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

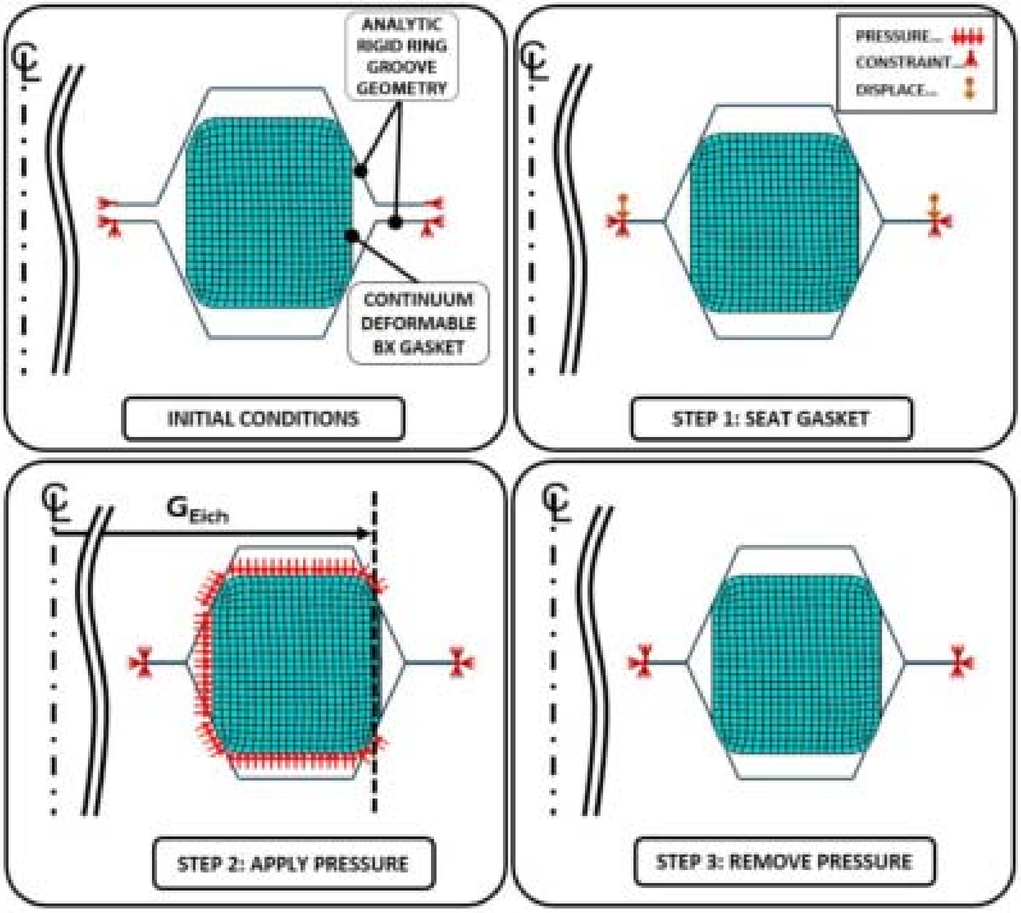
This annex demonstrates two different methods (axisymmetric and 3D) for verifying API 6BX gaskets are suitable for internal pressure requirements. The gasket material assessed was 316SS. This assessment was performed using room temperature properties. The methods in this annex have not been validated.

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.

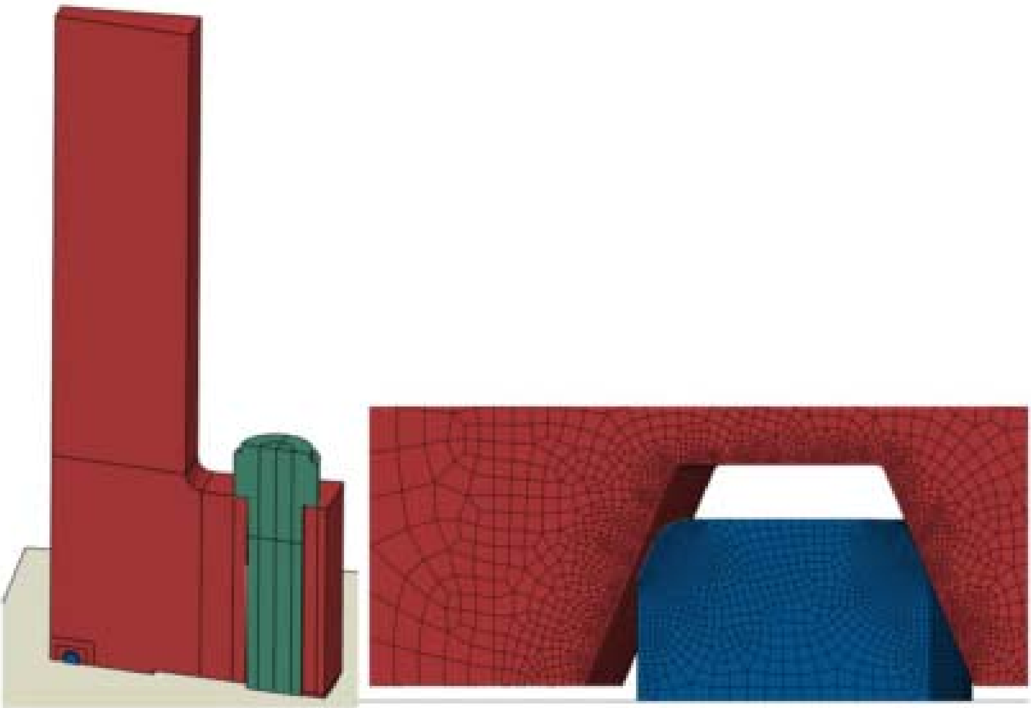


Figure A.2—3D Model of Flange Assembly

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.

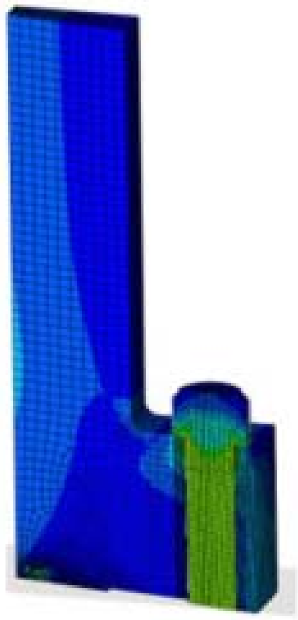


Figure A.3—Example Stress Plot of 3D Model

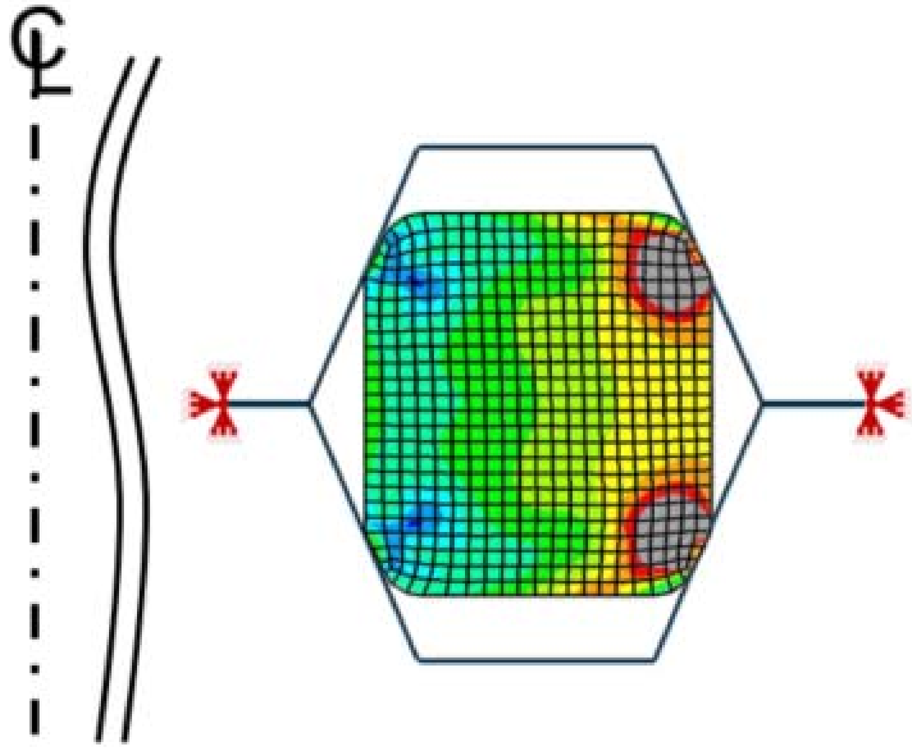


Figure A.4—Example Stress Plot of Axisymmetric Model