# Design Method Combining API and ASME Codes for Subsea Equipment for HP/HT Conditions up to 25,000-psi Pressure and 400°F Temperature

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## Summary

A current challenge in the offshore industry is the design of subsea equipment for pressures exceeding 15,000 psi and temperatures exceeding 250°F. This combination of pressure and temperature has been fairly accepted as the start of the high-pressure/high-temperature (HP/HT) region. The current American Petroleum Institute (API) standard for designing subsea equipment, *API Specification (SPEC) 17D* (2011), is limited to a working pressure of 15,000 psi and provides little guidance on temperature conditions exceeding 250°F. This paper demonstrates a design methodology that combines the API and American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessels Code (BPVC) for designing an example subsea pressure containing component for HP/HT conditions greater than 15,000 psi and 250°F.

This paper shows the evaluation of a combined load-capacity chart for an *API SPEC 17D* flange flow-loop [*API SPEC 6A* (2010), 4 in., 20 ksi] for a design pressure of 20,000 psi and a temperature of 350°F with external tension and bending loads. Both the linear elastic and elastic plastic methods for protection against plastic collapse are used to determine the structural capacity of the flange body. These methods combine the API material and design allowables and ASME design methods. Stress classification and linearization are used for the evaluation of design capacities with linear methods. Modified load-resistance design factors are used both to evaluate design capacities and to account for the difference in ASME and API hydrostatic-test pressures with elastic plastic methods. The structural capacity is combined with thermal analysis to determine the effects of high temperature on the flange capacity. To assess the cyclic-loading capacity of the flange, stress-based fatigue analysis and fracture-mechanics analysis are also compared.

The results obtained are comparable to existing *API Technical Report (TR) 6AF1* (1998) charts. This work has been performed to demonstrate both the acceptance of existing methods for HP/HT conditions and to introduce the advanced ASME design methods for designing *API SPEC 17D* subsea equipment. The methods presented are acceptable for designing equipment for working pressures up to 25,000 psi and temperatures up to 400°F.


## Introduction

The industry (i.e., API and the Association of American Wellhead Equipment Manufacturers) has a long and established history in the development of new standards and procedures for land- and platform-based equipment rated at or above 15,000 psi and 250°F. As described by Payne (2010), the original 15,000-psi wellhead specifications were developed in 1952. The first 20,000-psi wellhead systems were developed in 1972, quickly followed by 30,000-psi wellhead systems in 1974. These sytems have been successfully deployed both in land- and platform-based fields in the Gulf of Mexico and in the North Sea. In the years from 1960 to 1990, subsea wells rated at a working pressure of up to 5,000 psi at a water depth of 2,000 ft were being explored. In the 1990s, the working pressure increased to 10,000 psi at a water depth of 3,000 ft. From 2000 to 2012, subsea wells rated up to 15,000 psi and 350°F at a water depth of 8,000 ft were explored. The trend of the industry is moving toward wells being explored at 20,000 psi and 400°F or higher at water depths approaching or exceeding 10,000 ft. These wells require HP/HT-rated equipment for oil production, generating the need for a design method for such HP/HT-rated subsea production equipment.

The existing design methods use *API SPEC 17D* and the *ASME BPVC* as the primary design codes. The *ASME BPVC* was a result of a committee set up by ASME in 1911 for the purpose of formulating standard rules for the construction of steam boilers and other pressure vessels. This code is formulated for pressure vessels in the nuclear industry and for designing storage and transportation tanks. *API SPEC 17D* was formulated for standardization of subsea production systems. Because *API SPEC 17D* was formulated specifically for subsea equipment, it will be mandatory to meet the API material requirements, design allowables, and test requirements.

Subsea-wellhead and tree equipment rated up to 15,000 psi can be designed with methods documented in *API SPEC 17D*. However, this standard does not provide guidance for equipment rated at pressures greater than 15,000 psi and combined with temperatures greater than 250°F. The design methods given in *API SPEC 17D* refer to the design methods of *API SPEC 6A* (2010), which allows for the application of *ASME BPVC, Section VIII Division 2* (2010) design guidelines (as well as other codes) for equipment rated for working pressures up to 20,000 psi and for temperatures up to 650°F. *ASME BPVC, Section VIII Division 2* and *ASME BPVC, Section VIII Division 3* (2010) provide design methods for high-pressure vessels. These design methods can be used for designing the equipment; however, the design must meet the *API SPEC 17D* design allowable limits and material and test requirements.

The ASME and API codes differ in the material, hydrostatic-test, and nondestructive-examination (NDE) requirements in particular. The API codes provide temperature derating factors for elevated-temperature design. The standard hydrostatic-test requirement per *API SPEC 17D* is 1.5 times the rated working pressure, whereas in *ASME BPVC, Section VIII Division 2*, the hydrostatic-test requirement is 1.43 times the working pressure. Additionally, the hydrostatic-test pressure in ASME varies between divisions and has also fluctutated over time because of the experiences of the ASME community. The material requirements to satisfy various codes or standards are also not common or aligned. The different code-specific material requirements are provided in *API SPEC 6A, Section 5* and *ASME BPVC, Section VIII Division 2, Part 3* and *ASME BPVC,*

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*Section VIII Division 3, Part KM*. The primary choice of materials for these design recommendations will be based on the limitation of *API SPEC 6A, Section 5*. Specific guidance is provided in this paper to generate a conservative use of ASME methods with API materials. These approaches are based on the normalization of the hydrostatic-test pressures across the various codes. The NDE requirements in ASME can allow for larger flaw sizes as compared with the API allowable limits.

## Design Method

The design method refers to the procedures outlined in Kocurek et al. (2012). In addition to these results, it covers the methods to handle the high-temperature and combination of high-temperature and high-pressure design of subsea equipment. Components or assemblies classified as pressure-containing parts per *API SPEC 17D* will be evaluated by the following techniques. In addition, equipment not defined as pressure containing, but which experience a pressure greater than their working pressure (e.g., tubing hangers) during testing before deployment in the field, will be designed by these methods. These design methods include linear and elastic plastic protection against plastic collapse, linear and elastic plastic protection against local collapse, protection against cyclic loading, and closure and critical-bolting design. For protection against collapse from buckling, the same design methodology as suggested in *ASME BPVC, Section VIII Division 2, 5.4* is recommended.

To ensure that all modes of failure are addressed, three load cases are considered. A hydrostatic load case with a test-pressure value of at least 1.5 times the rated working pressure is considered per *API SPEC 17D* requirements. Additionally, a working load case with appropriate load scenarios and temperature (radial wall thermal gradients) distributions, as given by the design code used for calculations, is addressed. A reference table is supplied in *ASME BPVC Section VIII, Division 2,* Table 5.1. Last, a maximum load case for each component or assembly is evaluated for stress and structural stability. Along with each of the three load cases listed, each part must take into consideration any fatigue effects on the design. If a part falls under the designation of heavy section bodies (ratio of inner radius to thickness $R/t<4$), special requirements are outlined under the respective design methods. Worst material-tolerance conditions shall be used when evaluating structural capacities. To check for proper functionality of the equipment, the deflection of systems under load must be accounted for during the calculation of capacity. As consistent with API, there shall be no derating of material strengths for equipment in use up to 250°F. For anything greater than 250°F, the material strength should be derated per the factors in *API SPEC 6A, Annex G*. Fracture-toughness and fatigue-related data need to be evaluated for elevated temperatures because these are not readily available in the codes.

**Protection Against Plastic Collapse.** This design method addresses general plastic collapse in equipment because of gross distortion across its membrane as a result of applied loads. This failure mode can be analyzed by use of both linear methods and elastic-plastic analysis methods.

***Linear Methods.*** The linear methods include stress-analysis methods that use handbook solutions or other industry-accepted methods, as long as these solutions represent the component geometry and loading conditions appropriately. For temperatures greater than 250°F, thermal analysis shall be conducted to take into effect the temperature distributions and thermal strains. Numerical-analysis techniques, such as finite-element analysis (FEA), shall be used to determine the stresses in equipment for which handbook solutions cannot be applied appropriately. The calculation of stresses will be based on the stress-intensity equation on the basis of the Tresca yield surface and maximum-shear-stress theory. Thermal stresses will be classified as secondary stresses per *ASME BPVC, Section VIII Division 2, Section 5.12, paragraph 19* and per *API TR 6AF1, Section 5.2*.

Hydrostatic-test-design allowable-stress values will be based on *API SPEC 6A, Section 4.3.3.2* and/or *Section 4.3.3.3*. Hydrostatic-test-design allowable-stress values can also be based on *ASME BPVC, Section VIII Division 2, Section 4.1.6.2*. Working-load-design allowable-stress values will be based on *API SPEC 6A, Section 4.3.3.2*. Working-load-design allowable-stress values can also be based on *ASME BPVC, Section VIII Division 2, Section 4.1.6.1*. Thermal stresses, when considered, can use secondary stress allowable values on the basis of *ASME BPVC, Section VIII Division 2, Sections 5.15* and *5.5.6.1.d* and *API TR 6AF1, Section 5.2*. The material yield strength will need to be derated on the basis of the working temperatures per *API SPEC 6A, Annex G*.

If appropriate, in vessels that are $R/t\geq4$, the classification of stress can be performed on the basis of *ASME BPVC, Section VIII Division 2, Section 5.2.2.2* and *Figure 5.1*, the "Hopper Diagram." In vessels that are $R/t<4$, the classification of stress should not be used. Stresses may still be linearized, but all results should be considered as primary membrane stresses. If stress linearization is used, the peak-stress intensity derived through calculations shall not be allowed to exceed yield strength for more than 5% of the section thickness (path distance). This recommendation is based on *ASME BPVC, Section VIII Division 2, Section 5.2.1.3* (2010). This ensures that the stress distribution through the section of the material is not so extreme as to cause local failure in a portion of the section before full section failure might occur. The stress-linearization procedures should be aligned with *ASME BPVC, Section VIII Division 2, Annex 5.A*. Thermal loads, when considered in the analysis, can be directly classified as secondary stresses regardless of the $R/t$ ratio. In many scenarios, this will lead to a nonconservative load case, and, hence, a load case with no thermal loads will also need to be evaluated to ensure design acceptability.

Generally, the analyst must follow the requirements of *ASME BPVC, Section VIII Division 2, Part 5*, with the exception that stress intensity shall be used instead of equivalent stress. Although *ASME BPVC, Section VIII Division 2, Part 5* calculates stresses through equivalent-stress equations (e.g., von Mises stress), it is recommended to continue to use stress intensity to satisfy API design requirements. FEA should use a small-displacement theory. Thermal loading or thermal results can be calculated separately or together with the structural model, depending on the particulars of the problem. *ASME BPVC, Section VIII Division 2, Annex 5.A* established the preferred linearization procedure and path methodology. After evaluating the component or assembly by use of linear elastic criteria for protection against general plastic collapse, further evaluations may be addressed to protect against other failure modes.

***Elastic-Plastic Method.*** The elastic plastic method can be used to evaluate components and interactions for protection against plastic collapse. Generally, the analyst must follow the requirements of *ASME BPVC, Section VIII Division 2, Part 5*.

The von Mises yielding criterion and associated flow rules will be used. Material data will be reported as true stress-true strain. It is preferred to use actual-material test data for the elastic plastic methods, but a standard (and conservative) estimation of material behavior is given in *ASME BPVC, Section VIII Division 2, Part 3*. When temperatures greater than 250°F are considered, derated material yield strength can be used per *API SPEC 6A, Annex G* and curves can be estimated per *ASME BPVC, Section VIII Division 2, Part 3*. A load and resistance factor design (LRFD), as documented in *ASME BPVC, Section VIII, Table 5.5,* shall be used to determine if a structure is suitable for operation at a specific load case. However, given that the typical hydrostatic-test pressure established by *ASME BPVC, Section VIII, Part 8* is a minimum of 1.43 times the design pressure (maximum allowable working pressure), this is in conflict with the standard API hydrostatic-test case. Therefore, it is recommended that the LRFD factors be increased by the ratio established here:

$(1.50/1.43)=1.049$, ................................................................(1)

where 1.50 is the API hydrostatic-test multiplier, 1.43 is the ASME hydrostatic-test multiplier, and the factor 1.049 is the multiplier applied to increase the LRFD factors established by ASME. If the capacity of a component under engineering evaluation is to be established by use of the elastic plastic methods, the load-case inputs shall be increased uniformly until the solution is nonconvergent because of unbounded deformation. The loads at which nonconvergence is established are then decreased by the appropriate LRFD factors and the value set in Eq. 1.

As an example, if an internally pressurized vessel reached nonconvergence at 60,000-psi internal pressure, then the design pressure would be established by dividing 60,000 psi by 2.40 per *ASME BPVC, Section VIII Division 2 Table 5.5*, *Global Criteria 1,* and by 1.049 (per Eq. 1). This would establish the maximum allowable design pressure for the vessel at 23,832 psi, and an API hydrostatic-test pressure of 35,748 psi (calculated as 1.50 times the design pressure).

For temperatures greater than 250°F, when considering thermal loads because of temperature differentials, *ASME BPVC, Section VIII Division 2, Table 5.5* gives LRFD of 2.1 for thermal loads and the pressure and design loads; however, it is more conservative to use a factor of 2.4 for the pressure and design loads and to use the maximum possible differential temperature to which the equipment will be subjected. If there are appertunance live loads and other loads than just the pressure and design loads, then it would be appropriate to use the equation, with LRFD of 2.1 for the design loads and 2.7 for the appurtenance live loads.

After evaluating the component or assembly for protection against plastic collapse, further evaluations must be made to protect against local collapse.

**Protection Against Local Collapse.** This design method addresses local collapse in equipment in addition to protection against plastic collapse, as described in the previous subsections. This failure mode can be analyzed with both linear methods and advanced elastic-plastic analysis methods. In general, the analyst must follow the requirements of *ASME BPVC, Section VIII Division 2, Section 5.3* to address protection againt local collapse. When temperatures greater than 250°F are considered, thermal loads also need to be considered when evaluating for protection against local collapse. Material yield strength will need to be derated per *API SPEC 6A, Annex G* to determine the allowables.

***Linear Method.*** The linear elastic method is used in addition to the protection against plastic collapse, as described previously, to guard against local failure. *ASME BPVC, Section VIII Division 2, 5.3.2* outlines this linear method.

***Elastic-Plastic Method.*** The elastic plastic method is used with protection against plastic collapse to prevent local failure. *ASME BPVC, Section VIII Division 2, 5.3.3* describes the elastic plastic method used for protection against local failure. This method of analysis is for a sequence of applied loads on the basis of the load cases discussed previously.

**Protection Against Cyclic Loading.** After evaluation for general and local collapse, further evaluation may be made to determine that a failure mode generated by load fluctuations is not encountered. Examples of these failure modes are fracture formation or ratcheting of nonintegral connections.

Components or assemblies having undergone evaluation to prevent general plastic collapse shall adhere to the following criteria to protect against failure caused by cyclic loading. *ASME BPVC, Section VIII Division 2, 5.5* provides the fatigue-analysis approach. This is also the recommendation of *API SPEC 17D, Section 5.1.3.1*. *ASME BPVC, Section VIII Division 3, KD-3* is a traditional fatigue-analysis approach that is based on stress-life theory. It shall be used when a leak-before-burst mode of failure is established. If the leak-before-burst mode of failure cannot be shown to exist, then a fracture-mechanics approach to design shall be used. *ASME BPVC, Section VIII Division 3, KD-4* provides the step-by-step process for the fracture-mechanics approach. It involves estimation of design life cycles on the basis of crack propagation to critical crack depth, which is all defined in *ASME BPVC, Section VIII Division 3, KD-4*.

*ASME BPVC, Section VIII Division 3, KD-340* is a traditional fatigue-analysis approach that is based on strain-life theory. It shall be used when welds are present as structural supports. If the assembly has nonintegral components in the structural load path that could progressively distort through sequential load cycles, it must also be evaluated for protection against ratcheting failure. Examples of this are screwed-on caps, screwed-in plugs, shear-ring closures, and breech-lock closures. Other examples in the industry might include collet- or clamp-style connectors. Evaluation by use of the procedure outlined in *ASME BPVC, Section VIII Division 3, KD-234* shall be followed.

In most cyclic-load cases for pressure-controlling or -containing equipment with high internal-fluid temperatures, it is observed to have more allowable design cycles than the same component evaluated for conditions without temperatures. This is typically observed because of compressive stresses developed at the internal surfaces as a result of the temperature gradient. Hence, it is important to evaluate the component at a load case with no thermal conditions to ensure design acceptabiltity.

**Closure-Bolting Design.** Linear elastic methods outlined in *API SPEC 17D* and paralleled in *ASME BPVC, Section VIII Division 2, Part 5.7* are recommended for closure-bolting design. In analyzing the stress capacity of closure bolting, the maximum allowable tensile stress should be no more than 83% of the material's yield on the basis of the root area of the thread (*API SPEC 6A, 4.3.4*). According to *API SPEC 6A*, bolting stresses are to be determined with all loads acting on the closure area, including the pressure acting over the seal area, gasket loads, and any other mechanical and thermal loads. The maximum allowable stress is also determined in all conditions, such as working-pressure and hydrostatic conditions. When thermal loads are considered for bolting, thermal stresses shall use secondary-stress allowable values on the basis of *ASME BPVC, Section VIII Division 2, Sections 5.15* and *5.5.6.1.d*. Material yield strength will need to be derated on the basis of the factors provided in *API SPEC 6A, Annex G*. The change in bolt preload as a result of thermal expansions shall be captured during analysis by ensuring that the load step to apply the bolt pr load is before the load step at which thermal loads are applied.

Both closure and critical bolting require a preload to a high percentage of the material's yield strength. Closure bolting of all API 6BX and 17SS flanges is to be made up to approximately 67% of the bolt's material yield stress. Other studs, nuts, and bolts used on end connections of subsea equipment shall be made up to at least 50% of the bolt's material yield stress (*API SPEC 6A, Annex D*). Structural bolting is to be made up to the manufacturer's written specification. Studs, nuts, and other closure bolting for use in subsea service are often manufactured with corrosion-inhibiting coatings or platings that can affect the stud-to-nut friction factor dramatically. The manufacturer shall document the recommended makeup torque for their fasteners.

High-strength-alloy steel bolts, studs, and nuts shall be evaluated for cyclic operation by use of elastic stress analysis and equivalent-stress-amplitude loading established in *ASME BPVC, Section VIII Division 2, 5.5.3* and *Annex 3.F*. Stress concentrations at the root of standard ASME B1.1 threads are not to be included in the fatigue evaluation. In cases in which loading is shared between components that have deformed, such as a metal gasket, the controlling stress for the fatigue evaluation will be the effective total-equivalent-stress amplitude. This value is defined as one-half of the effective total-equivalent-stress range calculated for each cycle in the loading histogram.

As a safeguard to prevent fatigue failure of threaded components, a limit shall be placed on equipment so that no regular load encountered during working conditions causes a separation of the clamped joint of a threaded fastening system. An exception to this would be extreme scenarios, such as a drive-off or impact event. Clamped faces that separate during qualification of factory-acceptance testing shall have their fasteners either replaced or retensioned, as per the manufacturer's written specifications.

**Nondestructive Examination.** For NDE of the component, rules from *API SPEC 17D*, *API SPEC 6A*, and *API Recommended Practice (RP) 6HT* (2005) should be followed. There are no specific rules for equipment that would be subjected to temperatures greater than 250°F; hence, the same rules apply for high-temperature equipment.

## Example Analysis of an *API SPEC 6A,* 4-in. 20-ksi, 350°F 6BX-Type Flange

To demonstrate the application of the design methods outlined in the previous subsections, an *API SPEC 6A,* 4-in., 20-ksi, 6BX-type flange is analyzed when subjected to an internal-fluid temperature of 350°F and an external-fluid temperature of 35°F. The flange was assumed to be made from F22, low-carbon-alloy steel (2¼ Cr. 1 Mo.) with 75-ksi yield and 95-ksi ultimate tensile strength. The bolts were assumed to be made of high-strength steel with 105-ksi yield strength and preloaded at 67% of the bolt yield strength per the recommendation in *API SPEC 17D, Section 5.1.3.5*. The bolts are assumed to be open to seawater at a temperature of 35°F. Thermal conductivity of 24.85 Btu/hr ft$^2$ °F was assumed for both the flange and the bolts at all temperatures. The flange body and bolts were given a thermal coefficient of expansion of $7.3\times10^{-6}$/°F at all temperatures. Both the bolt and the flange are assumed to have modulus of elasticity of $29.7\times10^6$ psi at 70°F (room temperature) and 35°F (lower temperature) and $28.2\times10^6$ psi at 350°F (elevated temperature). A pressure-tension-bending moment (P-T-B) capacity chart, similar to that in *API TR 6AF1* (1998) and *API TR 6AF2* (2010), is generated by use of the design method of protection against plastic collapse by linear elastic analysis and elastic plastic analysis. Protection against cylic loading is demonstrated by evaluating the design cycles for extreme load cases on the P-T-B chart by both fatigue and fracture-mechanics analyses.

**Protection Against Plastic Collapse.** FEA was conducted by use of commercial software Abaqus 6.12™. A 3D, 180° model of the flange with 3D bolts and a rigid bottom surface was used for both linear and elastic plastic analysis, as shown in **Fig. 1.** The gasket was not included in the 3D model to avoid numerical instability resulting in premature failure of the model and because previously conducted linear elastic methods indicated that gasket groove was not an area of concern. The serviceability of the gasket is determined at the maximum working condition on the basis of contact stresses, and can be validated by performance testing on the basis of *API SPEC 17D* test methods. A predetermined gasket load by use of axisymmetric FEA was applied in the model to simulate the gasket for all analyses performed. Two flange-failure criteria were considered for the analyses: flange structural limit and bolt structural limit.

Linear elastic FEA for the hydrostatic-test pressure-load case was evaluated for the worst stress-classification line (linearized path) for membrane stress intensity and membrane-plus-bending stress intensity. The membrane stress intensity was limited to 83% of the yield strength and the membrane-plus-bending stress intensity was limited to the yield strength. The maximum structural capacity of the flange was determined to be 30 ksi, limited by the hub-neck area. The overall capacity of the flange was limited by the bolts and was 26 ksi.

Linear elastic FEA was conducted for the working conditions, which involved the determination of the P-T-B charts for the flange with and without thermal loads. When thermal conditions of internal-fluid temperature of 350°F and external-fluid temperature of 35°F are considered, stresses developed were considered as secondary stresses. Secondary stress allowable of 1.5×2/3×Yield Strength for membrane stress intensity and 3×2/3×Yield Strength for membrane-plus-bending stress intensity were used per *API TR 6AF1, Section 5.2*. The yield strength of the material was derated on the basis of factors in *API SPEC 6A, Annex G* for temperature of 350°F for F22. When temperature was not considered, the working allowables of 2/3×Yield Strength for membrane stress intensity and 1.5×2/3×Yield Strength for membrane-plus-bending stress intensity were used. Because the bolts are always at a temperature lower than 250°F, the allowable stress of 0.83×Yield Strength was used for tensile stress and bolts were not derated for yield strength.

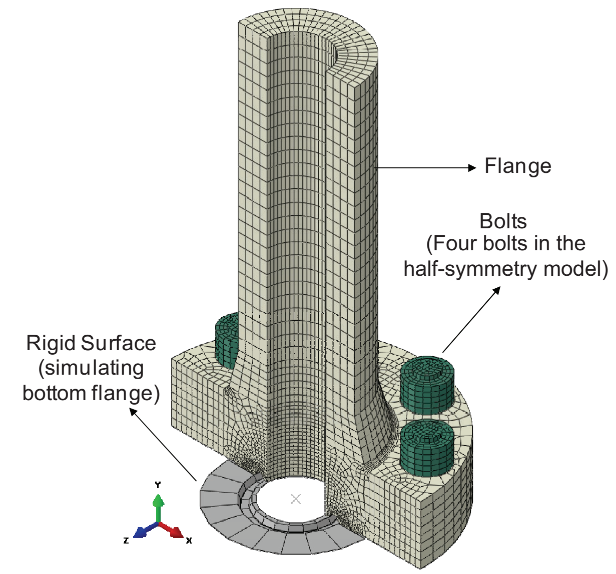


**Fig. 1—3D 180° FEA model of the 4-in., 20-ksi, 350°F API 6BX flange with 3D bolts and rigid surface simulating the bottom flange.**

Elastic-plastic FEA was conducted with the same meshed model. The true-stress/true-strain curves with minimum yield strength for 70 and 35°F and derated material yield strengths for elevated temperature of 350°F for the flange were calculated with *ASME BPVC, Section VIII Division 2, Annex 3.D* and *ASME BPVC, Section VIII Division 3 KD-231.4*. Linear elastic properties were used for the bolts. To respresent test conditions, the internal pressure was determined at the point at which the model failed to converge, which is called the plastic-collapse load. LRFD from *ASME BPVC, Section VIII Division 2, Table 5.5* (hydrostatic-test conditions) was used with the suggested hydrostatic-test factor of 1.049 (per Eq. 1) to determine the maximum allowable working pressure as 30 ksi. This is observed to be the same as linear elastic analysis methods.

Elastic-plastic FEA was also conducted for working conditions to determine the P-T-B capacity chart. A thermal-gradient equivalent to the elastic analysis was applied to the model just after the bolt preload. Then, a known external pressure and tension multiplied by the LRFD factor of 2.4 times the hydrotest factor were applied to the model. The bending moment that causes nonconvergence of the FEA because of structural instability was determined for that particular load case. Several cases were run, and results were compiled to determine the P-T-B chart for the flange. For elastic plastic analysis, LRFD of 2.4 times the hydrotest factor from *ASME BPVC, Section VIII Division 2, Table 5.5, Global Criteria 1* was used for all loads applied (P-T-B).

The treatment of thermal stress was examined for *ASME BPVC, Section VIII Division 2, Table 5.5 Global Criteria 1* and *Global*

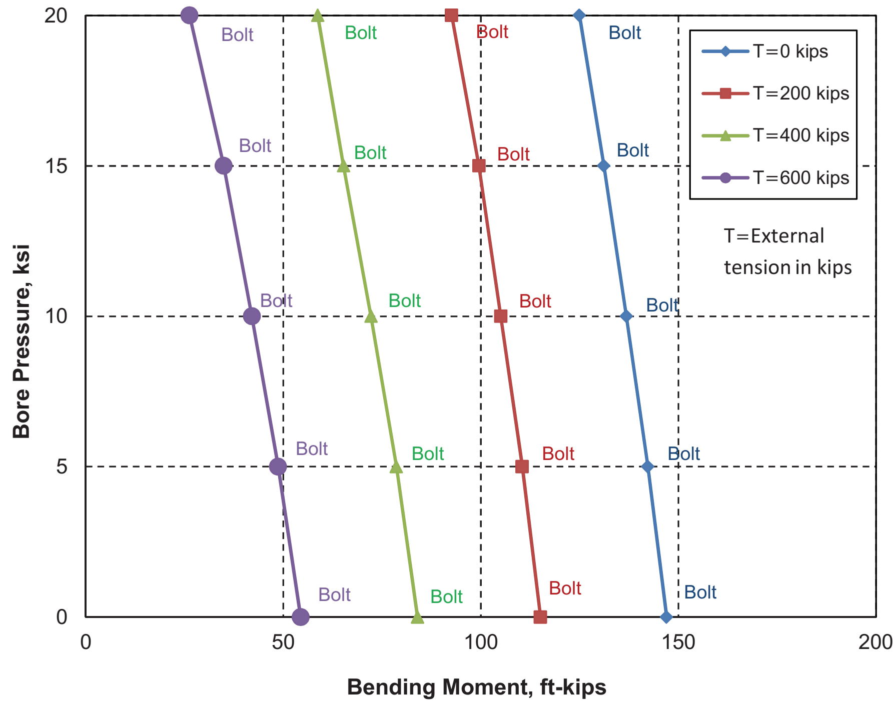


**Fig. 2—Overall P-T-B capacity of the 4-in., 20-ksi flange at 350°F by use of linear elastic- and elastic-plastic analysis methods, considering both bolt and flange structural limits.**

*Criteria 2*. The maximum differential temperatures (maximum thermal stresses) experienced by the component at the input conditions were applied to the model. These thermal stresses were combined with the 2.4 LRFD (Global Criteria 1) structural loads. This criterion was found to provide a more-conservative result than the 2.1 LRFD for all loads, including thermal loads (Global Criteria 2). The external loads were not considered as "live-appurtenance loads," and, hence, were not used with the LRFD of 2.7. Two charts have been presented in the paper. **Fig. 2** shows the P-T-B chart for the overall capacity of the flange, taking into account both flange and bolt structural limits, with both elastic-plastic- and linear elastic analysis methods. **Fig. 3** shows the P-T-B chart con-

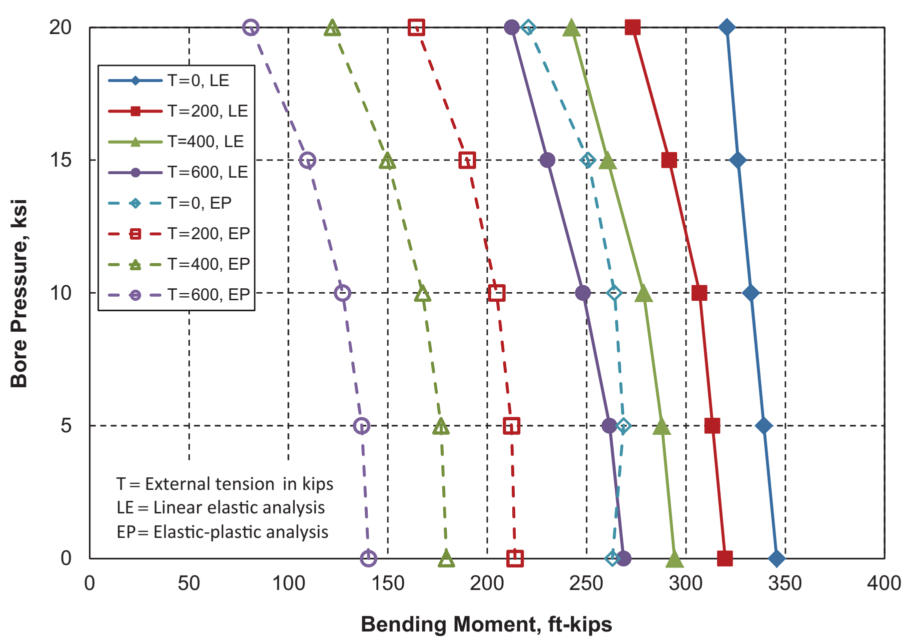


**Fig. 3—P-T-B capacity of the 4-in., 20-ksi flange at 350°F to compare the linear elastic and elastic-plastic analysis methods, considering only flange structural capacity.**

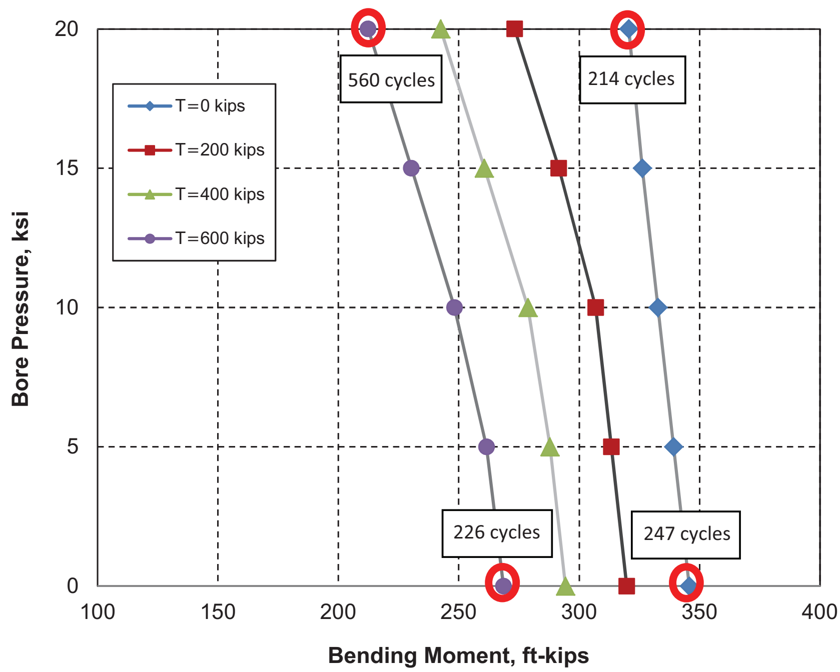


**Fig. 4—Design cycles by use of fatigue analysis for identified load cases of the P-T-B- capacity chart at 350°F, generated with linear elastic anlaysis for flange structural capacity.**

sidering only the flange structural limit for both elastic plastic and linear elastic methods for comparison.

It is observed from Fig. 2 that the capacity of the flange is limited by linear elastic analysis for all the load cases considered on the basis of bolt failure. The chart is comparable to the *API TR 6AF1* charts for the 4-in., 20 ksi, 6BX-type flange and shows slightly lower capacities. In contrasting these two methodologies, *API TR 6AF1* considers an axisymmetric FEA model to determine the P-T-B chart for this flange. The bolts are evaluated by use of an axisymmetric model, with the bolt stresses averaged for all eight bolts. The results reported in Fig. 2 use a 3D model to examine the stresses in each of the bolts. With external bending applied, the

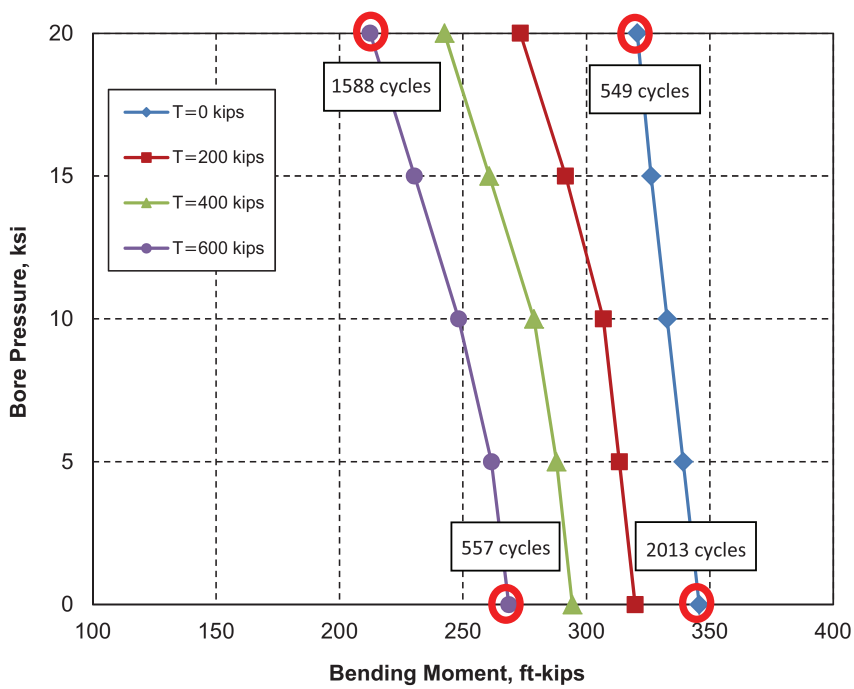


**Fig. 5—Design cycles by use of fracture-mechanics analysis for extreme load cases of the P-T-B capacity chart at 350°F, generated with linear elastic analysis.**

outermost bolts reach the maximum allowable stress before other bolts. This is the reason that the capacity observed is slightly less than that in the *API TR 6AF1* charts.

As seen in Fig. 3, the elastic plastic method generates a more-conservative result than the linear elastic methods. This is because for linear elastic anlaysis, the thermal loads are categorized as secondary stresses, whereas for elastic plastic analysis, they have been considered as design loads per *ASME BPVC, Section VIII Division 2, Table 5.5, Global Criteria 1*. Secondary stresses allow higher design allowables than the general-stress allowables, and the LRFD of 2.4 plus thermal stresses provide a conservative result. However, as observed from Figs. 2 and 3, the overall capacity of the flange is still limited by the linear elastic analysis on the basis of bolt-failure criteria.

**Protection Against Cyclic Loading.** To demonstrate the design methods for cyclic loading, the structural capacity of the flange was evaluated for extreme load cases from the P-T-B chart generated for flange stress capacity by use of linear elastic analysis. It was assumed that for one cycle, all the loads applied after bolt preload went from zero to the actual value and again went back to zero. For example, for a load case of 20,000-psi internal pressure, 600-kips external tension, 120-kips-ft external bending moment, and temperature of 350°F, one cycle of loading would be zero loads to 20,000-psi internal pressure, 600-kips external tension, 120-kips-ft external bending moment, and temperature of 350°F and back to zero loads. Design cycles were calculated for these cases by use of fatigue analysis and fracture-mechanics analysis per *ASME BPVC, Section VIII Division 3*.

The flange under consideration is assumed to be made from F22, low-carbon-alloy steel with 75,000-psi yield strength and 95,000-psi ultimate tensile strength. Published material properties have been used to determine the design cycles for both fatigue and fracture-mechanics analysis.

***Fatigue Analysis.*** Fatigue analysis was conducted per *ASME BPVC, Section VIII Division 3, KD-3*. Because *ASME BPVC, Section VIII Division 3, Table KD-320.1* provides the material-fatigue curve for low-carbon-alloy steel for ultimate tensile strength of 90,000 psi, this curve was used for fatigue analysis. Half of the maximum stress intensity from the linear stress-analysis results was used for the equivalent stress amplitude, and the number of design cycles was calculated. The allowable cycles were taken as one-tenth of the number of cycles calculated by use of the formulation from *ASME BPVC, Section VIII Division 3*. **Fig. 4** shows the design cycles for the extreme load cases considered.

***Fracture-Mechanics Analysis.*** Fracture-mechanics analysis was conducted per *ASME BPVC, Section VIII Division 3, KD-4*. For material properties, the fracture-toughness value of 138 ksi$\cdot\sqrt{\text{in.}}$ was used from the published report by Young (2008), which corresponds to the seawater-with-cathodic-protection environment and was determined per *ASME BPVC, Section VIII Division 3* by the crack-tip-opening displacement method. Recommended crack-growth-rate factors for steel in a marine environment with cathodic protection were used from *BS 7910* (2005), as listed in Eq. 2 in the form of the Paris-law equation:

$$\frac{\mathrm{d}a}{\mathrm{d}N}\frac{\text{in.}}{\text{cycle}} = 5.22\times10^{-7}\left(\Delta K\ \text{ksi}\ \sqrt{\text{in.}}\right)^{1.4}. \qquad (2)$$

For the crack orientation, both circumferential and axial crack orientations were considered. Crack locations on both the inside surface and the outside surface were based on the stress distribution. This was needed because of the compressive thermal stresses on the inside surface, which prevent crack propagation. The thermal stresses were considered as secondary stresses during calculations. For the load case with zero internal pressure and zero external tension, the flange was loaded in such a way that the stresses were predominantly hoop stresses. Therefore, an axially oriented crack on the outer surface, with a radial depth, yielded the most-conservative result. For the other three load cases identified, the flange was loaded in such a way that the stresses were predominantly axial stresses, and, hence, a circumferentially oriented crack on the outer surface, with a radial depth, yielded the most-conservative results. This places the largest initial crack perpendicular to the worst-loaded plane, and initializes crack propagation at the highest stress level in that plane. The crack was assumed to be semielliptical in shape, with a ratio of depth-to-surface length of 1:3 per *ASME BPVC, Section VIII Division 3, paragraph KD-411*. This initial flaw is placed along the failure path determined from the structural linear elastic analyses. An initial flaw length of $^{3}/_{16}$ in. was used for the analyses. Crack-face pressure was included in the evaluations when internal pressure was considered. Commercial software SIGNAL Fitness-For-Service version 4.0 by Quest Reliability was used for the analysis. The allowable final-crack-depth criteria per *ASME BPVC, Section VIII Division 3, paragraph KD-412* were used to determine the allowable cycles for a particular load case. **Fig. 5** shows the allowable design cycles by use of fracture-mechanics analysis for the load cases considered.

## Validation of Design Methods

Validation of the example analysis can be demonstrated by comparing the capacities with the published combined-load-capacity charts for the 4-in., 20-ksi, 6BX-type flange in *API TR 6AF1*. Similar capacity ratings were obtained when analyzed with these design methods. *API Report 86-21* (1987) demonstrates the validation of the flange-capacity charts by testing a $7^{1}/_{16}$-in., 10-ksi, 6BX-type flange for combined loadings. Cameron International Corporation (Cameron) has also successfully tested an 11-in., 25-ksi flange for hydrostatic test pressure of 37.5 ksi with the flange and bolts strain gauged, which provided accurate correlation to predicted values from the finite-element-model analyses. Cameron has successfully tested a $2^{9}/_{16}$-in., 25-ksi, Cameron Gate Valve for cyclic loading from 0 to 30 ksi for 500 cycles, validating no change in preload of the bonnet and end flanges. Additionally, post-test NDE inspections were performed to validate that no fatigue damage was experienced by the valve body as a result of crack growth. This same valve was also tested successfully for a hydrostatic test pressure of 37.5 ksi. The bonnet, bolts, and flanges were strain gauged, which provided accurate correlation to predicted values from the finite-element-model analyses. These test data demonstrate the validation of the design methodology outlined in this paper.

## Application

Two primary design methods have been presented and contrasted in the preceding sections, with both of the results approximating each other for the overall capacity. During the writing of this paper and while analyzing the example flange, a senior experienced analyst was used for approximately 2 months for analyzing, post-processing, verification, and documentation. During this period, approximately 8 days were spent on the linear elastic and fatigue portion of the work. Approximatly 60 days were spent on the elastic plastic and fracture-mechanics portion of the work. With both methods providing approximately similar results in both overall capacity and cycle life, the primary difference was a 650% increase in manpower needed to perform the elastic plastic and fracture-mechanics analyses. Even with the analyst's experience, the elastic-plastic analysis results were often subjected to numerical instability, boundary-condition influences, and interpretation of a point of "nonconvergence" as a result of unbounded deformation.

These results must be accounted for during the planning of a work package in which elastic plastic and fracture-mechanics analyses are required or anticipated. Staffing needs and experience levels must be planned, and the results/capacities must be reviewed. With the proliferation of FEA software and easy-to-use programs, the generation of results has become trivial. However, the establishment of the problem (boundary condition) and the interpretation of the results (nonconvergance) are still extremely difficult. The burden of establishing if a component is safe for use shifts from the design engineer to the analyst, resulting in a historic reversal of roles.

Given the geometries associated with pressure-containing equipment rated for more than 15,000 psi, the classification and treatment of stress needs to be evaluated carefully. In many cases, the elastic plastic methods presented previously are the only appropriate tools to use for the definition of component capacity. This is because stress linearization is not appropriate for heavy-walled vessels, especially around local or discontinuous regions. It is also seen that when large thermal gradients (stresses) are classified as secondary stresses, the capacity charts indicate the increase in capacities because of higher design allowables for the secondary stresses. However, it should also be noted that for the evaluation of more-standard load cases (e.g., P-T-B with no thermal considerations), the linear elastic methods provide equally conservative results. Both methods can be used together to provide efficient answers, with linear elastic methods being used initially and elastic plastic methods used to justify the linear elastic assumptions.

Reviewing fatigue and fracture-mechanics analyses for cyclic loading, both methods provide approximately equal results for allowable design cycles. The primary contrast between the methods lies in the material-data requirements. Fracture mechanics requires more material testing and characterization, even between the same material batches, while fatigue analysis relies on more generally accepted material assumptions with larger safety factors. Both methods are well posed to analyze ductile materials, which are required for pressure-containing subsea equipment.

## Conclusions

Several design-analysis methods are given in this paper to verify the capacity of components or assemblies under consideration, including protection against plastic collapse, protection against local collapse, and protection against cyclic loading. An approach to closure bolting is also presented. Both linear elastic and elastic plastic methods are given for the protection against local and plastic collapse. Using these methodologies, an example design analysis was presented for a 4-in., 20-ksi API 6BX flange at 350°F.

The linear elastic and elastic plastic methods of analyzing the sample flange gave the same pressure rating when only internal pressure load was applied. The overall capacity of the flange for hydrostatic-test condition is limited by bolt, which is analyzed using linear elastic methods. When external loads were used to determine the combined structural P-T-B capacity of the flange at 350°F, it was observed that the linear elastic methods gave the most-conservative results. When examining only the capacity of the flange and allowing the bolt stresses to exceed allowable values, the elastic plastic method gave more-conservative results than the linear elastic methods. This was based on two assumpations: the use of stress categorization in linear methods of thermal stresses as secondary stresses and the choice of *ASME BPVC, Section VIII Division 2, Part 5, Global Criteria 1*, which conservatively applied thermal loading in the elastic plastic methods. The overall capacity chart reported is comparable with the *API TR 6AF1* chart and verifies the results obtained.

Fatigue and fracture-mechanics analyses for cyclic loading are reported for the identified load cases on the P-T-B chart at 350°F. The reported cycles show that the flange design has a cyclic-loading capacity that meets the standard cycle requirements (200 cycles) for subsea equipment, but should be carefully evaluated for higher-cycle applications.

**Limitations of Work.** The work and results presented in this paper are not meant to provide a definitive solution to building equipment for working pressures greater than 15,000 psi and temperatures greater than 250°F. The purpose of this paper is to show that it is possible to combine the methods of both *API SPEC 17D* and *ASME BPVC* to achieve working pressures greater than 15,000 psi and temperatures greater than 250°F because there is not a published opinion. The intention of this work is to show that existing API methods are adequate for working pressures up to 25,000 psi and temperatures up to 400°F. The 400°F limit is currently based on the material-data availability. The recommendations presented in this paper are not meant to act as a new standard; an in-depth analysis should be performed on each component or assembly to verify that the proper requirements are met.

The methods published in this paper are limited to API Subcommitee 17, Subsea Production Systems, but these methods can be extended to other API subcommitees by use of similar methodology and changing the factors on the basis of respective applicable design codes subject to sufficient verification and validation.

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