High-pressure High-temperature (HPHT) Flange Design Methodology

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Introduction

This document is intended to provide design guidance for high-pressure high-temperature (HPHT) API 6BX style flanges. The current revision of this document focuses on recommending methods for quantifying flange capabilities subjected to combinations of pressure, bending, tension, and thermal loads. It intends to expand upon the work documented in API Technical Report 6AF2 by recommending methods using more advanced analysis modeling, such as 3D geometry, nonlinear material models, and large displacement theory. It also provides guidance on initial sizing of flange geometry based on the work presented in Robert Eichenberg's ASME paper 57-PET-23 "Design Considerations for AWHEM 15,000 psi Flanges" of 1957 and his Journal of Engineering for Industry paper of 1964. Eichenberg's work established the foundations for the API 6BX style flange.

It is not the intent of this document to restrict users from performing project or application specific analyses that could provide capabilities different to those using the methodology summarized herein. Alternative methods may be acceptable if justified by alternative industry accepted design codes. When other industry-approved HPHT design methods are employed, the methodology presented here shall not be viewed as an extra requirement nor is it intended to supplant other industry-approved HPHT design methodologies.

The intent of this design guideline is to enable the user to generate baseline capability charts similar to those seen in API Technical Report 6AF2, but using nonlinear FEA models, methods, and criteria.

The methodology is demonstrated on the API 6BX 5 in. 15K flange in Annex B. Annex C contains capability charts for all the 20K API 6BX flanges using a possible interpretation of the method described in the guideline.

At the time of writing, not all methodologies in this document have been validated. Therefore, this document serves as an example of the types of calculations and considerations necessary to define capabilities of API 6BX flanges.

Fatigue is intended be added to this document later under a future revision.

High-pressure High-temperature (HPHT) Flange Design Methodology

1 Scope

The scope of this document is to provide design guidelines for API 6BX style flanges used as end and outlet connectors in high-pressure, high-temperature (HPHT) surface and subsea applications. For this document, HPHT applications are intended to mean flanges assigned a temperature rating greater than 350 °F or a pressure rating greater than 15,000 psi.

Service temperature ratings above 550 °F (288 °C) are outside the scope of this document.

2 Normative References

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any addenda) applies.

API Specification 6A, *Specification for Wellhead and Tree Equipment*

API Standard 6X, *Design Calculations for Pressure-containing Equipment*

API Technical Report 6AF, *Technical Report on Capabilities of API Flanges under Combinations of Load*

API Technical Report 6AF1, *Technical Report on Temperature Derating on API Flanges under Combination of Loading*

API Technical Report 6AF2, *Technical Report on Capabilities of API Integral Flanges under Combination of Loading—Phase II*

API Technical Report 17TR8, *High-Pressure High-Temperature Design Guidelines*

ASME, *Boiler and Pressure Vessel Code Section VIII, Division 2—Alternative Rules*

ASME, *Boiler and Pressure Vessel Code Section VIII, Division 3—Alternative Rules for Construction of High Pressure Vessels*

3 Terms, Definitions, Acronyms, Abbreviations, and Symbols

3.1 Definitions

For the purposes of this document, the following terms and definitions, and those terms and definitions in API 6A, apply.

3.1.1
Stiffness ratio
Stiffness of the bolt divided by the stiffness of the bolt plus the stiffness of the flange body.

NOTE   Refer to B.2.5.

3.2 Acronyms, Abbreviations, and Symbols

For the purposes of this document, the following acronyms, abbreviations, and symbols apply.

FEA   finite element analysis

G	groove
H	height
HPHT	high-pressure high-temperature
ID	inner diameter
LRFD	load and resistance factor design
OD	outer diameter
P	working pressure
σ_{allow}	allowable stress
σ_{hoop}	hoop stress at working pressure
Sm	Design Stress
KB	Stiffness of Bolt
KJ	Stiffness of Flange Body

4   Summary of Methodology

The methodology comprises four distinct phases:

1)   Select a gasket for the internal pressure requirements.

2)   Calculate the bolting and flange dimensions.

3)   Determine the flange's capabilities with regards to pressure, bending, tension, and thermal loading.

4)   Validate the flange design in accordance with an industry accepted specification.

Loading conditions addressed by this guideline are pressure, bending, tension, and thermal effects. Loading conditions not addressed by this guideline include torsion and transverse shear.

Failure modes addressed by this guideline are flange body plastic collapse, flange body excessive deformation, and bolt failure under monotonic (as opposed to cyclic) loading conditions.

Failure modes not verified by the guideline include leakage, local strain, and fatigue. According to PN 90-21, API 6BX flanges (or bolting) will likely be overstressed before 6BX gasket leakage. For this reason, this document does not assess the 6BX gaskets for leakage. This is based on 6BX gaskets maintaining critical sealing contact stress after hub-separation. It assumes that pressure is applied prior to external loading. These findings were supported by validation testing on 4 $^{1}/_{16}$ and 7 $^{1}/_{16}$ 6BX flanges. Application where flanges are subjected to external loading prior to pressurization may require additional considerations.

5   HPHT End Flange Initial Sizing Methodology

5.1   General

This section provides guidance on how to develop the initial sizing of an API 6BX style flange using traditional hand calculations. These calculations should not be considered as a requirement during the verification process. Furthermore, use of these calculations does not exempt verification and validation of the final design.

5.2   Gasket Selection

Due to the complex behavior of seals, it is not possible to have a universal model, method or criterion for their design and verification.

NOTE   See Annex A for an example of how an API 6BX gasket might be verified for its capability.

5.3   Calculate Bolting and Flange Dimensions

5.3.1   Introduction

The method for calculating the bolting and flange dimensions is a five-step process. Each step establishes a parameter or dimension as listed below.

— Hub Thickness (wall of pipe)

— Pressure Loading

— Size and number of bolts

— Raised face diameter

— Flange thickness

NOTE   See Annex B for an example of the five-step process.

5.3.2   Hub Thickness (Wall of Pipe)

The wall thickness of the hub may be determined using an appropriate pressure vessel wall calculation.

NOTE   Eichenberg used Lame's formula to determine the minimum wall thickness of the hub. The allowable tangential stress on the inner fiber at working pressure was limited to 50 % of the specified minimum yield strength of the flange body. While this method was acceptable at the time of his work, later advancements in pressure vessel technology have provided more robust means for proper wall sizing of a pressure vessel wall. Examples of these can be found in publications such as API Standard 6X or API Technical Report 17TR8.

5.3.3   Pressure Loading

Calculate the total pressure end load generated by internal pressure acting on the flange and gasket.

5.3.4   Size and Number of Bolts

5.3.4.1   Determine the minimum required total stud thread root area by dividing the total pressure end load generated by internal pressure (see 5.3.3) at hydrostatic shell test condition by 83 % of the minimum specified bolt yield strength.

5.3.4.2   Select the size and number of bolts such that total stud thread root area is not less than the required total stud thread root area. The number of bolts should be a multiple of 4.

NOTE   Calculating the bolt circle requires determining the flange dimensions and establishing practical clearances between the bolts, and between the bolts and the hub, to allow tool access. Refer to Taylor Forge's "Modern Flange Design" for nominal dimensions [2].

5.3.5   Raised Face Outside Diameter

The raised face outside diameter may be calculated using the load generated by the chosen bolting assembled to 50 % of bolt yield strength assembly stress based on stress area.

The total bolt preload divided by the annular area defined by the raised face outside diameter and the outer edge of the seal groove should be less than 30,000 psi.

5.3.6 Flange Thickness

The flange's stiffness ratio may be used to determine the minimum acceptable flange thickness. Flange stiffness ratio is an indicator of resistance to bolt fatigue failure and loss of preload due to stress-relaxation, localized yielding, embedment, and other bolting phenomena.

NOTE   All standard API 6BX flanges were assessed, and their stiffness ratios were found to range from 0.50 to 0.37. Therefore, a stiffness ratio of 0.50 or less is recommended for API 6BX style flanges. See Annex B for an explanation of how the stiffness ratio is calculated.

6 HPHT End Flange Design Verification Analysis

6.1   Determine Flange Capabilities

Nonlinear analysis methods shall be used to assess the following failure modes for the loads discussed in Section 4:

— Flange body plastic collapse

— Service Criteria as applicable

— Excessive bolt stress

The methods and criteria used to assess these failure modes are explained in 6.2, 6.3, and 6.4.

6.2   High Temperature Effects

The reduction of material strength shall be included for wellbore temperatures greater than 250 °F up to 350 °F.

For temperatures above 350 °F, a heat transfer analysis shall be performed with the results incorporated into a structural analysis where the strength curves are adjusted for temperature and the coefficient of thermal expansion is included to assess thermal loads.

NOTE   API Technical Report 6MET contains details of material strength reduction at elevated temperatures for many different materials.

6.3   Flange Body Plastic Collapse

The flange body shall have a design margin supported by the applicable API document as verification that it does not fail by plastic collapse.

The flange body working condition plastic collapse criterion should be based on load and resistance factor design (LRFD).

NOTE 1   If performed, the steps in 5.3 verify the flange design at hydrostatic shell test condition.

NOTE 2   Elastic analysis is acceptable when in conformance with API Standard 6X.

6.4   Service Criteria

Service criteria limit the potential for unsatisfactory performance. Unsatisfactory performance may result due to excessive deformation, leakage, loss of preload, or other service criteria, as applicable.

Additional checks may be needed to verify local deformations do not result in an excessive loss of preload that may impact the functionality or performance of the design.

Examples of excessive deformation include onset of gross plastic deformation or interference with surrounding components. Refer to API 6X, ASME BPVC VIII-2, and VIII-3 for additional details regarding service criteria.

6.5 Excessive Bolt Stress

6.5.1 Assessment

Bolts shall be assessed for their maximum tensile stress. Bolting tensile stress shall be evaluated using the API 6A 83 % yield stress criterion.

6.6 Design Validation

Validation testing shall be performed and documented in accordance with an industry accepted specification.

7   Results

Results shall be in the form of capability charts like those found in API Technical Report 6AF, API Technical Report 6AF1, and API Technical Report 6AF2.

NOTE   See Annex C for examples of API 6BX flange capability charts using the Annex B methodology.

Annex A
(informative)

Verifying Gasket Suitable for Internal Pressure Requirements

A.1   Introduction

This annex demonstrates two different methods (axisymmetric and 3D) for verifying API 6BX gaskets are suitable for internal pressure requirements. The gasket material assessed was 316SS. This assessment was performed using room temperature properties. The methods in this annex have not been validated.

A.2   FEA Model

A.2.1   Axisymmetric FEA Model

The seal groove was modelled with rigid lines representing the least material condition. A friction factor of 0.1 was applied to the contact surfaces (see Figure A.1). The 6BX gasket was modelled in the least material condition with a true stress-true strain material model representing the minimum specified yield strength of the gasket. Rigid lines were moved to represent a fully face-to-face flange make-up condition.

NOTE   The least material condition results in the lowest interference condition.
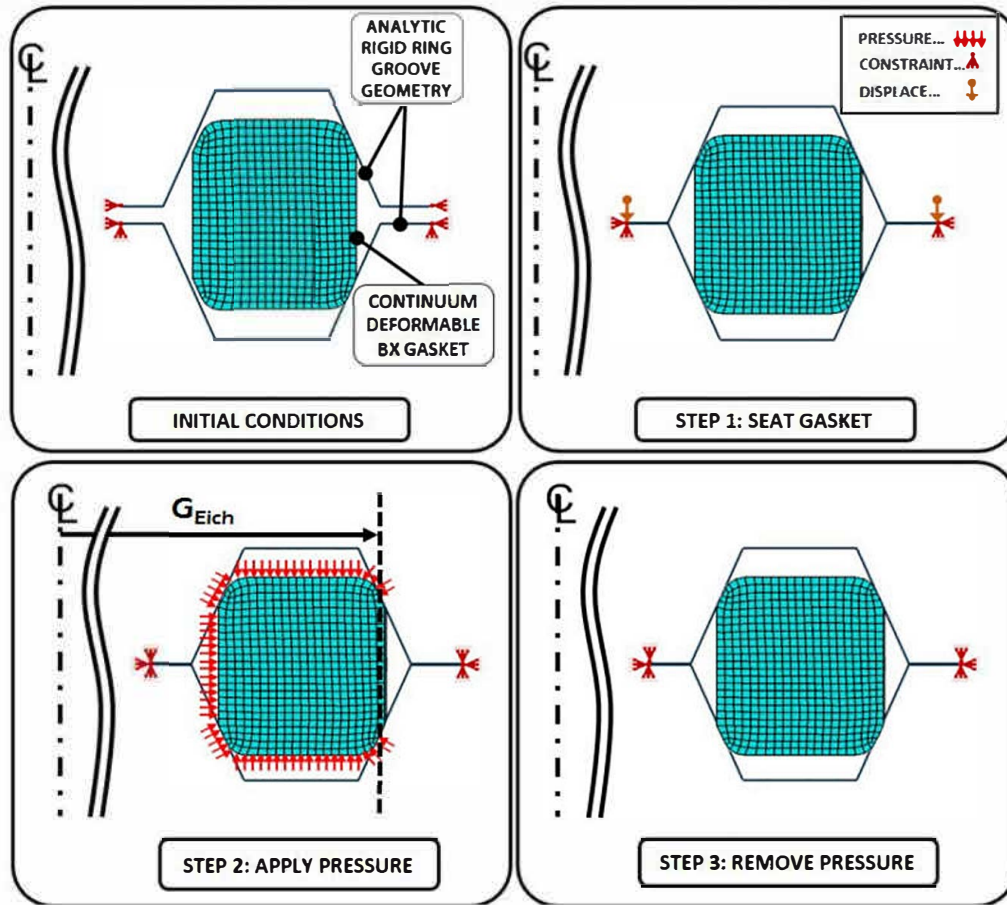
NOTE G_{Eich} is defined in Table B.2 of Annex B.

Figure A.1—Example Element Plot of 6BX Gasket—Initial Condition and Loading

A.2.2   3D FEA Model

A flange, gasket, and bolt were modeled using the least material condition (See Figure A.2). A friction factor of 0.1 was applied to the contact surfaces. Elastic-plastic material properties were applied to the flange 6BX flange to account for local yielding. Bolts were preloaded according to 50 % assembly stress for 95k bolts. True stress-true strain material models representing the minimum specified yield strength of the gasket and flange-were applied. A rigid surface was modeled at the horizontal plane of symmetry and contact was enforced between the rigid plane and the flange. Internal pressure and total pressure end load were applied to the flange when pressured. See Figure A.3 for an example of a Stress plot of a 3D model.
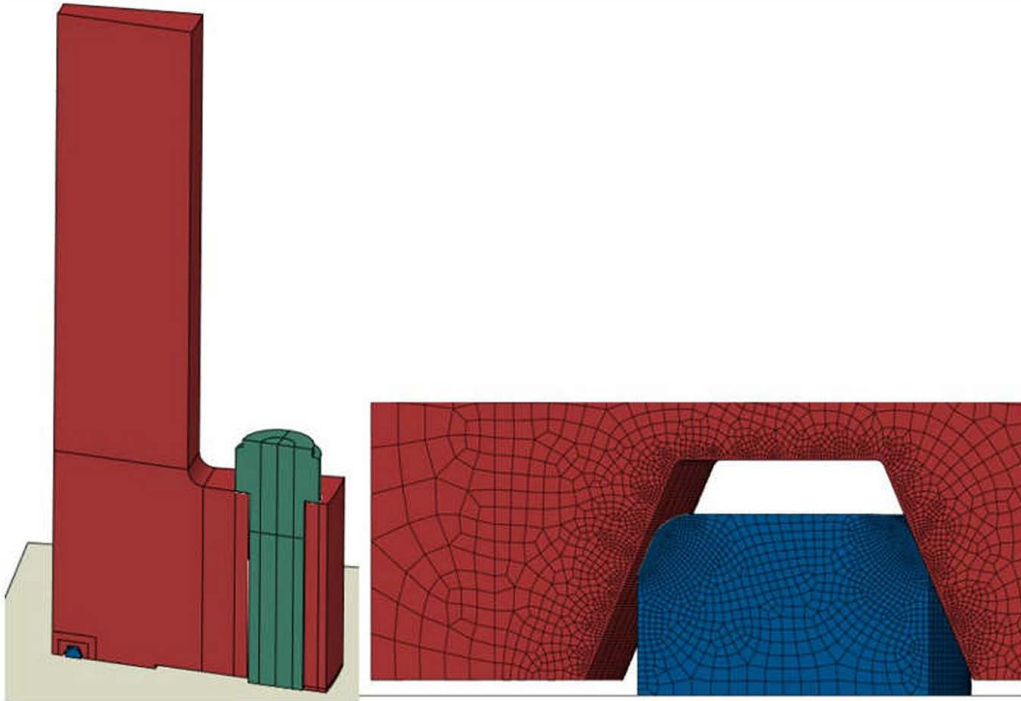
Figure A.2—3D Model of Flange Assembly

A.3   FEA Methods

The application of pressure loading was the same for both methods. Incremental pressures were then applied to the gasket from the inner diameter to the start of the outer sealing surface (see Figure A.1). Each incremental pressure was removed, and then contact force was recorded. This was repeated for each evaluated pressure. See Figure A.3 and Figure A.4 for a representative stress contour plot.
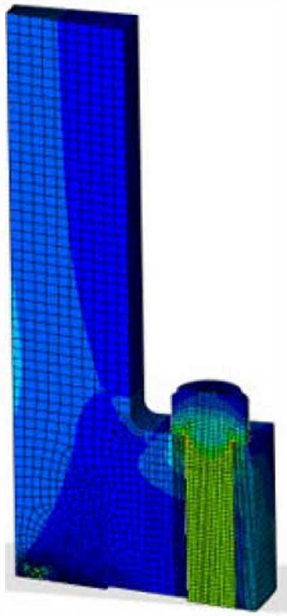
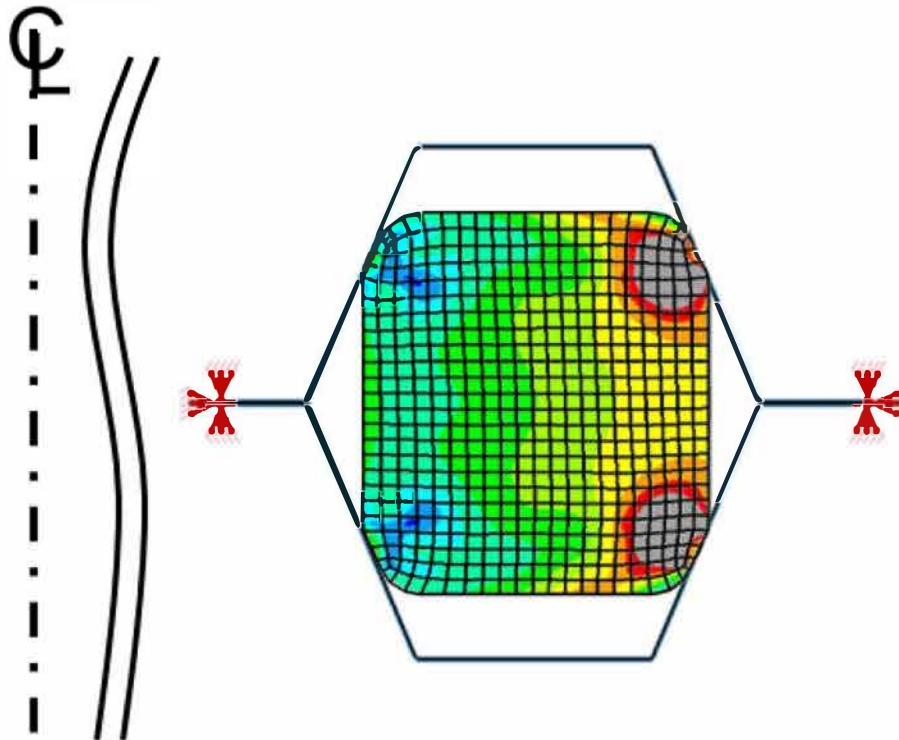**Figure A.3—Example Stress Plot of 3D Model**

Figure A.4—Example Stress Plot of Axisymmetric Model

A.4   Proposed Acceptance Criteria

All gaskets analyzed had similar behavior consisting of three regions when graphed (See Figure A.5 and Figure A.6). The first region of behavior is the upper shelf residual contact force. The second region of behavior is the reducing contact force region. The third region of behavior is the lower shelf residual contact force. Contact force for the rated working pressure of the gasket/flange assembly shall be in the upper shelf region, nearly equal to the subsequent gasket reaction force from 0 psi pressure applied. Subsequent contact force for the hydrostatic test pressure of the gasket/flange assembly shall be at least half the subsequent gasket reaction force from make-up only.

NOTE   These proposed acceptance criteria have not been validated by testing.
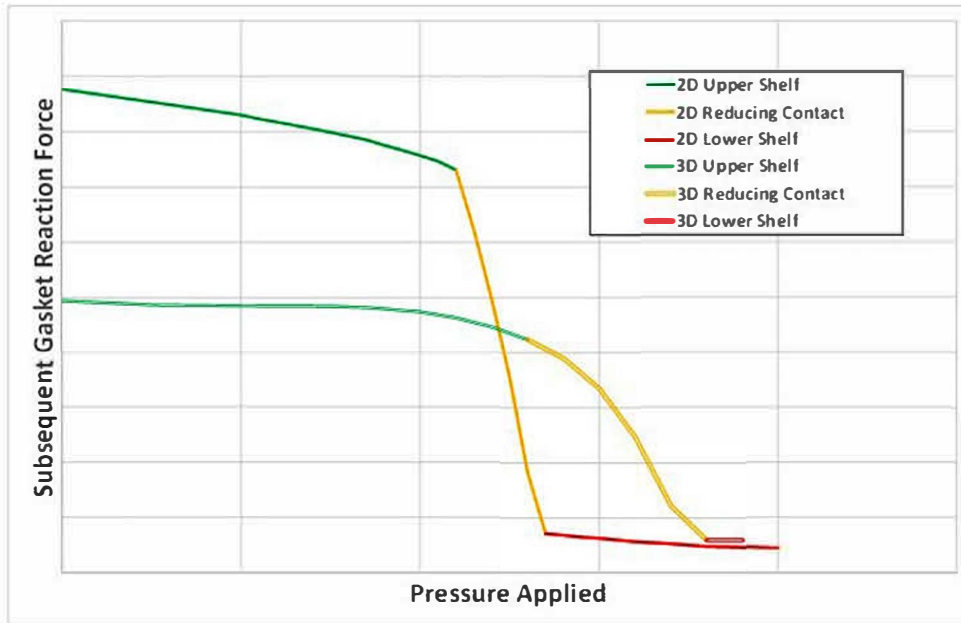

Figure A.5—2D and 3D Gasket Behavior Assessment

A.5   Results

A.5.1   Axisymmetric Results

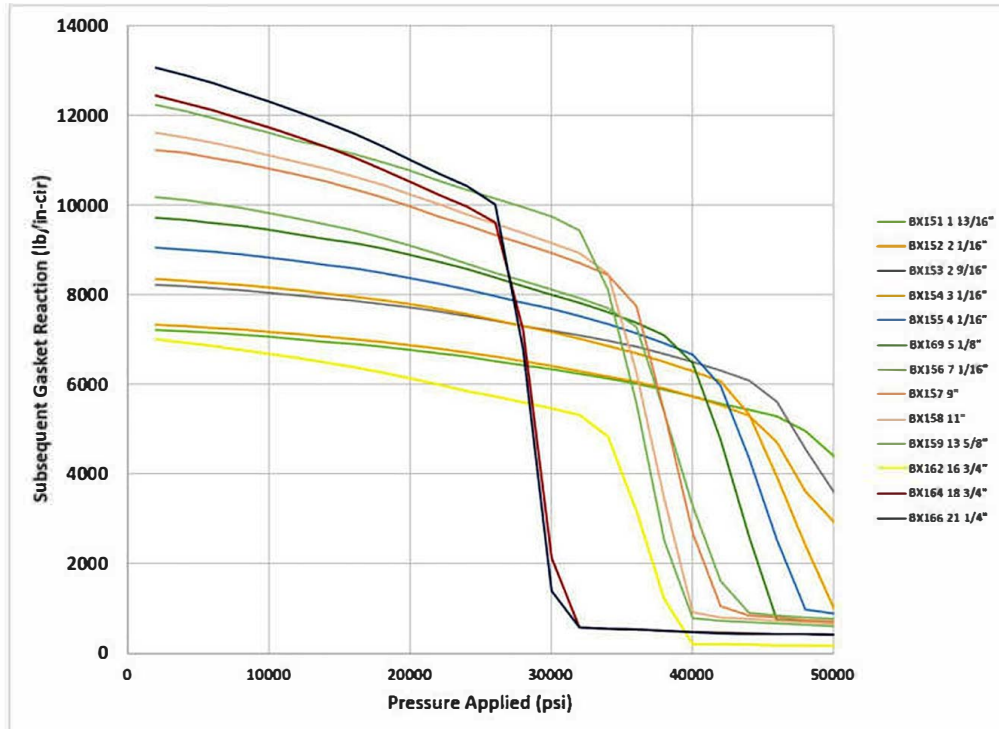The results of the axisymmetric analyses are shown in Figure A.6.

Figure A.6—Results of Gasket Assessment for Axisymmetric Model

A.5.2   3D Results

Results for a 3D analysis for the 6BX 164 18 ¾ in. are shown in Figure A.7.
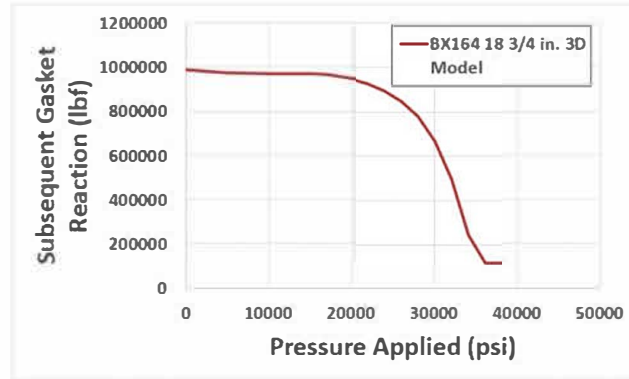
Figure A.7—Results of Gasket Assessment for 3D Model

Annex B

(informative)

API 6BX 5 ⅛ in. 15K Using Described Methodology

B.1 Introduction

Because API 6A defines the dimensions of a 6BX 5 ⅛ in. 5K flange, there is no need to calculate the bolt circle and flange dimensions. However, this annex follows the five HPHT Flange Methodology steps for sizing the flange as a demonstration of the procedure.

B.2 Calculate the Bolt Circle and Flange Dimensions

B.2.1 Hub Thickness (Wall of Pipe)

In this example, Lame's formula is used to determine the minimum wall thickness of the hub. The allowable stress for sizing the flange at Working Pressure is half of the flange body's minimum specified yield strength. These are the method and criterion used by Eichenberg when designing the original API 6BX flanges. The Hub Thickness calculation is summarized in Table B.1.

Table B.1—Calculation of Hub Thickness

Variable	Value	Description
P	15,000 psi	Working pressure of this flange.

Figure A.7—Results of Gasket Assessment for 3D Model

Annex B
(informative)

API 6BX 5 ⅛ in. 15K Using Described Methodology

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Table B.1—Calculation of Hub Thickness

Variable	Value	Description
P	15,000 psi	Working pressure of this flange.
Yield	75,000 psi	Minimum required yield strength of this flange.
ID	5.16 in.	Maximum bore of flange (from API 6A 21st edition, Table E.5: B).
OD	7.88 in.	Small diameter of hub (from API 6A 21st edition, Table E.5: J_2).
σ_{hoop}	37,520 psi	Hoop stress at working pressure. From Lame's thick wall formula: $((OD{\wedge}2 + ID{\wedge}2)xP)/(OD{\wedge}2 - ID{\wedge}2)$
σ_{allow}	37,500 psi	Allowable stress for sizing the flange.
Utilization	100 %	Percent utilization of allowable stress: $100\,x\,\left(\sigma_{hoop}/\sigma_{allow}\right)$

B.2.2   Pressure Loading

In this case, because an API 6BX gasket is being used, the loads generated by internal pressure are made up of two components: the total pressure end load generated by the pressure acting on an area defined by the Effective Sealing Diameter of the Gasket and a load generated by the pressure acting on the ID of the gasket creating a radial load that causes the gasket to wedge into the gasket gland generating a vertical load that tries to separate the joint. The pressure load on the Effective Sealing Diameter of the gasket and the axial component of the gasket wedge load are summed to determine the total pressure loading.

Figure B.1 illustrates the effective sealing diameter and the gasket wedging load.

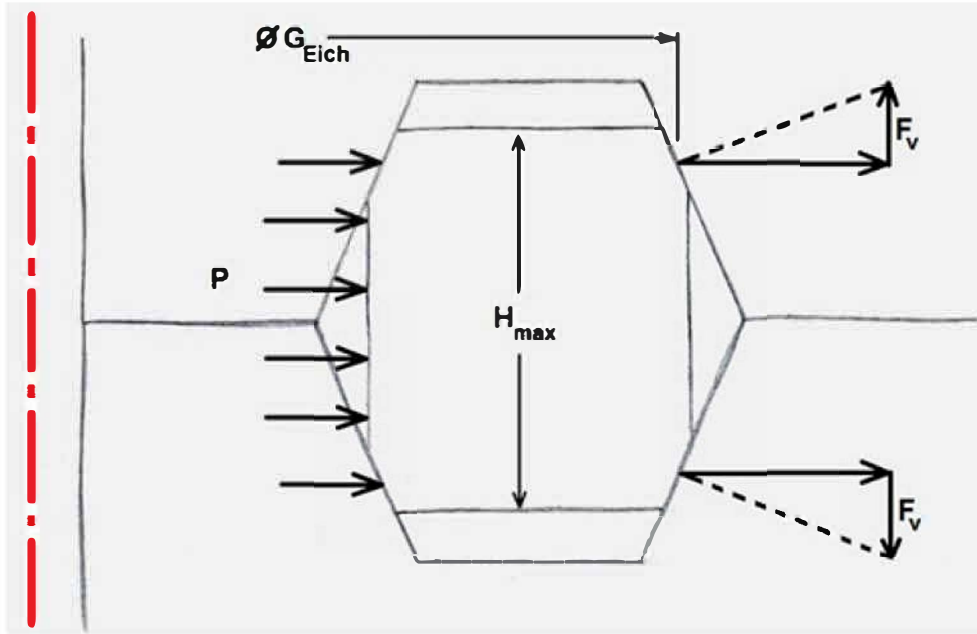Tables B.2, B.3, and B.4 demonstrate how to calculate the total pressure end load at hydrostatic test pressure.

Figure B.1—Effective Sealing Diameter and Gasket Wedging Load

Table B.2—Calculating Effective Sealing Diameter for Pressure End Load

Variable	Value (in.)	Description
G_{max}	6.959	Maximum OD of 6BX 169 Sealing Groove (from API 6A, 21st edition, Table E.11). The largest value is chosen as this will lead to the largest total pressure end load based on manufacturing tolerances: 6.955 + 0.004.
OD_{Tmax}	6.745	Maximum Diameter of 6BX 169 Gasket Flat (from API 6A, 21st edition, Table E.12). The largest value is chosen as this will lead to the largest total pressure end load based on manufacturing tolerances: 6.743 + 0.002.
H_{min}	0.624	Minimum Height of 6BX 169 Gasket (from API 6A, 21st edition, Table E.12). The minimum value is chosen as this will lead to the largest total pressure end load based on manufacturing tolerances.
OD_{max}	6.831	Maximum OD of unseated 6BX 169 Gasket (from API 6A, 21st edition, Table E12). The largest value is chosen as this will lead to the largest total pressure end load based on manufacturing tolerances.
G_w	0.132	Radial distance from OD of Sealing Groove to OD of 6BX 169 Gasket Flank after seating: $\left(\dfrac{H_{min}}{2}\right)x\mathrm{Tan}(23\ \mathrm{deg})$ 23° is the angle of the gasket sealing surface
G_{Eich}	6.780	Effective Sealing Diameter of 6BX 169 Gasket. The diametrical mid-point of the gasket sealing surface after seating: $G_{max} - 2Gw + \left(OD_{max} - OD_{Tmax}\right)$
F_p	P × 36.103	Total pressure end load generated by the pressure acting over the effective sealing diameter: $P\ x\ \pi\ x\left(G_{Eich}/2\right){}^{\wedge}2$

Table B.3—Calculation of Gasket Vertical Reaction Load

Variable	Value	Description
H_{max}	0.632	Maximum Height of 6BX 169 Gasket (from API 6A, 21st edition, Table E.12). The maximum value is chosen as this will lead to the largest Vertically Induced Reaction load "F_v" based on manufacturing tolerances: 0.624 + 0.008
F_v	P × 5.714	Vertical Gasket Reaction due to radial load generated by differential pressure across the gasket: $P \ x \ \pi \ x \ G_{r_{i,h}} x \ H_{max} \tan(23 \ deg)$

Table B.4—Calculate Total Pressure Loading

Variable	Value	Description
P	22,500 psi	Hydrostatic test pressure: 1.5 × 15,000
F_p	812,318 lbf	Total pressure end load generated by the pressure acting over the effective sealing diameter
F_v	128,565 lbf	Vertical Gasket Reaction due to radial load generated by differential pressure across the gasket
F_{total}	940,883 lbf	Total pressure loading at hydrostatic test pressure

B.2.3   Size and Number of Bolts

API Specification 6A requires that at hydrostatic test condition the bolt stress, based on root area, shall not exceed 83 % of yield. Also, typically, the number of bolts shall be a multiple of 4.

Calculating the bolt circle requires establishing practical clearances between the bolts, and between the bolts and the hub, to allow tool access. Taylor Forge tool clearance tables or modern literature may be used.

Eichenberg chose a hub length of approximately √2 × Bore and a taper of 1:4. In this case, the hub length is 3.22 in., which is very similar to √2 × Bore, and the taper is 1:4.16.

Table B.5 shows how the minimum required total root area may be calculated. Table B.6 shows how the optimal bolt circle may be determined. In this case, the optimal bolt circle, based on the smallest OD, was determined to be 12 × 1.50 in. The calculated bolt circle and OD were found to be 0.100 in. larger than the standard API Flange.

Table B.5—Calculate the Minimum Required Total Root Area

Variable	Value	Description
F_{total}	940,883 lbf	Total pressure loading at hydrostatic test pressure
Yield	80,000 psi	Minimum yield strength of closure bolting
$Bolt_{allow}$	66,400 psi	Maximum allowable stress at hydrostatic test pressure: 0.83 × Yield
$Root_{required}$	14.2 in.^{2}	Minimum required total root area of closure bolting threads: $F_{total} / Root_{required}$

Table B.6—Calculate Possible Bolt Circles and Flange ODs

Description	Value		
Bolt Size (in.)	1.375	1.50	1.625
Root Area (in.²)	1.155	1.405	1.680
Minimum Number of Bolts	12.3	10.1	8.5
Minimum Whole Number of Bolts (multiple of 4)	16	12	12
J1 (in.)	9.62	9.62	9.62
Minimum Radial Distance (from Taylor Forge) (in.)	1.875	2.0	2.125
Bolt Circle Based on Radial Clearance (in.)	13.37	13.62	13.87
Minimum Bolt Spacing (from Taylor Forge) (in.)	3.0625	3.25	3.50
Bolt Circle Based on Bolt Spacing (in.)	15.60	12.41	13.37
Minimum Possible Bolt Circle (in.)	15.60	13.62	13.87
Edge Distance (from Taylor Forge) (in.)	1.375	1.50	1.625
OD of Flange (in.)	18.35	16.62	17.12

B.2.4   Raised-face Diameter

The raised-face diameter was calculated using the load generated by the chosen bolt circle assembled to 50 % bolt stress based on stress area. This total bolt load divided by the annular area defined by the raised face diameter and the outer edge of the seal groove, shall be less than 30,000 psi.

B.2.5   Flange Thickness

Due to the complexity of the Eichenberg, Taylor Forge, and ASME methods of calculating the flange thickness, this methodology determines a flange thickness by assessing the flange's stiffness ratio. In this case, the stiffness ratio is found using FEA and comparing the stiffness of the closure bolting to the stiffness of the flange body. The largest stiffness ratio for existing API 6BX flanges is 0.50, meaning the closure bolting must be long enough to be at least as flexible as the flange body.

Various textbooks discuss flange stiffness ratio but there is no universally accepted method for quantifying a flange's stiffness ratio. Furthermore, it is difficult to calculate by hand the stiffness ratio of a flange with a raised face. For this methodology, the stiffness ratio was found by creating an axi-symmetric model of the flange profile with the studs represented by a beam element. The stiffness ratio was found by displacing the node at the base of the stud downwards 0.001 in. The resulting movement of the node at the top of the stud was recorded. The stiffness ratio was the stiffness of the bolt divided by the stiffness of the bolt plus the stiffness of the flange body, as shown in Bickford. All standard API 6BX flanges were assessed, and their stiffness ratios were found to range from 0.50 to 0.37. Therefore, a stiffness ratio of 0.50 or less is recommended for API 6BX style flanges. A thicker flange, with a lesser stiffness ratio, would be acceptable.

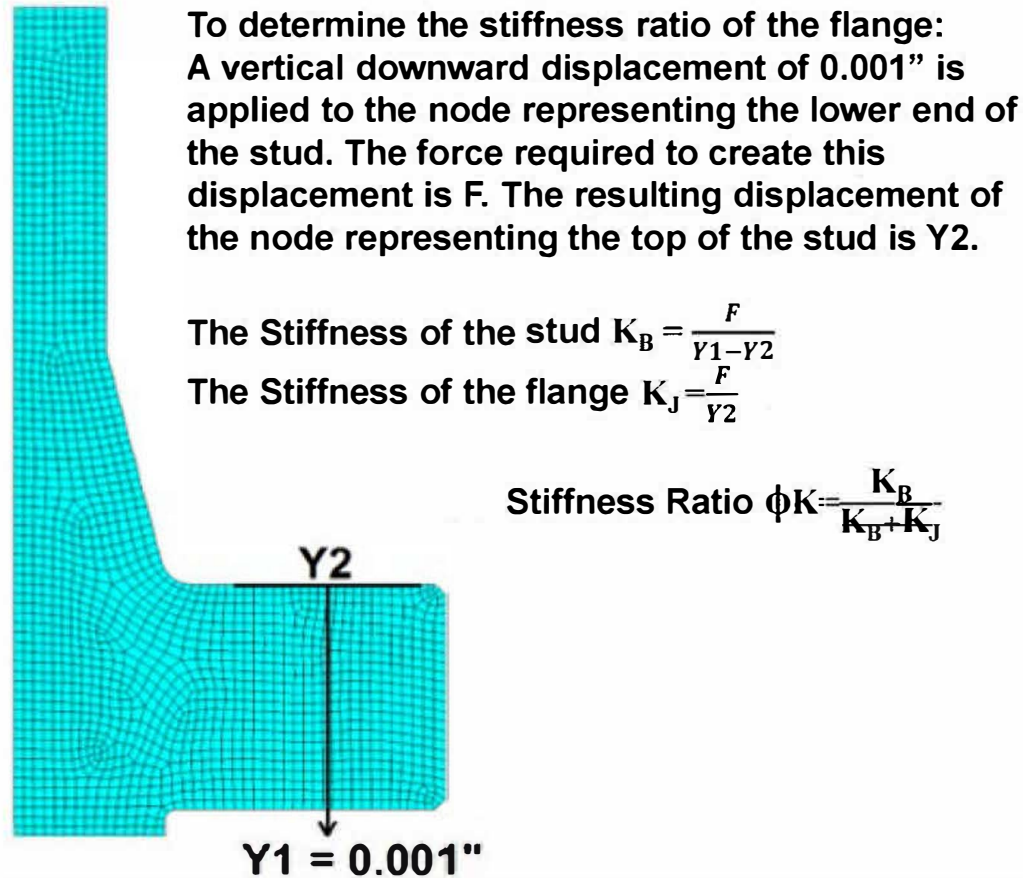Figure B.2 shows the methodology to determine the stiffness ratio of the flange.

To determine the stiffness ratio of the flange:
A vertical downward displacement of 0.001" is applied to the node representing the lower end of the stud. The force required to create this displacement is F. The resulting displacement of the node representing the top of the stud is Y2.

The Stiffness of the stud $K_B = \dfrac{F}{Y1 - Y2}$

The Stiffness of the flange $K_J = \dfrac{F}{Y2}$

Stiffness Ratio $\phi K = \dfrac{K_B}{K_B + K_J}$

Y2

Y1 = 0.001"

Figure B.2—FEA Model Used to Determine Stiffness Ratio

B.3   Flange Capability Determination

B.3.1   Introduction

This section describes an interpretation of the methodology that will use non-linear FEA to establish baseline capability charts for this flange.

The two aspects of flange performance addressed are:

1) Bolt Stress—The stresses in the closure bolting must be acceptable.

2) Flange Body—The stresses and deformations in the flange body must be acceptable.

B.3.2   FEA Model

To determine its bending capabilities, a 3D model representing 180° of the flange, gasket and bolting was created. The flange was modeled using maximum material condition for the gasket groove, and gasket was modeled as maximum material condition; the remaining dimensions were modeled from the API 6A tables. The maximum allowable raised face thickness was used. To avoid end effects, the hub was extended more than 2.5×sqrt(rt) beyond J3, where "r" is the average radius and "t" is the wall thickness of the hub end. The stud diameter represented the equivalent diameter of the stress area of the stud. The heavy hex nut was modelled as a cylinder with diameter equal to the nut width-across-flats, as it corresponds to the expected contact area between the nut and the flange. Threads were not included, and the nut and stud were modelled as volumes sharing common

areas. A friction factor of 0.1 was applied to the contact surfaces. A rigid surface was modeled at the horizontal plane of symmetry and contact was enforced between the rigid plane and the flange.

An elastic-perfectly plastic material model with a yield strength of 75ksi was chosen for the flange body. A true stress-true strain material model, representing 40/80ksi 316SS, was chosen for the gasket. A linear elastic material model was chosen for the closure bolting. Large Deflection effects were included. Figure B.3 shows an Element Plot of the model.

Top of Flange Hub where bending loads were applied
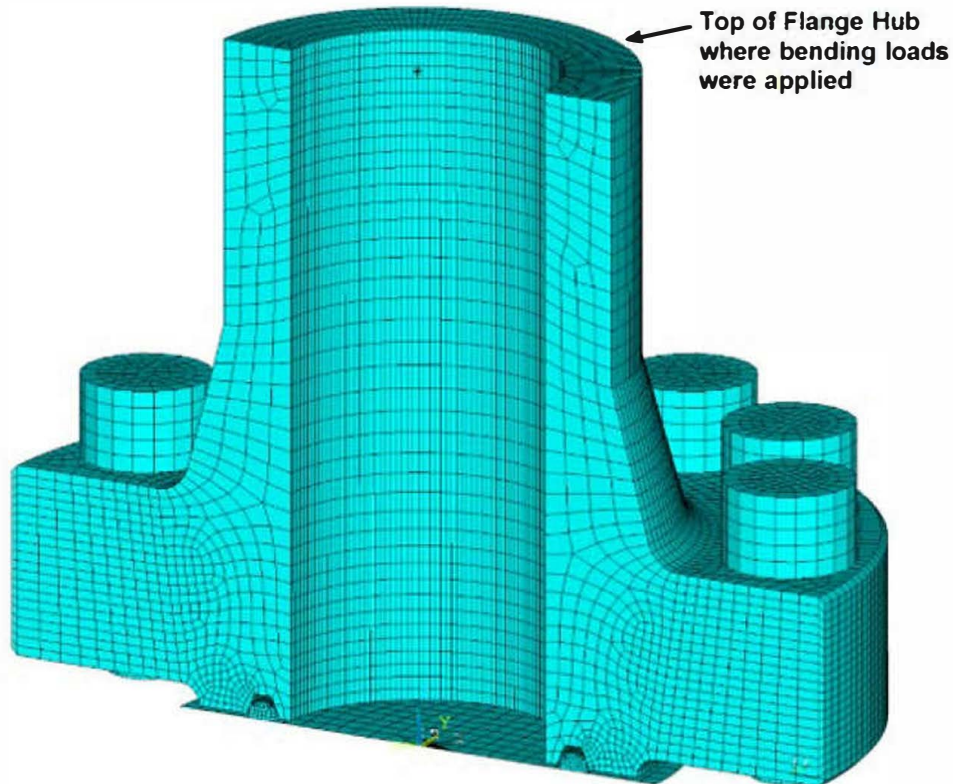
Figure B.3—Element Plot of Model

B.3.4   FEA Method

Bolt preload representing 50 % assembly stress was applied before any pressure or bend loading was applied.

When the loading scenario included internal pressure, it was applied to the bore of the flange, the raised face, and the seal groove out to a diameter defined by "G_{Eich}", the effective sealing diameter of the gasket. The pressure was also applied to the gasket. The total pressure end load that would have been generated by a blind end was compensated for by applying negative pressure to the upper surface of the flange body.

When the loading scenario included bend loading, the bending moment was generated by applying incremental angular rotations at the top of the flange hub via a pilot node. The moment required to generate each angular rotation was the applied bending moment (see Figure B.3).

B.3.5   Flange Body Criteria

The flange body was assessed for two failure modes: 1) plastic collapse, and 2) service criteria as described in 6.2 and 6.3.

The flange body working condition plastic collapse criterion was based on one of the accepted LRFD methods with a Load Factor of 1.5. This criterion was used for pressure or bend loading, or both.

The flange body working condition service criterion was determined to be at the onset of gross plastic deformation. The onset of gross plastic deformation was determined to be at the extent of the linear portion of the load vs displacement graph, similar in the approach used to establish the 0.2 % yield strength of metal. This criterion was only used for bend loading.

B.3.6 Bolt Stress Criteria

Bolts were assessed against the bolt stress criterion described in 6.4.

B.3.7 Results

Figure B.4 shows the results in the form of capability charts similar to those seen in API Technical Report 6AF2. The API 6AF2 results are also shown.
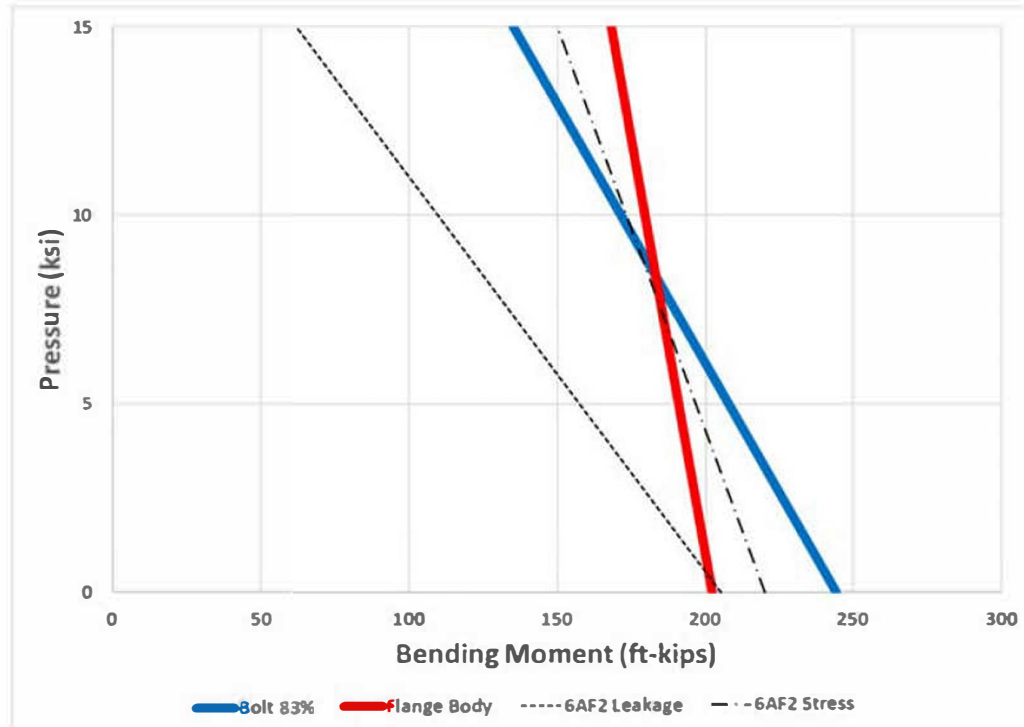
Figure B.4—5 ⅛ in. 15K 6BX Flange 80ksi Studs 50 % Assembly Stress

Annex C
(informative)

Examples of API 6BX Flange Capability Charts Using Annex B Methodology

Figures C.1 through C.9 show the capabilities of the 20K 6BX Flanges if assessed using the example method described in Annex B. The charts include the API 6AF2 capabilities for comparison.
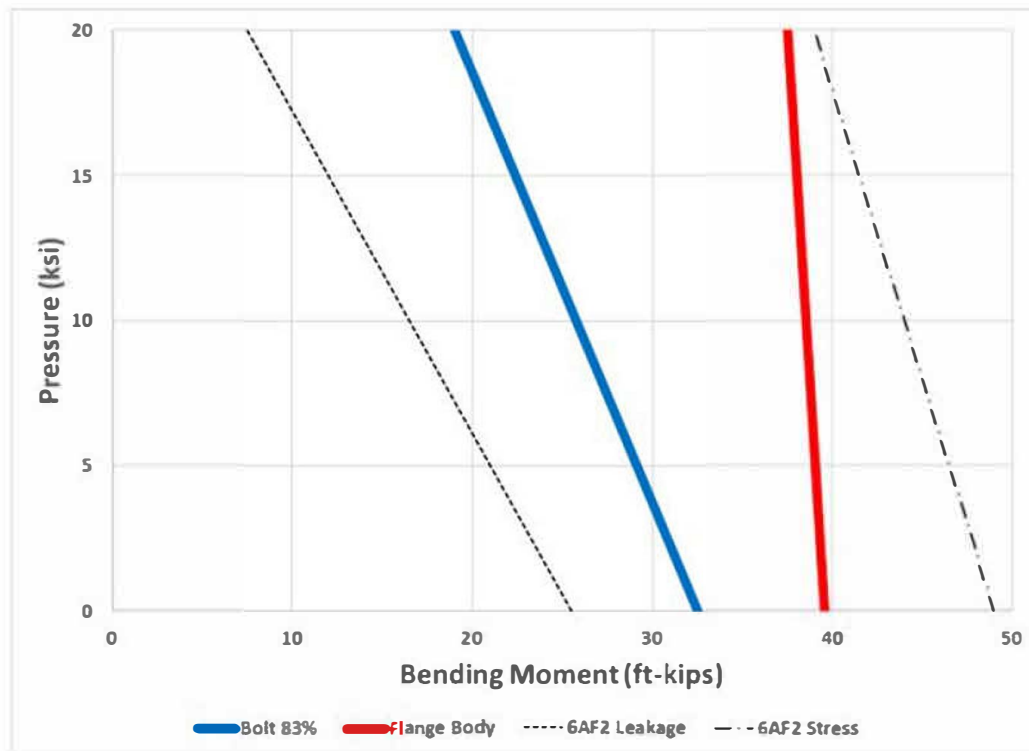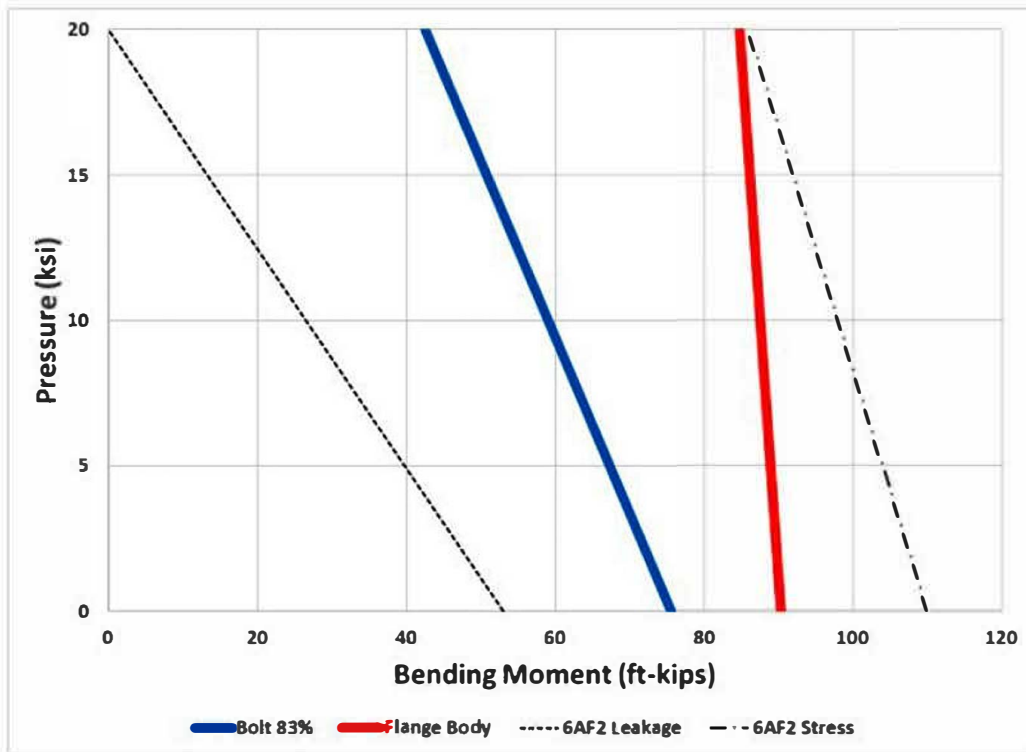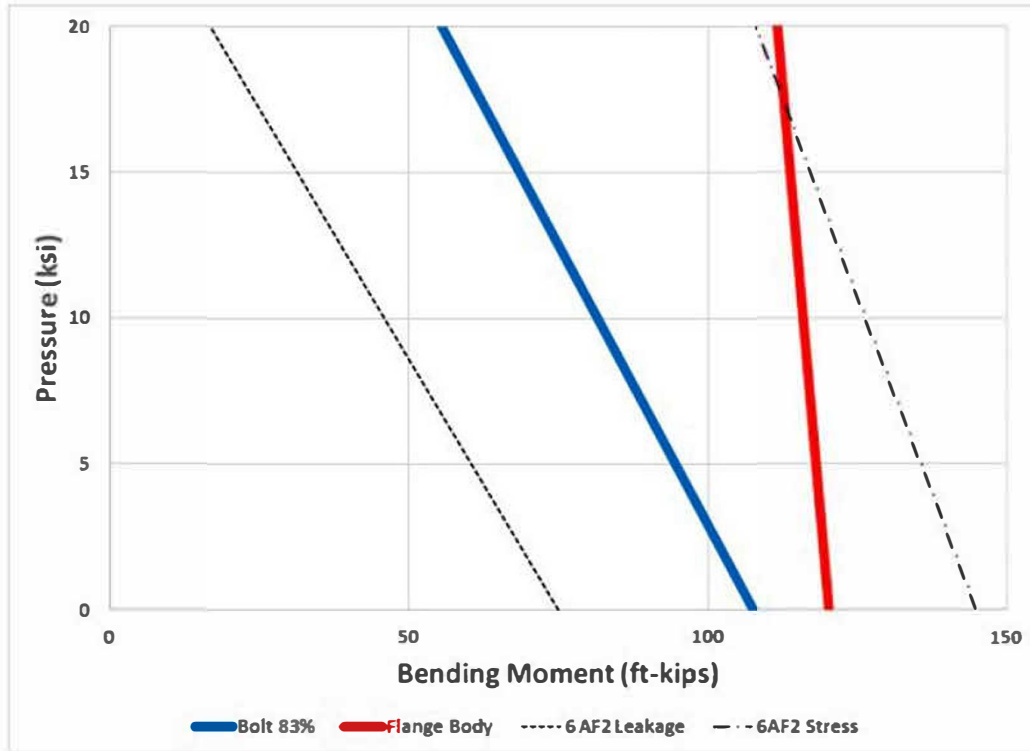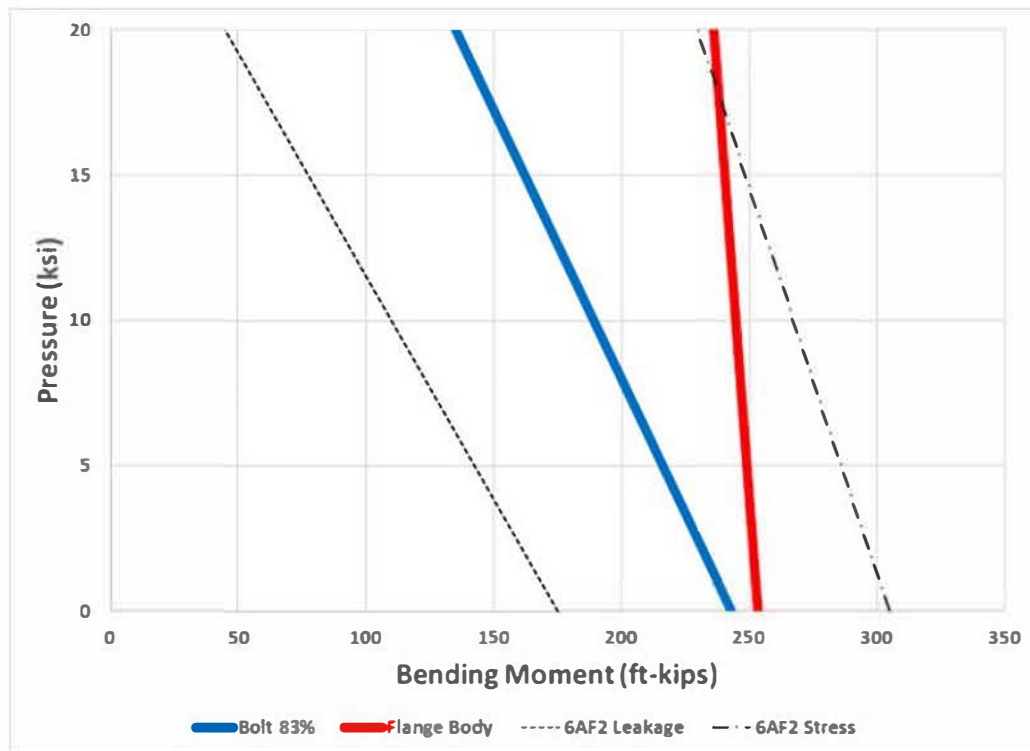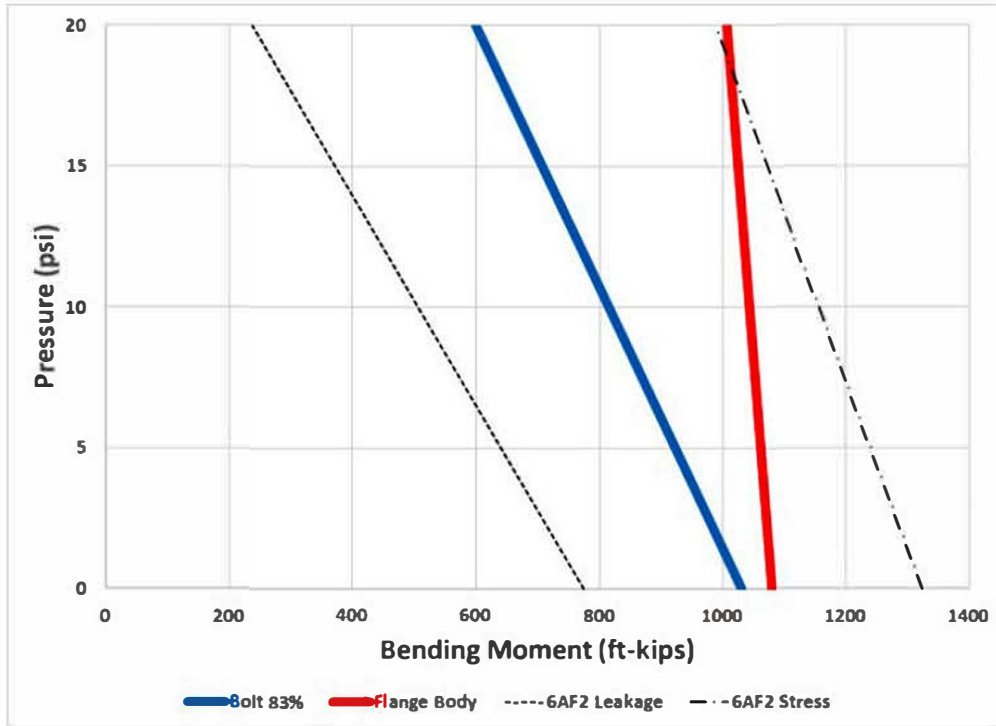

Figure C.1—1 $^{13}/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress
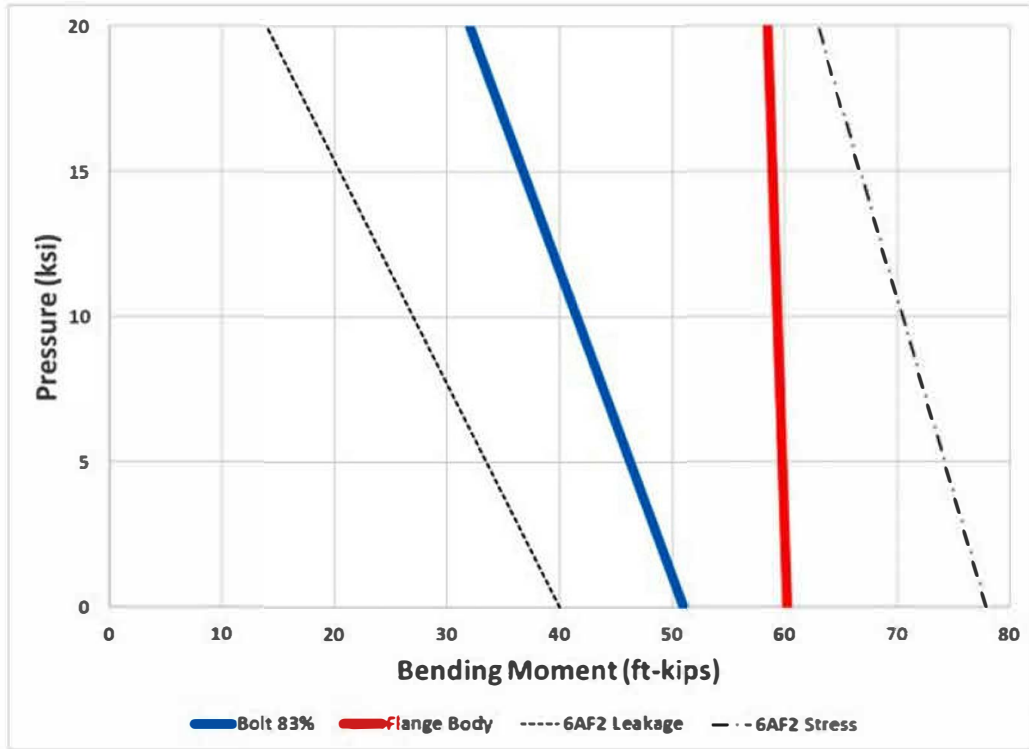
Figure C.2—2 $^1/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

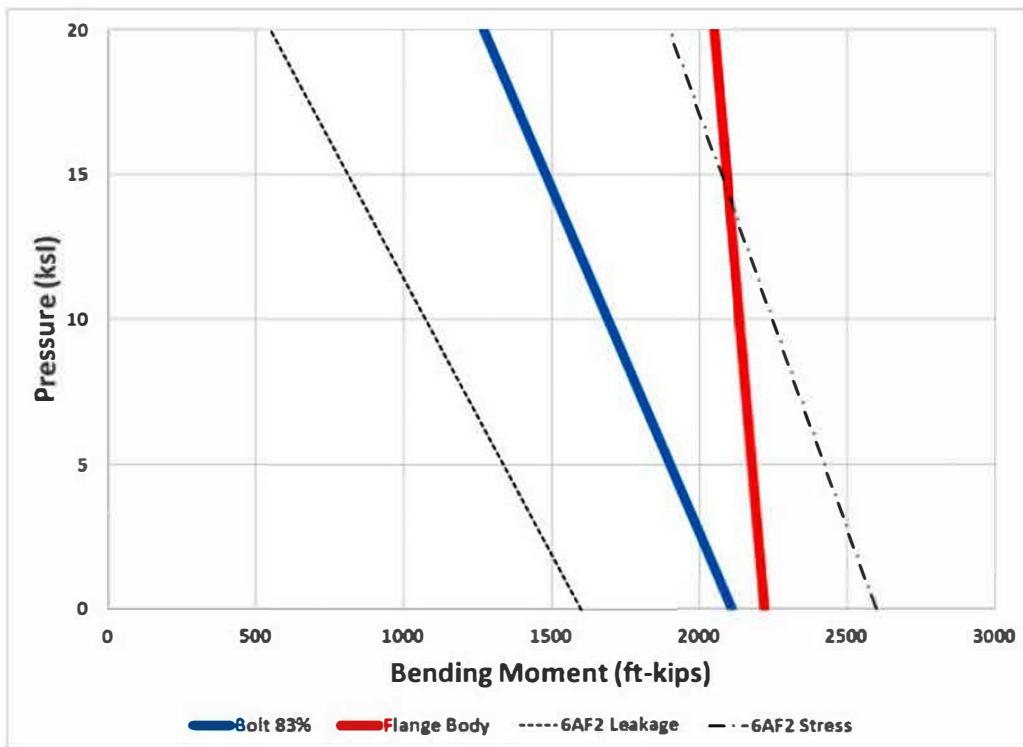Figure C.3—2 $^9/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

Figure C.4—3 $^1/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

Figure C.5—4 $^1/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

Figure C.6—7 $^1/_{16}$ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress
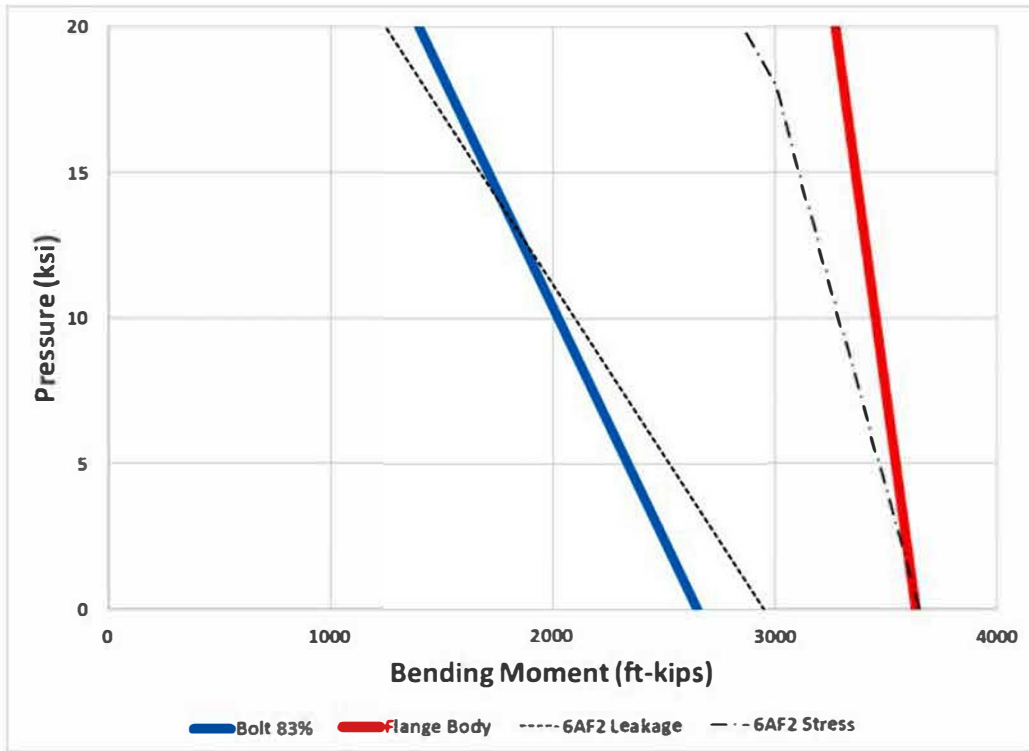

Figure C.7—9 in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

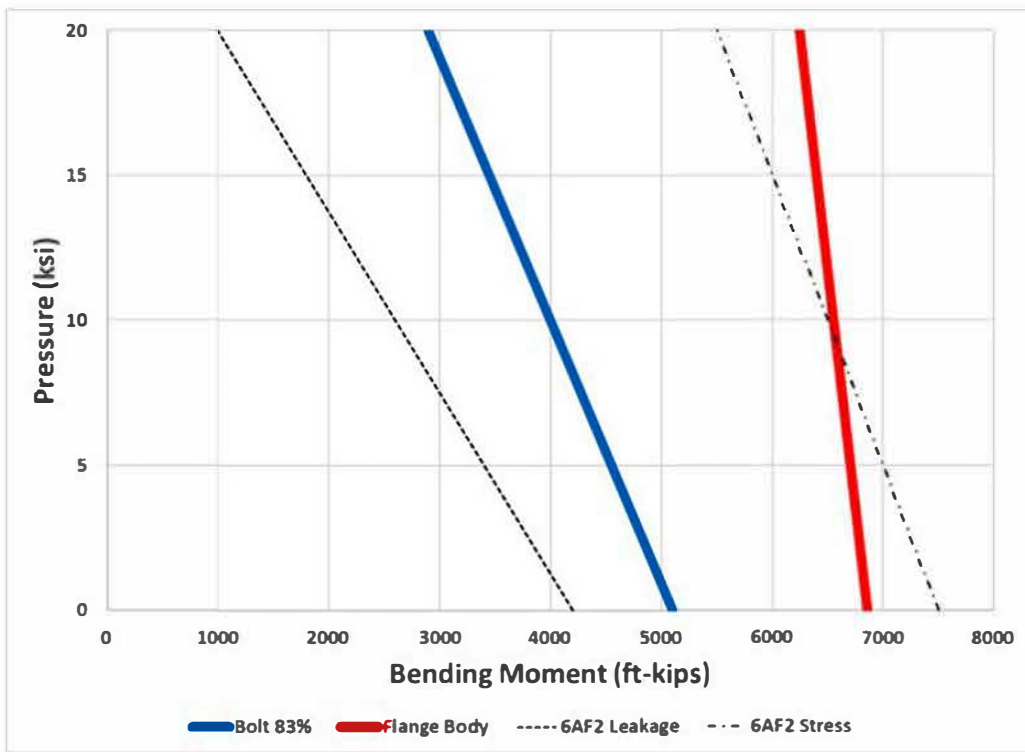**Figure C.8—11 in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress**

Figure C.9—13 ⅝ in. 20K 6BX Flange 80ksi Studs 50 % Assembly Stress

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