but not more than 90 deg and the same coefficients used. With segmental orifices, the pressure taps are always in the element of the pipe that is normal to the straight edge of the orifice.

II-III-12 Coefficients and Expansion Factors. The discharge coefficients for eccentric orifices are given by Fig. II-III-9 and for segmental orifices, by Fig. II-III-10.

The curves in Fig. II-III-9 were developed for use with the pipe Reynolds number, R_D, which is the manner in which they were originally reported [4]. When the throat Reynolds number, R_d, is evaluated, the corresponding value of R_D by which the coefficient may be selected is $R_D = \beta R_d$.

To illustrate the method of interpolating between the curves for $R_D = 10^4$ and 10^6: Assume an eccentric orifice in a 6-in. pipe with flange taps, a diameter ration $\beta = 0.64$, and an orifice Reynolds number, $R_d = 62{,}500$. Then,

$R_D = 0.64 \times 62{,}500 = 40{,}000$, and $\sqrt{R_D} = 200$, $1/200 = .005$. At $\beta = 0.64$ and $R_D = 10^4$ $C = 0.642$; also $1/\sqrt{R_D} = .01$. At $R_D = 10^6$ $C = 0.627$; also $1/\sqrt{10^6} = .001$. $0.642 - 0.627 = .015$, and $.015\,[(.01 - .005)/(.01 - .001)] = .015 \times 4/9 = .0067$. Therefore, $C = 0.642 - .0067 = 0.6353$, or 0.635.

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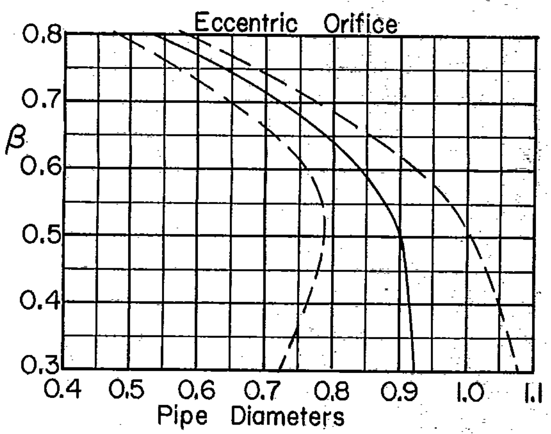

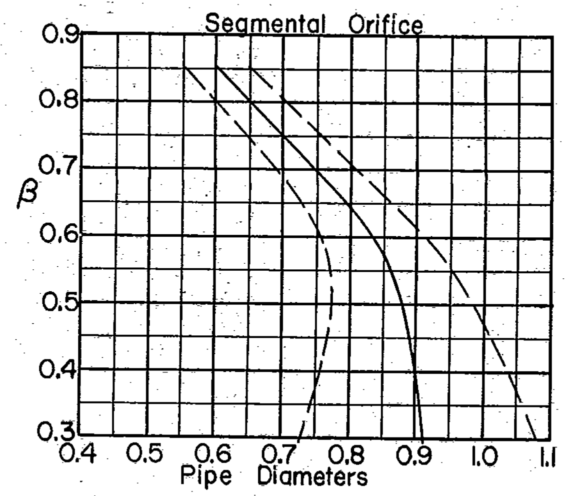

FIG. II-III-8 LOCATIONS OF OUTLET PRESSURE TAPS FOR VENA CONTRACTA TAPS WITH ECCENTRIC AND SEGMENTAL ORIFICES

With segmental orifices the pipe Reynolds number, R_D, is usually the value computed. Figure II-III-10 gives coefficients at only one value of R_D, as coefficients at other values of R_D have not been established.

II-III-13 Expansion factors to be used with eccentric or segmental orifices when metering compressible fluids are given by Figs. II-III-11 and II-III-12.

II-III-14 When using the discharge coefficient curves for segmental orifices (Fig. II-III-10) and the expansion factors (Fig. II-III-11), it must be remembered that $\beta = \sqrt{m}$. That is, for these orifices the area ratio, m, is the primary dimensional ratio, and the only significance of β is that of the diameter ratio of an equivalent circular orifice.

Note: The users of differential pressure meters are cautioned to be very careful to select the proper discharge coefficient or flow coefficient and the appropriate expansion factor for the particular differential producer being used. Carelessness in selecting these factors may introduce errors which would be difficult to trace.

Small Precision Bore Orifice Meters

II-III-15 Meter tubes of diameters smaller than 1.5 in. can be produced and duplicated provided special manufacturing care and procedures are used [5].

II-III-16 Meter Tubes. The meter tubes should be carefully selected from thick-wall stainless-steel stock of a type suitable for the service in which it will be used. Each tube should be welded to the orifice-holding flange, bored, ground and honed to a uniform inside diameter within a tolerance of ± 0.001 in. and to a surface finish of 5 to 10 microinch.

II-III-17 Orifice Plates. For most services orifice plates are made from stainless-steel sheet of 1/8-in. nominal thickness. Since surface roughness, flatness, orifice-edge squareness and thickness have relatively very large effects on reproducibility and flow-measurement reliability, the plate must be prepared very carefully. The plate should be flat within 0.001 in., and the surface roughness

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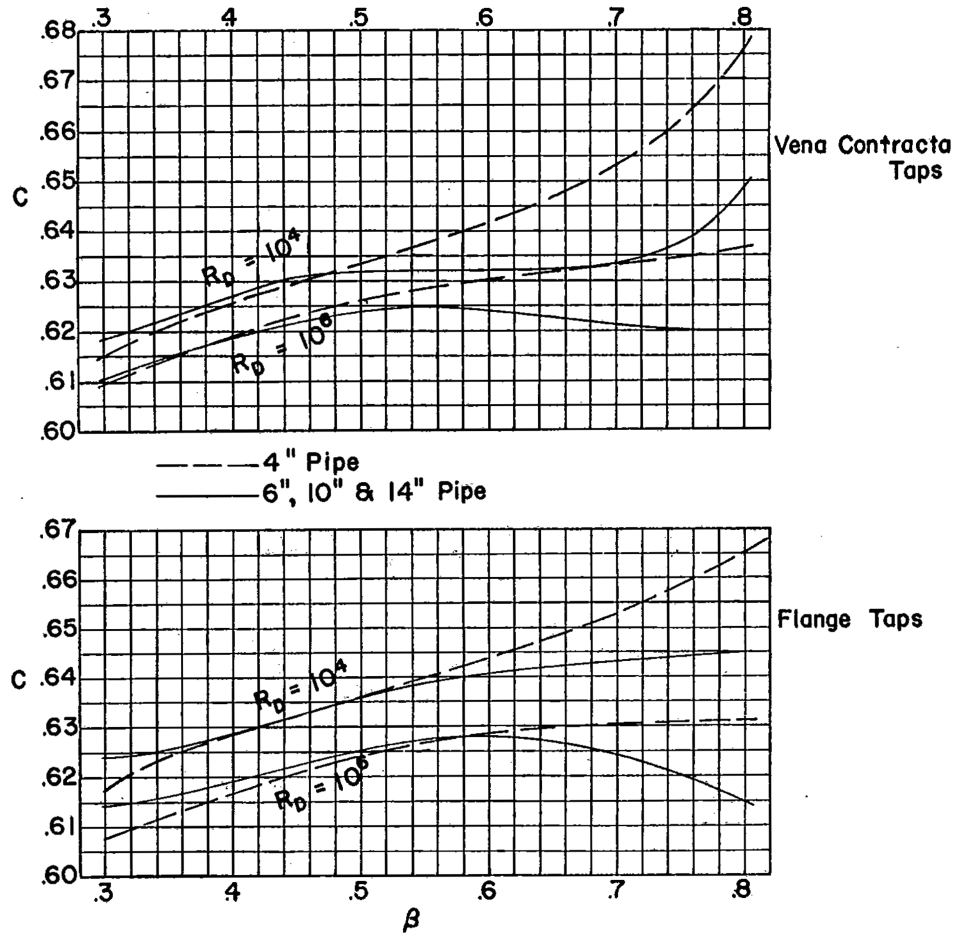
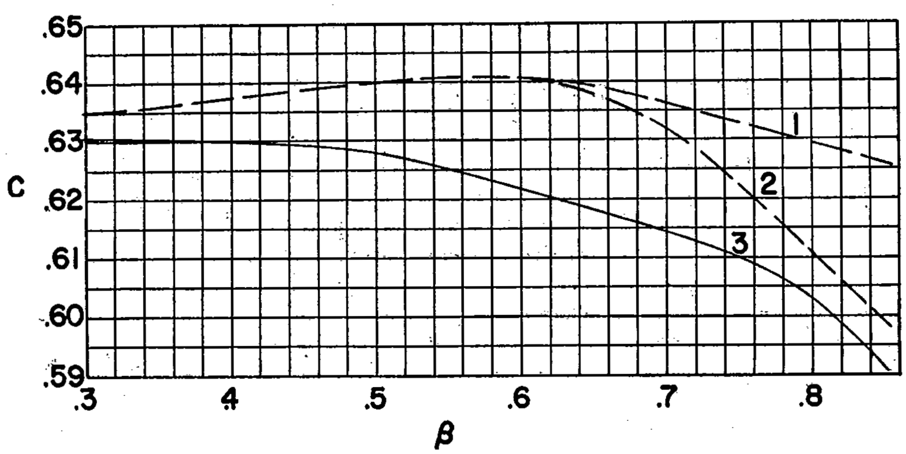


FIG. II-III-9 DISCHARGE COEFFICIENTS OF ECCENTRIC SQUARE-EDGED ORIFICES



Line 1, 4" pipe, vena contracta taps
Line 2, 4" pipe, flange taps
Line 3, 6", 10" and 14" pipe, flange and vena contracta taps.

FIG. II-III-10 DISCHARGE COEFFICIENTS FOR SEGMENTAL ORIFICES FOR VALUES OF $R_D = 10^4$

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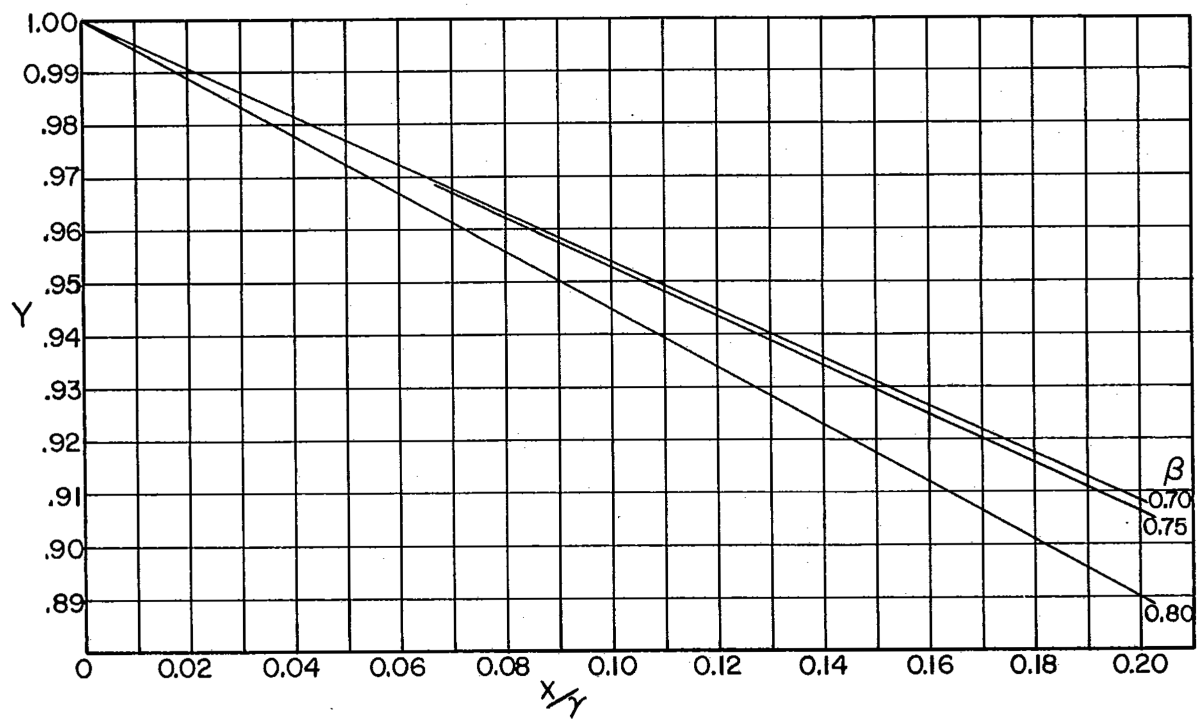

FIG. II-III-11 EXPANSION FACTORS FOR THIN-PLATE ECCENTRIC ORIFICES (STATIC PRESSURE MEASURED FROM UPSTREAM TAP)

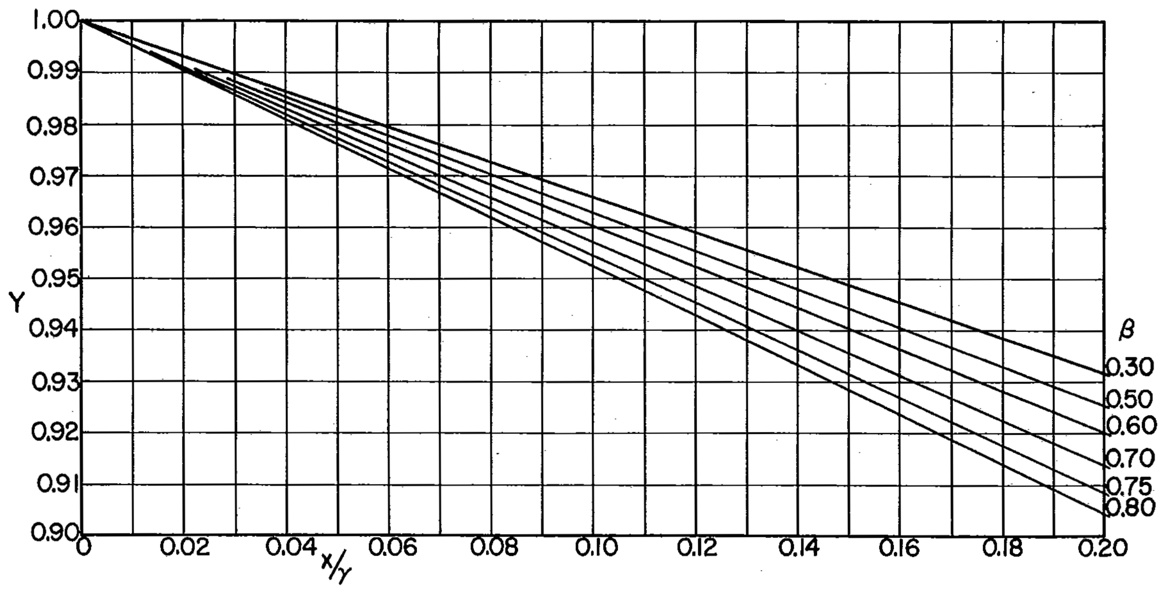

FIG. II-III-12 EXPANSION FACTORS FOR METERING COMPRESSIBLE FLUIDS WITH THIN-PLATE SEGMENTAL ORIFICES

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should not exceed 20 microinch. The thickness of the cylindrical face of the orifice should not exceed 0.02 D or 1/8 d, whichever is smaller. The upstream edge, or corner, must be square and sharp. Because of the thinness of the cylindrical face of the orifice, the safest way to measure the orifice diameter, d, without damaging the orifice edge, is with an optical comparator.

II-III-18 Pressure Taps. Pressures are measured from annular grooves on each side of the plate, as shown in Fig. II-III-13. This form of pressure opening is used since a 1-in. location from the downstream orifice face would be equivalent to one to two pipe diameters in the 1-in. and smaller tubes and, thus, in the varying pressure-recovery region. On the other hand, placing the downstream tap at 1/2 D would be impractical in these small sizes. At the upstream annular pressure groove there is a slight pressure buildup; and, although a 1-in. location could be used, the use of the groove makes for uniformity in manufacture.

II-III-19 Coefficients. The flow coefficient, K, for these small orifice meters may be computed by the equation

$$K = \left[0.5991 + \frac{0.0044}{D} + \left(0.3155 + \frac{0.0175}{D}\right)(\beta^4 + 2\beta^{16})\right] + \left[\frac{0.00052}{D} - 0.000192 + \left(0.01648 - \frac{0.00116}{D}\right)(\beta^4 + 4\beta^{16})\right]\lambda \qquad \text{(II-III-10)}$$

where

K = Flow coefficient

D = Inside diameter of meter tube (in.)

R_D = Pipe Reynolds number

R_d = Orifice Reynolds number

β = Ratio, (orifice diameter)/(tube diameter)

$\lambda = 1000/\sqrt{R_D} = 1000/\sqrt{\beta R_d}$

This equation has been found to give coefficients within ± 0.75 per cent of the values obtained from a calibration when pressures are measured from corner grooves as described above and when 0.5 in. $\lesseqgtr D \lesseqgtr$ 1.5 in., $0.1 < \beta < 0.8$, and $R_D > 1000$.

The equation that is to be used for computing the flow coefficient, when the meter tube has flange pressure taps located in the conventional (1-in.) position, is

$$K = 0.5980 + 0.468\,(\beta^4 + 10\,\beta^{12}) + (0.00087 + 0.0081\,\beta^4)\,\lambda \qquad \text{(II-III-11)}$$

and is applicable if 1 in. $< D <$ 1 1/2 in., $0.15 < \beta < 0.7$, and $R_D > 1000$.

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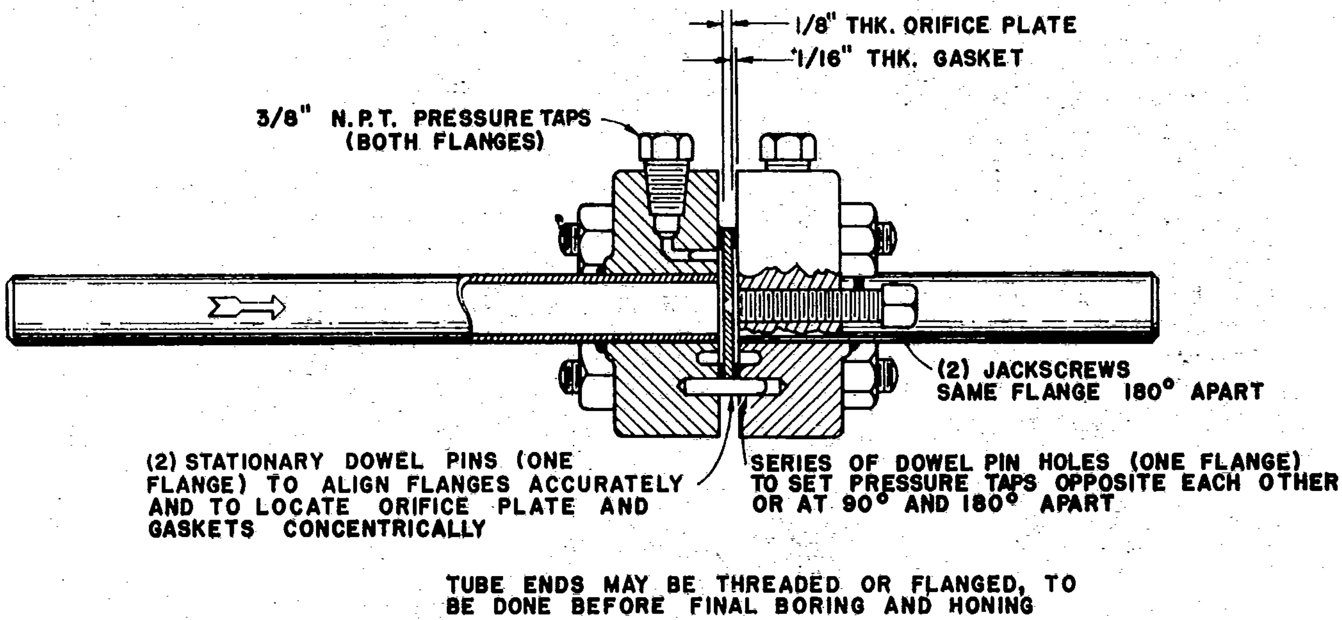

FIG. II-III-13 HONED SMALL-BORE ORIFICE FLOW SECTION

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Flow Nozzles

II-III-20 Material. Flow nozzles should be made of a corrosion-resistant material. For high-temperature service and many other services, stainless steel should be used. For water at temperatures below 400 F and pressures below about 250 psig, as well as for some other fluids including oils and gases, bronze may be used. Aluminum may be used for air, some other gases, and also liquids, if free of aluminum corroding elements, where temperatures and pressures do not exceed about 200 F and 200 psig.

II-III-21 Nozzle Form. The recommended form of flow nozzle is the "long-radius" or elliptical inlet nozzle, in which the curvature of the inlet to the nozzle throat is the quadrant of an ellipse. The proportions of the ellipse with respect to the pipe diameter and the nozzle throat diameter are shown by Fig. II-III-14. For the high β nozzles, diameter ratios between 0.45 and 0.80 both inclusive, the entrance curvature is the quadrant of an ellipse having a semi-major axis of $1/2\,D$ and a semi-minor axis of $1/2\,(D - d)$. For the low β nozzles, recommended for diameter ratios below 0.50, the semi-major axis is equal to the nozzle throat diameter, d, and the semi-minor axis is $5/8\,d$ to $2/3\,d$. The length of the cylindrical throat section of the high β nozzles should be $0.6\,d$ or $1/3\,D$, whichever is less. For the low β nozzles, the length of throat should be between $0.6\,d$ and $0.75\,d$ when pipe-wall taps will be used and $0.75\,d$ when the nozzle is made with throat taps.

The thickness of the nozzle wall and flange should be such as to prevent distortion of the nozzle throat from strains caused by the pipeline temperature and pressure, flange bolting or other methods of installing the nozzle in the pipeline. The outside diameter of the nozzle flange or the design of the flange facing should be such that the nozzle throat can be centered accurately in the pipe. (See Chapter II-II, Par. II-II-6.)

II-III-22 The throat of a flow nozzle should be as nearly cylindrical as possible. Any taper should not exceed the following negative amounts:

1. -0.001 in. for $d \leqq 3.00$ in.
2. -0.0015 in. for $3.01 \leqq d \leqq 6.00$ in.
3. -0.002 in. for $d \geqq 6.01$ in.

That is, any taper should be such that the throat diameter *decreases* toward the outlet end. There must be no bell mouth or diameter increase near the outlet end, especially within the last 1/4 in.

Any out-of-roundness of the nozzle throat should not exceed:

1. ± 0.002 in. for $d \leqq 3.00$ in.
2. ± 0.003 in. for $3.01 \leqq d \leqq 6.00$ in.
3. ± 0.004 in. for $d \geqq 6.01$ in.

The actual diameter of the nozzle throat should be determined by careful measurements after all machining and finishing has been completed. Measurements should be made on three or more diameters and desirably in two or more cross sections. When these measurements are made, care is necessary in order not to scratch or otherwise alter the surface of the throat.

II-III-23 Pressure Taps. Two pairs of pressure-tap locations are used with the ASME long-radius flow nozzles, namely, pipe-wall taps and nozzle-throat taps. Since flow nozzles may be used in a continuous pipeline, at the end of a pipe section or at the inlet or outlet of a plenum chamber, it will be convenient to describe the inlet and outlet tap locations separately rather than in pairs.

II-III-24 Inlet Pressure Tap. The same location of the inlet pressure tap is used with both pairs of pressure taps, namely, at one pipe diameter, D, preceding the plane of the nozzle elliptical inlet section, as shown in Fig. II-III-15. The same location applies when a nozzle is mounted at the open outlet end of a pipe section (Fig. II-III-16).

If a nozzle is installed at the inlet to a plenum chamber (Fig. II-III-17), no inlet pressure connection is required, and the value of the inlet pressure may be considered as atmospheric. If a nozzle is installed at the outlet of a plenum chamber, the inlet pressure (to the nozzle) will be the pressure in the plenum chamber.

II-III-25 Outlet Pressure Tap. When a high β nozzle is to be used in a continuous pipeline, the outlet pressure tap is located in the pipe wall $1/2\,D$ following the plane of the beginning of the elliptical inlet section of the nozzle. For a low β nozzle, if a pipe-wall tap is to be used, it should be located a distance of $1\text{-}1/2\,d$ (throat diameter) following the entrance plane of the nozzle. Likewise, for nozzles made with throat taps, the location of the tap or taps is $1\text{-}1/2\,d$ following the entrance plane.

If a nozzle is installed at the outlet end of a pipe section or the outlet of a plenum chamber, the outlet pressure may be measured with a barometer

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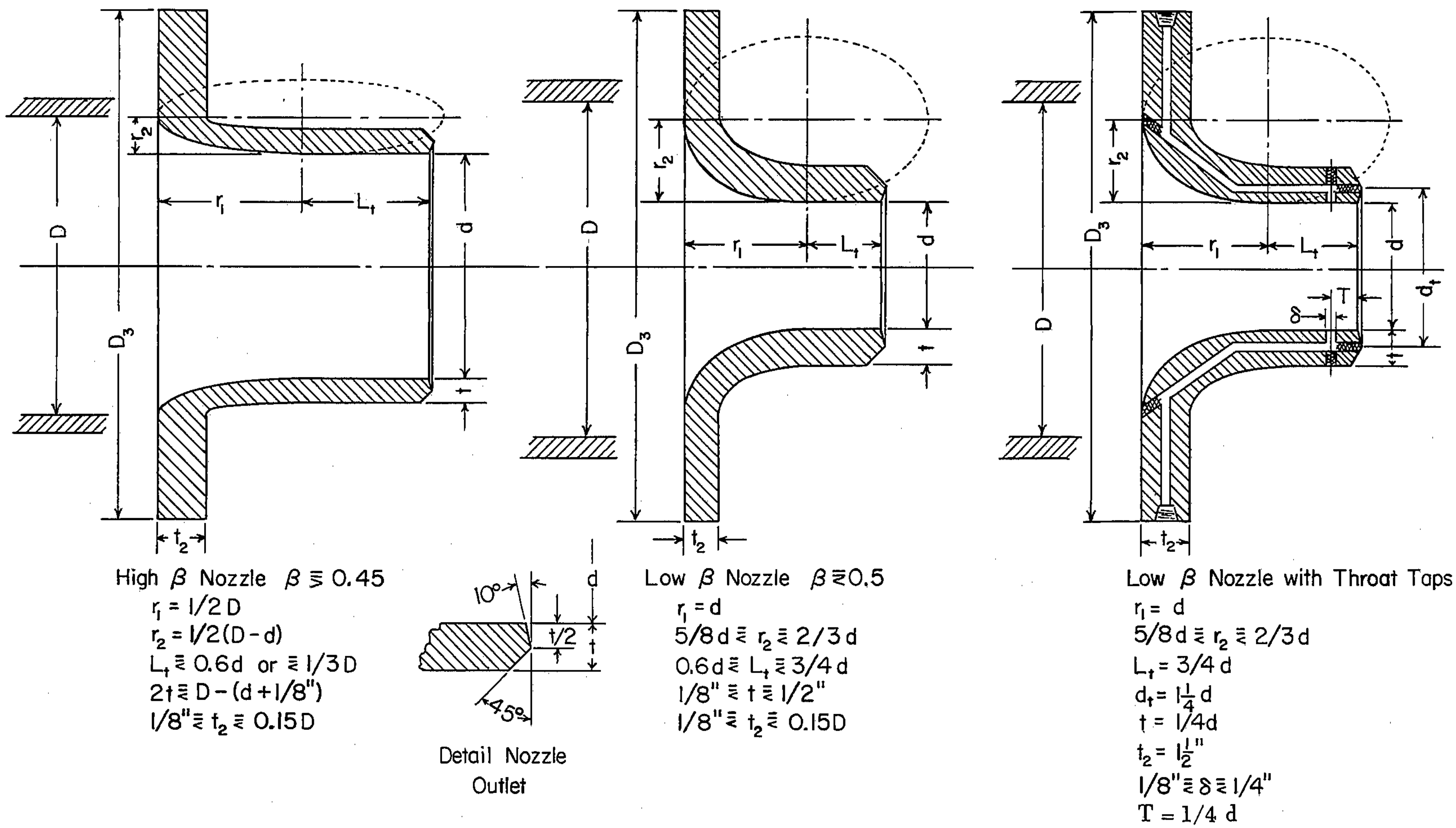

FIG. II-III-14 RECOMMENDED PROPORTIONS OF ASME LONG-RADIUS FLOW NOZZLES

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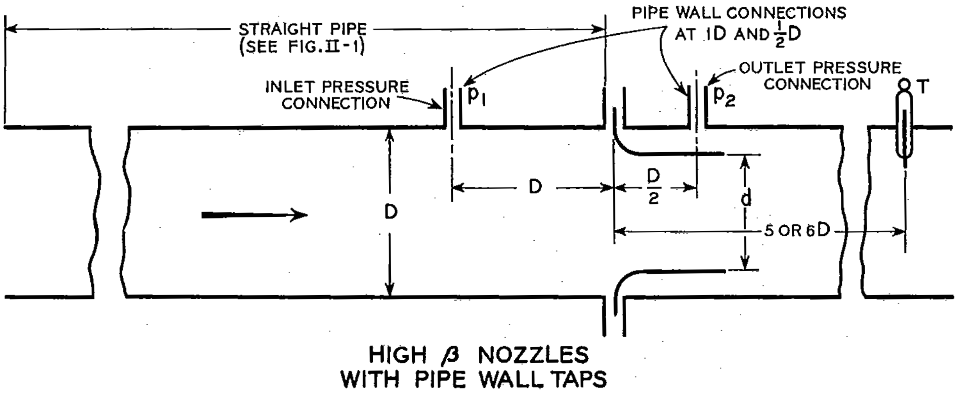

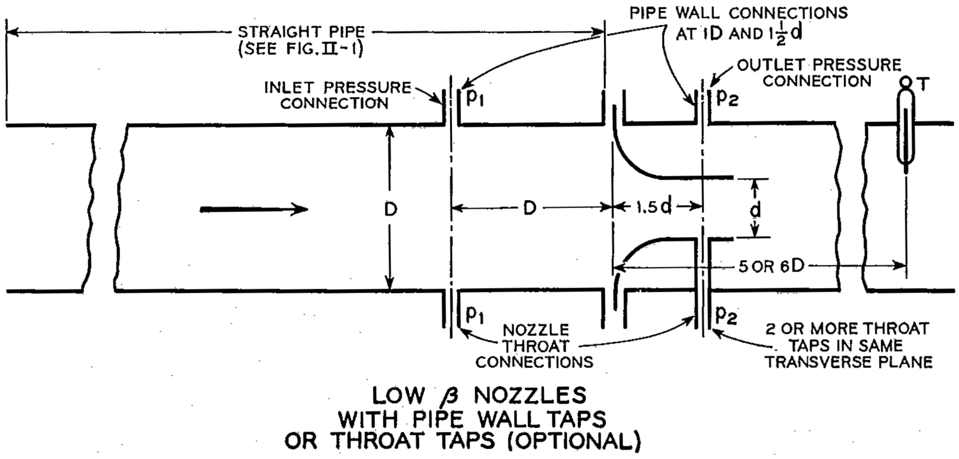

FIG. II-III-15 LOCATIONS OF PRESSURE TAPS USED WITH ASME LONG-RADIUS FLOW NOZZLES WHEN IN A CONTINUOUS PIPELINE (WHEN A THERMOMETER IS REQUIRED, THE WELL FOR IT MAY BE LOCATED AS SHOWN BY *T*.)

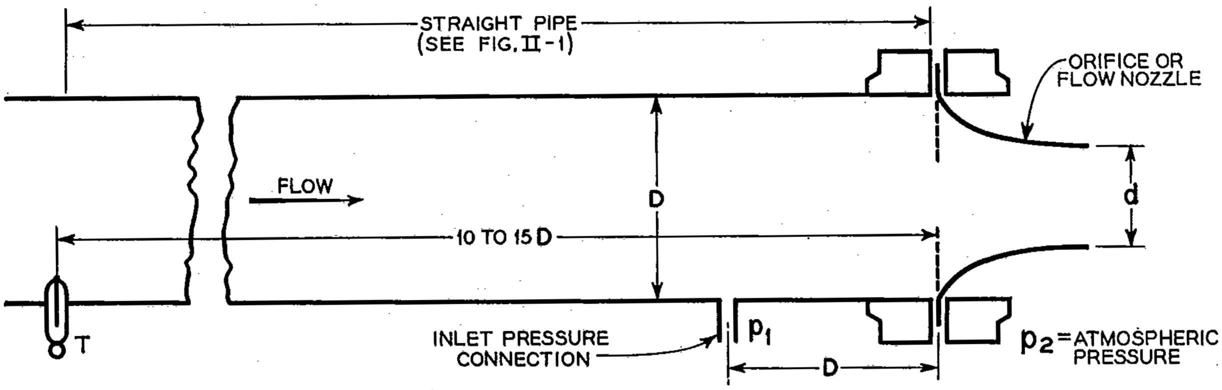

FIG. II-III-16 LOCATION OF INLET PRESSURE TAP FOR A NOZZLE MOUNTED AT OPEN OUTLET END OF PIPE

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located as near the nozzle outlet as possible, but not in the path of the emerging fluid. Alternately, if the nozzle is equipped with throat taps, the pressure in the throat may be measured with a manometer. If a nozzle is installed at the inlet to a plenum chamber, the pressure in the plenum chamber may be used as the nozzle outlet pressure. Of course, if the nozzle has throat taps, a special connection to these may be brought out from the chamber.

II-III-26 Construction of Pressure Taps. Pressure taps should be drilled (and preferably reamed) radially with respect to the pipe in which they are made. This drilling should be done after any coupling or other fitting for attaching of pressure tubing has been welded to the pipe. The hole where it breaks through the inner surface of the pipe must be free of burrs or wire edge, and the corner or edge of the hole left square and sharp or dulled (rounded) very slightly.

Special care and procedures are required for making pressure taps in the throat of a flow nozzle (Fig. II-III-14). The holes should be drilled and reamed before the final boring and polishing of the throat section. A plug sized for a press fit is pressed into the hole. The plug should be made with provision for removing it after the final machining and polishing of the nozzle surface. After removal of the plug, the edge of the hole should be free of burrs and square and sharp with the throat surface. Any slight burr may be removed by rolling a tapered piece of maple wood around the pressure tap hole.

It is recommended that throat tap nozzles be made with two or four tap holes; this is because these nozzles are used frequently in test work, and the additional taps are useful for multiple instrumentation. In such cases, it will be helpful to have a like number of inlet pressure taps.

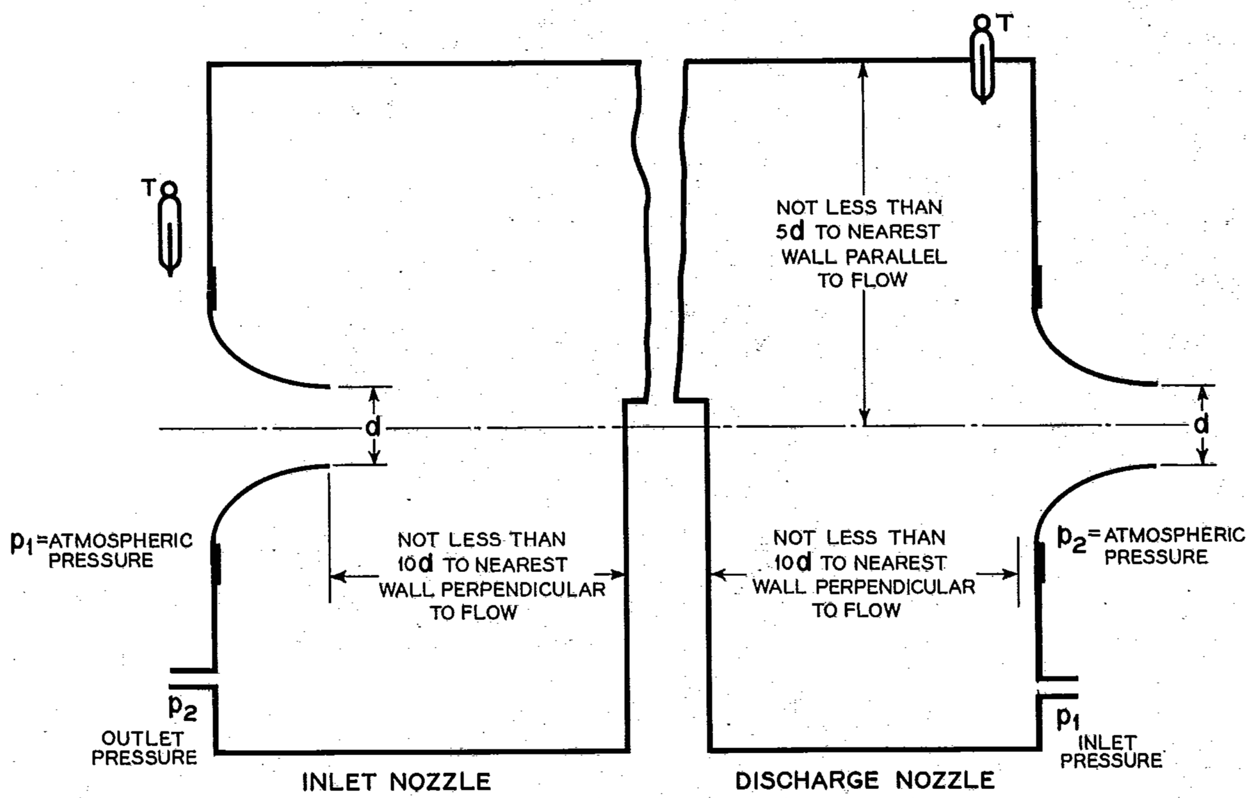

FIG. II-III-17 PRESSURE TAP LOCATIONS WHEN NOZZLES ARE USED AT INLET OR OUTLET OF A PLENUM CHAMBER

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II-III-27 Pressure Loss. The overall pressure loss with a flow nozzle is shown in Fig. II-III-18.

II-III-28 Coefficients. As with orifice meters, it is desirable whenever possible to calibrate a flow nozzle with the pipe section in which it is to be used. When a calibration is not made, the coefficient of discharge may be computed by equation (II-III-12) or obtained from Table II-III-5.

$$C = 0.99622 + 0.00059\,D - (6.36 + 0.13D - 0.24\beta^2)\frac{1}{\sqrt{R_d}} \qquad \text{(II-III-12)}$$

For low β flow nozzles with throat pressure taps, the discharge coefficient may be read from Fig. II-III-19 if the nozzle has not been calibrated.

The tables of coefficients of discharge for long-radius nozzles with pipe wall taps, Tables II-III-5 (a-d) are computed by equation (II-III-12). This equation was developed from data determined during commercial calibrations and a research program [11]. This resulted in a preponderance of the data being for relatively small pipe sizes and low Reynolds numbers. The test limits were:

D between 2 and 15.75 inches
R_d between 10^4 and 10^6
β between 0.15 and 0.75

Within these limits, equation (II-III-12) may be used for interpolation but *must never* be used for extrapolation outside of these ranges.

In the present state of the art it is suggested that an equation based on the flat plate boundary layer theory would be appropriate. A general form of such an equation is

$$C = A - B\left(\frac{R_{dt}}{R_d}\right)^a \qquad \text{(II-III-41)}$$

where $a = \frac{1}{2}$ *for* R_d less than R_{dt}

$= 1/5$ for R_d greater than R_{dt}

R_{dt} = throat Reynolds where the boundary layer changes from laminar to turbulent.

R_{dt} is more of a zone than a point since it may range from about 300,000 to about 3,000,000. It must be determined experimentally the same as A and B.

Where calibration data are not available the following equation has been suggested [12].

$$C = 0.9975 - 0.00653\,(10^6/R_d)^a$$
$$a = \tfrac{1}{2} \text{ for } R_d < 10^6$$
$$= 1/5 \text{ for } R_d > 10^6 \qquad \text{(II-III-42)}$$

Slight variations in form and dimension of either pipe or nozzle may affect the observed pressures, and thus cause the values of the exponent, a, and slope term, B, to diverge considerably from the values in equation (II-III-42) [13]. In view of this, it is possible that a tolerance greater than $\pm 2.0\%$ should be applied when any one of D, β, or R_d is outside the range of values listed above. Representative values of C by equation (II-III-42) are:

R_d	10,000	20,000	50,000	100,000	200,000
C	0.9322	0.9513	0.9683	0.9768	0.9829
R_d	500,000	10^6	5×10^6	10^7	10^8
C	0.9883	0.9910	0.9928	0.9934	0.9949

II-III-29 Expansion Factors. When metering a compressible fluid with a flow nozzle, the expansion factor, Y_a, to be used in computing the flow, given by equation (I-5-26), is

$$Y_a = \left[r^{2/\gamma}\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{1-r^{(\gamma-1)/\gamma}}{1-r}\right)\left(\frac{1-\beta^4}{1-\beta^4 r^{2/\gamma}}\right)\right]^{1/2} \qquad \text{(II-III-13)}$$

Values of Y_a for two values of γ are given by Figs. II-III-20 and II-III-21 and also by Tables II-III-6 and II-III-7.

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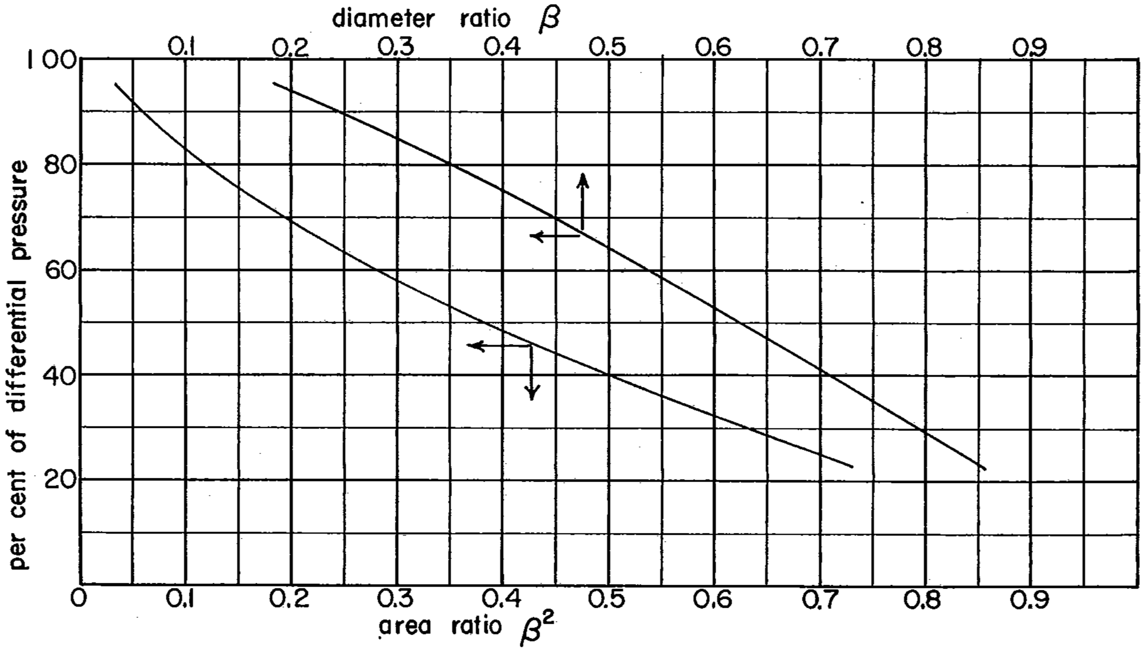

FIG. II-III-18 OVERALL PRESSURE LOSS ACROSS FLOW NOZZLES

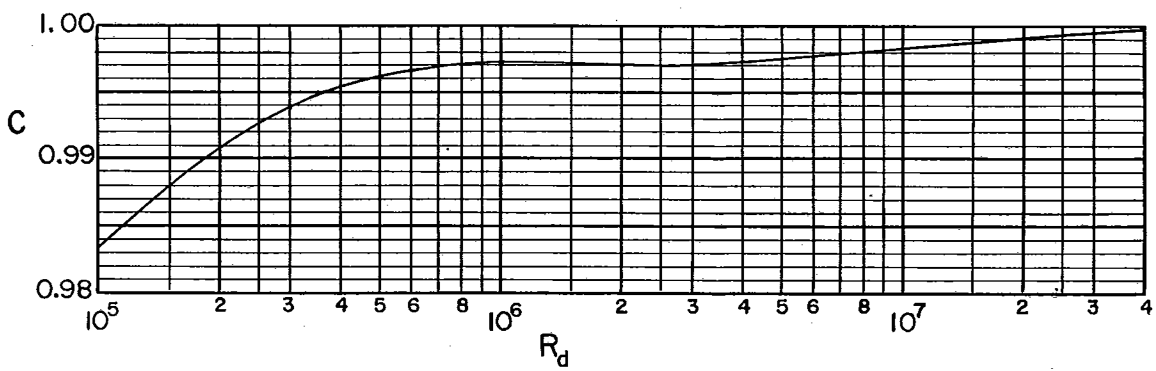

FIG. II-III-19 TYPICAL CALIBRATION CURVE OF A LOW-β FLOW NOZZLE WITH THROAT PRESSURE TAPS

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Table II-III-5 (a) Long-Radius Flow Nozzles: Discharge Coefficients, C, with Pipe Taps at 1 D and 1/2 D

(2-in. Pipe, D = 2.067 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.9312	.9369	.9414	.9450	.9480	.9506	.9555	.9592	.9643	.9678	.9732	.9765	.9881	.9908
.2000	.9312	.9370	.9415	.9451	.9481	.9506	.9556	.9592	.9643	.9678	.9733	.9765	.9881	.9908
.2500	.9313	.9370	.9415	.9451	.9481	.9506	.9556	.9592	.9644	.9678	.9733	.9765	.9881	.9908
.3000	.9313	.9371	.9416	.9452	.9482	.9507	.9556	.9593	.9644	.9679	.9733	.9765	.9881	.9908
.3500	.9314	.9372	.9416	.9452	.9482	.9508	.9557	.9593	.9644	.9679	.9733	.9766	.9881	.9908
.4000	.9315	.9373	.9417	.9453	.9483	.9508	.9557	.9594	.9645	.9680	.9734	.9766	.9881	.9908
.4500	.9316	.9373	.9418	.9454	.9484	.9509	.9558	.9594	.9645	.9680	.9734	.9766	.9881	.9909
.5000	.9317	.9375	.9419	.9455	.9485	.9510	.9559	.9595	.9646	.9681	.9734	.9767	.9881	.9909
.5500	.9319	.9376	.9420	.9456	.9486	.9511	.9560	.9596	.9646	.9681	.9735	.9767	.9882	.9909
.5750	.9319	.9376	.9421	.9456	.9486	.9511	.9560	.9596	.9647	.9681	.9735	.9767	.9882	.9909
.6000	.9320	.9377	.9421	.9457	.9487	.9512	.9561	.9597	.9647	.9682	.9735	.9767	.9882	.9909
.6250	.9321	.9378	.9422	.9458	.9487	.9512	.9561	.9597	.9648	.9682	.9736	.9768	.9882	.9909
.6500	.9322	.9378	.9423	.9458	.9488	.9513	.9561	.9597	.9648	.9682	.9736	.9768	.9882	.9909
.6750	.9322	.9379	.9423	.9459	.9488	.9513	.9562	.9598	.9648	.9683	.9736	.9768	.9882	.9909
.7000	.9323	.9380	.9424	.9460	.9489	.9514	.9563	.9598	.9649	.9683	.9737	.9768	.9882	.9909
.7250	.9324	.9381	.9425	.9460	.9490	.9515	.9563	.9599	.9649	.9684	.9737	.9769	.9882	.9909
.7500	.9325	.9382	.9426	.9461	.9490	.9515	.9564	.9599	.9650	.9684	.9737	.9769	.9883	.9909

Table II-III-5 (b) Long-Radius Flow Nozzles: Discharge Coefficients, C, with Pipe Taps at 1 D and 1/2 D

(4-in. Pipe, D = 4.026 in.)

β \ R_d	10,000	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.9298	.9358	.9404	.9442	.9473	.9499	.9551	.9589	.9642	.9678	.9735	.9768	.9889	.9917
.2000	.9298	.9358	.9405	.9442	.9473	.9500	.9551	.9589	.9642	.9678	.9735	.9768	.9889	.9917
.2500	.9299	.9359	.9405	.9443	.9474	.9500	.9551	.9589	.9642	.9679	.9735	.9769	.9889	.9917
.3000	.9299	.9359	.9406	.9443	.9474	.9500	.9552	.9590	.9643	.9679	.9735	.9769	.9889	.9917
.3500	.9300	.9360	.9406	.9444	.9475	.9501	.9552	.9590	.9643	.9679	.9736	.9769	.9889	.9917
.4000	.9301	.9361	.9407	.9445	.9475	.9502	.9553	.9591	.9644	.9680	.9736	.9769	.9889	.9917
.4500	.9302	.9362	.9408	.9445	.9476	.9502	.9553	.9591	.9644	.9680	.9736	.9770	.9889	.9918
.5000	.9303	.9363	.9409	.9446	.9477	.9503	.9554	.9592	.9645	.9681	.9737	.9770	.9889	.9918
.5500	.9305	.9364	.9410	.9447	.9478	.9504	.9555	.9593	.9645	.9681	.9737	.9770	.9890	.9918
.5750		.9365	.9411	.9448	.9479	.9505	.9555	.9593	.9646	.9682	.9737	.9771	.9890	.9918
.6000		.9365	.9411	.9448	.9479	.9505	.9556	.9593	.9646	.9682	.9738	.9771	.9890	.9918
.6250			.9412	.9449	.9480	.9506	.9556	.9594	.9646	.9682	.9738	.9771	.9890	.9918
.6500			.9413	.9450	.9480	.9506	.9557	.9594	.9647	.9683	.9738	.9771	.9890	.9918
.6750				.9450	.9481	.9507	.9557	.9595	.9647	.9683	.9739	.9772	.9890	.9918
.7000				.9451	.9482	.9507	.9558	.9595	.9648	.9683	.9739	.9772	.9890	.9918
.7250					.9482	.9508	.9558	.9596	.9648	.9684	.9739	.9772	.9890	.9918
.7500					.9483	.9509	.9559	.9596	.9648	.9684	.9739	.9772	.9890	.9918

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Table II-III-5 (c) Long-Radius FlowNozzles: Discharge Coefficients, C, with Pipe Taps at 1 D and 1/2 D

(8-in. Pipe, D = 7.981 in.)

β \ R_d	12,000	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.9334	.9384	.9424	.9458	.9486	.9541	.9582	.9639	.9678	.9739	.9775	.9905	.9935
.2000	.9334	.9384	.9425	.9458	.9486	.9542	.9582	.9640	.9679	.9739	.9775	.9905	.9935
.2500	.9335	.9385	.9425	.9459	.9487	.9542	.9583	.9640	.9679	.9739	.9776	.9905	.9935
.3000	.9335	.9385	.9426	.9459	.9487	.9542	.9583	.9640	.9679	.9740	.9776	.9905	.9935
.3500	.9336	.9386	.9426	.9460	.9488	.9543	.9584	.9641	.9680	.9740	.9776	.9905	.9935
.4000	.9337	.9387	.9427	.9460	.9489	.9544	.9584	.9641	.9680	.9740	.9776	.9905	.9936
.4500	.9338	.9388	.9428	.9461	.9489	.9544	.9585	.9642	.9680	.9741	.9777	.9905	.9936
.5000	.9339	.9389	.9429	.9462	.9490	.9545	.9585	.9642	.9681	.9741	.9777	.9905	.9936
.5500		.9390	.9430	.9463	.9491	.9546	.9586	.9643	.9681	.9742	.9777	.9906	.9936
.5750		.9390	.9430	.9464	.9492	.9546	.9587	.9643	.9682	.9742	.9778	.9906	.9936
.6000			.9431	.9464	.9492	.9547	.9587	.9644	.9682	.9742	.9778	.9906	.9936
.6250			.9432	.9465	.9493	.9547	.9587	.9644	.9682	.9742	.9778	.9906	.9936
.6500				.9465	.9493	.9548	.9588	.9644	.9683	.9743	.9778	.9906	.9936
.6750				.9466	.9494	.9548	.9588	.9645	.9683	.9743	.9779	.9906	.9936
.7000					.9494	.9549	.9589	.9645	.9684	.9743	.9779	.9906	.9936
.7250					.9495	.9549	.9589	.9646	.9684	.9744	.9779	.9906	.9936
.7500						.9550	.9590	.9646	.9684	.9744	.9779	.9906	.9937

Table II-III-5 (d) Long-Radius Flow Nozzles: Discharge Coefficients, C, with Pipe Taps at 1 D and 1/2 D

(16-in. Pipe, D = 15.25 in.)

β \ R_d	14,000	16,000	18,000	20,000	25,000	30,000	40,000	50,000	75,000	100,000	500,000	1,000,000
.1500	.9347	.9392	.9430	.9462	.9524	.9570	.9635	.9679	.9747	.9788	.9934	.9969
.2000	.9347	.9393	.9430	.9462	.9525	.9571	.9635	.9679	.9748	.9788	.9934	.9969
.2500	.9348	.9393	.9431	.9463	.9525	.9571	.9635	.9679	.9748	.9788	.9934	.9969
.3000	.9348	.9394	.9431	.9463	.9525	.9571	.9636	.9680	.9748	.9789	.9934	.9969
.3500	.9349	.9394	.9432	.9464	.9526	.9572	.9636	.9680	.9748	.9789	.9934	.9969
.4000	.9350	.9395	.9433	.9464	.9526	.9572	.9637	.9680	.9749	.9789	.9935	.9969
.4500	.9351	.9396	.9433	.9465	.9527	.9573	.9637	.9681	.9749	.9790	.9935	.9969
.5000	.9352	.9397	.9434	.9466	.9528	.9574	.9638	.9681	.9749	.9790	.9935	.9969
.5500		.9398	.9435	.9467	.9529	.9574	.9638	.9682	.9750	.9790	.9935	.9969
.5750		.9398	.9436	.9467	.9529	.9575	.9639	.9682	.9750	.9791	.9935	.9969
.6000			.9436	.9468	.9530	.9575	.9639	.9683	.9750	.9791	.9935	.9969
.6250			.9437	.9468	.9530	.9576	.9639	.9683	.9751	.9791	.9935	.9969
.6500				.9469	.9531	.9576	.9640	.9683	.9751	.9791	.9935	.9970
.6750				.9470	.9531	.9576	.9640	.9684	.9751	.9792	.9936	.9970
.7000					.9532	.9577	.9641	.9684	.9752	.9792	.9936	.9970
.7250					.9532	.9577	.9641	.9684	.9752	.9792	.9936	.9970
.7500						.9578	.9641	.9685	.9752	.9792	.9936	.9970

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Table II-III-6 Expansion Factors for Flow Nozzles and Venturi Tubes

$$Y_a = \left(r^{2/\gamma} \frac{\gamma}{\gamma - 1} \frac{1 - r^{(\gamma-1)/\gamma}}{1 - r} \frac{1 - \beta^4}{1 - \beta^4 r^2/\gamma} \right)^{1/2}$$

$\gamma = 1.4$

β	β^4	0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55
					r					
0.2	0.0016	0.9728	0.9448	0.9160	0.8863	0.8556	0.8238	0.7908	0.7565	0.7207
.3	.0081	.9726	.9444	.9154	.8855	.8546	.8227	.7896	.7552	.7193
.4	.0256	.9719	.9432	.9137	.8833	.8520	.8198	.7864	.7517	.7156
0.50	.0625	.9706	.9405	.9099	.8785	.8464	.8133	.7793	.7441	.7076
.55	.0915	.9694	.9383	.9067	.8745	.8416	.8080	.7734	.7378	.7010
.60	.1296	.9678	.9352	.9023	.8690	.8351	.8006	.7653	.7292	.6920
0.65	.1785	.9655	.9309	.8962	.8613	.8261	.7905	.7543	.7175	.6798
.70	.2401	.9622	.9247	.8876	.8506	.8136	.7765	.7392	.7016	.6633
.725	.2763	.9600	.9207	.8819	.8436	.8056	.7676	.7297	.6915	.6530
0.75	.3164	.9573	.9158	.8751	.8353	.7960	.7571	.7184	.6797	.6409
.775	.3608	.9540	.9097	.8669	.8252	.7845	.7445	.7050	.6657	.6266
.80	.4096	.9498	.9022	.8566	.8128	.7705	.7292	.6889	.6491	.6097
0.82	.4521	.9457	.8947	.8466	.8009	.7570	.7147	.6736	.6334	.5939
.84	.4979	.9405	.8856	.8344	.7864	.7409	.6975	.6557	.6152	.5755
.86	.5470	.9338	.8740	.8194	.7688	.7215	.6769	.6344	.5936	.5541

Table II-III-7 Expansion Factors for Flow Nozzles and Venturi Tubes

$$Y_a = \left(r^{2/\gamma} \frac{\gamma}{\gamma - 1} \frac{1 - r^{(\gamma-1)/\gamma}}{1 - r} \frac{1 - \beta^4}{1 - \beta^4 r^{2/\gamma}} \right)^{1/2}$$

$\gamma - 1.3$

β	β^4	.095	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55
					r					
0.2	0.0016	0.9707	0.9407	0.9099	0.8781	0.8454	0.8117	0.7768	0.7406	0.7030
.3	.0081	.9705	.9402	.9092	.8773	.8445	.8106	.7756	.7393	.7016
.4	.0256	.9698	.9390	.9074	.8750	.8417	.8075	.7722	.7357	.6978
0.50	.0625	.9683	.9362	.9034	.8700	.8358	.8008	.7648	.7278	.6896
.55	.0915	.9671	.9338	.9001	.8658	.8309	.7952	.7588	.7214	.6829
.60	.1296	.9654	.9305	.8954	.8599	.8240	.7876	.7505	.7126	.6738
0.65	.1785	.9629	.9259	.8889	.8519	.8146	.7771	.7392	.7007	.6614
.70	.2401	.9594	.9193	.8798	.8406	.8016	.7627	.7237	.6844	.6447
.725	.2763	.9570	.9150	.8739	.8333	.7933	.7535	.7139	.6742	.6343
0.75	.3164	.9542	.9098	.8667	.8246	.7833	.7426	.7023	.6622	.6221
.775	.3608	.9507	.9034	.8580	.8141	.7714	.7297	.6886	.6481	.6077
.80	.4096	.9462	.8955	.8473	.8013	.7570	.7141	.6723	.6313	.5908
0.82	.4521	.9418	.8876	.8368	.7888	.7431	.6992	.6568	.6155	.5750
.84	.4979	.9362	.8779	.8241	.7739	.7266	.6817	.6387	.5971	.5567
.86	.5470	.9292	.8658	.8084	.7557	.7067	.6608	.6172	.5756	.5353

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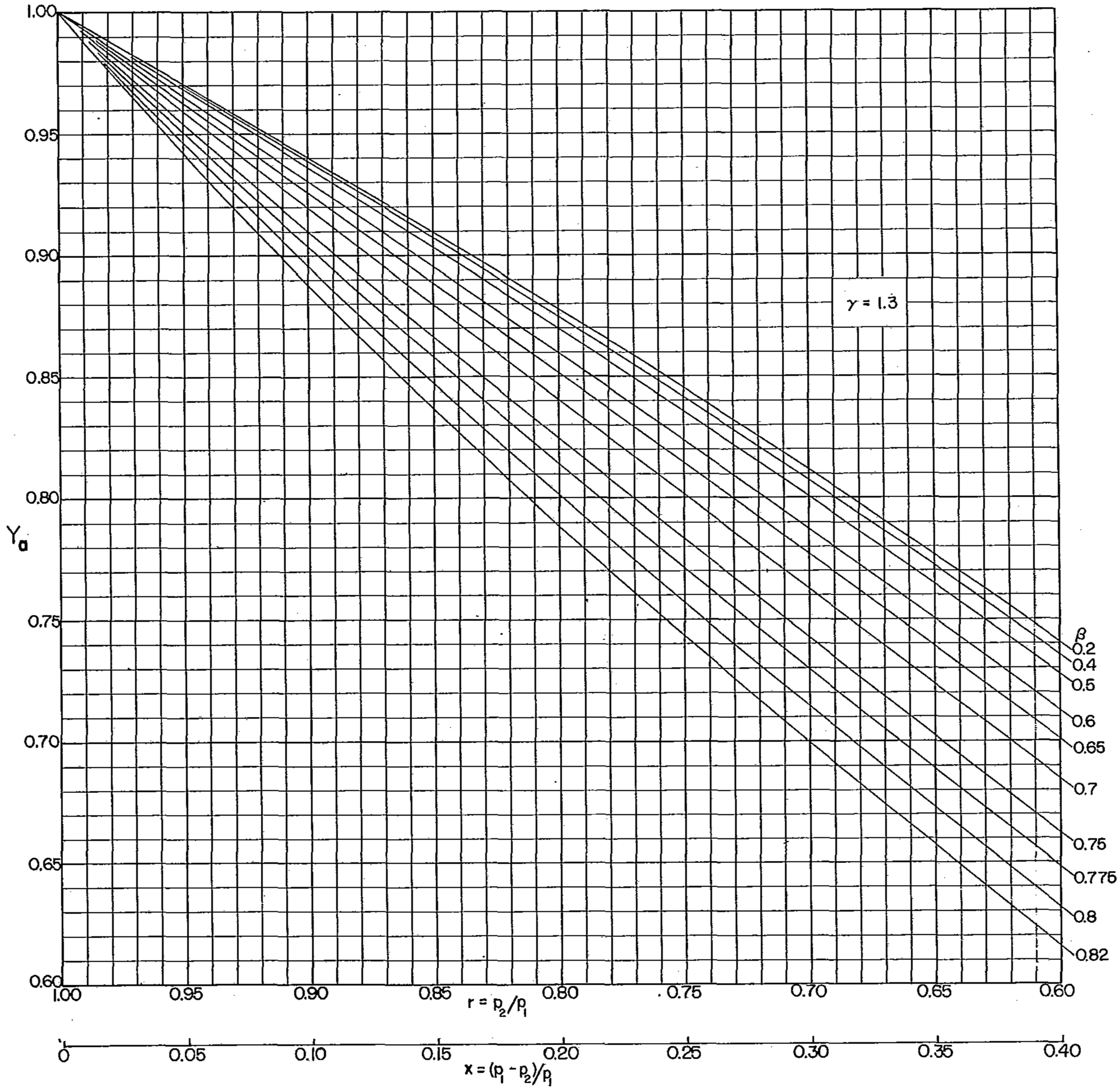

FIG. II-III-20 EXPANSION FACTORS FOR FLOWNOZZLES AND VENTURI TUBES, $\gamma = 1.3$

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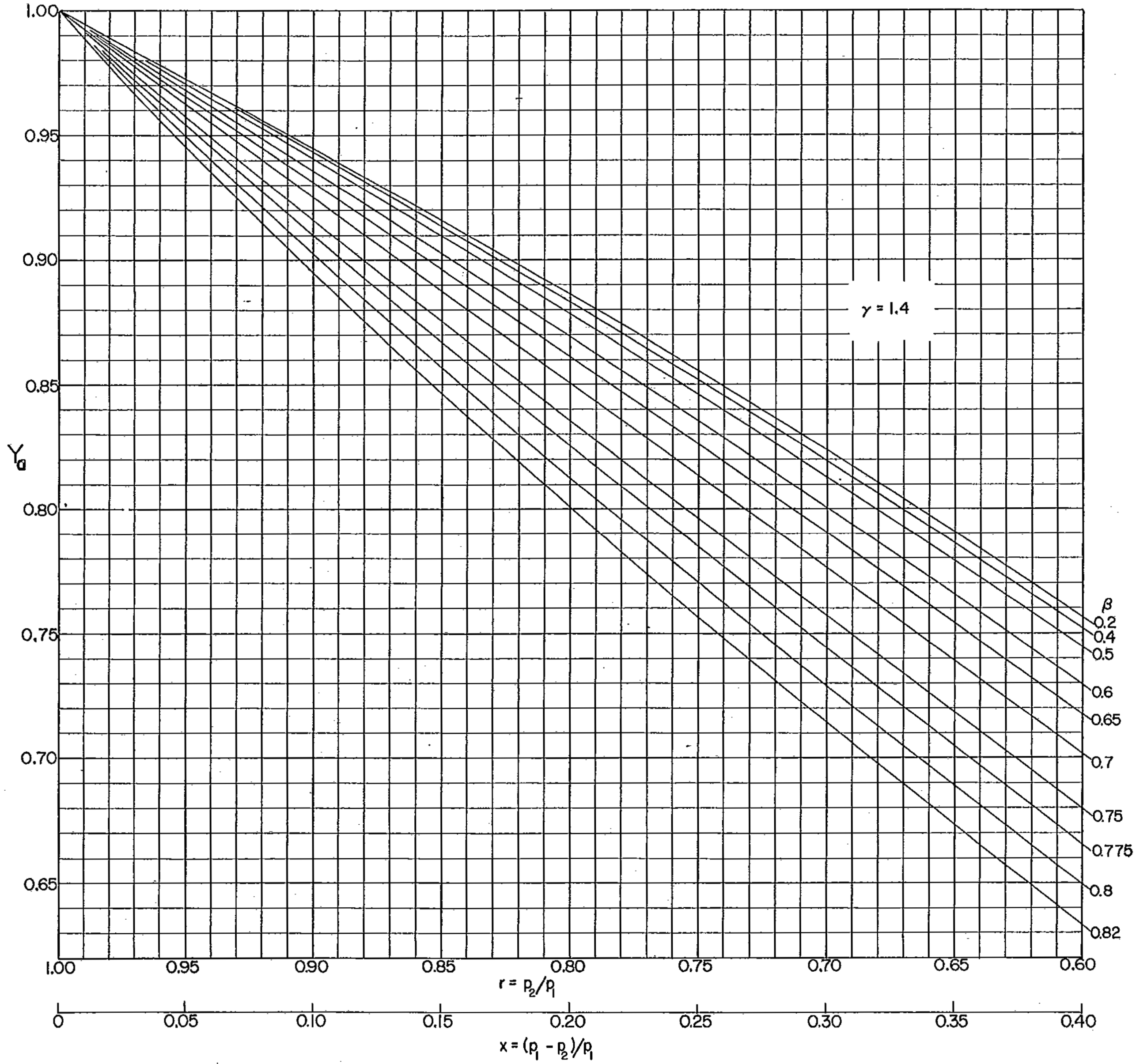

FIG. II-III-21 EXPANSION FACTORS FOR FLOWNOZZLES AND VENTURI TUBES, $\gamma = 1.4$

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II-III-30 1932 ISA Nozzle. Figure II-III-22 shows the form and proportions of the 1932 ISA flow nozzle (i.e., the flow nozzle adopted by the International Standards Association in 1932). It is used extensively abroad but hardly at all in this country. Pressures are measured from corner taps as shown by the figure.

Representative values of the flow coefficient, K, as a function of the pipe Reynolds number, R_D, are shown in Fig. II-III-23.

With compressible fluids, the expansion factor given by equation (II-III-13) applies.

For more detailed data on these flow nozzles, including tables of coefficients and correction factors, reference should be made to ISO Recommendation R541 [6].

II-III-31 Calibration of a Subsonic-Flow Nozzle. A Pitot tube or impact tip may be used to determine the discharge coefficient of a flow nozzle when other methods cannot be used. The procedure employs the use of a movable impact tip to measure the velocity pressure of the jet from the nozzle wall toward the center along one to four or more radial paths. The open end of the traversing tip should be very small and thin walled. Unless the nozzle is very large (e.g., over 12-in. throat diameter), a tip diameter of about 0.02 in. is a convenient size. The tip should be mounted so that, as it is moved along a radial path, it just clears the outlet end of the nozzle. Also, the mounting for the tip should incorporate some means for indicating the position of the center of the tip with reference to the axis of the nozzle or the inner surface of the nozzle throat. The amount of travel required in any test will depend some on the jet velocity, but it will not be necessary that the travel of the tip should be equal to the nozzle radius. (See Fig. II-III-24.)

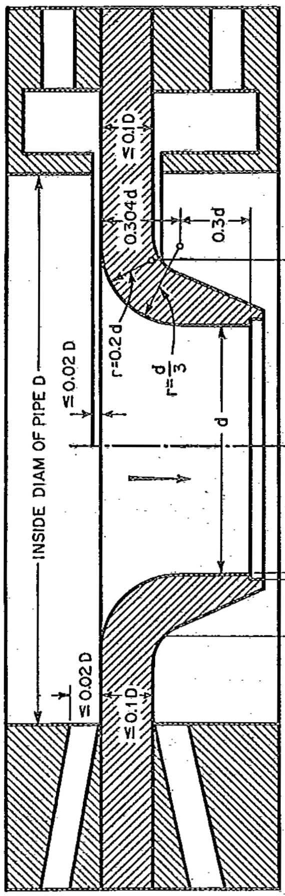

FIG. II-III-22 1932 FLOW NOZZLE OF INTERNATIONAL STANDARDS ASSOCIATION

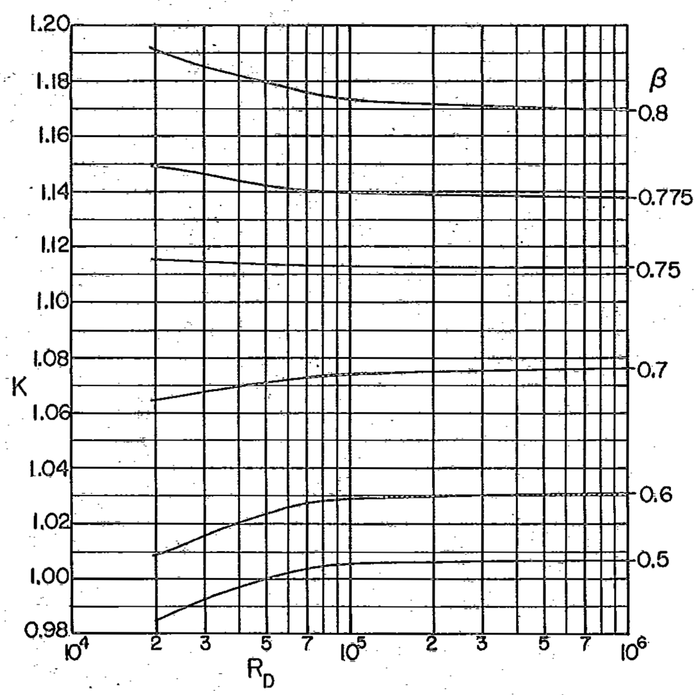

FIG. II-III-23 FLOW COEFFICIENTS, K, FOR THE 1932 ISA FLOW NOZZLE AS FUNCTION OF PIPE REYNOLDS NUMBER, R_D

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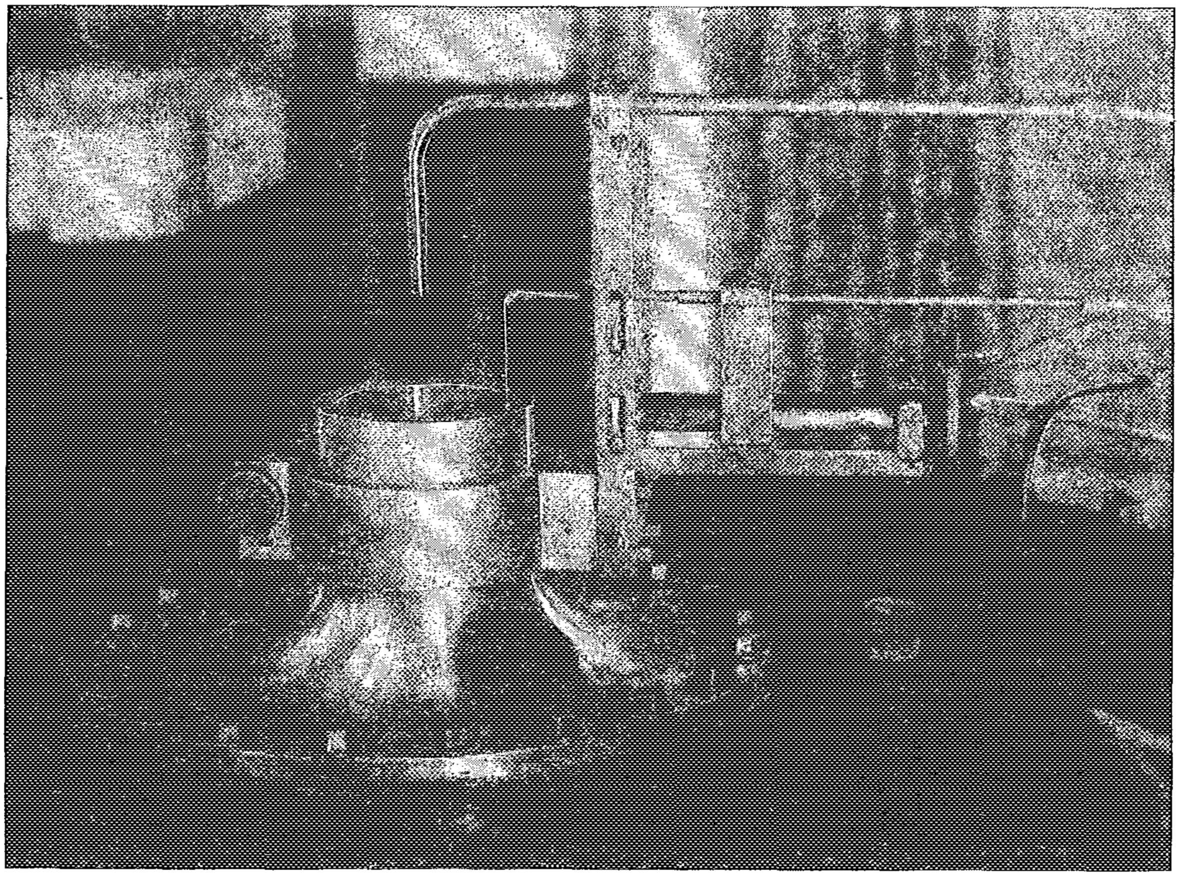

FIG. II-III-24 METHOD OF MOUNTING IMPACT TIPS FOR CALIBRATING A FLOWNOZZLE

If the nozzle is used as a discharge nozzle from a large plenum chamber, a pressure connection to the chamber will be required. In the more general case where the nozzle is mounted on the end of a pipe or within a pipe, a stationary central impact tip will be required. This tip must be mounted so that it does not interfere with the movement of the small traversing tip. The connection from the large tip is branched — one side going to a static pressure gage, usually a manometer. The other branch is connected to a differential pressure gage, the other side of which is connected to the small traversing tip. The range of this differential gage must be equal to the full velocity pressure (or impact pressure) of the nozzle jet, and the precision with which it can be read should be of the order of 0.5 per cent or less of the full range. Also, because of the smallness of the traversing-tip opening, the capacity of the connections between the differential gage and traversing tip should be kept as small as possible.

II-III-32 Procedure. Before starting a test, the traversing tip must be adjusted in its mounting so that when its center is exactly aligned with the (inner) surface of the nozzle throat the scale indicating the tip position reads "zero" (or an amount equal to the nozzle-throat radius). At the start of a test have the traversing tip withdrawn so that its inner edge is in line with the nozzle surface, i.e., the position scale reading is "minus" the value of the tip radius (assuming that a "plus" reading is toward the nozzle axis). Take readings of both the static and differential gages. Move the tip into the jet until its center is in line with the inner surface of the nozzle throat, i.e., the reading of the positioning scale is "zero;" observe and record the readings of the gages. Move the tip further into the jet by an amount equal to its radius. All of the tip is now just within the nozzle circumference. Observe and record the gage readings. Moving the tip by steps equal to its diameter, take two or three more sets of readings. Increase the amount of the

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tip movement to about four or five times its diameter, and continue taking readings of the pressure gages until a zero reading of the differential gage is obtained.

If the traversing tip can be moved to other positions around the nozzle to permit traverses at other meridians, these should be made in the manner as described, using the same tip positions as in the first traverse.

II-III-33 Computations. If traverses have been made at more than one position, compute the averages of the differential pressure for each separate radial setting of the traversing tip. The central impact pressure should have remained constant or very nearly constant throughout the series of readings. Now let

B = Radius of the cylindrical throat section of the nozzle

b = Radial distance between the geometrical axis of the nozzle throat and the geometrical center of the small traversing tip for any setting

c = Radius of the small traversing tip

Δ = Impact pressure measured by the larger central impact tip or the static pressure in the inlet chamber

δ = Impact pressure at the mouth of the smaller tip

= Δ – (the observed, or average, differential pressure)

Next, for each of the several positions of the traversing tip compute δ, $\sqrt{\delta/\Delta}$ and $(b/B)^2$. Plot the values of $\sqrt{\delta/\Delta}$ as ordinates against the corresponding values of $(b/B)^2$ as abscissae using suitable scales which will permit covering the entire range from 0 to 1.0 on the ordinate and a sufficient portion of the full abscissa scale to include all of the values of $(b/B)^2$. Figure II-III-25 shows a representative plot of such data.

II-III-34 If the diameter of the traversing tip could be diminished to a mere point, then when the axis of the tip is the plane of the nozzle wall, i.e., when $b = B$, the jet velocity would be zero and therefore δ would be 0.0 also. Hence, the ideal curve should pass through the point, (1.0, 0.0), which has been designated M in Fig. II-III-25. Compute $B - c$ and $[(B - c)/B]^2$, and draw a short line across the curve at this value of $[(B - c)/B]^2$. The point of intersection is designated N in Fig. II-III-25. Connect N with M with a straight line. Finally, let L represent the point vertically above M, i.e., at (1.0, 1.0), and let U represent the point where the curve becomes tangent to the $\sqrt{\delta/\Delta} = 1.0$ line. With a planimeter or by counting

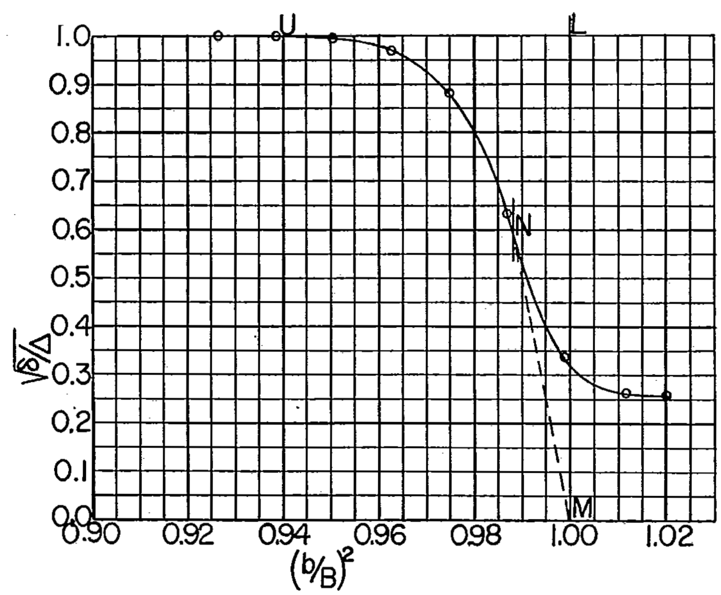

FIG. II-III-25 TYPE OF CURVE OBTAINED IN DETERMINING DISCHARGE COEFFICIENT OF FLOW NOZZLE WITH IMPACT TIP

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the squares, determine the area $\overline{MLUNM}$. Then if the area bounded by the four lines, $(b/B)^2 = 0$, $(b/B)^2 = 1.0$, $\sqrt{\delta/\Delta} = 0$ and $\sqrt{\delta/\Delta} = 1.0$, is called 1, the coefficient of the nozzle will be

$$C = 1 - (\text{area } \overline{MLUNM}) \qquad \text{(II-III-14)}$$

In Fig. II-III-25 the area $\overline{MLUNM} = 48$ squares very nearly. The area representing the full theoretical flow comprises 4000 squares. Hence, for the test there represented the coefficient is

$$C = \frac{4000 - 48}{4000} = 0.988$$

Note: For a detailed discussion of this procedure see References [7, 8].

Venturi Tubes

II-III-35 Fabrication. The classical or Herschel Venturi tube, as used in this country, is usually made of cast iron or cast steel in the smaller sizes. In very large sizes, the tubes may be made of rough-welded sheet metal [9].

The grouping of Venturi tubes used by the International Standards Organization Committee on Flow Measurement, is:

1. Tubes with a rough-cast unmachined surface of the entrance converging cone and recommended for use in 4- to 32-in. pipes.
2. Tubes with a machined converging entrance cone and seldom used in pipes larger than 10 in.
3. Tubes with a rough-welded sheet metal converging entrance cone and suitable for use in pipes up to 48 in.

Since the Venturi tubes used in this country are almost exclusively in group (1), this group will be the only one discussed in detail.

II-III-36 Venturi-Tube Proportions. As shown in Fig. II-III-26, the inlet section consists of a short cylindrical section joined by an easy curvature to a truncated cone having an included angle of 21 ±1 deg. The inlet cone is joined by another smooth curve to a short cylindrical section called the throat. The exit from this throat section leads by another easy curve into the exit or diffuser cone, the recommended included angle of which is 7 to 8 deg. If the inlet and throat sections are not a single unit or casting, the joint between them when assembled should be smooth, with neither step nor protruding gasket.

II-III-37 Pressure Taps. In the inlet and throat sections there should be four or more pressure holes leading into annular chambers, to which chambers the pipes to the secondary instruments (pressure gages) are connected. The cross-sectional area of the annular chambers, or tubes, should be not less than half the sum of the areas of the respective pressure holes. The recommended size of the pressure holes is between 5/32 and 25/64 in., inclusive, but never greater than 0.1 D or 0.13 d, respectively. Furthermore, within these limits the holes should be as small as suitable for use with the fluid being metered. The edge of the pressure holes with the inner surfaces of the inlet and throat must be free from burrs or nicks and may be square and sharp or rounded very slightly. (If rounded, the radius of rounding should be less than 0.1 the tap-hole diameter.)

II-III-38 Throat. The throat section may be lined with bronze or other corrosion-resistant material. It should be machined after being installed in the throat-section casting. When the tube is to be used at elevated temperatures, the thermal-expansion characteristics of the liner material should be as nearly as possible the same as that of the throat casting material.

It is recommended that the machining of the throat should produce a surface finish (roughness) of $(5 \times 10^{-6})\ d$ arithmetical mean deviation from the mean line of the profile (equivalent approximately to a 50-microinch finish). The machining should include the short curvature leading from the converging entrance section into the throat.

The throat diameter, d, should be measured very carefully in the plane of the throat pressure taps. The diameters should be near each pair of pressure taps and between the taps, with a minimum of four measurements. To determine whether the throat is cylindrical, diameters in other planes than that of the pressure taps should be measured. The same limitations regarding out-of-roundness as given for flow nozzles (Par. II-III-22) may be applied to the throat of a Venturi tube. The mean value of all diameters is to be used as the value of "d" in calculations of flow.

II-III-39 Other Features. In some cases it may be necessary to install a drain cock or vent in the pipe immediately preceding the Venturi tube to provide for removal of deposits or gases. These should be closed normally, especially when any important measurement is being made.

The diverging outlet cone angle may be as much as 15 deg, but the overall pressure loss will be greater, as shown in Fig. II-III-27. However, the 7-deg cone may be shortened, at the downstream end,

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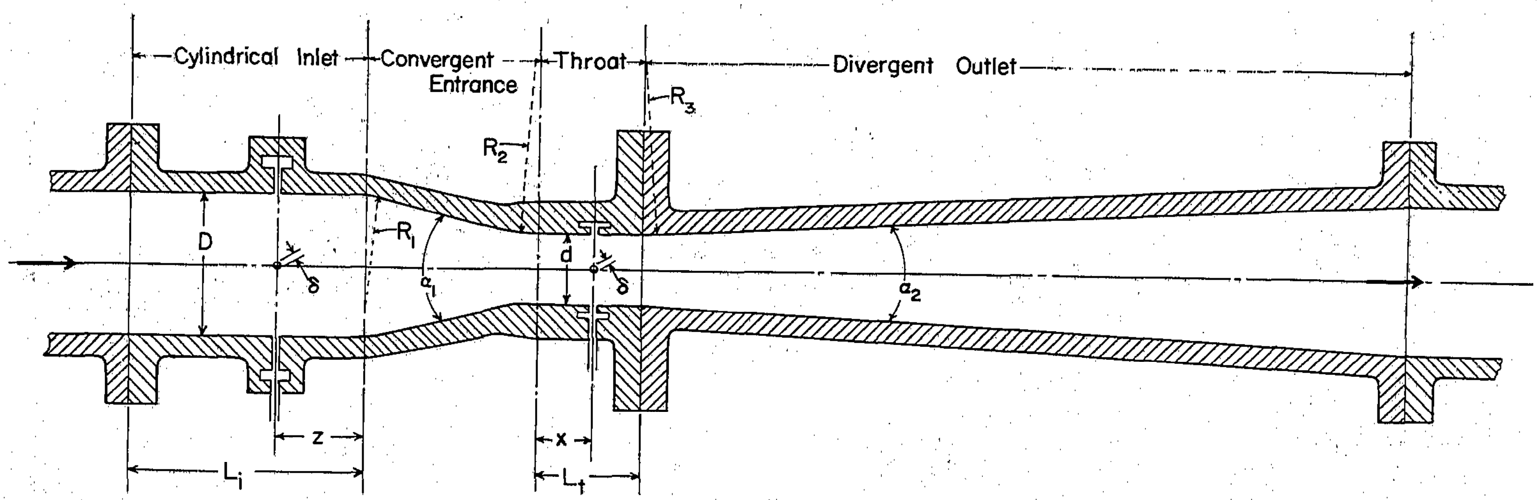

$L_i \lesseqgtr D$ or $L_i \lesseqgtr (D/4 + 10'')$
$z \lesseqgtr D/2 \pm D/4$ for $4'' \lesseqgtr D \lesseqgtr 6''$
$D/4 \lesseqgtr z \lesseqgtr D/2$ for $6'' \lesseqgtr D \lesseqgtr 32''$
$L_t \lesseqgtr d/3$
$y \lesseqgtr d/6$
$5/32'' \lesseqgtr \delta \lesseqgtr 25/64''$ and
$\delta < 0.1\,D$ or $0.13\,d$

$R_1 = 1.375\,D \pm 20\%$
$R_2 = 3.625\,d \pm 0.125\,d$
$5d \lesseqgtr R_3 \lesseqgtr 15d$
$\alpha_1 = 21° \pm 1°$
$7° \lesseqgtr \alpha_2 \lesseqgtr 8°$ or $7° \lesseqgtr \alpha \lesseqgtr 15°$

FIG. II-III-26 DIMENSIONAL PROPORTIONS OF CLASSICAL (HERSCHEL) VENTURI TUBES WITH A ROUGH-CAST, CONVERGENT INLET CONE

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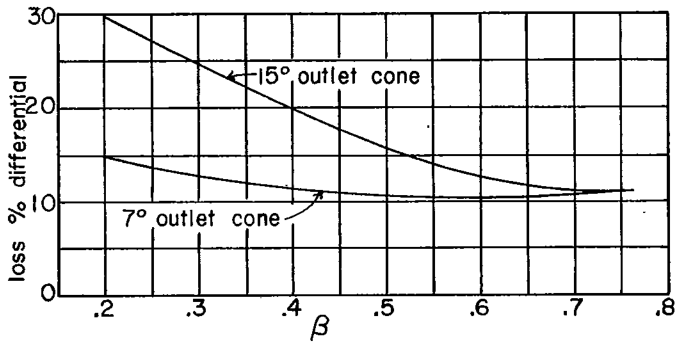

FIG. II-III-27 OVERALL PRESSURE LOSS THROUGH VENTURI TUBES

i.e., truncated by about 35 per cent of the normal length with only a very slight effect on the pressure loss.

As stated before, the dimensional relations shown in Fig. II-III-26 are those recommended when the converging entrance cone has a rough-cast surface. The proportions for tubes with a machined entrance cone or a rough-welded sheet-metal entrance cone are slightly different. However, it seems doubtful that the small dimensional differences would have any significant effect on flow measurements. To be sure, the character of the surface of the entrance cone does have an effect, as also the radius of curvature between the inlet cone and throat section.

II-III-40 Overall Pressure Loss. The overall pressure loss for Venturi tubes is shown by Fig. II-III-27.

II-III-41 Expansion Factors. The expansion factors to be used with Venturi tubes when measuring compressible fluids are the same as used with flownozzles, as given by the curves of Figs. II-III-20 and II-III-21 and by Tables II-III-6 and II-III-7.

II-III-42 Discharge Coefficients. For a classical Venturi tube with a rough-cast entrance cone,

C = 0.984 subject to a tolerance of ± 0.70%
when

$$4 \text{ in. } (100 \text{ mm}) \lesseqgtr D \lesseqgtr 32 \text{ in. } (800 \text{ mm})$$

$$0.3 \lesseqgtr \beta \lesseqgtr 0.75$$

$$2 \times 10^5 \lesseqgtr R_D \lesseqgtr 2 \times 10^6$$

For tubes with a machined entrance cone,

C = 0.995 subject to a tolerance of ± 1.00%
when

$$2 \text{ in. } (50 \text{ mm}) \lesseqgtr D \lesseqgtr 10 \text{ in. } (250 \text{ mm})$$

$$0.4 \lesseqgtr \beta \lesseqgtr 0.75$$

$$2 \times 10^5 \lesseqgtr R_D \lesseqgtr 1 \times 10^6$$

For tubes with a rough-welded sheet-metal entrance cone,

C = 0.985 subject to a tolerance of ± 1.50%
when

$$8 \text{ in. } (200 \text{ mm}) \lesseqgtr D \lesseqgtr 48 \text{ in. } (1200 \text{ mm})$$

$$0.4 \lesseqgtr \beta \lesseqgtr 0.70$$

$$2 \times 10^5 \lesseqgtr R_D \lesseqgtr 2 \times 10^6$$

II-III-43 Nozzle Venturi. The nozzle-Venturi (Fig. II-III-28) is used very little in this country. As made and used abroad, the cylindrical inlet and reducing conical sections of the classical Venturi are replaced with a single, short, curved inlet. The curvature of this inlet section is the same as that of the ISA 1932 nozzle, as shown in Fig. II-III-22. Beyond (downstream of) the throat of this inlet-nozzle section, there is an additional cylindrical section 0.4 d long, before the beginning of the

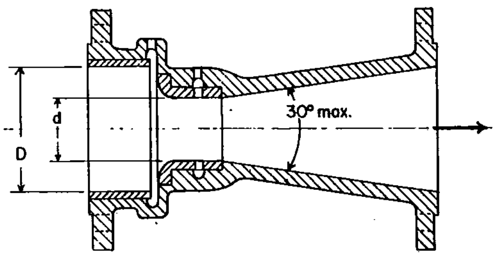

FIG. II-III-28 NOZZLE-VENTURI

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diverging outlet section. (Thus, the total length of the cylindrical "throat" is 0.7 d.) The included angle of the divergent outlet section may be 30 deg or less. The inlet pressure tap is a corner tap or annular slit at the intersection of the inlet face of the nozzle section with the pipe wall. The outlet or throat pressure taps are located 0.3 d from the entrance plane of the nozzle-throat portion and 0.4 D preceding the outlet end of the total throat section. For further data and coefficients for these nozzle-Venturis, refer to ISO 781 [9].

II-III-44 Computation of Rate of Flow: Subsonic Conditions. With all of the primary elements described in the preceding paragraphs used under *subsonic* conditions, the same general equation may be used for computing the rate of flow. As developed in Chapter I-5, this equation may be written in several forms, some of which are:

$$m(\mathrm{lb}_m/\mathrm{sec}) = 0.52502 \left(\frac{C\ Y\ d^2 F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\rho_1 (p_1 - p_2)} \qquad \text{(II-III-15a)}$$

$$= 0.52502\ K\ Y\ d^2 F_a\ \sqrt{\rho_1\ \Delta p} \qquad \text{(II-III-15b)}$$

$$= 0.099702 \left(\frac{C\ Y\ d^2 F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\rho_1\ h_w} \qquad \text{(II-III-15c)}$$

$$q_1\ (\text{cfs at } p_1,\ T_1) = 0.099702 \left(\frac{C\ Y\ d^2 F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\frac{h_w}{\rho_1}} \qquad \text{(II-III-16)}$$

$$m(\mathrm{lb}_m/\mathrm{hr}) = 1865.57\ K\ Y\ d^2 F_a\ \sqrt{\rho_1\ \Delta p} \qquad \text{(II-III-17a)}$$

$$= 358.93 \left(\frac{C\ Y\ d^2\ F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\rho_1\ h_w} \qquad \text{(II-III-17b)}$$

$$q_1\ (\text{cfh at } p_1,\ T_1) = 358.93 \left(\frac{C\ Y\ d^2\ F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\frac{h_w}{\rho_1}} \qquad \text{(II-III-18)}$$

For a segmental orifice, the d^2 would be replaced with (a = orifice area, in.^{2}), thus giving

$$m(\mathrm{lb}_m/\mathrm{hr}) = 457.0 \left(\frac{C\ Y\ a\ F_a}{\sqrt{1-\beta^4}}\right) \sqrt{\rho_1\ h_w} \qquad \text{(II-III-19)}$$

II-III-45 In the metering of compressible fluids, the general volume-pressure-temperature relation, equation (I-3-20) in Chapter I-3, in the form, $q_o = q_1\ [(p_1\ T_o\ Z_o)/(p_o\ T_1\ Z_1)]$, may be applied to equations (II-III-16) and (II-III-18) to give the corresponding volume rates at the reference conditions, p_o, T_o. Using the customary reference conditions for fuel-gas measurements of p_o = 14.73 psia, T_o = 60 F = 519.7 R and dry, equation (II-III-18) becomes

$$q_o(\mathrm{scfh}) = 7708\ K\ Y_1\ d^2 F_a Z_o \sqrt{\frac{h_w\ \rho_1}{G\ T_1 Z_1}} \qquad \text{(II-III-20)}$$

As stated in Chapter I-5, Par. I-5-14, this equation gives acceptable values if the metering conditions are not too far from the normal ambient conditions. For more precise measurements of particular gases and gas mixtures, equations similar to (II-III-20) should be developed from equations (I-5-27) and (I-5-33), using the gas constant, R, and the compressibility values for the particular gas.

II-III-46 The equation for the Reynolds number, of which the coefficients are a function, may be written

$$R_d = \frac{d\ V_2\ \rho_2}{12\ \mu} = \frac{48\ m}{\pi\ d\ \mu} \qquad \text{(II-III-21)}$$

and

$$R_D = \frac{D\ V_1\ \rho_1}{12\ \mu} = \frac{48\ m}{\pi\ D\ \mu} \qquad \text{(II-III-22)}$$

so that

$$R_D = \beta\ R_d \qquad \text{(II-III-23)}$$

The symbols and units applying to the above equations are:

a	Area of orifice, nozzle or Venturi throat	in.^{2}
C	Coefficient of discharge	ratio
d	Orifice diameter, also diameter of flow nozzle throat or Venturi throat	in.
F_a	Area thermal-expansion factor	Fig. II-I-3
G	Specific gravity, the ratio of molecular weights for gases	
h_w	Differential pressure	in. water at 68 F
K	Flow coefficient $= C/\sqrt{1-\beta^4}$	ratio

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m	Mass rate of flow	lb_m/sec or lb_m/hr
p	Pressure	psia
q	Volume rate of flow	cfs or cfh
R_d	Reynolds number using d	ratio
R_D	Reynolds number using D	ratio
Y	Expansion factor	ratio
Δp	Differential pressure	psi
μ	Viscosity of fluid, absolute	lb_m/ft-sec
ρ	Density of fluid	lb_m/ft^3

II-III-47 For use with metric units,

$$m(kg_m/sec) = 0.034783\ K\ Y\ d^2\ F_a\ \sqrt{\rho_1\ \Delta p} \qquad \text{(II-III-24)}$$

$$= 0.034752\ K\ Y\ d^2 F_a\ \sqrt{\rho_1\ h_w}$$

$$q_1(m^3/\text{sec at } p_1,\ T_1) = 0.000\ 034572\ K\ Y\ d^2 F_a \sqrt{\frac{h_w}{\rho_1}} \qquad \text{(II-III-25)}$$

In these two equations,

d	is in cm
h_w	is in cm of water at 20 C
m	is in kg_m/sec
p	and Δp are in gm_f/cm^2
q	is in m^3/sec
ρ	is in gm_m/cm^3

For evaluating the diameter ratio, β, both d and the pipe diameter, D, will be in cm. The Reynolds number with which to determine the coefficient may be evaluated by $R_d = 4000m/(\pi\ d\ \mu)$, in which the viscosity, μ, is in poise. With both p and Δp in the same units, the ratios, $x = \Delta p/p_1$ and x/γ, provide an index for the determination of the expansion factor, Y.

II-III-48 Proprietary Flow Tubes. There are a number of flow tubes that have constructional features that resemble both a flow nozzle and a Venturi tube. By utilizing a boundary layer effect upon the pressure sensed by either the high-pressure or low-pressure tap, these tubes may have a somewhat higher differential pressure than a conventional flow nozzle or Venturi tube of the same diameter ratio. For this reason, the overall pressure loss is rather low, as shown by Fig. II-III-29.

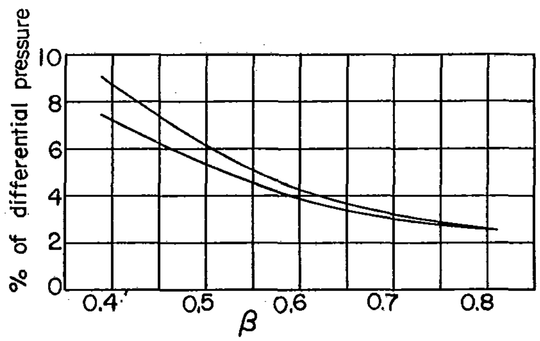

FIG. II-III-29 OVERALL PRESSURE LOSS THROUGH SOME PROPRIETARY FLOW TUBES

If one of these flow tubes is to be used, it should be calibrated with the piping section in which it is to be used and over the full range of rates of flow to which it will be subjected when in use.

Sonic-Flow Primary Elements

II-III-49 Materials and Form. The materials used for sonic flow nozzles are subject to the same considerations as given for subsonic flow nozzles (Pars. II-III-20 – II-III-22). The form or contour of the inner surface may be any of those shown in Figs. II-III-14 and II-III-22. If a minimal overall pressure loss is important, then the radial inlet Venturi (Fig. II-III-30) should be used. If both subsonic and sonic flows are to be measured with the same nozzle, it is recommended the form be one for which the coefficient is well known, such as the ASME long-radius nozzles (Fig. II-III-14).

II-III-50 Pressure Taps. As discussed in Chapter I-5, the inlet pressure and density only are required for determining the rate of flow. For this purpose, it is recommended that the inlet static pressure be

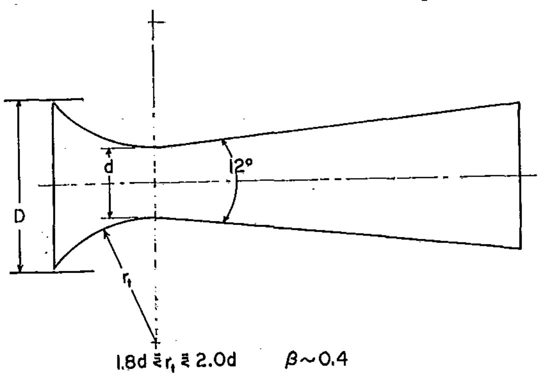

FIG. II-III-30 PROPORTIONS OF INTERIOR SURFACE OF CIRCULAR ARC OR RADIAL INLET VENTURI

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measured with a pipe-wall tap located one pipe diameter preceding the inlet of the nozzle. Alternatively, the total or stagnation inlet pressure may be measured with an impact or Pitot tube, mounted at the one pipe diameter distance. The axis of the open end of the impact tube should coincide with the nozzle axis if the diameter ratio, β, exceeds 0.3; but, if β is 0.3 or less, the opening may be anywhere within the central third of the cross-sectional area of the inlet channel.

The location of an outlet pressure tap will depend on the purpose for the outlet-pressure measurement. If the purpose is to insure that the outlet pressure is low enough so that sonic velocity is obtained at the nozzle exit, then the tap should be located in the pipe wall, as shown in Fig. II-III-15, or at the section of minimum cross-sectional area of a radial inlet Venturi. On the other hand, if the purpose is to determine the overall pressure loss, especially with the radial inlet Venturi, then the outlet pressure tap should be six to ten pipe diameters downstream from the throat section.

II-III-51 **Installation.** It is recommended that the installation requirements given for flow nozzles in subsonic service be followed with sonic flow nozzles in order to give consistent measurement over the entire flow range. A possible exception might be when the diameter ratio, β, is less than 0.2, so that the inlet velocity and flow pattern resemble that of a nozzle discharging from a plenum chamber.

II-III-52 **Temperature and Density.** For measurement of the fluid temperature, a thermocouple attached to the pipewall or a thermometer, bare or in a well, desirably a total temperature well, may be used where a simple instrument can be tolerated in terms of accuracy and safety. If a total temperature well is not used, then the stagnation temperature is to be obtained by applying the appropriate recovery factor to the thermometer reading. (See Chapter I-3, Par. I-3-17.)

The preferred location for the thermometer is on the outlet side of the sonic-flow element. (See Chapter I-5, Par. I-5-43.) If the thermometer is located on the inlet side of the element, it should be at least two pipe diameters, and preferably up to 200 stem or well diameters, upstream of the inlet pressure tap, so as to have as little effect on the pressure measurement as possible.

If the fluid density is determined with a densitometer or density cell, the instrument should be installed and operated in accordance with the manufacturer's instructions.

II-III-53 **Calibration.** As with all types of primary elements utilizing measurements of a pressure and a differential pressure, a sonic-flow primary element should be calibrated in the section of piping in which it will be used. If this is not possible, the actual piping should be duplicated as closely as possible, with particular care taken so that the inlet section is sized so as to maintain the diameter ratio, β, the *same*, if in service β is greater than 0.3. If in service β is less than 0.3, the calibration may be made with a pipe section giving a lower but not a higher value of β.

The reference method of measuring the fluid flow will depend on the size and flow capacity of the element, as well as on the facilities available. Some of the methods are volumetric gasometers, weighing containers and the pressure drop due to the discharge of gas from a container. All of these involve the measurement of a time interval.

Possibly the simplest procedure is to place the sonic-flow element in series with another meter whose flow characteristic is known from a history of reliable calibration data. This reference meter may be another sonic-flow nozzle operated under sonic or subsonic-flow conditions or maybe an orifice. Care must be taken to eliminate the effect of the discharge-velocity profile of the upstream element on the inlet-flow profile of the downstream element. Often screens or flow straighteners are placed between the two elements to restore a uniform velocity profile.

II-III-54 **Coefficients.** The discharge coefficient of a sonic-flow element should be obtained from a calibration as previously outlined. When this is not possible, an approximate coefficient value of 0.99 may be used. In this case, the uncertainty or tolerance to the coefficient should be between ± 2 to ± 3 per cent.

II-III-55 **Sonic-Flow Computations.** If the inlet static pressure is measured from one or more side-wall pressure taps, then first this value must be converted to total pressure using the equation

$$\left(\frac{p_1}{p_{1t}}\right)^{2/\Gamma} - \left(\frac{p_1}{p_{1t}}\right)^{(\Gamma+1)/\Gamma} = \beta^4 \, \frac{\Gamma-1}{2} \left(\frac{2}{\Gamma+1}\right)^{\frac{\Gamma+1}{\Gamma-1}} \qquad \text{(II-III-26)}$$

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However, if β is less than 0.5, then the following equation may be used

$$p_{1t} = \frac{p_1}{\left[1 - \beta^4 \frac{\Gamma}{2}\left(\frac{2}{\Gamma+1}\right)^{(\Gamma+1)/\Gamma-1)}\right]} \quad \text{(II-III-27)}$$

This conversion is not necessary if the total inlet pressure is measured with an impact tube.

With the values of the stagnation pressure and temperature, the mass rate of flow at throat sonic speed may be computed by

$$m\ (\text{lb}_m/\text{sec}) = C\ a\ \phi^*_i \left(\frac{\phi^*}{\phi^*_i}\right)\left(\frac{p_{1t}}{\sqrt{T_{1t}}}\right) \quad \text{(II-III-28)}$$

where

C = Discharge coefficient at sonic-flow conditions

a = Throat area of sonic-flow primary element, ft^2 or in.^{2}

p_{1t} = Inlet stagnation pressure, psfa or psia

T_{1t} = Inlet stagnation temperature of fluid, °R

ϕ^*_i = Sonic-flow function of an ideal gas having a constant ratio of specific heats, a molecular weight and structure identical to the real gas being metered, as evaluated by equations (I-5-103) and (I-5-105)

$\dfrac{\phi^*}{\phi^*_i}$ = Ratio of the real-gas sonic-flow function to the sonic-flow function of its ideal-gas counterpart, values from Tables II-III-8 through II-III-20

Where the equation of state is represented by tables of gas properties as in the case of steam, it may be more convenient and accurate to use

$$m\ (\text{lb}_m/\text{sec}) = C\ a\ B_F \left(\frac{F}{F_i}\right)\sqrt{\frac{p_{1t}}{v_{1t}}} \quad \text{(II-III-29)}$$

where

C = Discharge coefficient at sonic-flow conditions

B_F = $\sqrt{g_c}\ F_i$

F_i = Ideal isentropic expansion function, equation (I-5-104)

F = Real-gas isentropic expansion function, equation (I-5-124)

v_{1t} = Specific volume at inlet stagnation conditions

$\dfrac{F}{F_i}$ = Ratios given in Tables II-III-9 through II-III-21

Note: In equation (II-III-29), as given above, a is ft^2 and p is psfa; but, if a is in in.^{2} and p is psia, then the right side must be multiplied by 1/12.

II-III-56 In the special case of the sonic flow of a natural fuel gas, equation (II-III-28) can be modified by replacing $\phi^*_i\ (\phi^*/\phi^*_i)$ with $[(e_c j + b_c)/(e_z j + b_z)]$ $\sqrt{(gc\ MW)/R}$ so that

$$m\ (\text{lb}_m/\text{sec}) = C\ a\left(\frac{e_c j + b_c}{e_z j + b_z}\right)\sqrt{\frac{g_c\ MW}{R}}\ \frac{p_{1t}}{\sqrt{T_{1t}}} \quad \text{(II-III-30)}$$

where

$$j = X_{C_2H_6} + X_{CO_2} - \frac{1}{2}X_{N_2} + 2\ X_{C_3H_8} + 3\ X_{C_4H_{10}} \quad \text{(II-III-31)}$$

X = mole fraction of the subscripted gas and values of e_c, b_c, e_z and b_z are given by Tables II-III-23 through II-III-26 [10].

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Table II-III-8

Values of ϕ^*/ϕ_i, Ratio of the Real-Gas Sonic-Flow Function to the Sonic Flow Function of Its Ideal-Gas Counterpart

$$\phi_i^* = 0.53175$$

T_{1t}	Inlet Stagnation Pressure, p_{1t} (psia)									
(°R)	0	100	200	400	600	800	1000	1200	1400	1500
400	1.0002	1.0073	1.0145	1.0296	1.0458	1.0626	1.0809	1.0994	1.1180	1.1262
450	1.0002	1.0051	1.0100	1.0202	1.0306	1.0413	1.0519	1.0629	1.0733	1.0786
500	1.0002	1.0037	1.0072	1.0142	1.0212	1.0283	1.0353	1.0421	1.0488	1.0521
550	1.0001	1.0025	1.0050	1.0100	1.0150	1.0198	1.0245	1.0291	1.0335	1.0357
600	1.0000	1.0018	1.0035	1.0071	1.0105	1.0139	1.0172	1.0203	1.0232	1.0246
650	0.9998	1.0011	1.0024	1.0049	1.0072	1.0096	1.0119	1.0140	1.0160	1.0169
700	0.9995	1.0004	1.0013	1.0030	1.0049	1.0065	1.0079	1.0093	1.0106	1.0112

Table II-III-9

Values of F/F_i, Ratio of Real Gas to Ideal Gas Isentropic Expansion Function

$$F_i = 0.68473 \qquad B_F = 3.8839$$

T_{1t}	Inlet Stagnation Pressure, p_{1t} (psia)									
(°R)	0	100	200	400	600	800	1000	1200	1400	1500
400	1.0002	1.0019	1.0035	1.0076	1.0138	1.0204	1.0301	1.0418	1.0539	1.0581
450	1.0002	1.0019	1.0036	1.0081	1.0131	1.0195	1.0257	1.0344	1.0422	1.0465
500	1.0002	1.0021	1.0041	1.0081	1.0123	1.0177	1.0230	1.0295	1.0359	1.0390
550	1.0001	1.0016	1.0035	1.0075	1.0115	1.0162	1.0207	1.0262	1.0316	1.0343
600	1.0000	1.0016	1.0032	1.0074	1.0100	1.0157	1.0191	1.0238	1.0283	1.0304
650	0.9998	1.0014	1.0031	1.0066	1.0093	1.0138	1.0175	1.0219	1.0260	1.0279
700	0.9995	1.0012	1.0026	1.0057	1.0091	1.0129	1.0162	1.0200	1.0238	1.0256

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Table II-III-10

Values of ϕ^*/ϕ_i^*, Ratio of Real-Gas Sonic-Flow Function to Sonic Flow Function of Its Ideal-Gas Counterpart

$\phi_i^* = 0.40866$

T_{1t}	Inlet Stagnation Pressure, p_{1t} (psia)										
(°R)	0	1	100	200	400	600	800	1000	1200	1400	1600
600	1.0016	1.0018	1.0089	1.0181	1.0335						
650	1.0006	1.0011	1.0071	1.0139	1.0274	1.0424	1.0589	1.0754			
700	0.9996	1.0003	1.0052	1.0107	1.0218	1.0339	1.0461	1.0603	1.0813	1.1021	1.1230
750	0.9986	0.9995	1.0036	1.0082	1.0174	1.0272	1.0379	1.0487	1.0636	1.0785	1.0934
800	0.9971	0.9986	1.0018	1.0057	1.0136	1.0217	1.0312	1.0397	1.0514	1.0624	1.0738
850	0.9968	0.9978	0.9999	1.0034	1.0103	1.0176	1.0251	1.0326	1.0410	1.0507	1.0597
900	0.9959	0.9969	0.9986	1.0016	1.0074	1.0136	1.0199	1.0263	1.0338	1.0413	1.0487
950	0.9950	0.9960	0.9975	1.0000	1.0052	1.0104	1.0158	1.0212	1.0275	1.0337	1.0401
1000	0.9940	0.9952	0.9964	0.9986	1.0030	1.0076	1.0124	1.0172	1.0224	1.0277	1.0329
1050	0.9930	0.9945	0.9957	0.9976	1.0013	1.0053	1.0096	1.0138	1.0184	1.0229	1.0272
1100	0.9923	0.9935	0.9950	0.9967	1.0001	1.0036	1.0073	1.0109	1.0148	1.0187	1.0226
1150	0.9913	0.9928	0.9940	0.9956	0.9989	1.0019	1.0052	1.0084	1.0118	1.0152	1.0187
1200	0.9903	0.9918	0.9932	0.9946	0.9975	1.0004	1.0033	1.0062	1.0092	1.0123	1.0153
1250	0.9893	0.9908	0.9925	0.9938	0.9964	0.9989	1.0014	1.0038	1.0065	1.0093	1.0120
1300	0.9886	0.9898	0.9915	0.9927	0.9950	0.9973	0.9994	1.0016	1.0040	1.0065	1.0089
1350	0.9879	0.9884	0.9903	0.9914	0.9936	0.9956	0.9975	0.9994	1.0016	1.0038	1.0060
1400	0.9871	0.9879	0.9891	0.9901	0.9920	0.9939	0.9956	0.9974	0.9993	1.0012	1.0031
1450	0.9864	0.9869	0.9879	0.9888	0.9906	0.9922	0.9937	0.9952	0.9969	0.9986	1.0004
1500	0.9857	0.9859	0.9864	0.9872	0.9890	0.9905	0.9918	0.9932	0.9947	0.9962	0.9976
1550	0.9849	0.9847	0.9849	0.9857	0.9873	0.9887	0.9898	0.0010	0.9923	0.9937	0.9950
1600	0.9842	0.9837	0.9832	0.9840	0.9854	0.9869	0.9880	0.9891	0.9902	0.9913	0.9925

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Table II-III-11

Values of F/F_i, Ratio of Real-Gas to Ideal-Gas Isentropic Expansion Function

$F_i = 0.66727 \quad B_F = 3.7848$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 1	100	200	400	600	800	1000	1200	1400	1600
600	1.0018	1.0005	1.0000	0.9988						
650	1.0010	0.9998	0.9993	0.9984						
700	1.0003	0.9992	0.9987	0.9980	0.9971	0.9961	0.9950			
750	0.9995	0.9984	0.9981	0.9976	0.9969	0.9960	0.9952			
800	0.9987	0.9984	0.9980	0.9974	0.9967	0.9961	0.9954	0.9948	0.9943	0.9937
850	.9979	.9976	.9973	.9968	.9962	.9957	.9952	.9947	.9943	.9938
900	.9970	.9968	.9966	.9962	.9957	.9954	.9950	.9946	.9943	.9940
950	.9962	.9960	.9954	.9956	.9952	.9950	.9946	.9944	.9941	.9939
1000	.9954	.9953	.9952	.9950	.9947	.9945	.9943	.9941	.9939	.9937
1050	.9946	.9945	.9944	.9942	.9941	.9939	.9938	.9937	.9936	.9935
1100	.9938	.9937	.9935	.9934	.9934	.9932	.9932	.9932	.9932	.9932
1150	.9928	.9927	.9926	.9926	.9926	.9926	.9927	.9927	.9927	.9927
1200	.9917	.9918	.9918	.9919	.9920	.9920	.9921	.9922	.9923	.9925
1250	.9908	.9909	.9909	.9911	.9913	.9913	.9914	.9916	.9917	.9919
1300	.9899	.9901	.9901	.9903	.9905	.9906	.9908	.9910	.9911	.9913
1350	.9889	.9890	.9891	.9893	.9896	.9897	.9898	.9900	.9903	.9906
1400	.9879	.9880	.9881	.9884	.9886	.9888	.9890	.9893	.9895	.9898
1450	.9870	.9871	.9872	.9874	.9876	.9877	.9878	.9881	.9883	.9886
1500	.9860	.9861	.9862	.9865	.9865	.9868	.9870	.9871	.9873	.9875

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METHANE CH_4

Table II-III-12

Values of ϕ^*/ϕ_i^*, Ratio of Sonic-Flow Function of Methane to Sonic-Flow Function of Its Ideal-Gas Counterpart

$\phi_i^* = 0.3885$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.9993	1.0097	1.0209	1.0451	1.0734	1.1057	1.1429
500	0.9975	1.0050	1.0129	1.0297	1.0481	1.0683	1.0902
550	0.9952	1.0008	1.0065	1.0185	1.0315	1.0451	1.0594
600	0.9923	0.9965	1.0010	1.0099	1.0193	1.0289	1.0389
650	0.9891	0.9923	0.9958	1.0025	1.0093	1.0164	1.0237
700	0.9856	0.9882	0.9909	0.9961	1.0011	1.0065	1.0118

Table II-III-13

Values of F/F_i, Ratio of Isentropic Expansion Function to That of Its Ideal-Gas Counterpart

$B_F = 3.8135 \quad F_i = 0.6723$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.9993	0.9987	0.9989	0.9989	0.9999	1.0035	1.0100
500	0.9975	0.9974	0.9974	0.9981	1.0000	1.0032	1.0080
550	0.9952	0.9953	0.9956	0.9967	0.9986	1.0016	1.0054
600	0.9923	0.9926	0.9931	0.9944	0.9964	0.9990	1.0025
650	0.9891	0.9895	0.9901	0.9915	0.9934	0.9959	0.9990
700	0.9856	0.9862	0.9868	0.9883	0.9901	0.9925	0.9953

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Table II-III-14

Values of ϕ^*/ϕ_i^*, Ratio of Real-Gas Sonic-Flow Function to Sonic-Flow Function of Its Ideal-Gas Counterpart

$\phi_i^* = 0.52295$

T_{1t}	Inlet Stagnation Pressure, p_{1t} (psia)									
(°R)	0	100	200	400	600	800	1000	1200	1400	1500
400	1.0000	1.0067	1.0134	1.0275	1.0424	1.0578	1.0736	1.0895	1.1054	1.1131
450	1.0000	1.0046	1.0092	1.0187	1.0282	1.0380	1.0477	1.0572	1.0665	1.0711
500	1.0000	1.0032	1.0064	1.0130	1.0194	1.0259	1.0323	1.0384	1.0442	1.0469
550	1.0000	1.0023	1.0046	1.0090	1.0136	1.0179	1.0222	1.0263	1.0301	1.0320
600	0.9998	1.0015	1.0032	1.0064	1.0094	1.0124	1.0154	1.0180	1.0206	1.0219
650	0.9998	1.0010	1.0021	1.0044	1.0064	1.0084	1.0104	1.0122	1.0138	1.0148
700	0.9997	1.0005	1.0013	1.0028	1.0042	1.0055	1.0067	1.0078	1.0089	1.0093

Table II-III-15

Values of F/F_i, Ratio of Real-Gas to Ideal-Gas Isentropic Expansion Function

$F_i = 0.68473 \quad B_F = 3.8839$

T_{1t}	Inlet Stagnation Pressure, p_{1t} (psia)									
(°R)	0	100	200	400	600	800	1000	1200	1400	1500
400	1.0000	1.0030	1.0038	1.0084	1.0150	1.0229	1.0317	1.0431	1.0556	1.0617
450	1.0000	1.0020	1.0041	1.0091	1.0141	1.0210	1.0276	1.0360	1.0448	1.0491
500	1.0000	1.0019	1.0040	1.0090	1.0132	1.0191	1.0248	1.0316	1.0383	1.0423
550	1.0000	1.0019	1.0040	1.0080	1.0126	1.0176	1.0226	1.0284	1.0339	1.0367
600	0.9998	1.0017	1.0038	1.0079	1.0117	1.0163	1.0209	1.0258	1.0308	1.0333
650	0.9998	1.0015	1.0034	1.0075	1.0109	1.0151	1.0194	1.0238	1.0281	1.0306
700	0.9997	1.0014	1.0032	1.0069	1.0104	1.0143	1.0180	1.0221	1.0263	1.0280

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Table II-III-16

Values of ϕ^* / ϕ^*_i, Ratio of Real-Gas Sonic-Flow Function to Sonic-Flow Function of Its Ideal-Gas Counterpart

$\phi^*_i = 0.45620$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000	1200	1400	1500
400	0.9998	1.0082	1.0169	1.0355	1.0560	1.0786	1.1030	1.1294	1.1584	1.1734
450	0.9997	1.0056	1.0117	1.0244	1.0378	1.0520	1.0668	1.0819	1.0974	1.1052
500	0.9995	1.0039	1.0083	1.0172	1.0265	1.0361	1.0459	1.0558	1.0655	1.0705
550	0.9991	1.0022	1.0055	1.0121	1.0188	1.0257	1.0324	1.0393	1.0460	1.0493
600	0.9986	1.0010	1.0034	1.0083	1.0131	1.0180	1.0231	1.0280	1.0327	1.0350
650	0.9979	0.9997	1.0015	1.0051	1.0088	1.0124	1.0161	1.0196	1.0230	1.0247
700	0.9972	0.9985	0.9999	1.0026	1.0053	1.0079	1.0105	1.0131	1.0156	1.0168

Table II-III-17

Values of F/F_i, Ratio of Real-Gas to Ideal-Gas Isentropic Expansion Function

$F_i = 0.68473 \quad B_F = 3.8839$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000	1200	1400	1500
400	0.9998	1.0008	1.0013	1.0031	1.0083	1.0099	1.0157	1.0267	1.0430	1.0524
450	0.9997	1.0008	1.0020	1.0047	1.0088	1.0135	1.0193	1.0267	1.0345	1.0394
500	0.9995	1.0009	1.0022	1.0051	1.0085	1.0124	1.0168	1.0223	1.0286	1.0321
550	0.9991	1.0003	1.0019	1.0050	1.0082	1.0120	1.0158	1.0205	1.0250	1.0275
600	0.9986	1.0000	1.0014	1.0045	1.0073	1.0107	1.0145	1.0185	1.0223	1.0242
650	0.9979	0.9993	1.0006	1.0034	1.0063	1.0096	1.0130	1.0164	1.0198	1.0216
700	0.9972	0.9984	0.9999	1.0027	1.0054	1.0083	1.0112	1.0145	1.0177	1.0193

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Table II-III-18

Values of ϕ^* / ϕ^*_i, Ratio of Real-Gas Sonic-Flow Function to Sonic-Flow Function of Its Ideal-Gas Counterpart

$$\phi^*_i = 0.14029$$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000	1200	1400	1500
400	1.0096	1.0090	1.0086	1.0075	1.0063	1.0050	1.0039	1.0025	1.0012	1.0005
450	1.0062	1.0055	1.0048	1.0036	1.0022	1.0009	0.9994	0.9980	0.9965	0.9958
500	1.0039	1.0032	1.0025	1.0010	0.9996	0.9981	0.9966	0.9952	0.9937	0.9930
550	1.0023	1.0016	1.0009	0.9994	0.9979	0.9964	0.9949	0.9935	0.9920	0.9913
600	1.0013	1.0006	0.9998	0.9983	0.9969	0.9954	0.9939	0.9924	0.9910	0.9903
650	1.0005	0.9998	0.9991	0.9977	0.9962	0.9949	0.9933	0.9919	0.9904	0.9897
700	1.0001	0.9994	0.9987	0.9973	0.9959	0.9943	0.9930	0.9916	0.9902	0.9895

Table II-III-19

Values of F/F_i, Ratio of Real-Gas to Ideal-Gas Isentropic Expansion Function

$$F_i = 0.68473 \quad B_F = 3.8839$$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000	1200	1400	1500
400	1.0096	1.0112	1.0132	1.0169	1.0204	1.0243	1.0278	1.0314	1.0342	1.0370
450	1.0062	1.0075	1.0090	1.0125	1.0158	1.0189	1.0219	1.0251	1.0283	1.0299
500	1.0039	1.0052	1.0066	1.0093	1.0122	1.0149	1.0176	1.0208	1.0239	1.0255
550	1.0023	1.0036	1.0048	1.0074	1.0098	1.0122	1.0147	1.0175	1.0200	1.0213
600	1.0013	1.0025	1.0035	1.0057	1.0081	1.0103	1.0126	1.0148	1.0172	1.0184
650	1.0005	1.0016	1.0026	1.0047	1.0068	1.0091	1.0108	1.0129	1.0149	1.0160
700	1.0001	1.0010	1.0019	1.0039	1.0059	1.0075	1.0095	1.0116	1.0133	1.0143

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Table II-III-20

Values of ϕ^* / ϕ^*_i, Ratio of Sonic-Flow Function of Natural-Fuel Gas to Sonic-Flow Function of Its Ideal-Gas Counterpart

$\phi^*_i = 0.3947$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.9980	1.0091	1.0211	1.0476	1.0790	1.1164	1.1611
500	0.9961	1.0041	1.0123	1.0305	1.0507	1.0730	1.0976
550	0.9936	0.9994	1.0055	1.0185	1.0324	1.0472	1.0629
600	0.9906	0.9951	0.9997	1.0093	1.0192	1.0296	1.0404
650	0.9873	0.9906	0.9943	1.0015	1.0088	1.0164	1.0242
700	0.9839	0.9865	0.9893	0.9948	1.0002	1.0059	1.0116

*Composition by mole fraction: Methane (CH_4), 0.960; ethane (C_2H_6), 0.035; carbon dioxide (CO_2), 0.002; and nitrogen (N_2), 0.003.

Table II-III-21

Values of F/F_i Ratio of Isentropic Expansion Function of Natural-Fuel Gas to That of Its Ideal-Gas Counterpart

$B_F = 3.8135 \quad F_i = 0.6723$

T_{1t} (°R)	Inlet Stagnation Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.9980	0.9973	0.9968	0.9964	0.9976	1.0013	1.0090
500	0.9961	0.9959	0.9956	0.9961	0.9977	1.0008	1.0059
550	0.9936	0.9936	0.9937	0.9946	0.9964	0.9992	1.0032
600	0.9906	0.9909	0.9912	0.9924	0.9941	0.9967	1.0001
650	0.9873	0.9876	0.9882	0.9894	0.9912	0.9936	0.9967
700	0.9839	0.9843	0.9849	0.9863	0.9879	0.9903	0.9930

*Composition by mole fraction: Methane (CH_4), 0.960; ethane (C_2H_6), 0.035; carbon dioxide (CO_2), 0.002; and nitrogen (N_2), 0.003.

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For use in the calibration of displacement and other volume-indicating meters, equation (II-III-29) can be transformed to give volume rate of flow at the inlet conditions by multiplying all terms by the specific volume, v_{1t}, giving

$$q_1 \text{ (cfs)} = C\ a\ B_F \left(\frac{F}{F_i}\right) \sqrt{\frac{Z_1 R\ T_{1t}}{MW}} \qquad \text{(II-III-32)}$$

Let $F = (e_c j + b_c)$ and $\sqrt{Z_1} = (e_z j + b_Z)$.

Then,

$$q_1 \text{ (cfs)} = C\ a\,(e_c j + b_c)\,(e_z j + b_z) \sqrt{\frac{g_c R\ T_{1t}}{MW}} \qquad \text{(II-III-33)}$$

in which a, the nozzle throat area, is in *square feet* and $R = 1545.33$ (see Par. I-3-25). Obviously, the reciprocal of equation (II-III-33) will give the time in seconds for the discharge of 1 cu ft at inlet stagnation conditions.

II-III-57 For gases that are not tabulated here, the sonic flow can be computed with equal accuracy from equation (II-III-34) if Γ, Γ^* and Z are known.

$$m\ (\text{lb}_m/\text{sec}) = \frac{C\ a\ F \sqrt{g_c MW}}{\sqrt{Z_1 R}} \left(\frac{p_{1t}}{\sqrt{T_{1t}}}\right) \qquad \text{(II-III-34)}$$

$$= 0.1443\ \text{C a F} \sqrt{\frac{MW}{Z_1}} \left(\frac{p_{1t}}{\sqrt{T_{1t}}}\right)$$

where

$$F = \sqrt{\Gamma^* \left[1 + \frac{\Gamma}{2}\left(\frac{\Gamma - 1}{\Gamma}\right)\right]^{-[(\Gamma+1)/\Gamma-1)]}} \qquad \text{(II-III-35)}$$

The value of Z is fairly easily determined from tables for real gases, but the mean value of Γ between inlet stagnation and throat conditions is more difficult to determine. The value of Γ^* can be determined either from the acoustic velocity data

$$\Gamma^* = \frac{V_s^2}{Z\ R\ T\ g_c} \qquad \text{(II-III-36a)}$$

or from

$$\Gamma^* = Z(c_p/c_v) \qquad \text{(II-III-36b)}$$

when $p < 450$ psia and also from

$$\Gamma^* = \frac{Z\,\gamma}{1 - cp^2} \qquad \text{(II-III-37)}$$

where c is the coefficient in $Z = 1 + bp + cp^2$ and its value is about $10^{-6}/\text{atm}^2$.

Where the equation of state, or Z, is not well known or where accuracy requirements will permit, the value of Z used in equation (II-III-34) may be determined from "Reduced Coordinates Compressibility Charts" (Fig. II-III-31), which follow the tables. The reduced stagnation pressure is the ratio of the actual pressure to the critical point pressure, and the reduced stagnation temperature is the ratio of the actual temperature to the critical point temperature. Even though the curves in Fig. II-III-31 may not fit a particular gas exactly, they offer a significant improvement over using the ideal-gas equation. The critical properties of some commercial gases are given in Table II-I-5.

Often the deviation between Γ^* and Γ can be neglected. For example, a 7-per cent deviation between Γ^* and Γ would cause less than a 0.20-per cent error in the flow rate computed by equation (II-III-34). For those situations where the uncertainty due to using $\Gamma^* = \Gamma$ is acceptable, values of F_i (Table II-III-22) may be used. The most unfavorable approximation that might have to be accepted would result from computing the sonic flow of a gas using these values of F_i and the value of Z from the charts of Fig. I-III-31.

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Table II-III-22 Values of F_i for Selected Values of γ*

$$F_i = \sqrt{\gamma \left(\frac{1+\gamma}{2}\right)^{-\left(\frac{\gamma+1}{\gamma-1}\right)}}$$

γ	Increments of γ 0.00	0.02	0.04	0.06	0.08
1.1	0.6284	0.6325	0.6364	0.6404	0.6448
1.2	0.6483	0.6521	0.6562	0.6599	0.6638
1.3	0.6673	0.6705	0.6741	0.6776	0.6811
1.4	0.6848	0.6880	0.6914	0.6944	0.6978
1.5	0.7010	0.7039	0.7072	0.7128	0.7134
1.6	0.7165	0.7192	0.7222	0.7249	0.7281
1.7	0.7310	0.7337	0.7363	0.7390	0.7420

*These values may be used for real gases where $\Gamma^* = \Gamma = \gamma$.

Table II-III-23 Values of e_c

T_{1t} (°R)	Inlet Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	–0.0264	–0.0296	–0.0330	–0.0398	–0.0451	–0.0434	–0.0205
460	–0.0271	–0.0302	–0.0333	–0.0397	–0.0447	–0.0442	–0.0289
470	–0.0278	–0.0307	–0.0337	–0.0397	–0.0444	–0.0447	–0.0341
480	–0.0285	–0.0312	–0.0341	–0.0397	–0.0441	–0.0450	–0.0375
490	–0.0291	–0.0318	–0.0345	–0.0398	–0.0440	–0.0451	–0.0398
500	–0.0297	–0.0323	–0.0349	–0.0399	–0.0439	–0.0452	–0.0414
510	–0.0303	–0.0328	–0.0352	–0.0400	–0.0438	–0.0453	–0.0426
520	–0.0309	–0.0332	–0.0356	–0.0401	–0.0437	–0.0454	–0.0435
530	–0.0314	–0.0337	–0.0360	–0.0403	–0.0437	–0.0455	–0.0441
540	–0.0320	–0.0342	–0.0363	–0.0404	–0.0437	–0.0455	–0.0446
550	–0.0325	–0.0346	–0.0367	–0.0406	–0.0438	–0.0455	–0.0450
560	–0.0329	–0.0350	–0.0370	–0.0408	–0.0438	–0.0456	–0.0454
570	–0.0334	–0.0354	–0.0373	–0.0409	–0.0439	–0.0456	–0.0456
580	–0.0338	–0.0357	–0.0376	–0.0411	–0.0439	–0.0457	–0.0458
590	–0.0342	–0.0361	–0.0379	–0.0412	–0.0439	–0.0457	–0.0460
600	–0.0346	–0.0364	–0.0381	–0.0413	–0.0440	–0.0457	–0.0461
610	–0.0349	–0.0367	–0.0384	–0.0414	–0.0440	–0.0457	–0.0462
620	–0.0353	–0.0369	–0.0385	–0.0415	–0.0440	–0.0456	–0.0462
630	–0.0355	–0.0371	–0.0387	–0.0416	–0.0439	–0.0456	–0.0462
640	–0.0358	–0.0373	–0.0388	–0.0416	–0.0439	–0.0455	–0.0462
650	–0.0360	–0.0375	–0.0389	–0.0416	–0.0438	–0.0454	–0.0461
660	–0.0361	–0.0376	–0.0390	–0.0416	–0.0437	–0.0452	–0.0460
670	–0.0363	–0.0377	–0.0391	–0.0416	–0.0436	–0.0451	–0.0458
680	–0.0364	–0.0377	–0.0391	–0.0415	–0.0435	–0.0449	–0.0457
690	–0.0364	–0.0378	–0.0390	–0.0414	–0.0433	–0.0447	–0.0454
700	–0.0364	–0.0377	–0.0390	–0.0412	–0.0431	–0.0445	–0.0452

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Table II-III-24 Values of b_c

T_{1t} (°R)	Inlet Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.6719	0.6715	0.6713	0.6713	0.6723	0.6747	0.6791
460	0.6717	0.6714	0.6712	0.6713	0.6724	0.6748	0.6789
470	0.6714	0.6712	0.6711	0.6714	0.6725	0.6748	0.6786
480	0.6712	0.6710	0.6710	0.6713	0.6725	0.6747	0.6783
490	0.6709	0.6708	0.6708	0.6712	0.6724	0.6746	0.6780
500	0.6707	0.6706	0.6706	0.6711	0.6723	0.6745	0.6777
510	0.6704	0.6704	0.6704	0.6710	0.6722	0.6743	0.6774
520	0.6701	0.6701	0.6702	0.6708	0.6721	0.6741	0.6771
530	0.6698	0.6698	0.6699	0.6706	0.6719	0.6739	0.6768
540	0.6694	0.6695	0.6697	0.6704	0.6717	0.6736	0.6764
550	0.6691	0.6692	0.6694	0.6701	0.6714	0.6734	0.6760
560	0.6687	0.6689	0.6691	0.6699	0.6712	0.6731	0.6757
570	0.6684	0.6685	0.6687	0.6696	0.6709	0.6727	0.6753
580	0.6680	0.6681	0.6684	0.6692	0.6706	0.6724	0.6748
590	0.6676	0.6678	0.6680	0.6689	0.6702	0.6720	0.6744
600	0.6672	0.6674	0.6677	0.6686	0.6699	0.6717	0.6740
610	0.6668	0.6670	0.6673	0.6682	0.6695	0.6713	0.6735
620	0.6663	0.6666	0.6669	0.6678	0.6691	0.6709	0.6731
630	0.6659	0.6662	0.6665	0.6674	0.6687	0.6705	0.6726
640	0.6655	0.6657	0.6661	0.6670	0.6683	0.6700	0.6721
650	0.6650	0.6653	0.6657	0.6666	0.6679	0.6696	0.6717
660	0.6646	0.6649	0.6652	0.6662	0.6675	0.6691	0.6712
670	0.6641	0.6644	0.6648	0.6658	0.6671	0.6687	0.6707
680	0.6637	0.6640	0.6644	0.6653	0.6666	0.6682	0.6702
690	0.6632	0.6635	0.6639	0.6649	0.6662	0.6678	0.6697
700	0.6627	0.6631	0.6635	0.6645	0.6657	0.6673	0.6692

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Table II-III-25 Values of e_z

T_{1t} (°R)	Inlet Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	0.	−0.0252	−0.0530	−0.1179	−0.1988	−0.2991	−0.4162
460	0.	−0.0234	−0.0490	−0.1078	−0.1790	−0.2644	−0.3610
470	0.	−0.0218	−0.0454	−0.0989	−0.1622	−0.2360	−0.3175
480	0.	−0.0203	−0.0422	−0.0911	−0.1478	−0.2123	−0.2823
490	0.	−0.0190	−0.0393	−0.0842	−0.1353	−0.1923	−0.2532
500	0.	−0.0178	−0.0366	−0.0780	−0.1243	−0.1752	−0.2288
510	0.	−0.0166	−0.0342	−0.0724	−0.1147	−0.1604	−0.2079
520	0.	−0.0156	−0.0321	−0.0675	−0.1061	−0.1474	−0.1900
530	0.	−0.0147	−0.0301	−0.0630	−0.0984	−0.1360	−0.1743
540	0.	−0.0138	−0.0283	−0.0589	−0.0916	−0.1259	−0.1606
550	0.	−0.0130	−0.0266	−0.0551	−0.0854	−0.1169	−0.1485
560	0.	−0.0123	−0.0251	−0.0518	−0.0798	−0.1088	−0.1377
570	0.	−0.0116	−0.0236	−0.0487	−0.0748	−0.1015	−0.1281
580	0.	−0.0110	−0.0223	−0.0458	−0.0701	−0.0949	−0.1194
590	0.	−0.0104	−0.0211	−0.0432	−0.0659	−0.0889	−0.1116
600	0.	−0.0099	−0.0200	−0.0408	−0.0621	−0.0835	−0.1045
610	0.	−0.0094	−0.0190	−0.0385	−0.0585	−0.0785	−0.0980
620	0.	−0.0089	−0.0180	−0.0365	−0.0522	−0.0739	−0.0921
630	0.	−0.0085	−0.0171	−0.0346	−0.0522	−0.0697	−0.0867
640	0.	−0.0081	−0.0162	−0.0328	−0.0494	−0.0658	−0.0818
650	0.	−0.0077	−0.0154	−0.0311	−0.0468	−0.0623	−0.0772
660	0.	−0.0073	−0.0147	−0.0296	−0.0444	−0.0589	−0.0730
670	0.	−0.0070	−0.0140	−0.0281	−0.0422	−0.0559	−0.0691
680	0.	−0.0067	−0.0134	−0.0268	−0.0401	−0.0530	−0.0654
690	0.	−0.0064	−0.0128	−0.0255	−0.0381	−0.0503	−0.0621
700	0.	−0.0061	−0.0122	−0.0243	−0.0363	−0.0479	−0.0589

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Table II-III-26 Values of b_z

T_{1t} (°R)	Inlet Pressure, p_{1t} (psia) 0	100	200	400	600	800	1000
450	1.0000	0.9891	0.9780	0.9552	0.9315	0.9075	0.8837
460	1.0000	0.9899	0.9796	0.9585	0.9370	0.9152	0.8938
470	1.0000	0.9906	0.9810	0.9616	0.9419	0.9221	0.9028
480	1.0000	0.9912	0.9824	0.9644	0.9463	0.9283	0.9109
490	1.0000	0.9918	0.9836	0.9669	0.9503	0.9339	0.9181
500	1.0000	0.9924	0.9847	0.9693	0.9539	0.9389	0.9245
510	1.0000	0.9929	0.9857	0.9714	0.9573	0.9435	0.9304
520	1.0000	0.9933	0.9866	0.9734	0.9603	0.9477	0.9357
530	1.0000	0.9937	0.9875	0.9752	0.9631	0.9515	0.9406
540	1.0000	0.9941	0.9883	0.9768	0.9657	0.9550	0.9450
550	1.0000	0.9945	0.9891	0.9784	0.9681	0.9582	0.9490
560	1.0000	0.9949	0.9898	0.9798	0.9702	0.9612	0.9527
570	1.0000	0.9952	0.9904	0.9811	0.9723	0.9639	0.9562
580	1.0000	0.9955	0.9910	0.9824	0.9741	0.9664	0.9593
590	1.0000	0.9958	0.9916	0.9835	0.9759	0.9688	0.9622
600	1.0000	0.9960	0.9921	0.9846	0.9775	0.9709	0.9649
610	1.0000	0.9963	0.9926	0.9856	0.9790	0.9729	0.9674
620	1.0000	0.9965	0.9931	0.9865	0.9804	0.9748	0.9698
630	1.0000	0.9967	0.9935	0.9874	0.9817	0.9766	0.9719
640	1.0000	0.9969	0.9939	0.9882	0.9830	0.9782	0.9740
650	1.0000	0.9971	0.9943	0.9890	0.9841	0.9797	0.9758
660	1.0000	0.9973	0.9947	0.9897	0.9852	0.9812	0.9776
670	1.0000	0.9975	0.9950	0.9904	0.9862	0.9825	0.9793
680	1.0000	0.9976	0.9953	0.9911	0.9872	0.9838	0.9808
690	1.0000	0.9978	0.9956	0.9917	0.9881	0.9849	0.9823
700	1.0000	0.9979	0.9959	0.9922	0.9889	0.9861	0.9836

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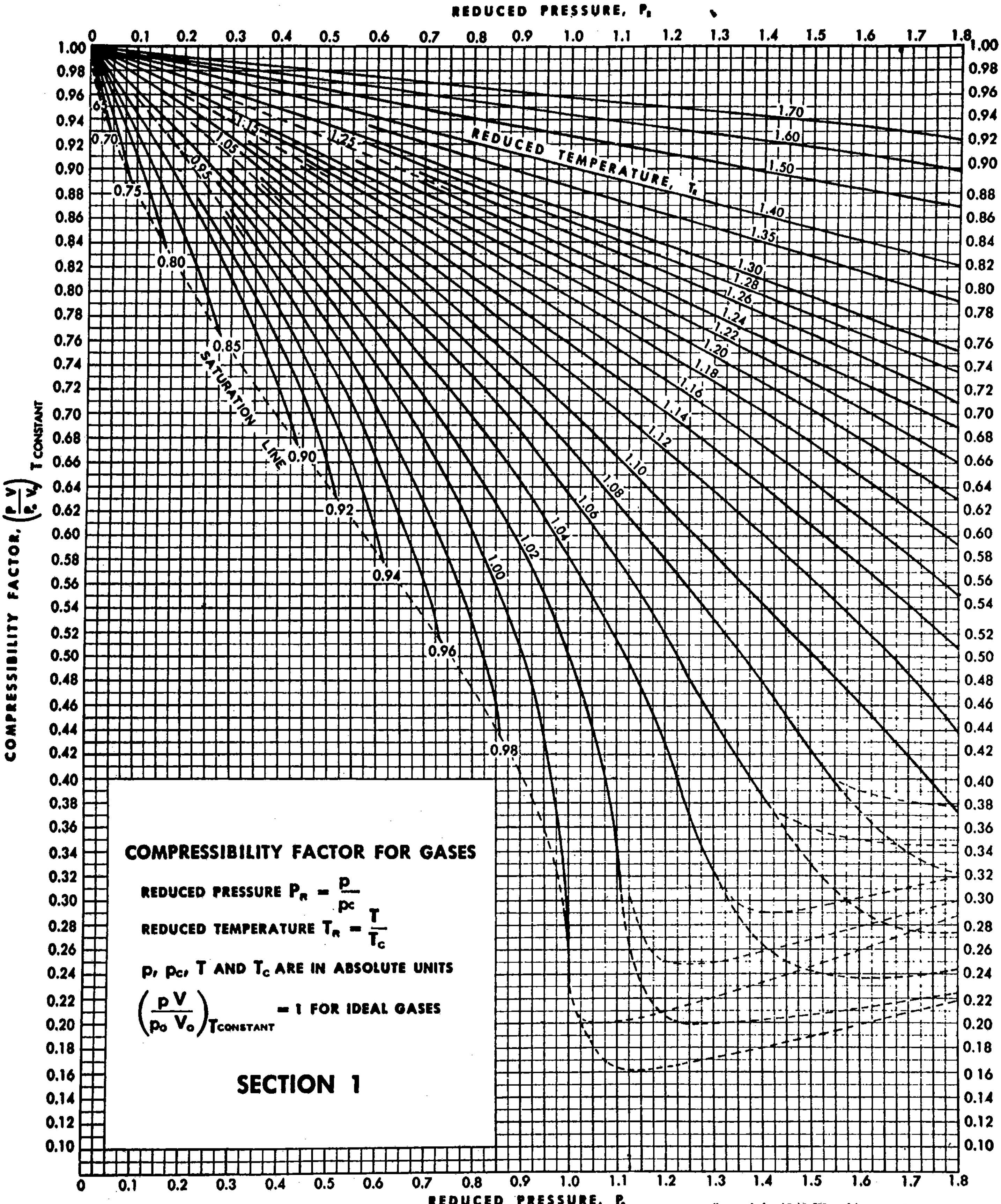

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FIG. II-III-31-1 VALUES OF COMPRESSIBILITY FACTOR, Z, AS FUNCTION OF REDUCED PRESSURE AND REDUCED TEMPERATURE

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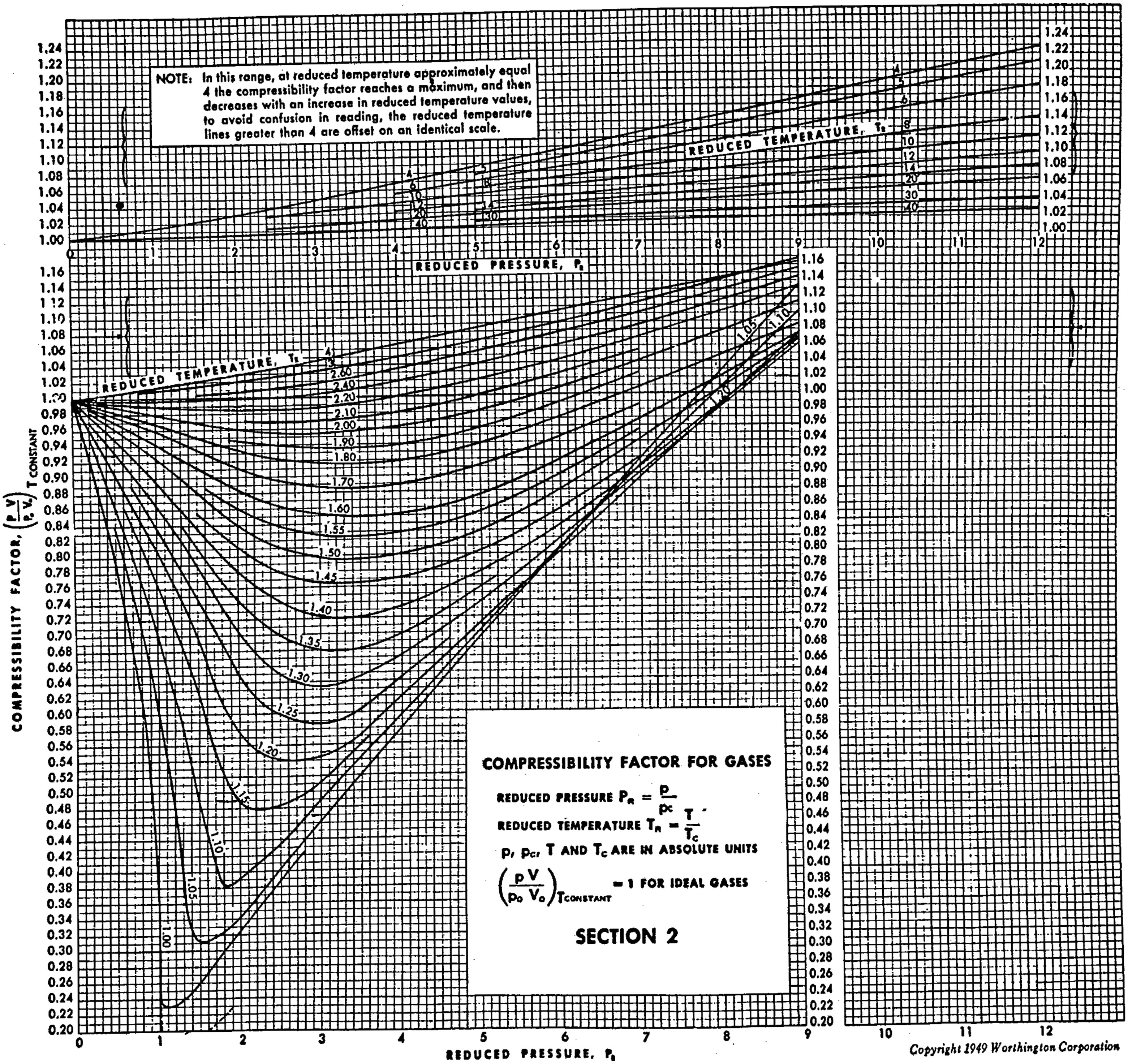

FIG. II-III-31-2 VALUES OF COMPRESSIBILITY FACTOR, Z, AS FUNCTION OF REDUCED PRESSURE AND REDUCED TEMPERATURE

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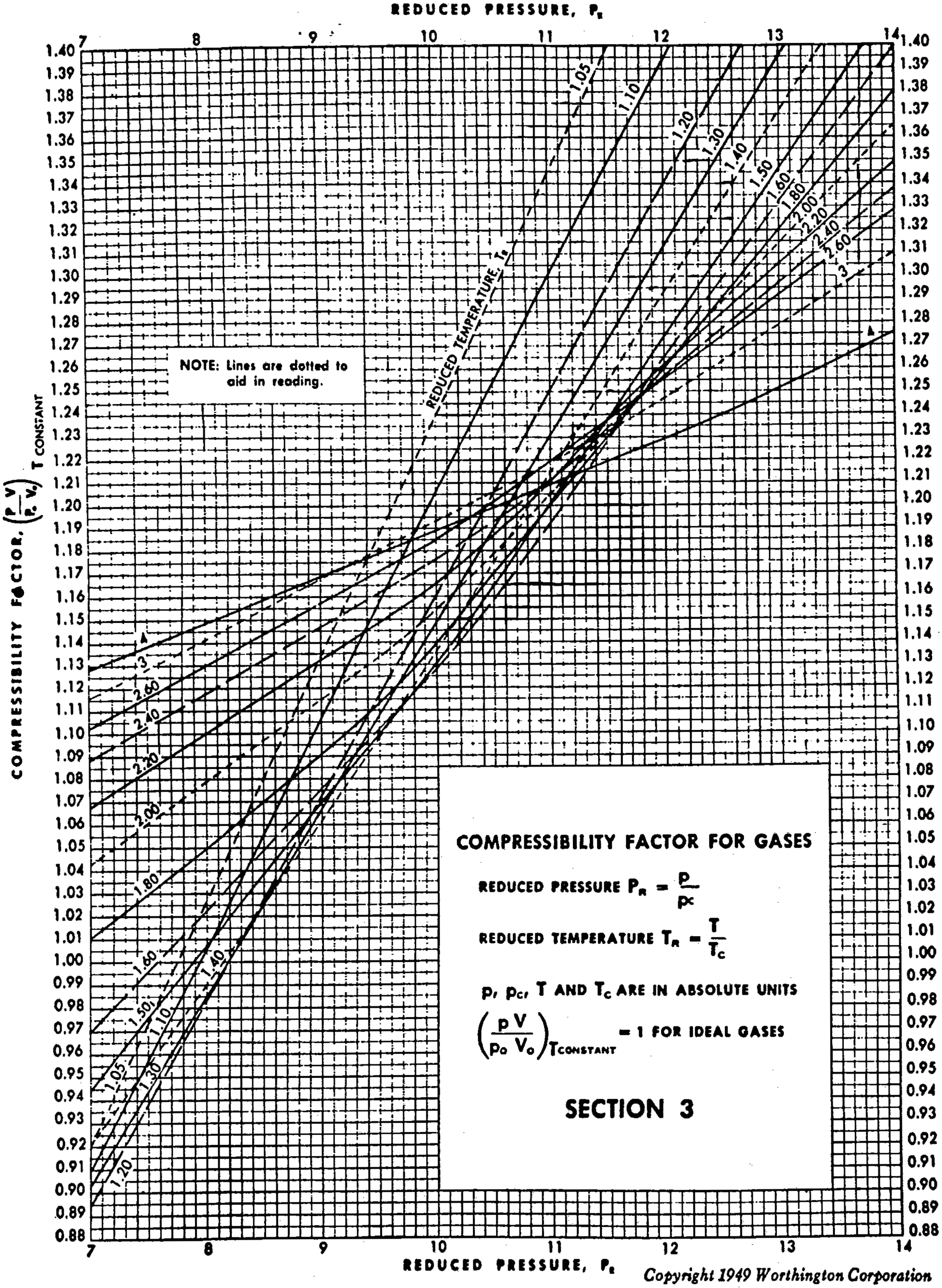

FIG. II-III-31-3 VALUES OF COMPRESSIBILITY FACTOR, Z, AS FUNCTION OF REDUCED PRESSURE AND REDUCED TEMPERATURE

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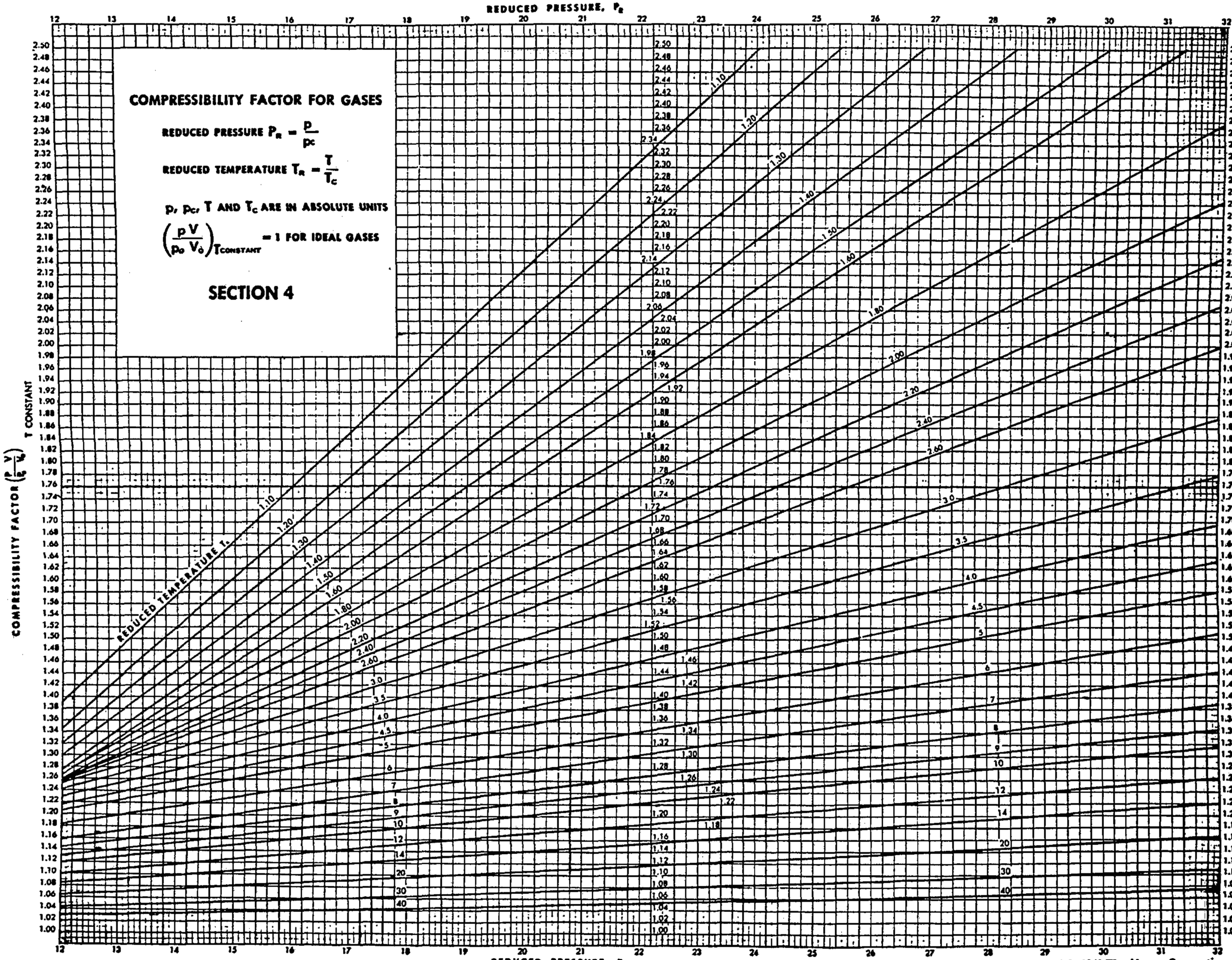

FIG. II-III-31-4 VALUES OF COMPRESSIBILITY FACTOR, Z, AS FUNCTION OF REDUCED PRESSURE AND REDUCED TEMPERATURE

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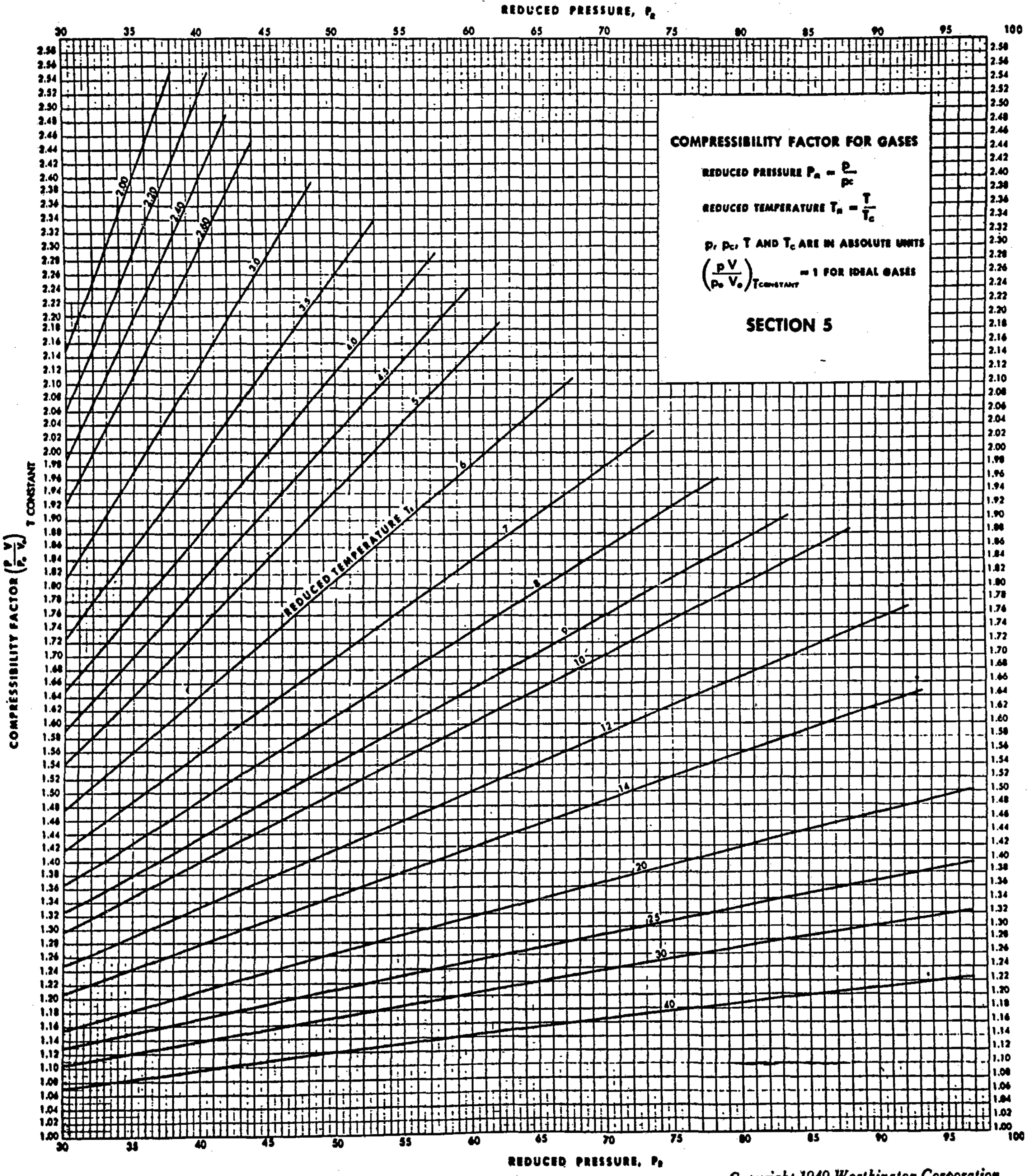

FIG. II-III-31-5 VALUES OF COMPRESSIBILITY FACTOR Z, AS FUNCTION OF REDUCED PRESSURE AND REDUCED TEMPERATURE

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II-III-58 Elbows. Ninety-degree elbows (Fig. II-III-32) may be used for monitoring the steadiness of a fluid flow. If such elbows are adequately calibrated, an accuracy within about ±0.5 per cent may be realized. Because the differential pressure obtained with an elbow is relatively low, their use is usually limited to measuring a liquid flow. If used for a gas flow, even the minimum stream velocity would be high (e.g., 300 to 500 fps).

II-III-59 The recommended locations of the pressure taps are in the outer and inner circumferences of the elbow midplane, 45 deg from the inlet end.

Although elbows may be located in either a horizontal or vertical pipeline, it is desirable that the velocity profile of the fluid stream entering the elbow be fairly uniform and free of swirls. For this reason, the same installation considerations should be followed as given by Fig. II-II-1 for orifices and flow nozzles of 0.80 diameter ratio.

II-III-60 The equation that may be used with elbow meters is

$$q \text{ (cfs)} = 0.37125\, KD^2 \sqrt{\frac{R}{D}\,\frac{\Delta p}{\rho}} \qquad \text{(II-III-38)}$$

or

$$m\ (\text{lb}_m/\text{sec}) = 0.37125\ KD^2 \sqrt{\frac{R}{D}\,\rho\Delta p} \qquad \text{(II-III-39)}$$

where

D = Diameter of pipe and elbow, in.

K = Flow coefficient, determined by calibration or by equation (II-III-40)

g_c = Proportionality factor between force and mass = 32.174

Δp = Differential pressure, psi

R = Radius of curvature of elbow center line, in.

ρ = Density of fluid in elbow, lb_m/ft^3

For uncalibrated, 90-deg elbows with pressure taps at the 45-deg section, tap-hole diameters, δ, as given in Table II-II-1, $R/D \geqslant 1.25$ (see Fig. II-III-32), and $10^4 \leqslant R_D \leqslant 10^6$, a value of the flow coefficient may be computed by

$$K = 1 - \frac{6.5}{\sqrt{R_D}} \qquad \text{(II-III-40)}$$

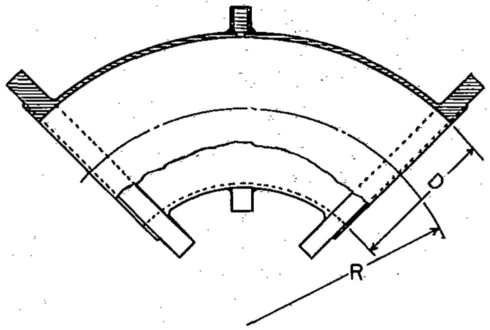

FIG. II-III-32 ELBOW METER

A flow evaluated with an elbow meter and K from equation (II-III-40) will be subject to a tolerance of about ± 4.0 per cent.

II-III-61 Electromagnetic Flowmeters. These meters are suitable for measuring the flow of liquids that have a conductivity greater than 20 micromhos (about 10 ppm of NaCl in water). Variations of the conductivity above this value do not affect the operation of the meter.

II-III-62 No special installation conditions are required inasmuch as the operation of the meter is unaffected by adjacent fittings. A helical flow pattern will have very little if not negligible effect upon the meter indications. Even pulsating flows up to about 10 or 12 cps can be measured with a suitably adjusted receiver. The important requirement is that the flow tube of the meter be completely filled with liquid all the time metering is in progress. Reverse flow can be metered by reversing the leads at the receiver, or a receiver can be adjusted to record flows in both directions.

The pressure loss through these meters is the same as for an equal length of pipe of the same diameter.

An accuracy of ± 1 per cent is to be expected over the normal 10 to 1 range of the unit. However, if these meters are specially adjusted, an accuracy of ± 0.5 per cent and possibly better can be attained over the range.

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Chapter II-IV

Examples

II-IV-1 As an aid to the use of the equations, tables and figures given in preceding chapters, the following example computations have been prepared. Also, as in the first example, reference and use are made of the computational procedures given in other publications [1, 2]. All of the examples apply to meters of the differential-pressure type.

II-IV-2 A contract for the sale of fuel gas calls for a maximum delivery rate of 14,000,000 scf*d* and a normal flow rate of 9,800,000 scfd. The orifice meter tube is to be a 6-in. schedule 40 pipe with flange pressure taps. The average flowing conditions are expected to be: pressure, measured at the upstream pressure tap, 250 psig; temperature, 81 F; specific gravity, 0.75; and average barometric pressure, 14.70 psia. The secondary element is to be a mercury-type differential recorder, with a range of 100-in. water and using a L-10 charter (i.e., a chart ruled on a square-root scale of 0–10). The reference or base conditions for the measurement are to be 14.70 psia, 60 F and dry.

Wanted: the diameter of the orifice required to provide a direct-reading flowmeter scale.

(a) The solution to this example can be worked readily by using procedures given in the *Flowmeter Computation Handbook* [1]. The equation and table used from that handbook are designated by an asterisk (*). Since only the differential-pressure pen will be read to infer the flow rate, the tacit assumption is that the conditions of measurement will be stable and/or any occasional small variations of pressure, temperature or composition can be neglected.

(b) From Chapter 5*, equation (5-4)*,

$$q_h \text{ (scfh)} = 0.6085 \frac{D^2 \; I}{\sqrt{G}} \; \frac{T_b}{P_b} \sqrt{\frac{h_w p_f}{T_f}}$$

where

I = principal meter constant
Subscript b refers to the reference condition
Subscript f refers to the flowing condition

(c) $q_h = \dfrac{14{,}000{,}000}{24} = 583{,}333$ scfh

$D = 6.065$ in.

$p_f = 250 + 14.70 = 264.7$ psia

$T_f = 459.7 + 81 = 540.7\ R$ and $T_b = 519.7\ R$

(d) $583{,}333 = I\left[0.6085 \dfrac{6.065^2}{\sqrt{0.75}} \; \dfrac{519.7}{14.70} \sqrt{\dfrac{100 \times 264.7}{540.7}}\right]$

$I = 91.2413$

(e) $\beta = 0.62185$ (from Table 1-5*)

(f) $d = 6.065 \times 0.62185 = 3.772$ in. as the (trial) diameter of the orifice

(g) A more exact determination of the required orifice diameter, if desired, may be made by applying a Reynolds number factor, F_r, an expansion factor, Y_1, and an area thermal expansion factor, F_a, as multipliers to equation (5-4)*. (These factors are explained in Appendix C*.) For doing this, the maximum flow rate will be used as above.

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(h) $R_D = 0.00424\, w_h/D\,\mu$ (equation A-34*)

(i) w_h (= m_h lb$_m$/hr) = 583,333 (0.75 × 0.0764) = 33,430 lb$_m$/hr

(j) μ = 0.00 000 77 lb$_m$/ft-sec (Fig. II-I-8)

(k) R_D = (0.00424 × 33,430)/(6.065 × 0.00 000 77) = 3,035,000

(l) Using the value of β from (e) above, the Reynolds number adjustment factor, $F_r = 0.997$ (Fig. C-1–4*).

(m) For Y_1, $\dfrac{h_w}{p_f} = \dfrac{100}{264.7} = 0.378 \quad Y_1 = 0.9953$ (Fig. C-3-1*)

(n) F_a for 81 F = 1.0002 (Fig. II-I-3 or Fig. C-2-1*)

(o) Adjusted value, $I' = I/\ F_r \times Y_1 \times F_a$

$$= 91.2413/(0.997 \times 0.9953 \times 1.0002)$$

$$= 92.15$$

(p) $\beta' = 0.62405$ (Table 1-5*)

(q) $d' = 6.065 \times 0.62405 = 3.785$ in.

(r) If the normal rate of flow were used instead of the maximum, the value of Y_1 would become 0.9977; the effect on F_r would be too small to read; and no change would occur in F_a, giving an adjusted value of $I'' = 91.70$, for which β'' = 0.62315 and d'' = 3.779 in.

(s) The example may be solved by using the computational procedures given in Gas Measurement Committee *Report No. 3* [2]. The equations, tables and figures from *Report No. 3* will be indicated by a dagger (†). The basic equation is equation (1)† from that report:

$$q_h\ (\text{scfh}) = C' \sqrt{h_w\, p_1}$$

(t) $C' = F_b \cdot F_r \cdot Y \cdot F_{pb} \cdot F_{tb} \cdot F_{tf} \cdot F_g \cdot F_{pv} \cdot F_m \cdot F_a$ (equation (2)†)

(u) F_b = basic orifice factor. It includes the orifice diameter which is sought.

(v) F_r = Reynolds number factor = $1 + b/\sqrt{h_w\, p_1}$ (Table 5†).

The factor b depends on the orifice and pipe diameters; therefore, for this example, it is necessary to *assume* a value of d for the first determination; then, if necessary, a closer value will be used for a second detemination. Assuming $d = 3.875$ in., then for 6-in. pipe $b = 0.0505$ (Table 5†) and

$$F_r = 1 + \frac{0.0505}{\sqrt{100 \times 264.7}} = 1.0003$$

(w) Y_1 = expansion factor = 0.9953 as in (m) above

(x) F_{pb} = pressure base factor = $\dfrac{14.73}{14.70}$ = 1.0020 (Table 12†)

(y) F_{tb} = temperature base factor = 1.000 (Table 13†)

(z) F_{tf} = flowing temperature factor = 0.9804 (Table 14†)

(aa) F_g = specific gravity factor = 1.1547 (Table 15†)

(bb) F_{pv} = supercompressibility factor = 1.0175 (Table 16†)

(cc) F_m = manometer factor (mercury type) = 0.9983 (Table 17†)

(dd) F_a = orifice thermal expansion factor = 1.0002 as in (n) above

(ee) $C' = F_b \times 1.0003 \times 0.9953 \times 1.0020 \times 1.000 \times 0.9804 \times 1.1547 \times 1.0175 \times 0.9983 \times 1.0002 = 1.1474\, F_b$

(ff) $583{,}333 = 1.1474\, F_b \sqrt{100 \times 264.7}$

F_b = 3124.9 for a 6-in. schedule 40 pipe. (This corresponds to an orifice diameter.) d = 3.751 in.

II-IV-3 Compressed air is flowing in a 10-in. pipe under a line pressure of 125 psig, a temperature of 90 F and water vapor saturation; it is metered with a type 316, stainless-steel, concentric, thin-plate, square-edged orifice. The orifice diameter is 6.250 in.; the i.d. of the meter-tube inlet section is 10.02 in., with pressure taps at D and $1/2\ D$. The differential pressure is 30-in. water, and the barometric pressure is 14.7 psia.

Required: the rate of flow in cu ft per hr at the reference conditions of 30-in. Hg abs., 60 F and water vapor saturation.

(a) β = 6.25/10.02 = 0.62375

(b) E = 1.0860 (Table II-I-1 or Fig. II-I-4)

(c) F_a = 1.0005 (Fig. II-I-2)

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(d) $p_1 = 125 + 14.7 = 139.7$ psia

(e) Since the temperature of the manometer or gage by which the differential pressure is measured is not given, it will be assumed to have been 68 F. Then $x = (30 \times 0.03606)/139.7 = 0.00775$ and $x/\gamma = 0.00775/1.4 = 0.00553$, with which $Y_1 = 0.997$ (Fig. II-I-5).

(f) At 140 psia and 90 F, $Z_1 = 0.998$ (Fig. II-I-11).

(g) The saturation pressure of water vapor at 90 F is 0.698 psia, and the specific humidity is

$$S = \frac{0.622}{1} \times \frac{0.698}{(139.7 - 0.698)} = 0.00312 \quad \text{(I-3-38)}$$

(h) $$\rho_1 = 2.6991\ (1.0 + 0.00312)\ \frac{(139.7 - 0.7)}{549.7 \times 0.998} \times 1$$

$$= 0.6860\ \text{lb}_m/\text{ft}^3 \quad \text{(I-3-40)}$$

(i) $$m = 0.99702 \times 1.086 \times 1.0005 \times 0.997 \times 6.250^2$$

$$C\ \sqrt{30 \times 0.6860} = 19.14\ C\ \text{lb}_m/\text{sec}$$

(II-III-16)

(j) At 90 F, the viscosity of air is $\mu = 0.0000127$ (Fig. II-I-8).

(k) For a value of R_d with which to locate the correct tabulated value of C, a trial value of $C = 0.62$ is assumed; then,

$$m = 11.87 \text{ and } R_d = \frac{48 \times 11.87}{\pi \times 6.250 \times 0.0000127}$$

$$= 2{,}283{,}000$$

(l) $C = 0.6070$ (equations (II-III-3 and -4) or Table II-III-3)

(m) $$q_1 = \frac{19.14 \times 0.6070}{0.6858} \times 3600 = 60{,}990 \text{ cfh at}$$

the flowing conditions

(n) If the total pressure of this air-water vapor mixture is reduced from 139.7 psia to 30 in. Hg = 14.735 psia, the partial pressure of the water vapor portion will be reduced proportionally, i.e., from 0.698 psia to 0.0736 psia. Now the saturation pressure of water vapor at 60 F is 0.256 psia, so that at 30-in. Hg and 60 F the mixture will not be water-vapor saturated. However, the volume rate of flow of this mixture at 30-in. Hg and 60 F will be

$$q_o = 60{,}990 \left(\frac{139.7 - 07}{14.735 - 0.0736}\right)\left(\frac{519.7}{549.7}\right)\left(\frac{1.000}{0.998}\right)$$

$$= 547{,}800 \text{ cfh}$$

(o) On the hypothesis that there would be a condition of water vapor saturation at the reference state given as 30 in. Hg, 60 F and water vapor saturation, the equivalent rate of flow would be

$$q_o' = 60{,}990 \left(\frac{139.7 - 0.7}{14.735 - 0.256}\right)\left(\frac{519.7}{549.7}\right)\left(\frac{1.000}{0.998}\right)$$

$$= 554{,}600 \text{ cfh}$$

Note: If the presence of the water vapor is neglected at both the initial and final conditions, the apparent rate of flow would be $q_o'' = 547{,}750$ cfh.

II-IV-4 Fuel oil flowing in a 6-in. pipe is measured with an orifice meter under the following conditions:

1. I.D. of pipe at orifice meter section = 6.065 in.
2. Orifice plate 304 stainless steel, orifice bore = 3.750 in.
3. Differential pressure between vena contracta taps 137-in. water
4. Temperature of the flowing oil = 180 F
5. Oil data, by supplier, viscosity = 150 SSU at 180 F and specific gravity = 0.939 at 180/60, and 0.980 at 60/60

Required: rate of flow in gpm at 60 F, the measurement reference temperature.

(a) $m = 0.099702\ (C\ E\ Y\ F_a d^2)\ \sqrt{\rho_1 h_w}$ (equation II-III-15)

(b) $\beta = 3.570/6.065 = 0.5886$

(c) $E = 1.066$ (Table II-I-1 or Fig. II-I-4)

(d) $Y = 1.000$ for a liquid

(e) $F_a = 1.002$ (Fig. II-I-3)

(f) $\rho_1 = 0.939 \times 62.3707 = 58.566\ \text{lb}_m/\text{ft}^3$

(g) $\rho_o = 0.980 \times 62.3707 = 61.123\ \text{lb}_m/\text{ft}^3$

(h) $$m = 0.099702 \times 1.066 \times 1.000 \times 3.570^2 \times 1.002\ C\ \sqrt{58.566 \times 137}$$

$$= 121.58\ C\ \text{lb}_m/\text{sec}$$

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(i) To compute the Reynolds number with which to determine the value of C, a preliminary value of m may be computed by using an estimated value of C such as $C = 0.61$, thus giving $m \cong 132.98 \times 0.61 \cong 81.12$.

(j) For 150 SSU, $\nu = 0.000345$ ft²/sec (equation (II-I-6) or Fig. II-I-10 and equation (II-I-3)).

(k) $\mu = 0.000345 \times 58.566 = 0.0202$ lb_m/sec-ft

(l) $R_d = (48 \times 81.12)/(\pi \times 3.57 \times 0.0202) = 17{,}180$

(m) $C = 0.6237$ (equations (II-III-5 and 6) or Table II-III-4)

(n) $m = 132.98 \times 0.6237 = 82.94$ lb_m/sec

(o) gpm $= [m \times 60 \times (7.48052 \text{ gal/ft}^3)]/\rho_o$

$$q_o = (82.94 \times 60 \times 7.48052)/61.123$$

$$= 609.03 \text{ gpm at 60 F}$$

II-IV-5 Calculate the maximum designed capacity, in pounds per hour, of a transmitter of 212-in. water maximum differential pressure range, wet calibrated, used with a 5 per cent chrome-moly steel long-radius, high-ratio flow nozzle of 5.674-in. throat diameter, installed in a pipe of 7.683-in. i.d. and fitted with pipe-wall taps at D and $1/2\ D$, for measuring steam flow at 900 psig and 900 F.

(a) $m(lb_m/hr) = 1865.6\ C\ E\ Y_a d^2 F_a \sqrt{\rho_1\ \Delta p}$ (equation II-III-17)

(b) $\beta = 5.674/7.683 = 0.73853$

(c) $E = 1.1931$ (Table II-I-1)

(d) $F_a = 1.0118$ (Fig. II-I-3)

(e) Assume the atmospheric pressure is 14.6 psia; then $p_1 = 914.6$ psia; and at 900 F the specific volume of steam is 0.8362 ft³/lb_m (*1967, ASME Steam Tables*, Table 3), and $\rho_1 = 1/0.8362 = 1.196$ lb_m/ft³.

(f) Assume the transmitter temperature is 70 F; then

$\Delta p = 212 \times 0.03605 = 7.643$ psi (Fig. II-I-2)

(g) At the maximum range of the transmitter, $x = 7.643/914.6 = 0.00836$, and $Y_a = 0.993$ (Fig. II-III-21).

(h) For the purpose of evaluating the designed capacity, it will suffice to assume that $C = 0.99$. Then,

(i) $m_{max} = 1865.6 \times 0.99 \times 1.1931 \times 0.993 \times 5.674^2 \times 1.0118 \times \sqrt{1.196 \times 7.643}$

$= 215{,}500$ lb_m/hr, which is approximately 60 lb_m/sec

(j) Since in use the flow rate may never be at the maximum meter capacity, it is realistic to assume that under-average flow conditions the differential pressure will be about 119-in. water (= 4.29 psi), corresponding to approximately 75 per cent of maximum flow rate. Also, an approximate value of R_d is sufficient for the purpose of establishing the value of C.

(k) Thus, using $m = 45$ lb_m/sec and μ

$= 0.0000192$ (Fig. II-I-7)

$R_d = (48 \times 45)/(\pi \times 5.674 \times 0.0000192)$

$= 6{,}310{,}000$

(l) Although this value of R_d is above the range of values on which equation (II-III-9) and Table II-III-5 were based, an extrapolation by means of equation (II-III-9) may be made along with the application of a larger tolerance. On this basis, $C = 0.993 \pm 2.0$ per cent.

(m) For $\Delta p = 4.29$ psi, $x = 0.00469$ and $Y_a = 0.997$.

(n) $m = 1865.6 \times 0.993 \times 1.1931 \times 0.997 \times 5.674^2 \times 1.0118 \times \sqrt{4.29 \times 1.196}$

$= 162{,}595$ lb_m/hr assumed normal flow rate

(o) Using $C = 0.993$ in step *(i)* in place of 0.99 gives as the maximum capacity of the transmitter $m_{max} = 216{,}150$ lb_m/hr.

II-IV-6 Condensate from a steam turbine is metered with a long-radius, low-ratio flow nozzle of type-304 stainless steel. Pressure taps are located at 1 D upstream and in the nozzle throat. The nozzle had been calibrated, with the calibration going up to a maximum R_d of 4,500,000, at which the value of C was 0.9955. The primary element dimensions and the metering data are:

meter section pipe i.d.	= 12.090 in.
nozzle throat diameter	= 5.000 in.
line pressure at upstream tap	= 250 psig
flowing temperature at nozzle	= 300 F
differential pressure by mercury manometer	= 40.5 in.

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temperature of manometer 90 F

barometric pressure, in. mercury at 32 F 29.40

Required: the rate of flow in pounds per hour.

(a) β = 5.000/12.090 = 0.41356

(b) E = 1.01495 (Table II-I-1)

(c) Y = 1.000 for liquids

(d) F_a = 1.0042 (Fig. II-I-3)

(e) Barometric pressure = 29.40 × 0.4912 = 14.44 psia (Fig. II-I-2), p_1 = 250 + 14.44 = 264.44 psia

(f) v_1 = 0.01744 ft³/lb$_m$, (*1967 ASME Steam Tables,* Table 3), and

ρ_1 = 1/0.01744 = 57.339 lb$_m$/ft³

(g) Δp = 40.5 x 0.45235 = 18.32 psi (Fig. II-I-2)

(h) m = 0.525 C × 1.015 × 5.00² × 1.0042 × $\sqrt{18.32 \times 57.339}$ = 433.59 C lb$_m$/sec

(i) If the value of C by the calibration applies approximately at the actual flowing conditions, then m = 433.59 × 0.9955 = 431.64

(j) μ = 0.000125 (Fig. II-I-5)

(k) R_d = (48 × 431.64)/(π × 5.000 × 0.000125) = 10,550,000

(l) Extrapolation of the calibration curve parallel to the typical curve for a nozzle with throat taps (Fig. II-III-19) shows C = 0.9969 at R_d = 10,550,000.

(m) Pounds per hour = 3600 × 433.59 × 0.9969 = 1,556,100.

II-IV-7 To calculate the rate of flow in pounds per hour of water at 60 F and 95 psig, flowing through a 6.00 × 4.00-in. cast iron Venturi tube which will produce a differential pressure of 100-in. water.

(a) m (lb$_m$/hr) = 358.93 $C\ E\ Y\ d^2\ F_a\ \sqrt{\rho_1\ h_w}$ (equation (II-III-17)

(b) β = 4.00/6.0 = 0.6667

(c) C = 0.984 (Par. II-III-34)

(d) E = 1.1163 (Fig. II-I-4)

(e) Y = 1.000 for liquids

(f) F_a = 1.000 (room temperature)

(g) ρ = 62.3707 (Table II-I-4)

(h) m = 358.93 × 0.984 × 1.1163 × 4.00² × $\sqrt{62.3707 \times 100}$ = 498,190 pounds per hr

(i) μ = 0.00076 (Fig. (II-I-5)

(j) $R_D = (48\ \frac{498190}{3600}) / (\pi \times 6.0 \times 0.00076)$ = 463,500. Since both R_D and β are within the limits given in Par. II-III-34, the coefficient value of 0.984 is valid.

II-IV-8 The flow of superheated steam through a 3/4-in. bleed line is both controlled and metered with a sonic-flow radial inlet Venturi. The i.d. of the pipe upstream of the Venturi inlet is 0.742 in., and the throat diameter of the Venturi is 0.2569 in. The pressure is measured from a sidewall tap 1 D upstream from the Venturi inlet, and the temperature is measured with a total temperature well and thermometer located downstream. The inlet line pressure is 1200 psig, and the temperature is 950 F. From a calibration the Venturi coefficient is 0.994.

Required: the maximum rate of flow in lb$_m$/sec.

(a) $m = C^*\ (a/12)\ B_F\ (F/F_i)\ \sqrt{p_{1t}/v_{1t}}$ (equation (II-III-29))

(b) a = 0.05183 in.²; $a/12$ = 0.004319

(c) β = 0.2569/0.742 = 0.346

(d) Since β is less than 0.5, equation (II-III-27) is applicable for calculation of p_{1t}

$$p_{1t} = \frac{p_1}{\left[1-\beta^4 \frac{\Gamma}{2}\left(\frac{2}{\Gamma+1}\right)^{(\Gamma+1)/(\Gamma-1)}\right]}$$

(e) p_1 = 1200 + 15 = 1215 psia, assuming 15 psia as the barometric pressure.

(f) Γ = 1.285 (*1967 ASME Steam Tables,* Fig. 11, p. 298).

(g)
$$p_{1t} = \frac{1215}{1-(0.346^4)\left(\frac{1.285}{2}\right)\left(\frac{2}{1+1.285}\right)^{\frac{2.285}{0.285}}}$$

$$= \frac{1215}{1-0.01433 \times 0.6425 \times 0.3434}$$

= 1220 psia

(h) B_F = 3.7848 (Table (II-III-11)

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(i) $F/F_i = 0.9945$ Table (II-III-11)

(j) $v_{1t} = 0.644$ *(1967 ASME Steam Tables,* Table 3)

(k) $m = 0.994 \times 0.004319 \times 3.7848 \times 0.9945$

$\times \sqrt{1220/0.644}$

$= 0.7033\ lb_m/sec$ or $2531.96\ lb_m/hr$

II-IV-9 Propane (C_3H_8) is measured through a long-radius, low-ratio flow nozzle at sonic-flow conditions. The nozzle is mounted in a pipe of 2.90-in. i.d. and has a throat diameter of 1.1574 in. The total inlet pressure, $p_{1t} = 800$ psia, is measured with an impact tube located 200 stem diameters upstream of the nozzle inlet. The temperature, T_{1t}, measured with a total temperature well located downstream of the nozzle, is 340.3 F (= 800 R). The nozzle has not been calibrated.
Required: rate of discharge in lb_m/sec.

(a) Since the factors for propane are not tabulated, the sonic-flow rate must be computed using the "Reduced Coordinates Compressibility Charts" (Fig. II-III-29), an F_i factor from Table II-III-22 in conjunction with equation (II-III-34) and

(b) $m\ (lb_m/sec) = 0.1443\ C\ a\ F_i\ p_{1t} \sqrt{\dfrac{MW}{Z\ T_{1t}}}$
(equation (II-III-34))

(c) For propane, C_3H_8, from Table II-I-5,

molecular weight, MW = 44.0972

critical pressure = 617.4 psia

critical temperature = 666 R

(d) Using $\Gamma^* = \Gamma = \gamma = 1.33$, $F_i = 0.6723$ (Table II-III-22).

(e) Reduced pressure = 800/617.4 = 1.295 (use 1.3-)

Reduced temperature = 800/666 = 1.2

$Z = 0.748$ (Fig. II-III-29-1)

(f) $a = \dfrac{\pi \times 1.1574^2}{4} = 1.052\ in.^2$

(g) Nozzle coefficient is assumed to be 0.99 ± 1.0 per cent.

(h) $m = 0.1443 \times 0.99 \times 1.052 \times 0.6723 \times 800$

$\sqrt{44.0972/(0.748 \times 800)}$

$= 21.942\ lb_m/sec$

II-IV-10 A natural fuel gas is discharged from a sonic-flow nozzle connected to the outlet of a displacement meter. The composition of the gas as given in mole fractions is:

methane	CH_4	0.960
ethane	C_2H_6	0.035
carbon dioxide	CO_2	0.002
nitrogen	N_2	0.003

The conditions under which the nozzle is operated are:

stagnation temperature = 80.3 F (=540 R)

stagnation pressure = 385.6 psig

barometric pressure = 29.3-in. Hg.

From a calibration with air, the product, Ca, the effective area of the sonic-flow nozzle, was reported to be 0.1930 in.^{2}.

Required: the time in seconds for 1 ft^3 at the displacement meter outlet conditions to be discharged from the displacement meter through the sonic-flow nozzle.

(a) The outlet conditions of the displacement meter are taken to be the same as the inlet conditions to the sonic-flow nozzle.

(b) $q_1 = C\ a\ (e_c\, j + b_c)\ (e_z\, j + b_z) \sqrt{g_c\ T_{1t}\ \dfrac{R}{MW}}$
(equation (II-III-33))

(c) The molecular weight of the gas is (using Table II-III-27)

$(0.960 \times 16.043) + (0.035 \times 30.0701)$

$+ (0.002 \times 44.01) + (0.003 \times 28.013)$

$= 16.6258$

(d) Barometric pressure $= 29.3 \times 0.4912 = 14.4$ psia (Fig. II-III-2)

(e) $p_{1t} = 385.6 + 14.4 = 400.0$ psia

(f) $j = 0.035 + 0.002 - ½(0.003) = 0.0355$ (equation (II-III-27))

(g) $e_c = -0.0404$; $b_c = 0.6704$; $e_z = -0.0589$;

$b_z = 0.9768$ (Tables II-III-23 through II-III-26)

(h) $(e_c\, j + b_c) = (-0.0404 \times 0.0355 + 0.6704)$

$= 0.6690$

$(e_z\, j + b_z) = (-0.0589 \times 0.0355 \times 0.9768)$

$= 0.9747$

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(i) $q_1 = \frac{0.1930}{144} \times 0.6690 \times 0.9747$

$$\sqrt{32.174 \times 540 \times \frac{1545.32}{16.6258}} = 1.1106 \text{ cfs}$$

(j) Time to discharge 1 $ft^3 = 1/1.1106$

$= 0.9004$ sec/ft^3

II-IV-11 The flow of a fuel oil in a 3-in. pipe is monitored at a long-radius welded elbow, which has been fitted with pressure taps in the 45-deg plane. The pipe is schedule 40. The flowing temperature of the oil is 110 F, and the specific gravity is 0.79 at 110/60. The average differential pressure is 16 in. of water. The elbow was not calibrated.

Required: the approximate rate of flow in barrels per hour at 60 f.

(a) $q \text{ (cfs)} = \frac{\pi D^2}{4} K \sqrt{g_c \frac{4R}{\pi D} \frac{\Delta p}{\rho}}$
(equation (II-III-38))

(b) $D = 3.068$ in.

$R = 4.5$ in.

$p = 16 \times 0.03605 = 0.577$ psi (Fig. II-I-2)

$\rho = 0.79 \times 62.3707 = 49.27\ lb_m/ft^3$
(Table II-I-4)

(c) Since the elbow was not calibrated, the flow coefficient, K, is to be evaluated by

$$K = 1 - \frac{6.5}{\sqrt{R_D}} \qquad \text{(II-III-36)}$$

This requires assuming a value for R_D and, after obtaining a first value of K and completing a computation of q, computing a value of R_D to be compared with the assumed value. Assuming $R_D = 50{,}000$,

$$K = 1 - \frac{6.5}{\sqrt{50{,}000}} = 0.970$$

(d) $q \text{ (cfs)} = \frac{\pi\, 3.068^2}{4} \times 0.970$

$$\sqrt{32.174 \frac{4 \times 3.068}{\pi \times 4.5} \times \frac{0.577}{49.27}}$$

$= 4.101$ cfs

(e) $R_D = 48m/(\pi D \mu)$. Since no information is given on the viscosity of the oil, a value of 0.009 lb_m/ft-sec will be assumed; then

$$R_D = \frac{48\ (4.101 \times 49.27)}{\pi \times 3.068 \times 0.009} = 111{,}800$$

(f) $K' = 1 - \frac{6.5}{\sqrt{111{,}800}} = 0.981$

(g) $q' = 4.101 \frac{0.981}{0.970} = 4.147$ cfs at 110 F

(h) The observed specific gravity of 0.79 at 110/60 corresponds to a specific gravity of 0.8094 at 60/60 (Table 23 of Ref. [3]).

(i) The volume reduction factor to convert a volume at 110 F to the corresponding volume at 60 F for a specific gravity of 0.8094 is 0.9753 (Table 24 of Ref. [3]).

$4.147 \times 0.9753 = 4.043$ cfs at 60 F

(j) cu ft $\times$ 0.17811 = U.S. barrels (Table 1 of Ref [3]).

$4.043 \times 0.17811 \times 3600 = 2592.4$ bbl/hr at 60 F

(k) Since the value of K is subject to a tolerance (uncertainty) of $\pm$ 4.0 per cent, the rate of flow is subject to a tolerance of $\pm$ 4.0 per cent or more, and 2592 $\pm$ 4.0 per cent bbl/hr would be the reported rate of flow.

References

[1] "Flowmeter Computation Handbook," ASME, New York, 1961.

[2] "Orifice Metering of Natural Gas," Gas Measurement Committee Report No. 3, American Gas Association, New York, 1969.

[3] "ASTM-IP Petroleum Measurement Tables;" ASTM, Philadelphia, Pa., 1952.

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Chapter II-V

Tolerances

II-V-1 Tolerances, Their Significance. Except by accident, no two meters, even of the same type, are likely to give *exactly* the same indication when the same quantity of fluid is flowing through each. The degree to which this applies is not the same for all types of meters, applying least to the displacement types and more to the differential-pressure types. For this reason, "tolerances" are assigned to the values of the factors entering into the metering of fluids. (The expressions, "limit of accuracy" or "per cent uncertainty," might well be substituted for "tolerance.") Tolerances have to do with those practically unavoidable differences between ostensibly duplicate primary elements. They do *not* refer to accidental errors of observation, concerning which no general predictions are possible.

In any one measurement, the probability is very small that the departures from 100 per cent accuracy in the individual items will all affect the final result in the same direction; hence, from mathematics, the overall tolerance will be the square root of the sum of the squares of the tolerances on (departures of) the individual factors. In other words, an overall tolerance determined in this way is the most probable amount of departure from the actual quantity, with there being as much chance that the departure will be smaller than larger than this amount.

II-V-2 There have been a number of procedures used for evaluating or assigning tolerances with the result that the per cent uncertainty assigned to an item by one worker has not been exactly comparable to that assigned by another to the same item. In order to provide a uniform basis for assigning numerical values to tolerances, the committee on Fluid Flow Measurement of the International Organization for Standardization (ISO/TC-30) has adopted the following procedure:

1. The numerical value of a tolerance shall be twice the standard deviation.
2. The standard deviation is to be computed as follows: Sum up the squares of the deviations with respect to *the most probable value;* divide by the number of observations minus one; take the square root of this quotient.

This procedure has been followed in evaluating the tolerances given in this edition of *Fluid Meters.* The *most probable values* of the discharge coefficients of square-edged orifices are, to date, the values computed by equations (II-III-1) through (II-III-6), or read from Tables II-III-2, II-III-3 and II-III-4. Similarly, for flow nozzles used with pipe-wall taps, the most probable values are those computed by equation (II-III-12) or read from Table II-III-5. For low-ratio nozzles with the downstream tap in the throat, the most probable values are those read from the curve of Fig. II-III-19. For Venturi tubes, the most probable values are given in Pars. I-5-35 and II-III-42.

The tolerance values given in Tables II-V-1 and II-V-2 are those recommended as applying to uncalibrated primary elements. When a primary element is calibrated, the tolerance to be used should be computed from the calibration data by the procedure described above.

II-V-3 Prior to the editing of the fifth edition of *Fluid Meters,* tolerance values given by this committee and also by the Gas Measurement Committee of the American Gas Association in their Report

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No. 3 were not derived by an evaluation of the standard deviation. Instead, the arithmetic average of the departures of the test values from the correlation curves was computed, and this value, without being doubled, was reported as the tolerance for the particular item. It is of interest that those arithmetic average values are very close to the values of σ obtained in the recent correlation, which is the basis for some of the tolerances given here [1].

II-V-4 The application of the tolerances in the tables and the computation of the overall tolerance to which the measurement of the flow of a fluid may be subject are illustrated by two examples. In doing this the extent or power to which the separate factors affect the total tolerance is taken into account.

Item	*Tolerance (per cent)*	*Effect Factor*	*Square*
Tolerance for Example II-IV-2			
Orifice diameter, d	± 0.08	2	0.0256
Differential pressure, h_w	± 0.25	½	0.0156
Evaluation of density, ρ_1	± 0.50	½	0.0625
Coefficient, C	± 1.1	1	1.21
Expansion factor, Y_1	± 0.5	1	0.25
Area factor, F_a	± 0.02	1	0.0004
			1.5641
Overall tolerance	± 1.25		
Tolerance for Example II-IV-6			
Throat diameter, d	± 0.08	2	0.0256
Differential pressure, h_w	± 0.10	½	0.0025
Value of density, ρ	± 0.10	½	0.0025
Coefficient, C	± 0.70	1	0.49
			0.5206
Overall tolerance	± 0.72		

II-V-5 As may be seen from these examples, the overall tolerance will always be greater than that of the item having the largest tolerance. To say this another way, the final result of a flow-measurement computation cannot be more exact or have a smaller per cent uncertainty than the factor having the greatest uncertainty. Thus, where one factor, usually the coefficient, has a tolerance ranging from ± 0.4 to ± 4.0 per cent, the use of numbers with four to six significant digits does not imply a corresponding high degree of exactness. The use of so many digits improves the agreement between two or more computers and aids in the "rounding off" of the final result.

Reference

[1] "A Statistical Approach to the Prediction of Discharge Coefficients of Concentric Orifice Plates," R. B. Dowdell and Yu-Lin Chen; *Trans. ASME, Journal of Basic Engineering*, vol. 92, no. 3, Sept. 1970.

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Table II-V-1 Tolerances for Discharge Coefficients and Flow Coefficients

Primary Element	Coefficient from	Pipe Size, D	R_d or R_D	β	Tolerance (per cent)
Square-Edged Concentric Orifices Flange taps	Equations (II-III-1), (II-III-2) or Table II-III-2				
D & ½ D taps	Equations (II-III-3), (II-III-4) or Table II-III-3	$D \geqq 2.0$ in.	$R_d \geqq 5000\,D$	$0.20 \leqq \beta < 0.70$ $0.11 \leqq \beta \leqq 0.20$ $0.70 \leqq \beta \leqq 0.75$	± 1.0 or less ± 2.25 to ± 1.0 linearly with β ± 1.0 to ± 2.25 linearly with β
Vena contracta taps	Equations (II-III-5), (II-III-6) or Table II-III-4			V.C. taps only $0.70 \leqq \beta \leqq 0.80$	± 1.0 to ± 2.5 linearly with β
As above		$1.0 < D < 2.0$ in.		As above	Above tolerances to be multiplied by a factor of 1 to 2 increasing linearly as D decreases
As above			$4000 \leqq R_d < 5000\,D$	As above	Above tolerances to be multiplied by a factor of 1 to 2 increasing linearly as R_d decreases
Long-Radius Flow Nozzle (Fig. II-III-14) Pipe-wall taps at D & ½ D	Equation (II-III-12) or Table II-III-5	$2.0 \leqq D \leqq 16$ in.	$10^4 \leqq R_d \leqq 2.5 \times 10^6$	$0.2 \leqq \beta \leqq 0.8$	± 2.0
Long-radius Flow Nozzle (Fig. II-III-14) Taps at 1 D and nozzle throat	Calibration (See Par. II-IV-6)	$2.0 \leqq D \leqq 16$ in.	$R_d \leqq 10^5$	$0.2 \leqq \beta \leqq 0.5$	As determined (or ± 0.8)
1932 ISA Flow Nozzle (Fig. II-III-22) Corner taps	K by Fig. II-III-23	$2 \leqq D \leqq 40$ in.	$2 \times 10^4 \leqq R_D \leqq 10^6$	$0.32 < \beta < 0.8$	± 1.0
Venturi Tube Rough-cast inlet cone	Par. II-III-38	$4 \leqq D \leqq 32$ in.	$2 \times 10^5 \leqq R_D \leqq 10^6$	$0.3 \leqq \beta \leqq 0.75$	± 0.75
Venturi Tube Machined inlet cone	Par. II-III-38	$2 \leqq D \leqq 10$ in.	$10^5 \leqq R_D \leqq 10^6$	$0.4 \leqq \beta \leqq 0.75$	± 1.0
Venturi Tube Welded sheet metal inlet cone	Par. II-III-38	$8 \leqq D \leqq 48$ in.	$2 \times 10^5 \leqq R_D \leqq 2 \times 10^6$	$0.4 \leqq \beta \leqq 0.7$	± 1.5
Eccentric Orifice Flange taps Vena contracta taps	Fig. II-III-9	$4 \leqq D \leqq 14$ in.	$10^4 \leqq R_D \leqq 10^6$	$0.3 \leqq \beta \leqq 0.8$	$D = 4$ in. ± 1.9 $D > 4$ in. ± 1.4
Segmental Orifice Flange taps Vena contracta taps	Fig. II-III-10	$4 \leqq D \leqq 14$ in.	$10^4 \leqq R_D \leqq 10^6$	$0.35 \leqq \beta \leqq 0.85$	± 2

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Table II-V-2 Tolerances for Expansion Factors

Primary Element	Factor from	D	β	x or $h_{w/p}$	Fluids	Tolerance* (per cent)
Square-Edged Concentric Orifices Flange taps	Equations (II-III-4), (II-III-5)	$D \lesssim 1.0$	$0.11 \lesssim \beta \lesssim 0.75$	$x < 0.4$ $h_{w/p} < 11.1$	Gases for which γ (or Γ) is known	0.0 to ± 0.75
D & ½ D taps Vena contracta taps	Figs. II-III-5, II-III-6			$h_{w/p_2} < 18.5$	Gases for which γ (or Γ) is uncertain	0.0 to ± 1.5
Flow Nozzles Venturi Tubes	Equation (II-III-10) Tables II-III-6, II-III-7	$D \lesssim 1½$ in.	$0.1 \lesssim \beta \lesssim 0.8$	$x < 0.4$ $h_{w/p_1} < 11.1$	Gases for which γ is (or Γ) known	0.0 to ± 0.4
	Figs. II-III-20, II-III-21				Gases for which γ is (or Γ) uncertain	0.2 to ± 1.2
Eccentric Orifice	Fig. II-III-11	$4 \lesssim D \lesssim 16$	$0.3 \lesssim \beta \lesssim 0.8$	$x < 0.3$	Gases for which γ is known	0.0 to ± 1.0
Segmental Orifice	Fig. II-III-12			$h_{w/p_1} < 8.3$	Gases for which γ is uncertain	± 1.0 to ± 4.0

*Tolerances increase linearly as x increases.

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