Technical Report on Performance of API and ANSI End Connections in a Fire Test According to API Specification 6FA

API TECHNICAL REPORT 6F1
THIRD EDITION, APRIL 1999

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Upstream Segment

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FOREWORD

An analytical procedure to predict the performance of API Spec 6A and API Spec 6D standard end connections in the API Fire Test was developed in several research projects. In this Report these procedures are applied to API Spec 6A connections from $2^1/_{16}$ in. to $7^1/_{16}$ in., and API Spec 6D (ANSI) connections from 2 in. to 6 in.

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0 Introduction

The Production Department of API first published in 1978 a procedure for fire testing valves. The current procedure is API Spec 6FA, *API Specification for Fire Test for Valves* (1)[1]. Basically, the Specification calls for a fire with an average temperature of 1400°F–1800°F to be applied for 30 minutes to a valve pressurized with water to detect leakage. Since end connections usually are included in these tests, they become potential failure points for a fire-tested valve assembly.

Since end connections are standard products, API has sponsored several research projects to define the ability of end connections to pass the fire test. Ten standard end connections of various sizes, types, and pressure ratings were tested in 1979 (2). (PRAC-79-21)

Two subsequent research projects (3, 4) in 1980–81 (PRAC 80-33) and 1981–82 (PRAC 81-33) resulted in procedures to analytically predict the performance of flanged and clamped connections to the fire environment.

This procedure basically is composed of four parts: (a) predicting the temperature distribution in the flange, (b) predicting the preload loss, (c) predicting the performance of the various seals with the reduced preloads which occur in a fire, and (d) predicting whether or not yielding of bolts is likely to occur. Significant yielding will lead to leakage either during the fire or shortly afterward.

A fourth project (PRAC-83-33) (5) evaluated the standard end connections in API Specifications 6A (6) and 6D (7) using these analytical procedures.

This report summarizes the results of all the projects. In addition, the appendixes present the analytical procedures used to generate the performance prediction of section 3.

1 Scope and Applicability

The sizes of interest are shown in Table 1. The ANSI (8) sizes are adopted by API Spec 6D. The API sizes refer to those in API Spec 6A.

Table 2 shows the materials and gasket types which have been studied. As can be seen, there are a large number of sizes and combinations of materials and gaskets.

One very important point is that this Technical Report, and the research projects which preceded it, are based on the fire test procedure in API Spec 6FA. Whether or not API Spec 6FA adequately defines a "true" fire is totally outside the scope of this bulletin. The test simulates a closed condition with no flow through the connection. If there were flow through the connector, the heat transfer response would likely be different. Also, the external loads from dead weight, piping system thermal expansion, wind, etc., could significantly affect a joint's performance in a fire. These effects could be included in the design procedures reported herein, but they are not in the scope of this report.

1.1 EFFECTS OF FIRE ENVIRONMENT

Exposure to a fire environment around a flanged or clamped joint will tend to reduce the joint preload required for many seals to function. In a flanged joint, the flange exterior and bolts will heat up quicker than the seal and interior flange portion. This thermal gradient across the joint will cause preload to be lost. If enough preload is lost, the seal may unseat and leak; lose the contact pressure necessary to maintain a seal; fail if the seal is not strong enough to carry pressure load without the restraint from the adjacent contact surfaces, or fail if the temperature capacity of the seal material is exceeded.

1.1.1 In addition to preload loss from the thermal gradient, there is a preload loss due to the reduction of joint stiffness resulting from increasing temperature. Another factor which can cause preload loss is yielding of the gaskets, studs, or flanges. Bolt yielding results in permanent stretching which means permanent preload loss and likely leakage, either hot or cold.

Table 1—Sizes of Interest

ANSI Flanges Analyzed					
Nominal Pipe Size, in.	Class				
	150	300	400	600	900
2	X	X		X	C1 1500
$2^1/2$	X	X		X	C1 1500
3	X	X		X	X
$3^1/2$	X	X		X	
4	X	X	X	X	X
5	X	X	X	X	X
6	X	X	X	X	X

Clamps and API Flanges Analyzed							
Size (Bore)	6B 2000	6B 3000	6B 5000	6BX 10000	6BX 15000	Clamp 5000	Clamp 10000
$2^1/_{16}$	X	X	X	X	X	X	X
$2^9/_{16}$	X	X	X	X	X	X	X
$3^1/_{16}$				X	X		X
$3^1/_8$	X	X	X			X	
$4^1/_{16}$	X	X	X	X	X	X	X
$5^1/_8$	X	X	X	X		X	
$7^1/_{16}$	X	X	X	X	X	X	X

[1]Numbers in parentheses refer to corresponding items in section 2, References.

Table 2—Materials and Gaskets of Interest

Materials and Gaskets ANSI Flanges		
	Combinations	
	1	2
Flange Material	SA 105	316 SS
Bolt Material	SA 193 B7	SA 193 B8
Gasket Type	1. Spiral Wound 2. RTJ Type R	RTJ Type R
Gasket Material	1. Stainless 2. Soft Iron	Stainless

Materials and Gaskets ANSI Flanges		
	Combinations	
	1	2
Flange Material	4130	410 SS
Bolt Material	SA 193 B7	SA 453 Gr 660
Gasket Type	RTJ	RTJ
Gasket Material	Carbon Steel	Stainless Steel

1.2 SIGNIFICANT VARIABLES

Variables which are significant to this problem are as follows:

1.2.1 Size and Geometry of the Joint

The size will greatly affect the thermal gradient through the joint, with larger joints having much larger thermal gradient than smaller joints for a thermal event such as the standard API Fire Test. However, the larger joints do not get nearly as hot as the smaller joints for the fire test conditions of this Technical Report. Figures 1and 2 illustrate this.

1.2.2 Type of Seal

Of particular importance for the seals is the seating load, the retaining load; amount of pressure energization; whether the seal is used in joints that make up face-to-face, or with standoff; and the gasket material.

1.2.3 Material of Construction

Carbon steel has a thermal conductivity approximately three times as large as that of austenitic stainless steel, and a lower thermal expansion coefficient. Thus, the thermal response and preload change of the joints are a very strong function of the material.

In addition, a critical variable is the yield strength of the bolts and flange at the high temperature of the test. If a large amount of yielding occurs, the remaining preload will be very small.

1.2.4 Joint Preloads and Internal Pressures

Flanged and clamped joints are normally designed with preloads based on two considerations: (a) the amount of preload required to seat the seal, and (b) the amount of preload required for the gasket retaining load, pressure end loads, and any external forces or moments. Joints in which the seating load is controlling (highest) will tend to have more preload margin for the pressure and retaining load case and, thus, may tend to be more leak resistant in the fire test. Further, the API Fire Test is at about 75% of the rated pressure. For high pressure connections, this 75% of rated pressure is a very large number compared to the retaining load. Thus, it might be expected that higher pressure joints are more resistant to the fire.

Another consideration is the variability of preload in actual field joints.

2 References

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   Specification for Fire Test for Valves, American Petroleum Institute, latest edition.
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5. Fowler, Joe R. and David S. Young
   Prediction of Performance of 2"–6" ANSI Flanges and $2^1/_{16}$"–$7^1/_{16}$" Clamps and Flanges in a Standard Fire Environment, prepared for the American Petroleum Institute, July 1984 (PRAC-83-33).
6. API Spec 6A
   Specification for Wellhead and Christmas Tree Equipment, American Petroleum Institute, latest edition.
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   Specification for Pipeline Valves (Gate, Ball, Plus and Check Valves), American Petroleum Institute, latest edition.
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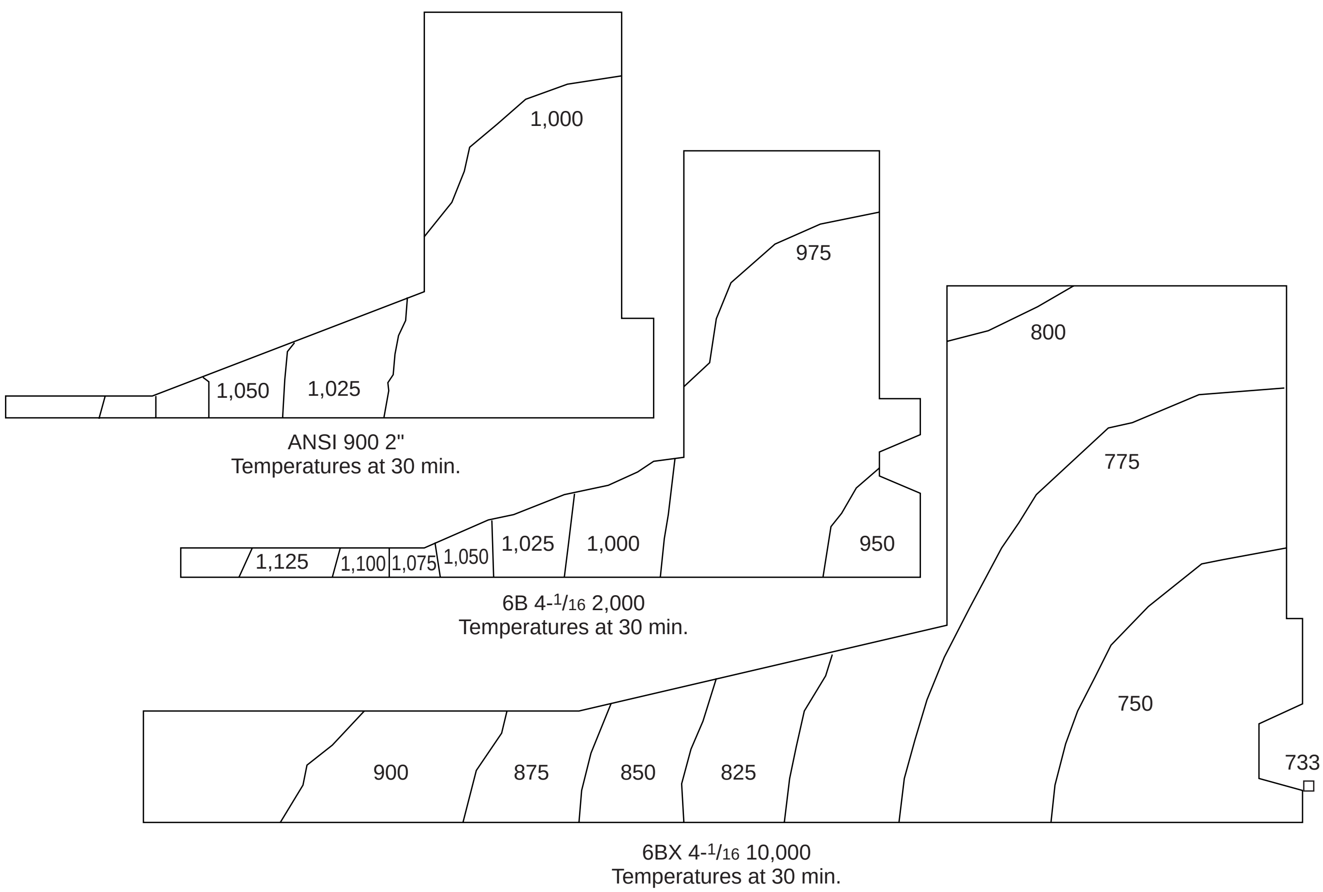


Figure 1—Typical Calculated Temperatures of Joints

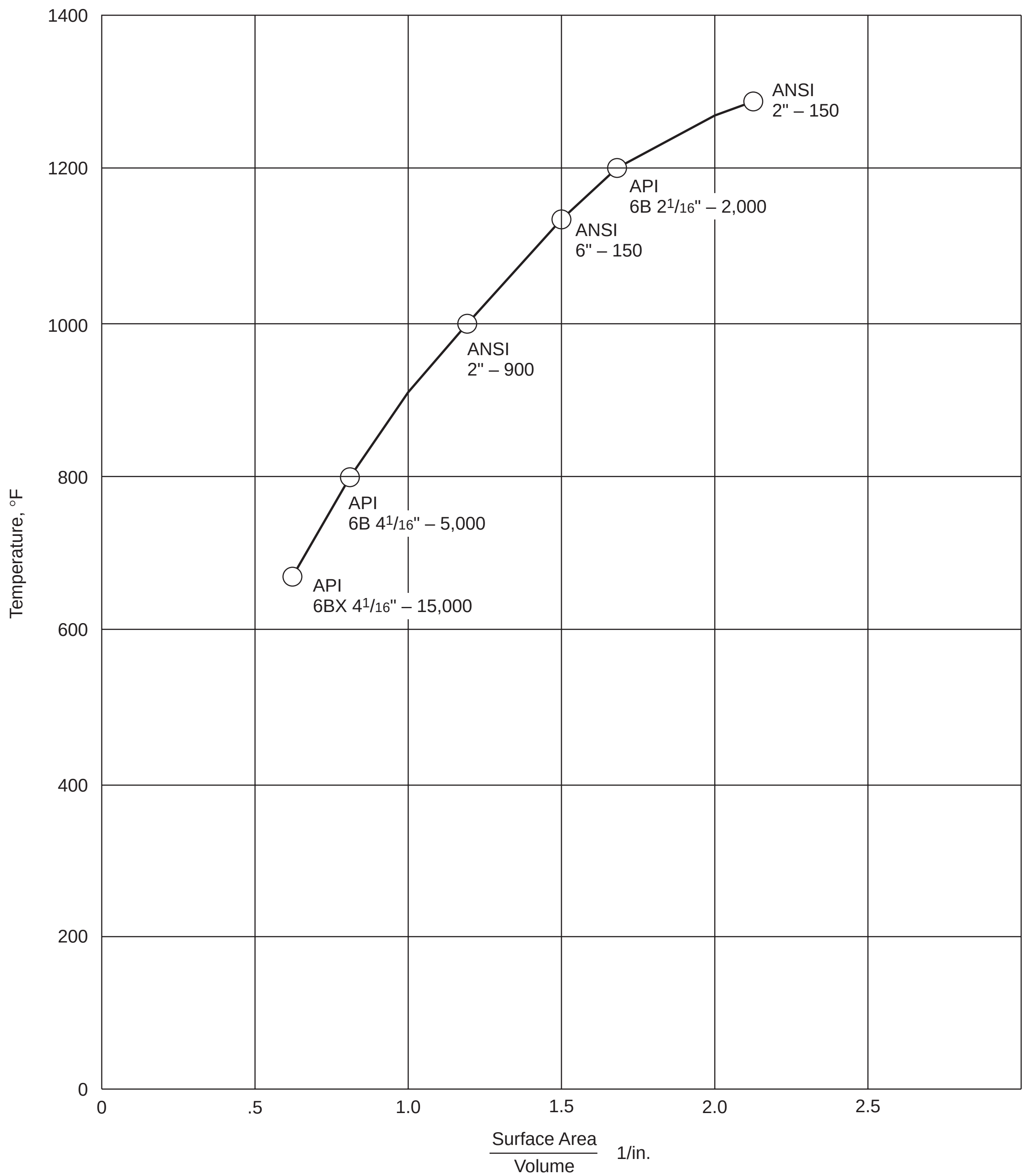


Figure 2—Average Calculated Temperatures of Various Joints Carbon Steel at 30 Minutes

3 Performance of API and ANSI End Connections in the Standard Fire Test of Specification 6FA

3.1 INTRODUCTION

The prediction of whether or not a particular joint will fail in the fire test is based on the three following considerations:

a. Does the joint lose so much preload that it does not have enough left for the required seal retaining load?

b. Does the joint get so hot that the bolts or the connection yield?

c. Do the BX gaskets have enough elastic "springback" to keep from unseating?

Table 3 shows the classification adopted in reporting possible leakage.

3.2 MAKEUP ASSUMPTIONS

3.2.1 For the ANSI flanges, the assumptions contained in the *ASME Boiler and Pressure Vessel Code* (9) regarding flange makeup were used. The makeup assumption is that the joint is made up to a bolt load corresponding to the average of the required and actual bolt area, times the bolt allowable at room temperature.

3.2.2 For the API flanges, the bolts were assumed to be made up to one-half the yield strength.

3.2.3 The required makeup clamping load for clamp type connections is the greater of the gasket seating load and the gasket retaining load, plus design pressure end load. A positive angle of friction of 5.7 degrees is used for the makeup condition. Seating loads and retaining loads for RX gaskets are shown in Table A-1 of Appendix A.

3.2.4 Tables 4 through 6 summarize the findings.

Table 3—Classifications

A = When made up to normal specifications the joint has adequate retaining load and the BX gaskets will not unseat. No leakage is predicted.

B = When made up to normal specifications, the retaining load is between 50% and 100% of the recommendation of the *ASME Boiler and Pressure Vessel Code (BPV)*. Leakage is possible.

C = Retaining load is less than 50% of the recommendation of the BPV. Leakage is likely.

D = BX gaskets will unseat. Leakage is likely.

E = The yield strain of the bolt at temperature is between 75% and 100% of the makeup bolt strain. Therefore, leakage is likely.

F = The yield strain of the bolt at temperature is less than 75% of the makeup bolt strain. Leakage is almost certain.

3.3 GENERAL PREDICTIONS

3.3.1 The spiral wound stainless steel gaskets have very high retaining and seating loads. Even though they easily stand the temperature, it is more difficult to maintain a seal with them. The RTJ gaskets are better from a retaining-load standpoint.

3.3.2 The low pressure and small size carbon steel and 4130 joints are the most susceptible to leakage under the no-flow conditions of the test.

3.3.3 Actual required retaining loads to maintain a seal are not well known. The *ASME Boiler and Pressure Vessel Code* requirements are guidelines, but they are likely to be conservative. Most joints that are classification B (50% to 100% of required retaining load) probably won't leak.

Table 4—Classification Summary of ANSI Carbon Steel Flanges Performance Prediction in Standard Fire Test

Nominal Pipe Class Size in.	Class 150	300	400	600	900
2	F	F, B	—	E, C	A
$2^1/_2$	F	F	—	B	A
3	F	E, B	—	C	C
$3^1/_2$	F	E, B	—	C	—
4	E	B	B	C	B
5	F	B	B	C	B
6	F	B	B	C	B

Spiral Wound Stainless Steel Gaskets

Nominal Pipe Class Size in.	Class 150	300	400	600	900
2	F	F	—	C,E	A
$2^1/_2$	E	E	—	B	A
3	E	A	—	C	C
$3^1/_2$	A	A	—	B	—
4	A	B	A	C	A
5	A	C	B	C	A
6	A	C	A	C	B

RTJ Soft Iron Gasket

Notes:

A = No leakage.

B = Retaining load 50% to 100% of required—probably won't leak.

C = Retaining load less than 50% of required—leakage likely.

D = BX gasket unseats—leakage likely.

E = Bolts yield small amount—leakage possible.

F = Bolts yield large amount—leakage likely.

Table 5—Classification Summary of 316 Stainless ANSI Flanges Performance Prediction in Standard Fire Test

Nominal Pipe Class Size in.	Class				
	150	300	400	600	900
2	A	A	—	B	A
$2^1/_2$	C	A	—	C	C
3	C	C	—	C	C
$3^1/_2$	C	C	—	C	—
4	C	C	C	C	C
5	C	C	C	C	C
6	C	C	C	C	C

Stainless RTJ Gasket

Notes:
A = No leakage.
B = Retaining load 50% to 100% of required—probably won't leak.
C = Retaining load less than 50% of required—leakage likely.
D = BX gasket unseats—leakage likely.
E = Bolts yield small amount—leakage possible.
F = Bolts yield large amount—leakage likely.

Table 6—Classification Summary of API Flanges and Clamps[a] Performance Prediction in Standard Fire Test

Size (Bore)	6B 2,000	6B 3,000	6B 5,000	6BX 10,000	6BX 15,000	Clamp[a] 5,000	Clamp[a] 10,000
$2^1/_{16}$	F	E	A	E	A	E	F
$2^9/_{16}$	F	A	A	A	A	E	F
$3^1/_{16}$	—	—	—	A	A	—	A
$3^1/_8$	F	A	A	—	—	A	—
$4^1/_{16}$	A	A	A	A	A	A	A
$5^1/_8$	A	A	A	A	—	A	—
$7^1/_{16}$	A	A	A	A	A	E	A

4130 Material—B7 Studs

Size (Bore)	6B 2,000	6B 3,000	6B 5,000	6BX 10,000	6BX 15,000	Clamp[a] 5,000	Clamp[a] 10,000
$2^1/_{16}$	C	C	C	D	D	C	C
$2^9/_{16}$	C	C	C	D	D	C	C
$3^1/_{16}$	—	—	—	A	D	—	C
$3^1/_8$	C	C	C	—	—	C	—
$4^1/_{16}$	C	C	C	A	A	C	C
$5^1/_8$	C	C	C	—	—	C	—
$7^1/_{16}$	C	C	C	A	A	C	C

410 SS Material—A286 Studs

Notes:
A = No leakage.
B = Retaining load 50% to 100% of required—leakage possible.
C = Retaining load less than 50% of required—leakage likely.
D = BX gasket unseats—leakage likely.
E = Bolts yield small amount—leakage possible.
F = Bolts yield large amount—leakage likely.

[a]Clamp-type connections are covered in API Specification 16A, *Specification for Drill Through Equipment*. (17)

3.3.4 BX gaskets tend to unseat in small sizes with 410 stainless bodies and A286 studs (SA 453 Gr 660). There is no such tendency for 4130 bodies with B7 studs.

3.3.5 API 6B flanges and API clamps with 410 stainless bodies and A286 studs are very poor in fire resistance. This is because of the greater thermal expansion with temperature of the bolts compared with the bodies. No work was done for 410 stainless bodies and B7 studs.

3.3.6 The API flanges with 4130 materials (60 ksi), and B7 studs, and soft iron RTJ gaskets are very fire resistant except for the smallest sizes and pressure range.

3.3.7 Although in many cases yielding of the flanges is predicted, this was usually ignored in the classification. This is because the calculation procedures for flange stresses are usually conservative; and because yielding at the surface of a flange or clamp does not result in significant permanent deformations until the yield is exceeded by a good margin.

4 Comparison of Predictions and Test Data

4.1 Table 7 shows the materials of the carefully instrumented tests of Reference (4); and Table 8 shows the results. The leakage results are consistent with the predictions of section 3.

4.2 Table 9 shows the results of testing of the original screening tests [Reference (2)]. Materials were carbon steel for the ANSI connection, 4130 API Type 2 for the API connections, and B7 studs for all.

4.3 Also shown are the leakage predictions from section 3 of this Technical Report.

4.4 The API 6B $2^1/_{16}$ in. 2,000 connection did not leak even though it has an F rating (bolts yield). However, the connection only reached 848°F, which is very low. For this reason, the same joint configuration was tested in the instrumented tests, where the connection reached 1,180°F and leaked.

4.5 Other tests which are inconsistent with the predictions are the $2^1/_{16}$ in. 15,000 6BX (leaked, but was not predicted to leak); and the $7^1/_{16}$ in. 5,000 6B (leaked, but was not predicted to leak). However, the $7^1/_{16}$ in. 5,000 6B had a much hotter flame temperature than the test required, and this may explain the difference.

4.6 The remainder of the tests are consistent with the predictions.

Table 7—Materials of Test Flanges (Reference 4)

Joint	Stud Material	Flange Material	Gasket Material
6" 150 lb. ANSI	SA 193 B7	A 105 GrII	Spiral Wound 347 SS w/impregnated lubricant
$2^1/_{16}$" 2,000 lb API 6B	SA193 B7	API Type 2 4130	R23 Carbon Steel
$2^1/_{16}$" 1,000 lb API 6BX	A286 SA453 Gr660	410 SS API Type 2	BX 152 316 Stainless
6" 900 lb ANSI	SA 193 B7	A 105 GrII	Spiral Wound 347 SS w/impregnated lubricant
$4^1/_{16}$"15,000 lb API 6BX	A286 SA453 Gr660	410 SS API Type 2	BX 155 316 Stainless

Table 8—Results of Testing Carefully Instrumented Tests (Reference 4)

Joint	Bolt Maximum Temperature	Bolt Yielding	Leakage	Rating from Section 3
6" 150 lb ANSI	1,300°F	Yes	Leak began at 25 min. Bolts loose after test.	F
$2^1/_{16}$" 2,000 lb	1,200°F	Yes	Leak began at 57 min. (burn finished at 30 min.) Breakout torque about half makeup torque. Flange temperature 1,180°F.	F
$2^1/_{16}$" 10,000 lb	1,028°F	No	BX gasket unseated at 19 minutes. Repressured at 1:38 at 270ºF and held.	D
6" 900 lb ANSI	850°F	No	None	B
$4\ ^1/_{16}$" 15,000 lb API 6BX	679°F	No	None	A

Notes:
A = No leakage.
B = Retaining load 50% to 100% of required—probably won't leak.
C = Retaining load less than 50% of required—leakage likely.
D = BX gasket unseats—leakage likely.
E = Bolts yield small amount—leakage possible.
F = Bolts yield large amount—leakage likely.

Table 9—Results of Testing Original Screening Tests

Joint Description	Gasket	Leakage	Rating from Section 2	Comments
6" 150 ANSI	Asbestos	Yes	F	
6" 900 ANSI	Asbestos	No	B	
2" 2,000 6B	R23	No	F	Flange Temp of 848° is very low.
6" 600 ANSI	Asbestos	Yes	C	
$2^1/_{16}$" 5,000 Clamp	RX23	No	E	
$2^1/_{16}$" 10,000 6BX	BX152	Gasket Unseated	E	
$2^1/_{16}$" 15,000 6BX	BX152	Gasket Unseated	A	
$7^1/_{16}$" 5,000 6B	R46	Yes Resealed	A	Flame Temperatures were 1600°F–1700°F.
$4^1/_{16}$" 5,000 6B	R39	No	A	
$2^1/_{16}$" 5,000 6B	R27	No	A	

Notes:
A = No leakage.
B = Retaining load 50% to 100% of required—probably won't leak.
C = Retaining load less than 50% of required—leakage likely.
D = BX gasket unseats—leakage likely.
E = Bolts yield small amount—leakage possible.
F = Bolts yield large amount—leakage likely.

5 Conclusions and Guidelines

5.1 Small diameter, low pressure connections in carbon steel are very likely to leak in the standard fire test with no flow. This is because they get too hot; and yielding of the B7 bolts occurs.

5.2 Two ways that a user could significantly improve protection against leakage of these joints in a fire are by insulating the joints, or by using a joint that is one or two pressure classes higher than what is required for pressure alone. There are other ways of reducing the likelihood of a flange fire that are not in the scope of this Technical Report.

5.3 Bodies of 410 stainless with A286 studs and BX gaskets in sizes smaller than $4^1/_{16}$ in. have poor leakage resistance in the fire test. These gaskets are likely to unseat.

5.4 API 6B flanges and clamps with 410 stainless bodies and A286 (SA 453 660) bolts are very likely to leak in a fire test because the bolts expand much more than the bodies. As a general rule, it is good practice to match the expansion coefficients of the bolts and bodies.

5.5 The actual seat retaining loads required to prevent leakage are not well known. The retaining loads required by the *ASME Boiler and Pressure Vessel Code* may be conservative.

5.6 There is possibly a wide variation in results from those predicted here because of the variability of test facilities; uncertainties of makeup; variations in actual material strengths; and variations in gaskets. However, those results should be indicative of comparative strengths.

5.7 API flanges with 4130 bodies and carbon steel RTJ gaskets are predicted to be relatively good in the fire test (except for the small, low pressure sizes). This is because the predicted retaining loads of these gaskets are small, and because the makeup bolt loads are much higher than the equivalent ANSI flanges.

5.8 Of the 172 connections studied, the predicted breakdown is as follows:

Category	Number	Percent
A = No leakage	54	32
B = Low retaining load, leakage possible	16	9
C = Very low retaining load, leakage likely	67	39
D = BX Gasket unseated, leakage likely	5	3
	(5 of 22, or 23% of BX joints)	
E = Bolts yield some, leakage possible	13	8
F = Bolts yield a lot, leakage likely	15	9

5.9 Things that may change the predicted results include change in the ASME seal retaining load and flow in the pipe. Rapid flow of a relatively cool liquid could keep most joints from leaking. However, slow flow of a gas would have little effect.

APPENDIX A—ANALYSIS OF ANSI AND API FLANGES

In the References (3) and (4), the detailed analysis procedures are described. For sake of completeness, the entire procedure is presented here, even though it involves extensive repetition of the material in Reference (4). The step-by-step procedure is shown below.

Step 1: Predict Average Joint Temperature

This is done by the following equation:

$$\frac{T_\infty - T}{T_\infty - T_o} = e^{-\left(\frac{hA}{\rho CV}\right)\theta} \tag{A-1}$$

where

T_∞ = flame temperature (1500°F),

T = average temperature of joint, °F,

T_o = starting temperature (70°F),

h = external convection coefficient,

$$10\frac{Btu}{\mathrm{hr - ft^2 - °F}}$$

A = all joint surface areas exposed to flame, ft^2

ρ = density, lbm/ft^3

C = specific heat Btu/lbm °F,

V = joint metal volume, tapered hub plus flanges, ft^3

θ = time, hours.

It has been shown in Reference (3) that solutions of the above equation were in agreement with the predicted average joint temperature obtained using the finite element method.

Step 2: Predict Temperature Gradient Across the Joint

It has been found that the temperature gradient across the joint is dependent upon the ratio A/V, where A is the surface area exposed to the flame and V is the volume of steel in the joint. Figure A-1 shows the temperature difference (ΔT°F) times thermal conductivity $\left(k\frac{\mathrm{Btu}}{\mathrm{hr - ft^2 - °F}}\right)$ plotted against A/V (1/in.) for the time of 624 seconds, 1,368 seconds, and 1,800 seconds.

To determine the ΔT, the average temperature should first be found from Equation A-1. Then the thermal conductivity for that temperature and the material in question should be used with the KΔT read from Figure A-1 to find the ΔT across the flange.

Then the bolt and seal/seat temperatures can be approximated by adding ΔT/2 to and subtracting ΔT/2 from the average joint temperature (Step 1), respectively.

Step 3. Calculate the Retaining Loads

There are several different types of gaskets used in the flange joints. They can be divided into two separate categories.

Category 1: Type R Ring Joint and Raised Face Flanges (No contact outside Bolt Circle)

These gaskets do not have radial interference or initial makeup. They seal by maintaining a contact stress greater than the internal pressure.

To predict whether or not this gasket will leak, it is recommended to use the procedures of the *ASME Boiler and Pressure Vessel Code*, Section VIII, Division 2, Article 3-3, “Flange with ring type gaskets.” Two conditions must usually be checked—operating and seating. The required preload, W_{m1}, and gasket seating load, W_{m2}, are given as follows:

$$W_{m1} = \frac{\pi}{4}G^2P + \underbrace{(2b \times \pi GmP)}_{\text{Retaining Load}} \tag{A-2}$$

$$W_{m2} = \pi bGy \text{ seating load} \tag{A-3}$$

where

G = gasket diameter (seal diameter),

P = internal pressure,

b = effective gasket seating width,

m = gasket factor,

y = gasket seating stress.

The gasket factors, seating stresses, and seating width procedures are from the *ASME Boiler and Pressure Vessel Code*, Section VIII, Division 1, App 2, Table 2-5.1 and 2-5.2 (1986).

Category 2: RX and BX Gaskets

API RX and BX gaskets have initial radial interference between the seat and seal when made up. The normal flange joints with RX gaskets do not make up face-to-face, i.e., the flange faces do not touch and the preload is all through the gasket. For BX gaskets and RX gaskets in API clamps, the joints do make up essentially face-to-face. All preload other

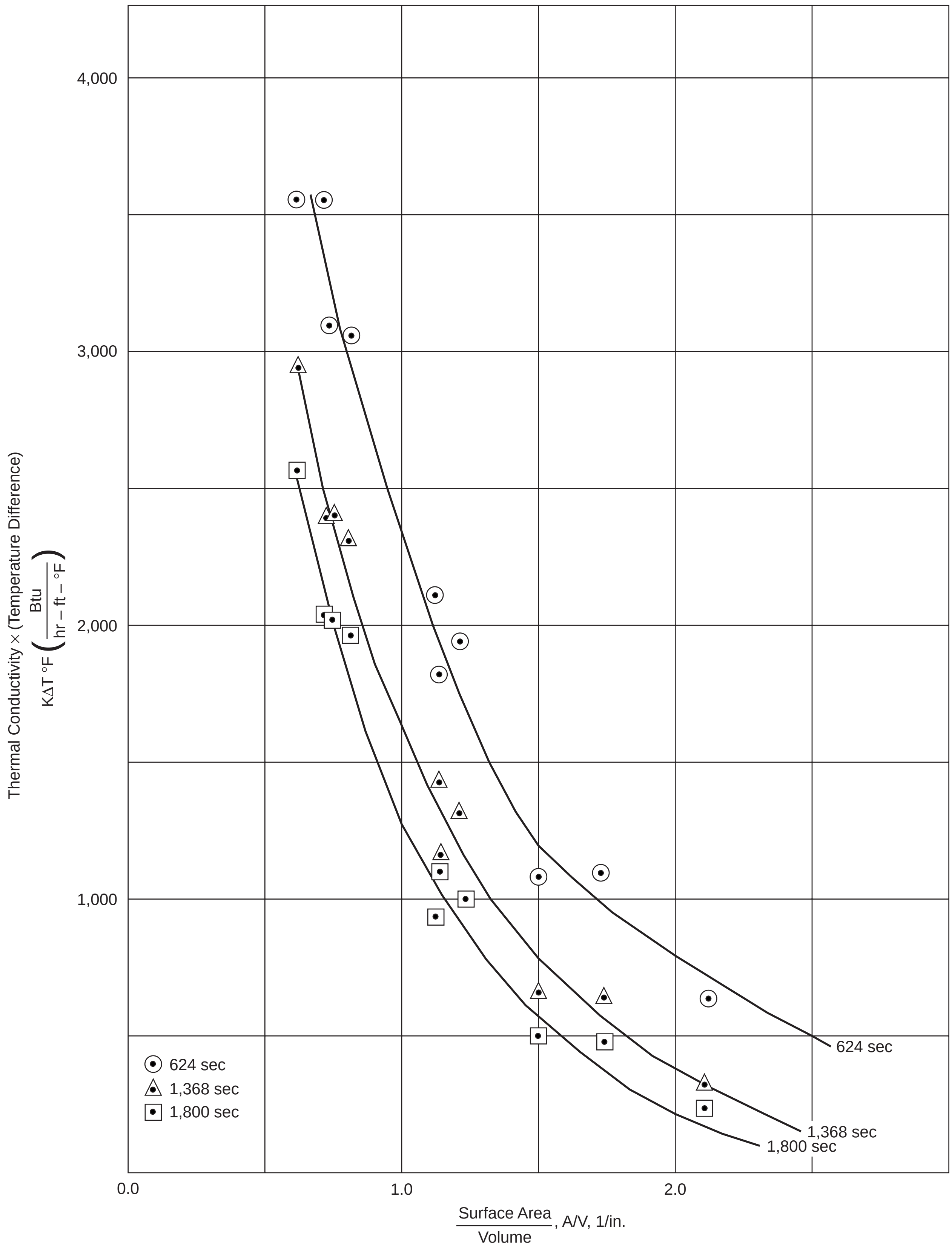


Figure A-1—Correlation of Temperature Gradients Across Flanges than that to retain the seal is reacted at the outside of the raised face for flanges, and at the outside of hubs for clamp connections.

Both RX and BX gaskets require a radial compressive load between the seat and seal on the outside of the gasket to maintain the interference. The retaining load is simply the axial load required to maintain the radial load on the seals, 23 degrees outside shoulder.

For RX gaskets these retaining loads have been computed in a report done for the Association of Wellhead Equipment Manufacturers (10). These are shown in Table A-1.

The equations for predicting the BX gasket retaining loads are derived and included in Appendix C. Also included are the equations to predict whether or not BX gaskets will unseat due to an expansion difference between the bolt and flange.

Table A-1—API Type RX Pressure Energized Ring Gaskets Predicted Values for Seating and Retaining Loads

		Retaining Load vs. Internal Pressure (lb)			
Gasket	Seating Load (lb)	2000 psi	3000 psi	5000 psi	10000 psi
RX 20	40777.	23851.	25351.	28352.	35853.
RX 23	74517.	42999.	45448.	50345.	62587.
RX 24	74517.	43656.	46448.	52012.	65921.
RX 26	74517.	43979.	46917.	52793.	67483.
RX 27	74517.	44312.	47417.	53626.	69150.
RX 31	74517.	45166.	48698.	55762.	73422.
RX 35	74517.	45833.	49699.	57429.	76756.
RX 37	74517.	46500.	50699.	59096.	80090.
RX 39	74517.	47167.	51699.	60763.	83424.
RX 41	74517.	48167.	53199.	63264.	88425.
RX 44	74517.	48834.	54199.	64931.	91759.
RX 45	74517.	49751.	55575.	67223.	96343.
RX 46	66803.	47285.	53849.	66977.	99796.
RX 47	172449.	109103.	119566.	140492.	192808.
RX 49	74371.	52760.	60126.	74858.	111688.
RX 50	93251.	66279.	75578.	94175.	140669.
RX 53	61930.	49232.	58015.	75581.	119497.
RX 54	77555.	61796.	72866.	95006.	150355.
RX 57	52611.	47468.	57751.	78318.	129735.
RX 63	211856.	154711.	177904.	224288.	340249.
RX 65	42632.	47033.	59650.	84885.	147971.
RX 66	53286.	58972.	74834.	106560.	185873.
RX 69	37546.	47766.	62051.	90619.	162040.
RX 70	73317.	84420.	107886.	154817.	272146.
RX 73	53464.	66460.	86021.	125144.	222950.
RX 74	66906.	85476.	111109.	162375.	290539.
RX 82	74517.	41666.	43447.	47011.	55919.
RX 84	74517.	41999.	43947.	47844.	57586.
RX 85	78077.	44652.	47018.	51748.	63573.
RX 86	105440.	60032.	63091.	69210.	84506.
RX 87	105440.	60595.	63935.	70616.	87319.
RX 88	137654.	70526.	84100.	93242.	116098.
RX 89	139198.	79693.	83952.	92469.	113763.
RX 90	229327.	133306.	141328.	157373.	197485.
RX 91	203761.	131023.	144441.	171276.	238365.
RX 99	74517.	51001.	57450.	70349.	102595.

Step 4. Predict the Preload Loss

Preload of a flange joint that has not yielded will be lost because of two reasons: the average temperature, and the temperature gradient.

First, the joint is made up at room temperature by the initial stretch, Δ, of the bolts. The bolt force, F_b, and stretch, Δi, are related by the following (see Figure A-2):

$$F_b = \frac{K_b K_f}{K_b + K_f}\Delta i \qquad \text{(A-4)}$$

Both K_b and K_f are linearly related to the modulus of elasticity, E. As the temperature increases, and E decreases as shown in Table A-2, the bolt force, F_b, must decrease since the bolt stretch is constant.

Second, the temperature gradient across the flanges causes a stress distribution-compressive at the outside, tensile at the inside across the line A-A of Figure A-2.

This causes a flange rotation which also tends to relieve preload.

The first type of preload loss can be predicted simply by knowing the average joint temperature. This can be computed from Equation A-1.

The second type was assumed to be predicted by following the equation:

$$P = CE\alpha\Delta T \qquad \text{(A-5)}$$

where

P = preload loss, lb,

C = preload constant to be determined from the finite element data. Figures A-3, A-4, and A-5 contain these C factors.

E,α = Modulus of elasticity, coefficient of thermal expansion computed as mean value in going from 70°F to the average temperature (see Table A-2),

ΔT = The temperature gradient across Section A-A (Figure A-1).

Joints that have the same C factor do not necessarily have the same preload loss, of course, as the equation also includes the ΔT across the flange, which is generally greater for the thicker flanges.

The final load in the joint at the gasket is computed by starting with the initial reload F_i. The final gasket load, Ff is as follows:

$$F_f = (F_i - CE\alpha\Delta T)\frac{E_T}{E_{RT}} - \frac{\pi}{4}G^2P \qquad \text{(A-6)}$$

where

F_f = final gasket load,

F_i = initial gasket load before the test or pressure is applied,

$CE\alpha\Delta T$ = from Equation (A-5),

E_T = modulus of elasticity at average joint temperature from Equation (A-1),

E_{RT} = modulus of elasticity at room temperature,

G = gasket diameter (seal diameter),

P = fire test pressure.

The initial preload F_i for an ANSI flange is equal to the bolt allowable stress at room temperature times the average of available bolt area and required bolt area. For an API flange, the initial preload is equal to the total available bolt area times half of he bolt yield strength.

If this final gasket load of F_f is less than that required to maintain a seal for the particular gasket in question, then leakage may be expected.

Table A-2—Moduli of Elasticity psi $\times 10^6$

Temperature °F	SA193 B7	A105 GrII	API TP2 4130	410 SS	SA453 Gr660
70	29.9	27.9	29.9	29.2	31.7
500	27.4	26.4	27.4	27.0	29.6
600	26.7	25.7	26.7	26.0	29.2
800	23.8	23.4	23.8	23.1	27.9
900	21.5	18.5	21.5	21.1	27.1[b]
1,000	18.8	15.4	18.8	18.6	26.3[b]
1,100	15.0	13.0	15.0	15.6	25.5[b]
1,200	11.2	10.6[a]	11.2	12.2	24.7[b]
1,300	7.4[a]				24.0[b]
1,400					
1,500					

[a]Extrapolated.
[b]Taken as 1.15 X Austenitic Stainless Steel Reference *ASME Boiler & Pressure Vessel Code*.

Table A-2—Coefficients of Thermal Expansion Mean in Going From 70° to Temperature in./in. $\times 10^{-6}$/°F (Continued)

Temperature °F	SA193 B7	A105 GrII	API TP2 4130	410 SS	SA453 Gr660
70	5.73	6.5	5.73	5.98	8.24
500	7.06	7.34	7.06	6.48	8.82
600	7.28	7.42	7.28	6.53	8.92
700	7.51	7.59	7.51	6.60	9.06
750	7.61	7.68	7.61	6.64	9.11
800	7.71	7.76	7.71	6.67	9.17
1,000	8.11	8.08	8.11	6.79	9.41
1,200	8.51	8.40	8.51	6.91	9.65

Step 5: Perform a Stress Analysis of the Joint at Ambient and Operating Temperatures

Stress analysis procedures for flanges are given in the *ASME Boiler and Pressure Vessel Code*. Reference (15) gives stress analysis procedures for clamp-type connectors.

Figure A-6 shows a typical set of high temperature stress strain curves for SA 193 B7 studs. When the studs are made up, they are elastically stretched a certain amount, resulting in a given stress, say 50 ksi. To a first approximation, this stretch (and strain) remains the same as the joint heats up unless the temperature of the studs gets so high that yielding occurs. For example, the strain corresponding to 50 ksi at 100°F is 1,700 microstrain. At 1,200°F the yield strain is only 1,400 microstrain. Therefore, if the stud reaches 1,200°F, some yielding must occur. The testing (4) showed that if yielding occurs, leakage is very likely.

Table A-3 shows the yield strengths assumed for this work.

Table A-3—Yield Strengths

Temperature °F	SA193 B7[a]	A105 GrII[b]	API TP2 4130[c]	410 SS[d]	SA453 Gr660[e]	316 SS[f]
70	105	36	60	60	85	30
500	—	29.1		51.1	85	19.9
700	86.3	—	49.3	—	85	18.1
800	78.15	23.0		46.5	85	17.6
900	70	20.0	40	41.8	85	17.3
1,000	48.5	18.0	27.7	35.4	85	17.1
1,100	27.0	13.0	15.4	27.5	85	17.0
1,200	15.8	10.0	9.03	18.9	85	—
1,300	4.6	6.0	2.63	12.0	85	—
1,350	—	—	—	—	69	—
1,400	—	3.0	—	8.6	—	—
1,500	—	—	—	6.6	32	—

[a]Reference (11).
[b]Reference (12).
[c]Follows B7, corrected for *y*.
[d]Reference (13).
[e]Reference (14).
[f]SA 182 F316.

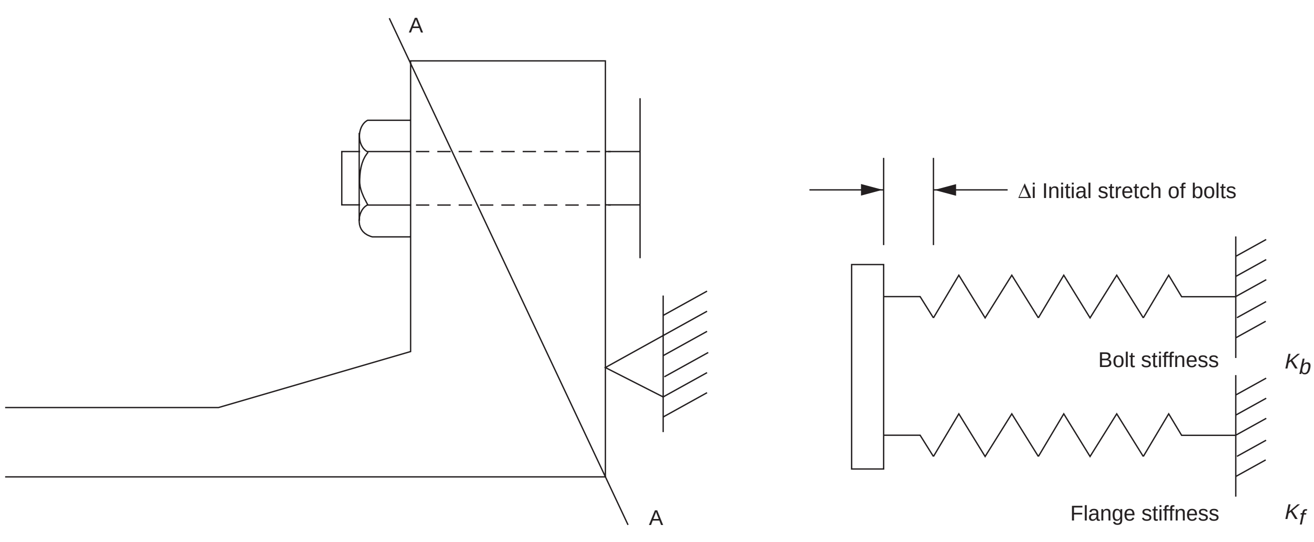


Figure A-2—Idealized Model of Flanged Joint

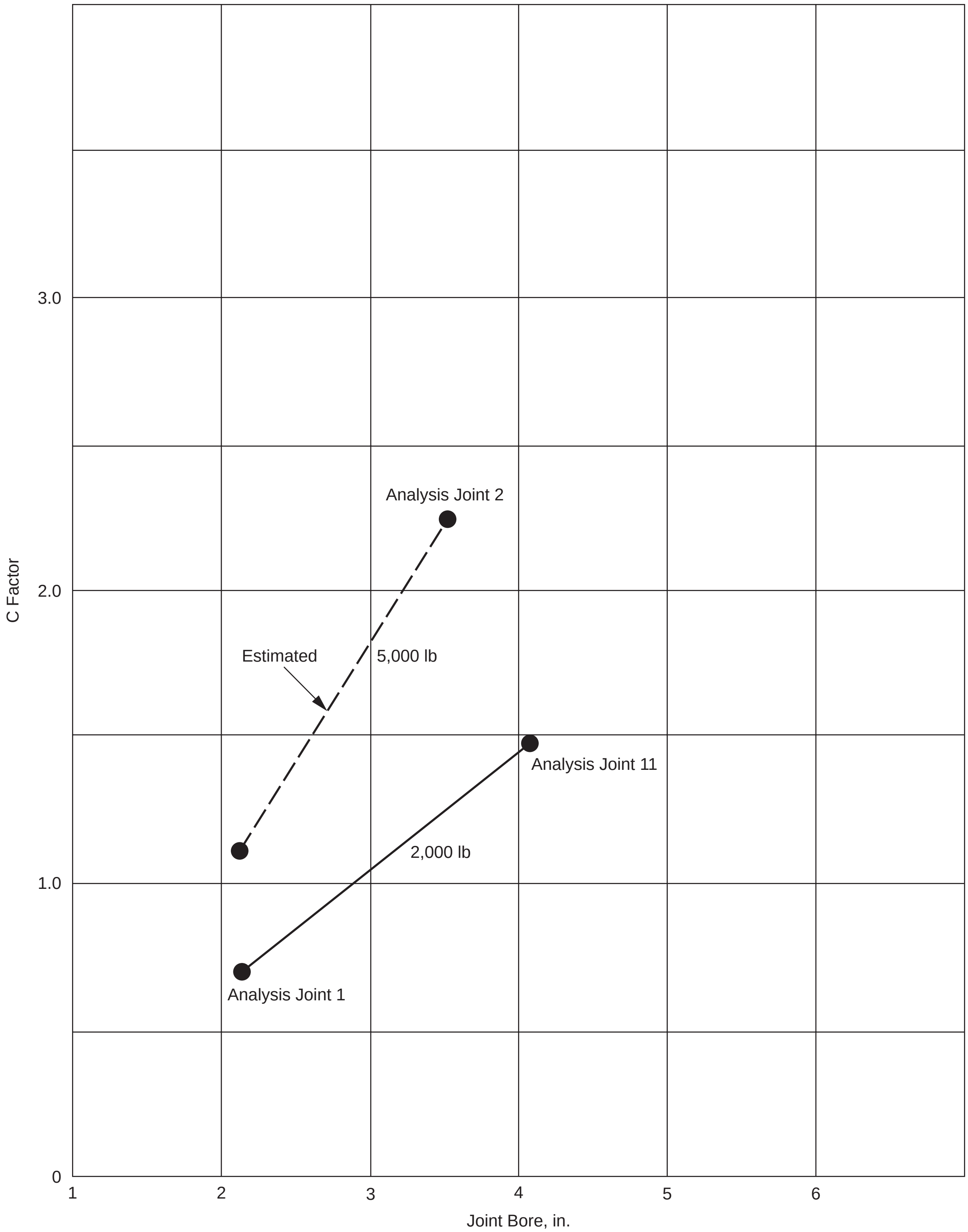


Figure A-3—C Factor for Equation A-5, 6B Flanges

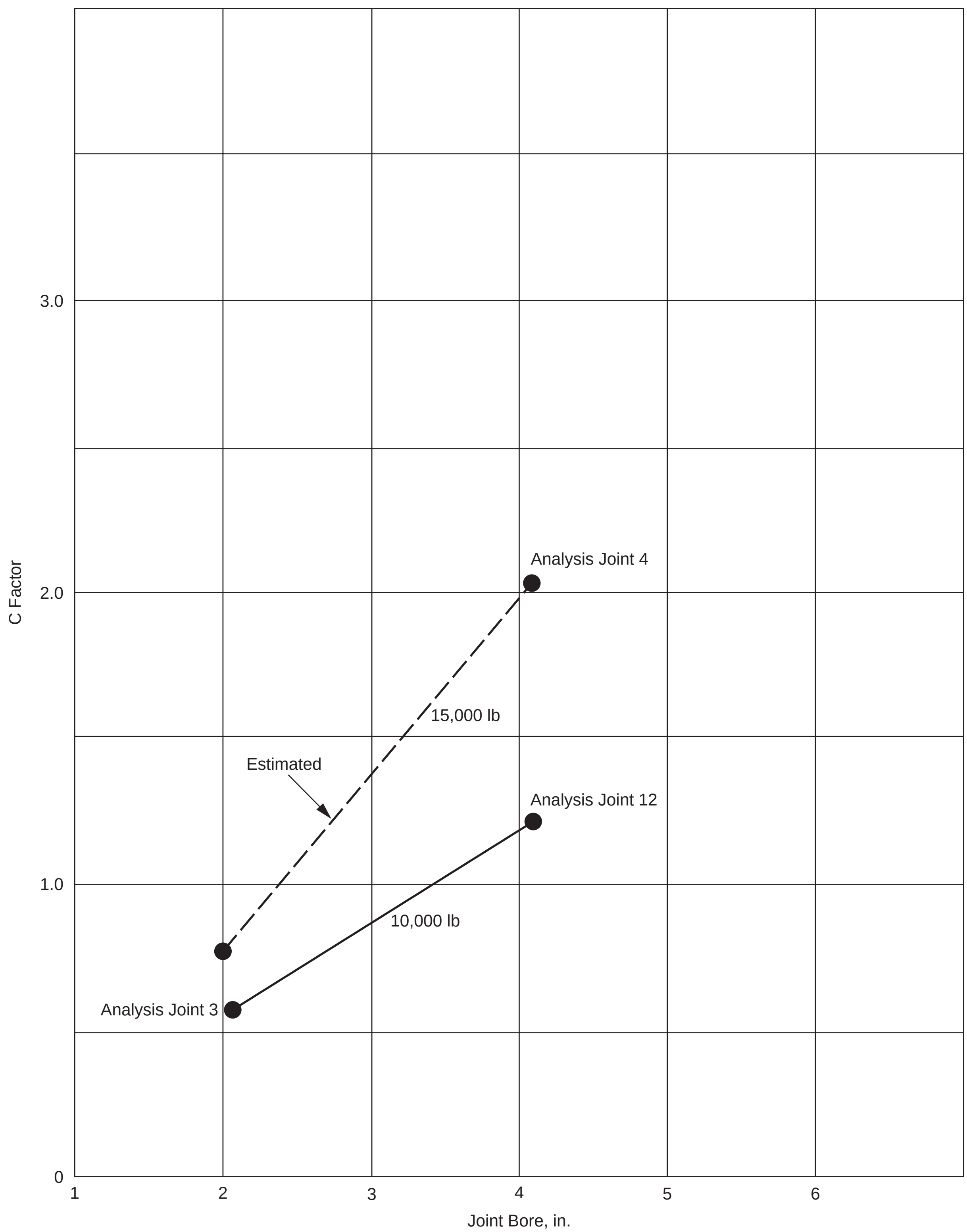


Figure A-4—C Factor for Equation A-5, 6BX Flanges

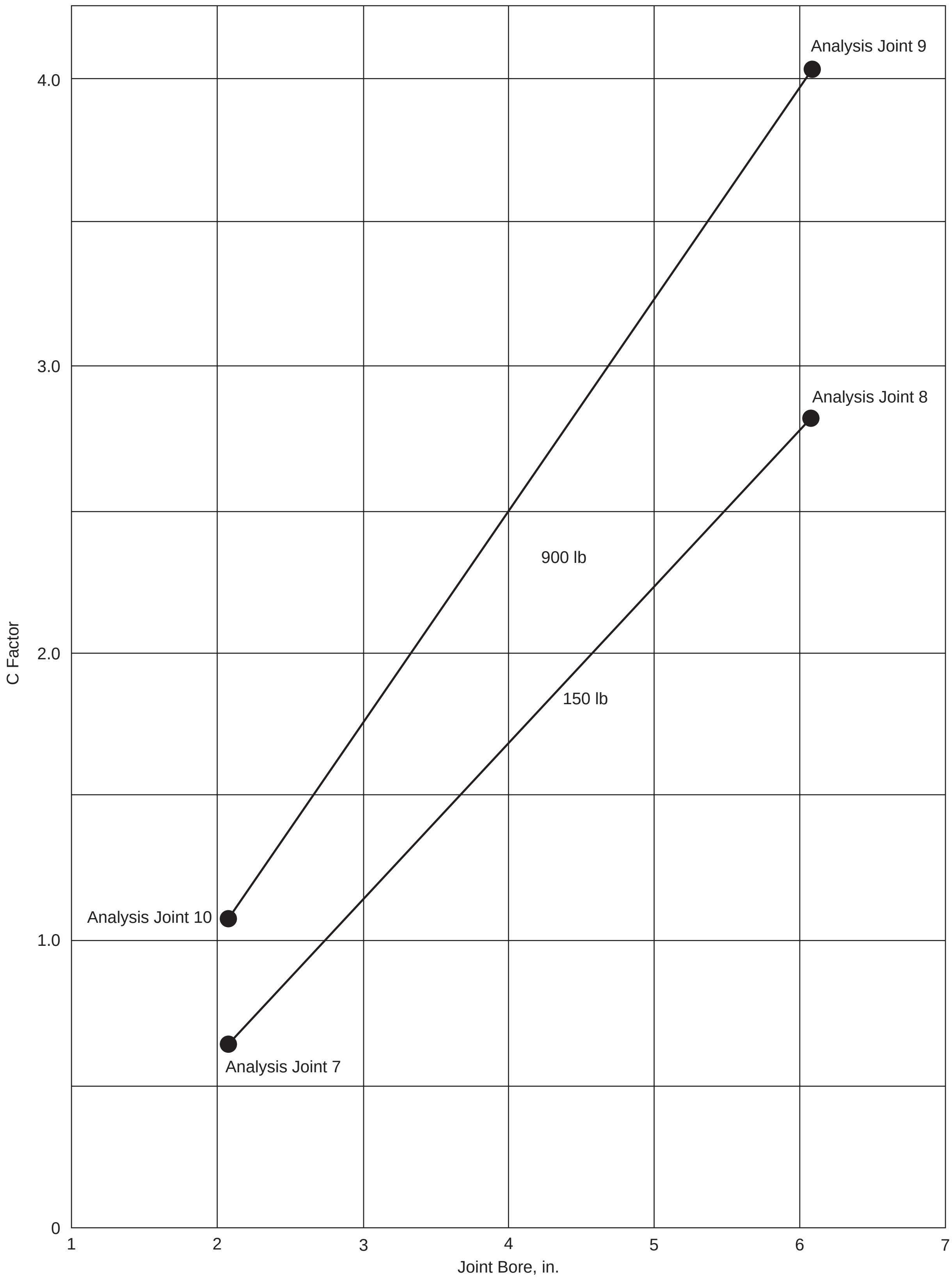


Figure A-5—C Factor For Equation A-5, ANSI Flanges

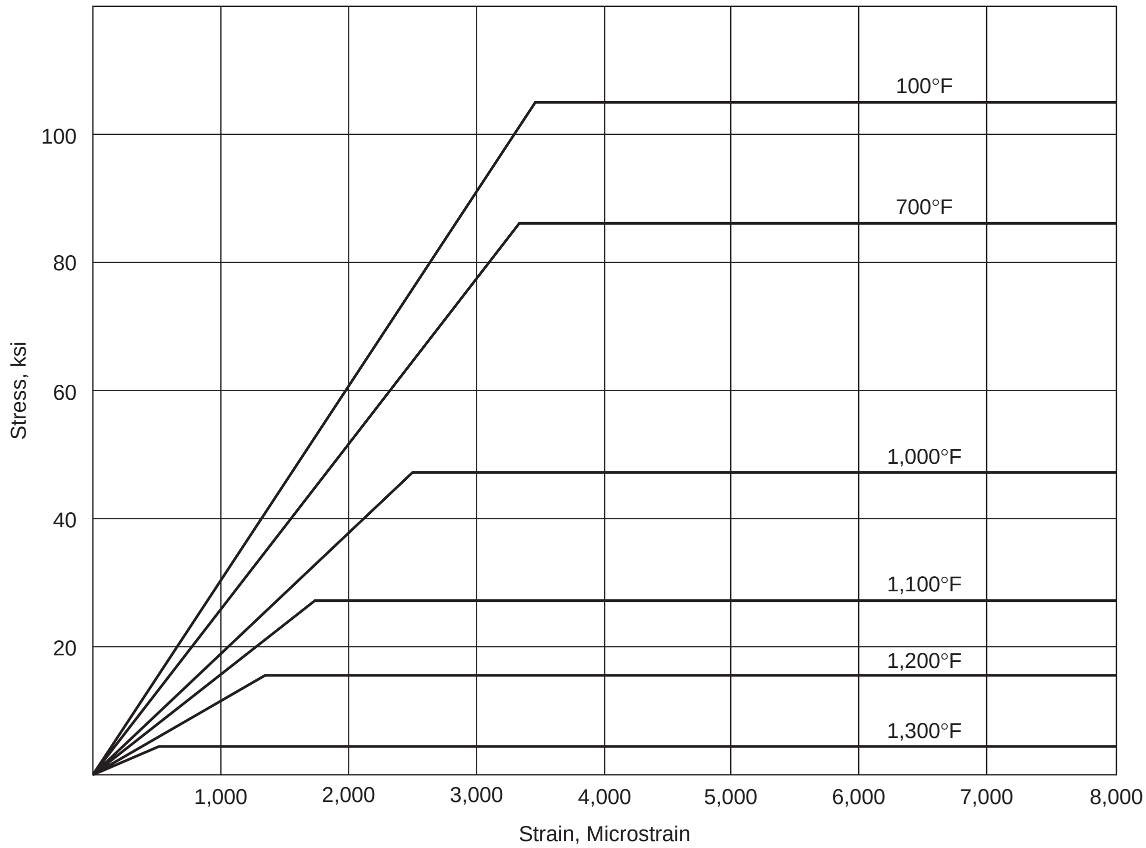


Figure A-6—Stress/Strain SA 193 B7

Expansion Coefficient Difference Between Bolts and Connectors

One potentially significant variable not included in standard stress analysis procedures for flanges or clamps is a difference in thermal expansion coefficients between bolts and flanges or between bolts and clamps.

Flanges

For flanges, initial makeup represents stretch of the studs to a given amount depending on the preload. If the flange and bolts do not expand at the same rate during the fire test, the joint can either be loosened, or the bolts overstressed, by the expansion difference. To accurately analyze these effects, a finite element analysis should be performed on a case-by-case basis. However, an approximate analysis can be used to predict general results that should be representative of the influence of differential expansion.

Referring to Figure A-2, a difference in expansion between bolts and flanges results in a change in bolt force in both the bolt "spring" and the flange "spring." Analysis work has consistently shown that the typical bolt and flange geometries, the flange stiffness K_f is 5 to 20 times greater than the bolt stiffness. Therefore, as a first approximation, it can be assumed that all of the differential expansion goes to change the load in the bolt.

The initial stretch, Δ_i, of the bolt is:

$$\Delta i = \frac{\sigma_b L}{E_{RT}} \tag{A-7}$$

where

σ_b = initial bolt stress, psi,

L = bolt length between nuts, inches,

E_{RT} = room temperature modulus of elasticity, psi.

The change in bolt stretch, Δ_c, due to differential expansion between the bolt and flange is:

$$\Delta_c = (\alpha_f - \alpha_b)(L)(\Delta T) \tag{A-8}$$

where

α_f = coefficient of thermal expansion of flange, in./in./EF,

α_b = coefficient of thermal expansion of bolt, in./in./°F,

ΔT = temperature change of the joint due to the fire.

When the bolt expands more than the flange ($\alpha_b > \alpha_f$) the bolt will lose all its preload when $\Delta_i + \Delta_c = 0$.

When this occurs

$$\frac{\sigma_b L}{E_{RT}} + (\alpha_f - \alpha_b)(L)\Delta T = 0 \qquad \text{(A-9)}$$

Solving for ΔT

$$\Delta T = \frac{\sigma_b}{E_{RT}(\alpha_b - \alpha_f)} \qquad \text{(A-10)}$$

As an example, consider 410 SS bodies with SA 453 660 bolts.

Taking α_b = 52,500 psi, E_{RT} = 31.7 × 106 psi, the following expansion coefficients from Table A-2 are produced along with the calculation of *T* to cause all bolt preload to be lost.

Temperature	α_f	α_b	ΔT
°F	in./in./°F × 10[6]		°F
500	5.98	8.24	733
600	6.48	8.82	708
750	6.64	9.11	671
800	6.67	9.17	662
1,000	6.79	9.41	632

Assuming the joint was made up at 70°F, at a joint temperature of about 750°F (671° + 70° = 741°) the bolts would lose all initial preload.

If the gasket used is a nonpressure energized gasket such as a Type R, then leakage would be expected. If it is a gasket such as the BX, which can accommodate some movement, leakage would not occur until a high temperature is reached. These calculations are shown in Appendix C.

APPENDIX B—ANALYSIS OF CLAMP TYPE CONNECTORS

The same equations and techniques for flanges discussed in Appendix A apply to the clamp-type connections as well. In this case, however, the ΔT refers to the difference in average temperature between the clamp and hub as opposed to flanges where the ΔT is the temperature gradient across the flanges. These temperature differences can be found from Figure B-1.

Instead of using C factors to predict the preload loss, the procedures given in Reference (16) will be used, which are described in the following:

T_{clamp} = Average connection temperature + $\Delta T/2$

T_{hub} = Average connection temperature – $\Delta T/2$

From Figure B-2, the radial differential thermal expansion due to gross axial expansion of the connector is calculated as:

$$\begin{aligned}\Delta r_a &= \Delta X \cos 25° \sin 25° \\ &= (\Delta X_{hub} - \Delta X_{clamp}) \cos 25° \sin 25° \quad \text{(B-1)} \\ &= [\alpha_{hub}(T_{hub} - 70°)L - \alpha_{clamp}(T_{clamp} - 70°\text{F})L] \\ &\quad \cos 25° \sin 25°\end{aligned}$$

where α_{hub} and α_{clamp} are the thermal expansion coefficients of hub and clamps at their respective temperatures.

Similarly for gross radial thermal expansion of the connector:

$$\begin{aligned}\Delta r_r &= \Delta y \sin 225° \\ &= (\alpha_{hub}\, T_{hub} - \alpha_{clamp}\, T_{clamp}) R_m \sin 225° \quad \text{(B-2)} \\ &= [\alpha_{hub}(T_{hub} - 70°) - \alpha_{clamp}(T_{clamp} - 70°)] \\ &\quad R_m \sin 225° \quad \text{(B-3)}\end{aligned}$$

The total radial differential thermal expansion is:

$$\Delta r = \Delta r_a + \Delta r_r \quad \text{(B-4)}$$

The clamp hoop strain is:

$$\varepsilon_h = \frac{\Delta r}{R_m} \quad \text{(B-5)}$$

The hoop tension force is:

$$F_h = \varepsilon_h A_{clamp} E_{eff} \quad \text{(B-6)}$$

where E_{eff} is the effective hoop stiffness of clamp/bolt assembly and is derived from Appendix D.

The axial clamping load loss due to the thermal differential between clamp and hub is:

$$F_a = \frac{\pi F_h}{\tan 25°} \quad \text{(B-7)}$$

This preload loss can then be used in Equation A-6 to determine the remaining retaining load on the gasket.

To perform a stress analysis of the clamp-type connection, the procedures referenced in (15) should be used. An example run of the AWHEM computer program is illustrated in Figure B-3.

The differential thermal expansion between clamp and bolts also affects the preload changes and this effect was not included in the computer program. The calculation of preload changes due to this effect is rather straightforward and is illustrated in the following for 5,000 psi, 410 stainless steel connection size $2^{1}/_{16}$ with SA 453 Gr 660 bolts.

Assuming that clamps and bolts are at the same temperature in a fire environment, the differential thermal expansion between the clamp and bolt is:

$$\Delta L = (\alpha_c - \alpha_b) L_b \Delta T \quad \text{(B-8)}$$

where

ΔL = thermal expansion differential, in.,

α_c = thermal expansion coefficient of $2^{1}/_{16}$ in clamp (6.79 in./in./°F $\times 10^{-6}$ @ 1,000°F),

α_b = thermal expansion coefficient of $^{7}/_{8}$ in. bolt (9.41 in./in./°F $\times 10^{-6}$ @ 1,000°F),

L_b = bolt length, 5.25 in.,

ΔT = temperature increase from ambient temperature,

ΔL = $(9.41 - 6.79)(5.25)(1{,}000 - 70)(10^{-6})$,

= 0.012792 in.

Considering the clamp/bolt assembly as springs in series, then the equivalent spring rate of the assembly K_e, is:

$$K_e = \frac{K_c K_b}{K_c + K_b} \quad \text{(B-9)}$$

where

$$\text{clamp } K_c = \frac{A_c E_c}{L_c} \quad \text{(B-10)}$$

where

A_c = the clamp cross-section area in hoop direction,

E_c = Young's Modulus at temperature,

L_c = Circumferential length of clamp,

$$K_c = \frac{(3.023)(18.6)10^6}{2\pi(3.15625)},$$

$$= 2.83 \times 10^6 \text{ lb/in.},$$

bolts; $K_b = \frac{A_b E_b}{L_b} = \frac{(0.4263)(26.3)10^6}{5.25} =$

2.135×10^6 lb/in.

Therefore

$$K_c = \frac{K_c + K_b}{K_c K_b} = 1.21 \times 10^6 \text{ lb/in.}$$

The reduction of hoop force as a result of clamp/bolt differential thermal expansion is:

$$F_h = K_e \Delta L = 15.566 \text{ lb}$$

The reduction of axial clamp load is (reference Equation (B-7):

$$F_a = \frac{\pi F_h}{\tan 25°} = 104{,}870 \text{ lb}$$

This is greater than the initial preload of 89,332 lb, and therefore the makeup preload is completely lost and the connection will leak.

Similar calculations can be performed for other sizes of clamp type connectors by following the above procedures.

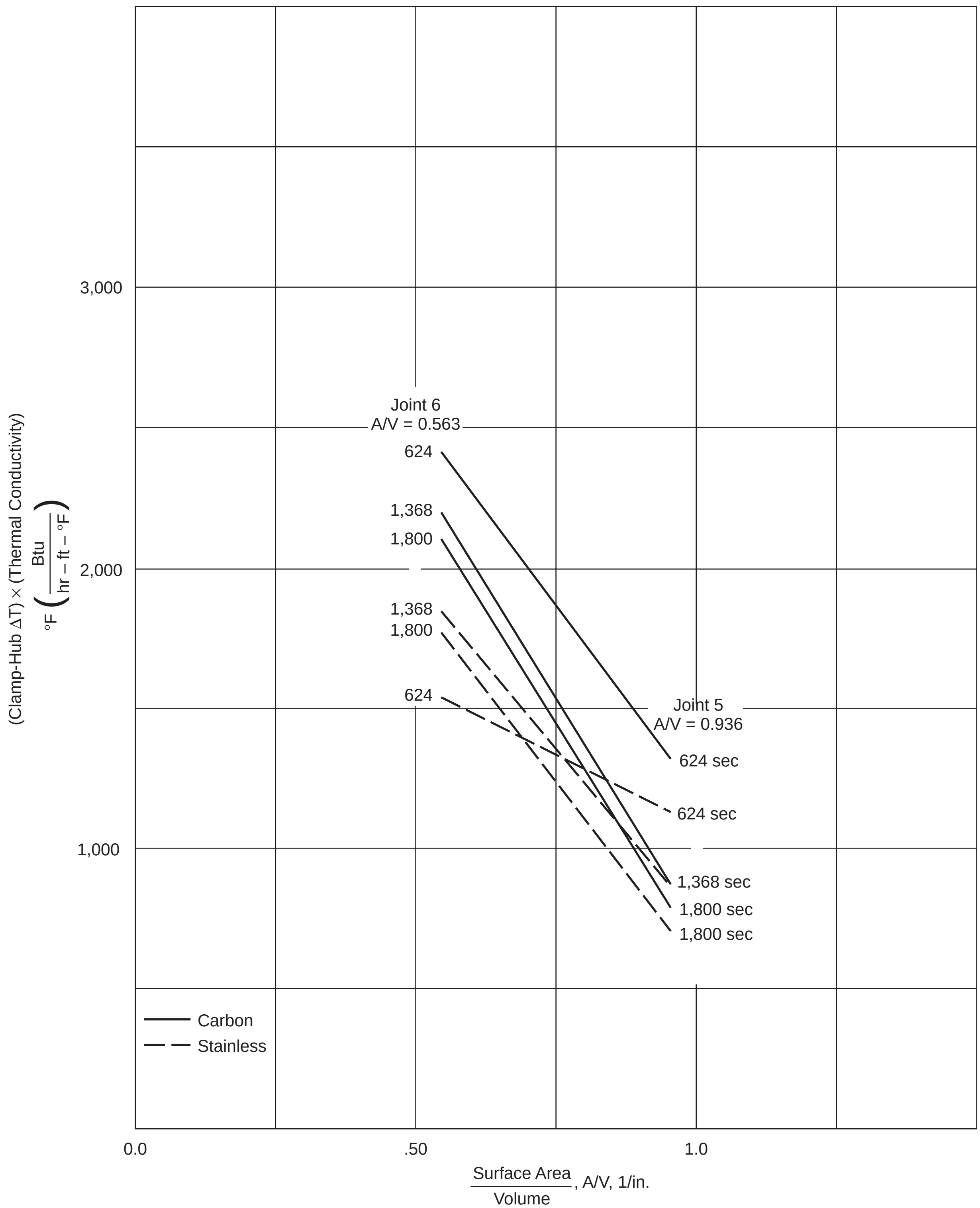
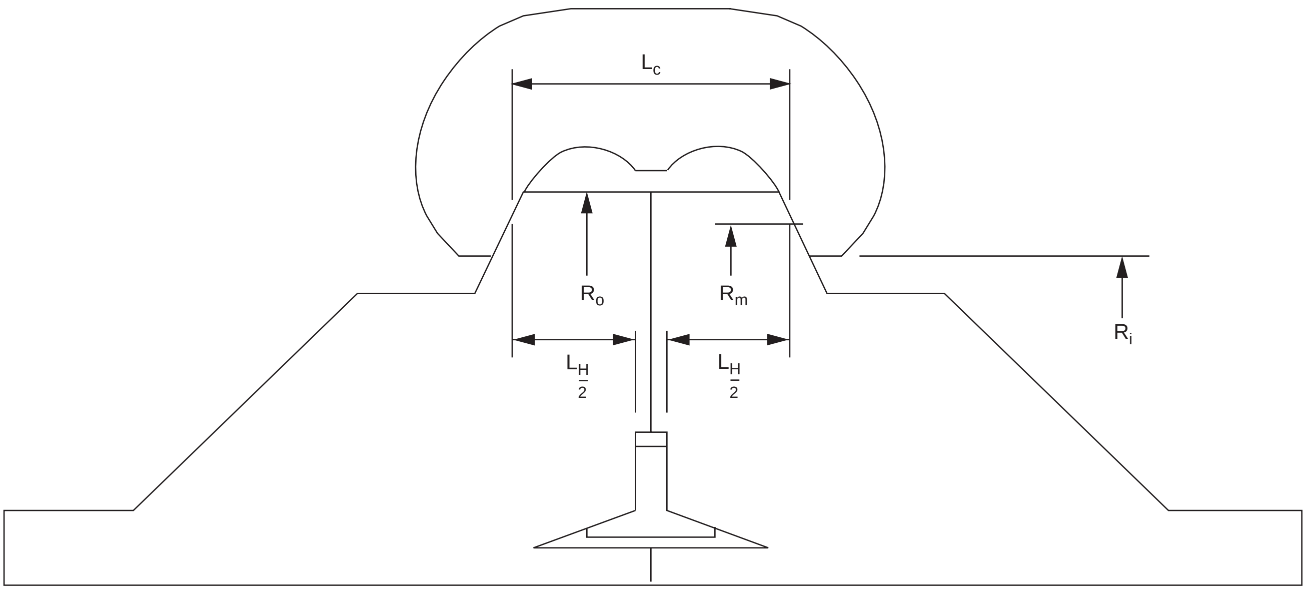


Figure B-1—Correlation of Temperature Difference Between Clamp and Hubs



Note: $L_c = L_h$ for nonrecessed hubs

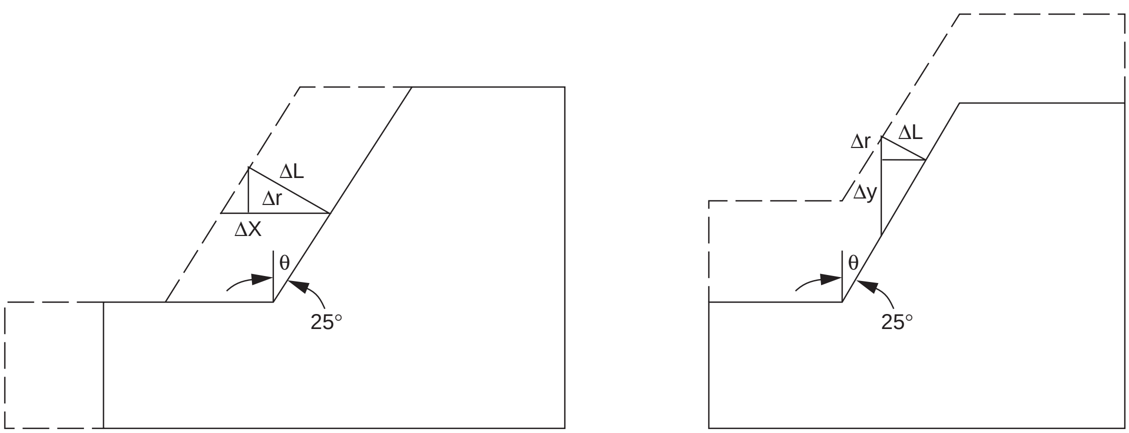


Figure B-2—Radial Expansion Differential Calculations

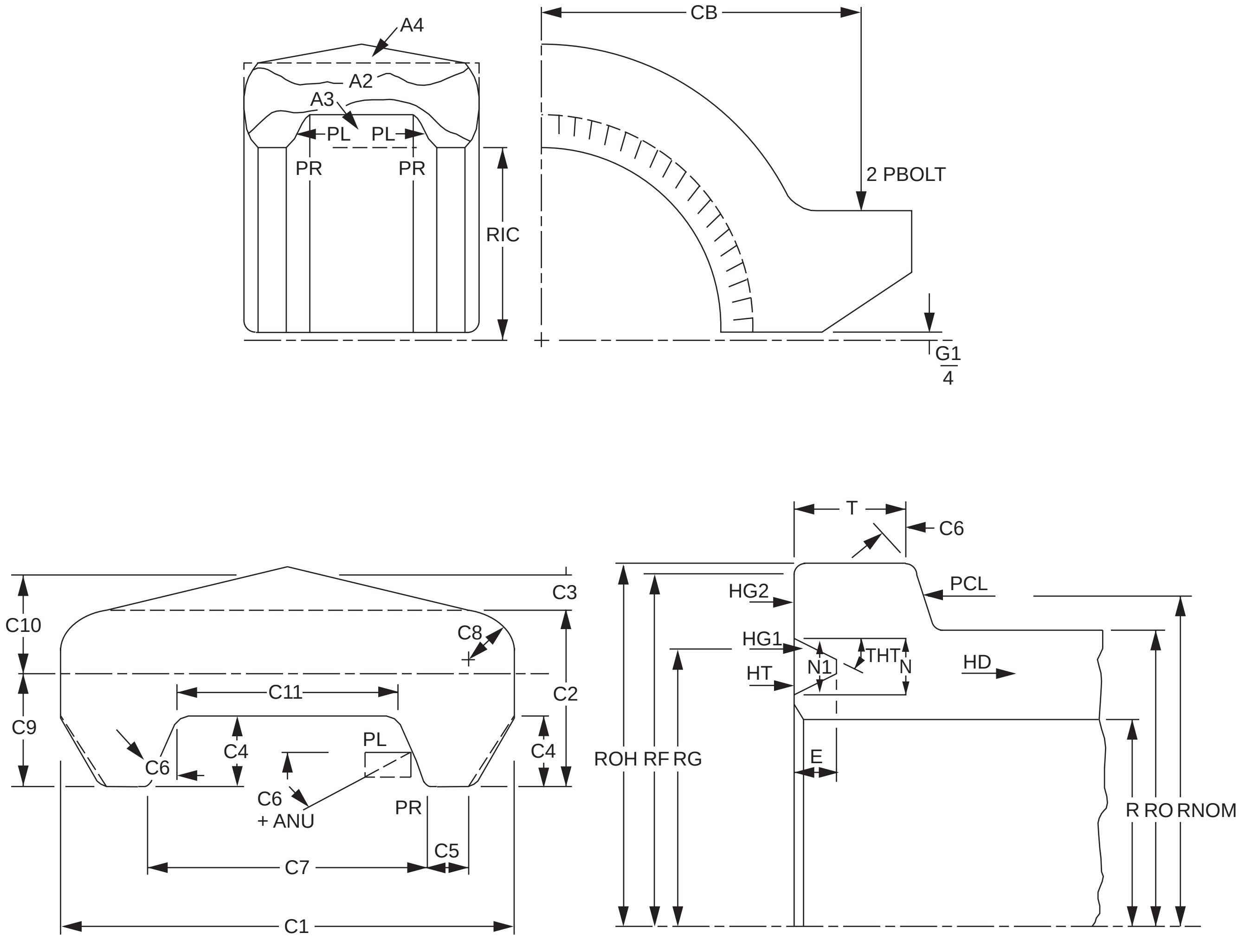


Figure B-3(a)—AWHEM Clamp-Type Connection Major Dimensions

Design Stress Calculations for Clamp Type Connectors							
$13^{5}/_{8}$—5 MSP AWHEM Connector							
Input Data—Geometry							
R =	6.812500	RO =	9.500000	RG =	7.710900	XN +	0.673300
THT =	23.000000	E =	0.553500	RIC =	9.625000	C6 =	25.000000
ROH =	10.312500	T =	1.866000	RF =	10.187500	C1 =	7.687500
C2 =	2.956200	C3 =	0.325100	C4 =	0.812500	C5 =	1.25000
C7 =	4.375000	C8 =	2.000000	G1 =	1.500000	G2 =	0.000000
C8 =	13.875000						
Input Data-Actions (Case No. 1 of 3)							
P =	5000.0	F =	0.0	PBOLT =	180600.0	PCL =	1911132.8
ANU =	5.7	HG1 =	78318.0	MX =	18344000.0		
Clamp Stresses							
SC1 =	159669.6	SC2 =	–51665.6	SC3 =	–121217.5	SC4 =	71640.2
SC5 =	–46461.7	SC6 =	–8966.6	SC7 =	–8455.0	SBR =	44381.1
Hub Stresses							
SH1 =	31601.9	SH2 =	–21015.6	SH3 =	–8701.3	SH4 =	8569.6
SH5 =	3735.6	SH6 =	15586.3	MMX =	4480215.0	SH7 =	143480.1
Input Data-Actions (Case No. 2 of 3)							
P =	5000.0	F =	0.0	PBOLT =	180600.0	PCL =	2433474.4
ANU =	0.0	HG1 =	78318.0	MX =	18344000.0		
Clamp Stresses							
SC1 =	159669.6	SC2 =	–51665.6	SC3 =	–121217.5	SC4 =	89966.5
SC5 =	–57906.4	SC6 =	–3164.6	SC7 =	–14256.9	SBR =	56511.1
Hub Stresses							
SH1 =	31601.9	SH2 =	–21015.6	SH3 =	–8701.3	SH4 =	8569.6
SH5 =	3735.6	SH6 =	15586.3	MMX =	7083756.0	SH7 =	143480.1
Input Data-Actions (Case No. 3 of 3)							
P =	5000.0	F =	0.0	PBOLT =	180600.0	SH4 =	8569.6
ANU =	–5.7	HG1 =	78318.0	MX =	18344000.0		
Clamp Stresses							
SC1 =	159669.6	SC2 =	–51665.6	SC3 =	–121217.5	SC4 =	118275.1
SC5 =	–75584.9	SC6 =	5797.5	SC7 =	–23219.1	SBR =	75248.2
Hub Stresses							
SH1 =	31601.9	SH2 =	–21015.6	SH3 =	–8701.3	SH4 =	8569.6
SH5 =	3735.6	SH6 =	15586.3	MMX =	11105426.0	SH7 =	143480.0

Figure B-3(b)—AWHEM Clamp-Type Connection Computer Output

APPENDIX C—RETAINING LOADS AND UNSEATING CONDITIONS FOR BX GASKETS

The BX gasket is sketched in Figure C-1. The dimensions *A*, *N*, *C*, *OD*, and *G* are from Table 904.3 of API Spec 6A, 16th Edition (6).

The axial engagement, *X*, of the seat in the seal is given by

$$N = 2X \tan 23° + C \tag{C-1}$$

$$X = \frac{N - C}{2 \tan 23°}$$

The seat radius at the contact point

$$= \frac{G}{2} - X \tan 23° = R_{\text{final}} \tag{C-2}$$

The radial interferences is the difference between this radius and the origin radios of the seal at that point.

$$R_{\text{orig}} = \frac{OD}{2} - \frac{A - C}{2} \tag{C-3}$$

The radial interference *I*, is then

$$R_{\text{orig}} - R_{\text{final}}$$

The retaining load, F_a is determined using the following equation:

$$F_a = \left[2\pi \frac{E(AR)I}{R} + \frac{1}{2} PH\pi I.D.\right]\beta \tag{C-4}$$

where

E = elastic modulus at temp (modified to consider yielding),

AR = $^1/_2$ ring cross-sectional area = $^1/_2$ *AH*,

I = radial interference (considering thermal effects),

R = mean radius,

P = internal pressure,

H = ring height,

$I.D.$ = ring inner diameter = $OD - 2A$,

β = tan 23°.

To account for friction, simply modify β as shown below:

$$\beta = \frac{\sin 23° - \mu \cos 23°}{\cos 23° + \mu \sin 23°}$$

μ = coefficient of friction

There are two important cases to consider:

μ = 0 then β = tan 23° as before,

μ = tan 23° then β = 0 i.e., no axial force is required to maintain radial interference.

To consider yielding, replace *EI/R* in equation C-4 with the yield strength. Yielding occurs if *EI/R* is greater than the yield strength of the seal.

To consider gasket movement due to a differential expansion between the flange and bolts (bolts expand more), develop a stress strain curve for the gasket. The seal strain corresponding to the yield strain at temperature is that available for elastic recovery to allow the gasket to maintain contact with the seat as the seat pulls away axially. The seat and flange will be forced by pressure to follow the bolt as it expands.

This expansion can be calculated from

$$\Delta L = \frac{1}{2}(Lb)(\alpha_b - \alpha_f)(\Delta T) \tag{C-5}$$

where

Lb = effective bolt length,

α_b = coefficient of thermal expansion of bolt,

α_f = coefficient of thermal expansion of flange,

T = temperature difference between ambient and seat/seal temperature.

The axial differential expansion, ΔL, has an equivalent radial component, ΔR_t, of

$$\Delta R_t = \Delta L \tan 23°$$

The hoop strain, corresponding to elastic recovery, required for the seal to follow the seat is

$$\varepsilon = \frac{\Delta R_t}{R} \tag{C-6}$$

If this is greater than σ_y/E (yield strength divided by modulus of elasticity) both at temperature, there is not enough elastic springback available to cause the seal to follow the seat as it retracts and leakage is likely.

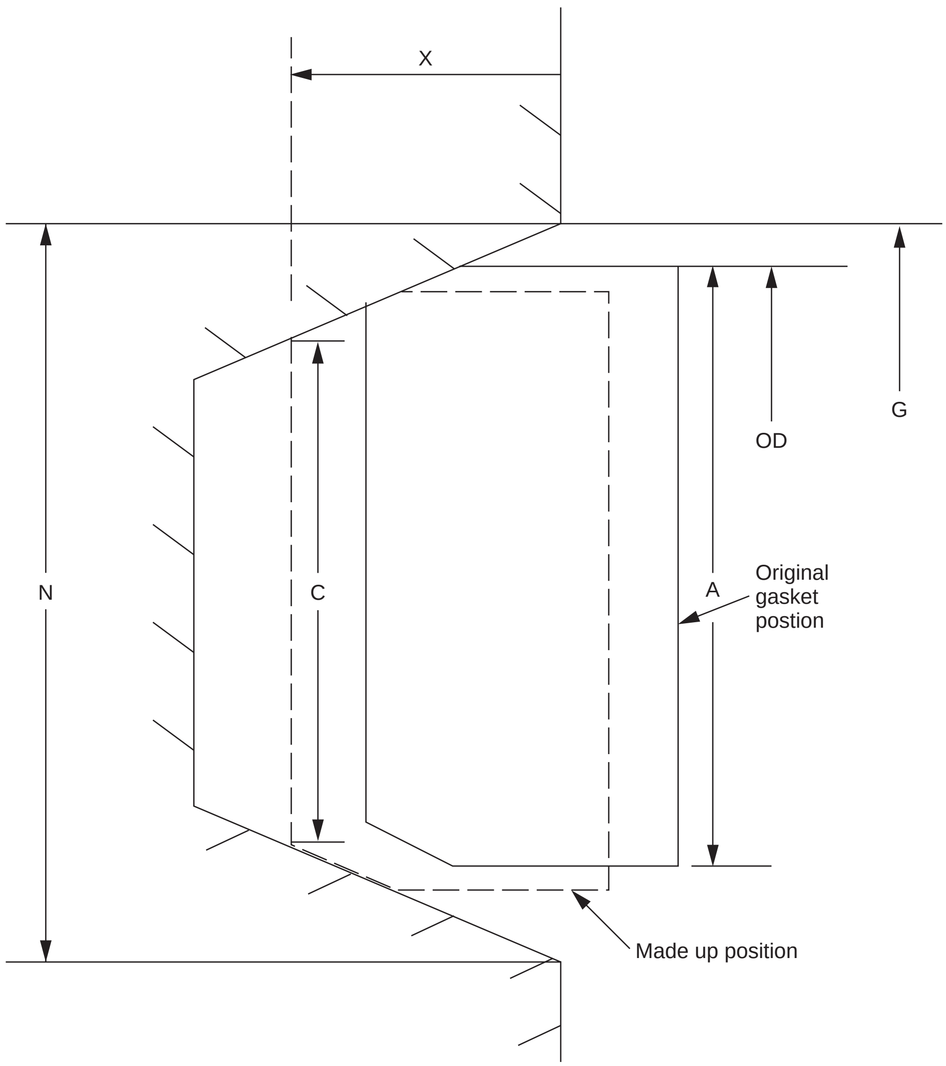


Figure C-1—Sketch of BX Gasket

APPENDIX D—EFFECTIVE CLAMP STIFFNESS CALCULATIONS

To represent a clamp-bolt assembly by a fully axisymmetric clamp cross-section, the equivalent hoop stiffness of the clamp must be determined first. This equivalent hoop stiffness can then be incorporated into the clamping load calculations by utilizing an effective elastic modulus in the hoop direction.

Consider the true hoop stiffness of the clamp-bolt assembly as shown by Figure D-1 loaded by an internal line load, P. The clamp outside hoop stress is approximated as:

$$\sigma_{ho} = \frac{2P\,R_i^2}{T_h(R_o^2 - R_i^2)} \quad \text{(D-1)}$$

where

R_i = the clamp inside radius,

R_o = the clamp outside radius,

T_h = an equivalent clamp thickness.

The equivalent clamp thickness is defined as:

$$T_h = \frac{A_c}{R_o - R_i} \quad \text{(D-2)}$$

where

A_c = the clamp cross-sectional area.

The radial strain is:

$$\varepsilon_r = \frac{\Delta R_o}{R_o} = \frac{\sigma_{ho}}{E} \quad \text{(D-3)}$$

where

E = Young's Modulus,

ΔR_o = the radial expansion of the clamp at the outside radius.

Solving for ΔR_o and inserting the expression for σ_{ho} yields:

$$\Delta R_o = \frac{2R_oPR_i^2}{T_h(R_o^2 - R_i^2)E} \quad \text{(D-4)}$$

The bolt stiffness, K_b, is:

$$K_b = \frac{A_bE}{L_b} \quad \text{(D-5)}$$

where

A_b = the bolt area per bolt,

L_b = the bolt length between nuts. The bolt elongation is:

$$\Delta L_b = \frac{F_b}{K_b} \quad \text{(D-6)}$$

where

F_b = the bolt force per bolt. Conservatively assuming the bolt force is equal to $\frac{1}{2} PR_o$, the bolt elongation may be written as:

$$\Delta L_b = \frac{PR_oL_b}{2A_bE} \quad \text{(D-7)}$$

Thus, the total circumferential growth C at the clamp outside diameter is:

$$\Delta C = 2\Delta L_b + 2\pi\Delta R_o \quad \text{(D-8)}$$

$$= \frac{1}{E}\left[\frac{PR_oL_b}{A_b} + \frac{4\pi R_oR_i^2P}{T_h(R_o^2 - R_i^2)}\right]$$

Recall that the total circumferential growth represented by a fully axisymmetric clamp body is:

$$\Delta C = \frac{4\pi R_oR_i^2P}{T_h(R_o^2 - R_i^2)E_{eff}} \quad \text{(D-9)}$$

where

E_{eff} = an equivalent Young's Modulus in the hoop direction. Therefore, the equivalent Young's Modulus is given as:

$$E_{eff} = E \times \left[\frac{\dfrac{4\pi R_i^2}{T_h(R_o^2 - R_i^2)}}{\dfrac{L_b}{A_b} + \dfrac{4\pi R_i^2}{T_h(R_o^2 - R_i^2)}}\right] \quad \text{(D-10)}$$

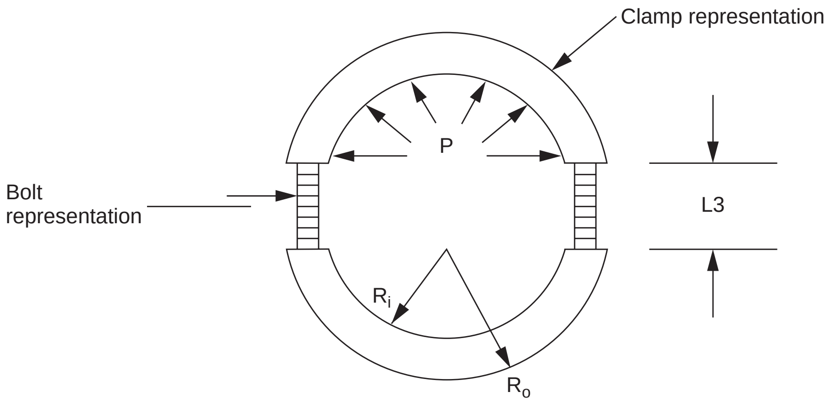


Figure D-1—Equivalent Clamp Hoop Stiffness

APPENDIX E—METRIC CONVERSIONS

E.1 English units are preferred in all cases and shall be standard in this specification. The following factors are from API Std 2564:

Length:	1 inch (in.)	= 25.4 millimeters (mm)
Pressure:	1 pound per square inch (psi)	= 0.06894757 bar inch (psi)
Stress:	1 pound per square inch (psi)	= 0.006894757 Megapascals (MPa)
Energy:	1 foot-pound (ft-lb)	= 1.355818 Joule (J)
Torque:	1 foot-pound (ft-lb)	= 1,355818 Newton-meter (N-m)
Mass:	1 pound mass	= 0.453524 kilogram (kg)

Temperature Conversion: The following formula may be used to convert degrees Fahrenheit (°F) to degrees Celsius °C):

$$°C = (5/9)\ (°F–32)$$

E.2 In addition to the above conversions, the designations PN for nominal pressure and DN for nominal diameter are sometimes used in the designation of valves. For the purposes of this specification, the PN designations relate to the pressure classes, and the DN designations relate to NPS, or nominal pipe sizes, as follows:

Class 150	= PN 20	Class 300	= PN 50
Class 400	= PN 64	Class 600	= PN 110
Class 900	= PN 150	Class 1500	= PN 260
Class 2500	= PN 420		
NPS 2	= DN 50	NPS $2^1/_2$	= DN 65
NPS 3	= DN 80	NPS 4	= DN 100

For NPS 4 and greater listed sizes, multiply the NPS by 25 to obtain the DN, except that there is no equivalent DN for NPS 36.

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