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SUBSEA FLANGES, COMPARISON BETWEEN COVENTIONAL API 6A TYPE 6BX FLANGE AND SPO COMPACT FLANGE DESIGNS

Przemyslaw LUTKIEWICZ[1],David ROBERTSON
Freudenberg Oil & Gas Technologies
Metal Sealing Solutions - Vector Products
Tollbugata 49
3044 Drammen (Norway)

Sam (Kwok Lun) Lee
TechnipFMC
11740 Katy Freeway
Houston, Texas 77079 US

ABSTRACT

The API flange design is a well-known commonly used solution. The flange concept was developed in late 1920s and 1930s by Waters and Taylor. The design methodology of the flange was published in 1937[1], well known as the "Taylor Forge method". This is still the basis of the present ASME flange calculation. The design is based on the simple elastic principles and linear stress analysis/calculations. The conventional flange type dimensions are described in API 6A [2] and analyzed in API 6AF [3] and 6AF2 [4]. On the other hand, the Compact Flange concept was presented first by Webjørn in 1989 VCF joint [5]. It is based on plastic theory equations and plastic collapse capacity. In 1989 the initial concept was adopted by the Steel Product Offshore (SPO) company for oil industry by equipping flange with HX seal ring for raiser and subsea use. After that a topside budget version (with simpler IX seal ring) was prepared by SPO and presented on PVP 2002 conference [6][7][8]. The Compact Standardized and simplified flange design with IX seal ring is defined and described in ISO-27509 [9]. As for today, along ASME B.16.5 [10] pressure classes range, SPO CF 5K, 10K, 15K and 20K rating flange classes were designed and are in use. The main advantages for CF design are reliability, low weight/compact dimensions and static behavior compared to the conventional design. The design is already well known and commonly uses for European region (mostly Norway). Despite its benefits, CF is still rare outside Europe region. A comparison between those two different concepts will be presented in this paper followed by the examples and Finite Element Analysis (FEA). In case of FEA the Compact Flange design is more suited to the plastic collapse analysis than to elastic stress evaluation as it is for API, therefore comparison between different FEA approaches will be studied in addition.

Keywords: FEA, Compact Flange, API flange

NOMENCLATURE

2D	two dimensional
3D	three dimensional
APDL	ANSYS Parametric Design Language
BCO	Boundary Conditions
BM	Bending Moment
CF	Compact Flange
DNV	Det Norske Veritas
FE	Finite Elements
FEA	Finite Elements Analysis
HPHT	High Pressure High Temperature
P	Pressure
R&D	Research & Development
SCL	Stress Classification Line
SCF	Stress Concertation Factor
SPO CF	SPO Compact Flange

[1] Contact author: przemyslaw.lutkiewicz@fogt.com

INTRODUCTION

The history for CF designs starts at year 1967. In that year Haagen [11] describes first time the flange design which was based on the modified raised face flange. The same year Webjørn [5] introduced a gasket free CF design. This design did not include a gasket or seal ring. The high contact forces on the bore diameter for a taper flange face contact is high enough to ensure tightness. The design also uses higher bolt pretension than conventional flanges (80% of bolt strength). The design was proved to work and described further by other authors: Schneider [12], Hyde et al [13]. Finally, generally accepted, CF design was introduced to the market by SPO company. To ensure the high level of safety HX the seal ring was introduced as an additional seal. That also fulfilled the double seal requirements for some first projects, where SPO CF design was used subsea. During the field experience it appears even if theoretically the seal ring is not needed, practically because of the scratching possibility on the taper flange face, the seal ring is more reliable as a primary seal.

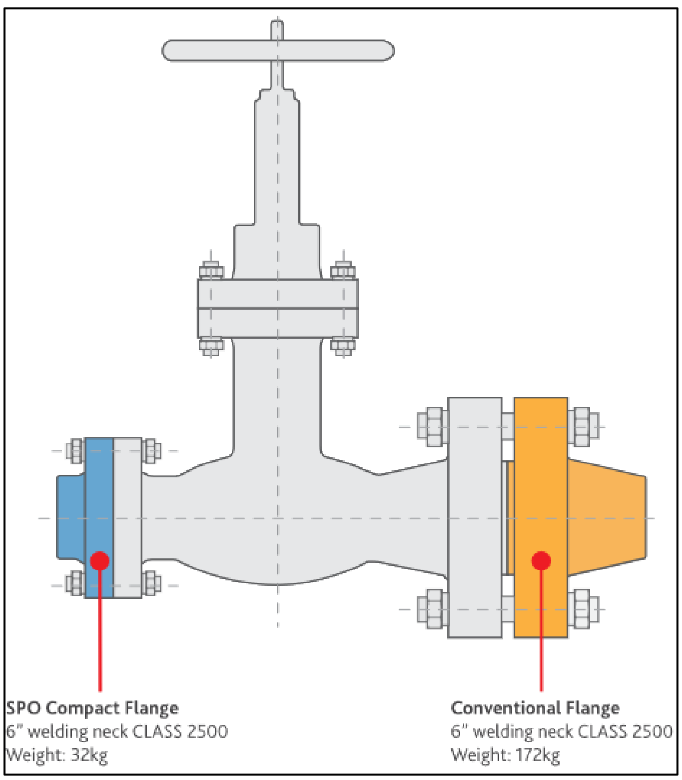

FIGURE 1: Simple comparison between SPO CF and ASME B.16.5 flange (6" size, 2500 pressure class).

The original scope for SPO CF design was to cover ASME B.16.5 [10] pressure classes. Comparing to the conventional B16.5 flanges, SPO CF typically weight 70-82% less (see FIGURE 1). The first SPO CF was designed for Saga Petroleum's riser for the Ekofisk field. After that 1100 flanges were delivered for Snorre TLP project. Those flanges were equipped with HX seal ring type to aimed dual barrier requirement for subsea connections. Based on good experience, Norwegian Oil industry together with SPO decide to release the topside simplified and standardized design with IX seal ring in form of the national NORSOK L-005 [14] standard in 2001 and international ISO-27509 [9] standard in 2012. The scope and methodology behind those standards were presented at ASME PVP 2002 [6][7][8] conference to further propagation of the design.

Further SPO CF development included the introduction of 5K, 10K, 15K and lately 20K pressure class design. The 20K range was type approved by DNVGL to meet the requirements of API 17TR8 [15] in 2018. In addition, the fugitive emission tests were conducted successfully [16][17] and SPO CF design was tested and confirmed as reliable as a 12m pipe with one butt weld [18] [19]. The tightness level of $1x10^{-5}$ to $1x10^{-6}$ cm^3/sec/mm sealing diameter was also proved by testing.

The SPO CF flanges are used often in the offshore industry especially in critical application where reliability and compactness are the main interest. Risers are the perfect example, where extreme environmental loading are often combined with high pressure loads and affected by cycling (fatigue). The other proved area of use is related with high [14] or low temperature applications. The temperature range where SPO CF is still reliable vary from cryogenic (see FIGURE 3) up to 1328°F (720°C). High reliability even in high temperature make SPO CF good alternative for welded connection, especially when difficult austenitic to ferritic pipe connection needs to be made.

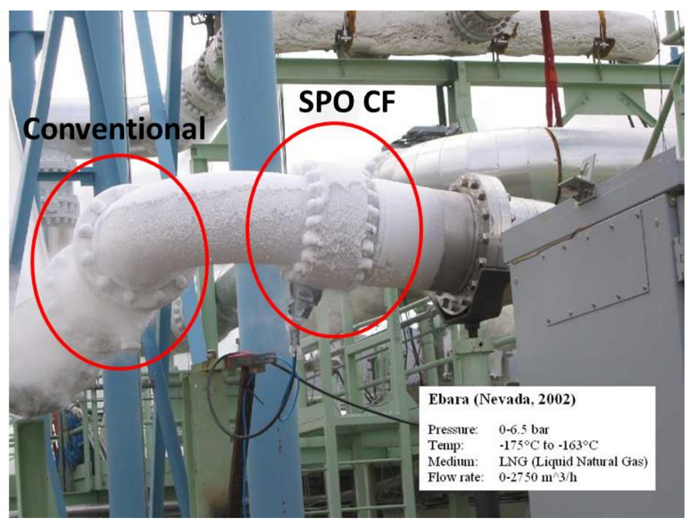

FIGURE 2: Cryogenic SPO CF application (Ebara, 2002).

Prior to 1950's, the design of using traditional methods results in the flanges that were impractically large. Flanges with pressure-energized ring gaskets were developed by the standardization Committee of the Association of Wellhead Equipment Manufacturers (AWHEM) for 15,000 psi and 10,000 psi working pressure [20][21] and later adopted as standard by the American Petroleum Institute (API). They are fully described in the API standard 6A, second edition, 1963. API included

15,000 psi and 20,000 psi flanges in more recent editions of API Spec 6A. API classes all flanges that accept BX ring gaskets as 6BX flanges. Type R and RX gaskets shall be used on 6B flanges. Both R and RX gaskets are not used on flanges with pressure rating above 5000 psi. API 6BX open face flanges must have raised faces, while API 6BX Studded Face flanges may have their raised faces omitted. The raise face flange (type 6BX flange) intends to carry some of the bolt load, which results in less bending of the flange. These flanges are primarily used for wellheads and valves in the oil field service. All API flanges require ring type joint facings with the proper gaskets for optimal integrity of their application.

The first edition of API Spec 17D [22] published in 1992. They use the same dimension of API 6A flange with BX ring gaskets in subsea applications. It requires all flanges used for subsea applications to have ring grooves manufactured from, or inlayed with, corrosion resistant alloy, such as UNS NO6625 grade. Style BX pressure energized Ring Joint Gaskets are designed for use on pressurized systems up to 20,000 PSI. manufactured in accordance with API 6A [2]. R gaskets, which are not designed to be pressure energized and RX gaskets, which are pressure energized without face to face make-up.

API TR 6AF [3] was done with 2D finite element analysis in order to find the bending and tension capacity of API flange. Later, API 6AF [3] was done to determine the effects of high temperature, 350°F (177°C) or 650°F (343°C), on API flange. It shows both the stress criteria and the leakage criteria. The leakage criteria were too conservative as shown by subsequent testing. API TR 6AF2 [4] was done with 3D analysis of the same flange. The results of 6AF [3] and 6AF2 [4] are generally in agreement.

There are many different sizes, pressure ratings, gasket types (R, RX, and BX) and flange types (integral, blind or welding neck flanges) for the API flanges. Type 6B flanges with maximum working pressure of 5000 psi and type 6BX flanges up to 20,000 psi are very common for subsea and surface application in the oil and gas industry.

DESIGN CONCEPT DIFFERENCES

There are several differences in the design principles between conventional API 6A [2] and CF. Conventional API 6A [2] flanges are designed for internal bore pipe diameter and sizing is made around it. On the other hand, SPO CF design is based on the pipe outside diameter and sizing is made around this dimension. The high-pressure classes, size range is much more reach for SPO CF and existing for larger flanges, especially for weld neck flange type. It can be seen easily that for 5K pressure class SPO CF sizing trends to be more compact (see FIGURE 3) and using lower size bolting. This trend will disappear for 15K (see FIGURE 4) and 20K flanges, however for SPO CF, large sizes are still available (24in for 15K and 18in for 20K).

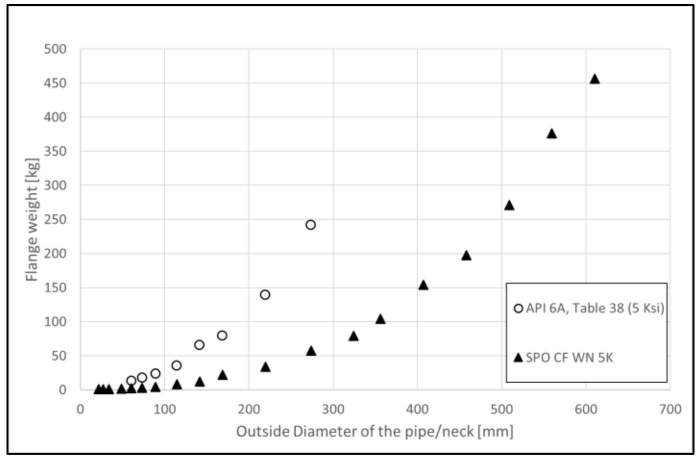

FIGURE 3: Weight comparison (in kg) between conventional API 6A and CPO CF for 5K pressure class.

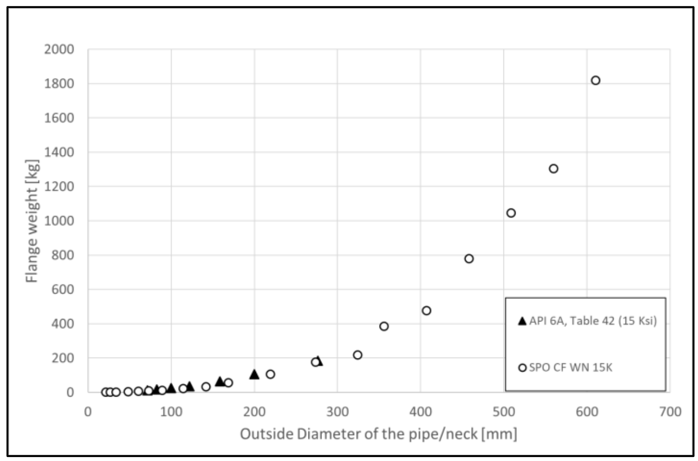

FIGURE 4: Weight comparison (in kg) between conventional API 6A and CPO CF for 15K pressure class.

The other difference is that for convectional API 6A [2] flanges were designed for room temperature pressure rating. Bolts are not considered. The design was checked by FEA and findings were described in the API technical Reports 6AF [3] and 6AF2 [4]. The SPO CF design sizing is based on conservative assumption of the uniform 482°F (250°C) temperature for flange and bolts. The exception is SPO CF 20K pressure class, which was designed for 350°F (177°C). The SPO CF utilization at max. temperature and under maximum pressure related with pressure class is set to 50% of the flange capacity. In that way there is still plenty of capacity left to accommodate external loads. The common practice is to use one pressure class higher flange dimensions, for the convectional API 6A [2] design, in case of use in elevated temperatures. It should be also highlighted that for high pressures (15K and 20K) API flanges need to use strong, min 75K in yield strength material. For SPO CF it is still allowed to use 65K in yield flange material. This reflects the fact that SPO CF design (flange dimensions) are based on 65K material properties. The use of 65ksi material is to open the flange to market outside of wellheads as defined in API 6A [2].

Elastic & Elastic-Plastic

The conventional flange type calculation was developed by Waters et al in 1937 [1]. At 1943 the gasket factors were introduced by Rossheim and Mark [23]. This elastic based calculation method is commonly known as Taylor Forge method and is bounded by several limitations (see list in PD 6438:1969 [24]). ASME VIII, div.2 [25] is using this method for a convectional flange calculation (section 16.4) and is the most recognizable. Another variation of the method can be found in EN 13445-3 [25] code.

ASME VIII [25] was first published in 1915. The code is based on the allowable stress concept which takes into account safety factor and material strength. Originally the safety factor was 5.0 in relation to the material tensile strength. After it come down to 3.0 value and lately to 2.4. To be more precise allowable stresses are based on the min of yield strength by 1.5 and tensile strength by 2.4, however from the practical point of view the 1.5 safety factor over yield is covered mostly for austenitic materials only. The same 2.4 safety factor value is used for EN 13445-3 [26] code.

The stress criteria for the hub, flange and bolt are based on ASME code stress categories using the basic allowable membrane stress intensities defined by API 6A [2]. The allowable membrane stress intensity is 67% of the minimum yield strength of the flange/hub. The allowable membrane stress intensity for the hydrostatic pressure at room temperature is defined as 83% of the yield strength in the flange/hub location.

The SPO CF design is based on the design with tapered flange face, which is in contact outside the bolt circle (environmental wedge seal). Therefore, Taylor Forge method presented in e.g. ASME VIII, div.2 [25], section 16.4, is not applicable.

For the SPO CF, leakage will appear only in case of large flange separation, which will be triggered by excessive yielding. Unstable fracture failure is covered by material selection and quality check. Fatigue failure is covered by the static behavior and low SCF for the elliptical transition. Therefore, the main failure mode for SPO CF is by excessive yielding (gross plastic deformation). The ISO-27509 [9] code defines the CF structural capacity based on empirical equations. The methodology is based on the tensile plastic capacity of the warped flange ring. The pipe/neck interaction (but not capacity) and prying effect for wedge contact is also considered. Maximum allowable design loads are set to be 2/3 (=1/1.5) of flange structural capacity. This is similar value as 1.5 safety factor used in API 6A [2]. ISO-27509 [9] methodology is based on global approach as the safety factor is applied to the CF capacity and not stress evaluation. This analytical method has a good correspondence to the Limit Load FEA and is still conservative to test as no strain hardening is utilized.

Gasket & Seal Ring

It is important to distinguish between the gasket and seal ring concept. Both have the same purpose, to seal the connection, but the way how they work, is completely different.

The API conventional flange type is using the gasket concept. The gasket sealing ability is related with axial compression introduced by the flange bolt tightening. By high axial compression stress between the flange groove and gasket itself is tightening is provided. The gasket is force controlled (tightness depends on the compression force). The loads needed to activate the seal are quite high compare to the bolt pre-tension. Gasket compression is in the axial direction which is in parallel to the main load direction (end trust from the pressure load, external tension load and bending moment). Although, large part of load is transferred from flange to flange by the gasket and therefore the loading conditions are influencing the tightness.

During the API 6A [2] conventional flange make-up, gasket always makes a contact in the outside diameter of the groove first before touching the inside diameter of the groove later [28]. The gasket will be plastically deformed until the raised face contacts when the bolt is fully preloaded. When the internal pressure is applied, the gasket is energized against the groove. However, the pressure end load will start to separate the flanges when the preload is overcome. The contact pressure will decrease until the internal pressure leaks.

The CF design uses a seal ring (Primary Seal). The seal ring is acting on the radial direction, which is perpendicular to the main loads. It is displacement driven as it is radially distorted (self-energized) during the bolt pre-tensioning. The seal ring tightening force/contact pressure is provided by the elastic energy stored in the ring by radial compression. The bolting force needed for tightening (pre-energizing) the seal ring is low (around 5% for medium size). Seal rings do not transfer loads from flange to flange. Loads are transferred by flange to flange through the flange face contact areas; heel and wedge (see FIGURE 5). Therefore, for the CF, the seal ring is not influenced by the load variation until the flange to flange contact is maintained. This is also related with static connection feature, which will be described later in the detail. The seal ring is pressure energized when exposed to the pressure load. It means, that with higher pressure, higher contact stresses are created between seal ring and seat. Different seal ring types are available depending on the application. As the seal ring is located in a seal ring groove and connection is static, the seal ring is protected from corrosion, wear and fretting during operation. For the topside use, the IX seal ring is a common choice. The IX seal ring dimensions for simplified and standardized topside CF can be found in ISO 27509 [9]. For subsea applications the bi-directional HX and DuoSeal type seal rings are offered for SPO CF design.

It should be remembered that the seal ring is a primary seal for the CF. In addition, the heel contact itself is a barrier/seal which works in the similar way as gasket. It seals in the axial direction and is affected by the loading. As this is less robust and reliable seal type (easy to damage the contact surface) it is defined as a Secondary Seal for CF. Undamaged flange heel is in contact for most flanges up to 1.5 of the flange pressure rating. It will be active for maximum allowable design loads and may not seal at any extreme load conditions. In the capacity hand calculations and FEA analysis is often conservatively assumed as not tight. At the end the environmental seal made by the wedge contact should be mentioned. The wedge seals the bolts against the outside corrosive agents. The wedge closure is also used as a practical/visual check for proper flange assembly. Most of the bolt pre-tension load is transferred by the heel contact which is within the bolt circle. Environmental wedge seal takes only around 10 to 20% of bolt pre-tension. The other benefit from such force balance is the insignificant bolt prying effect compare to conventional flange type.

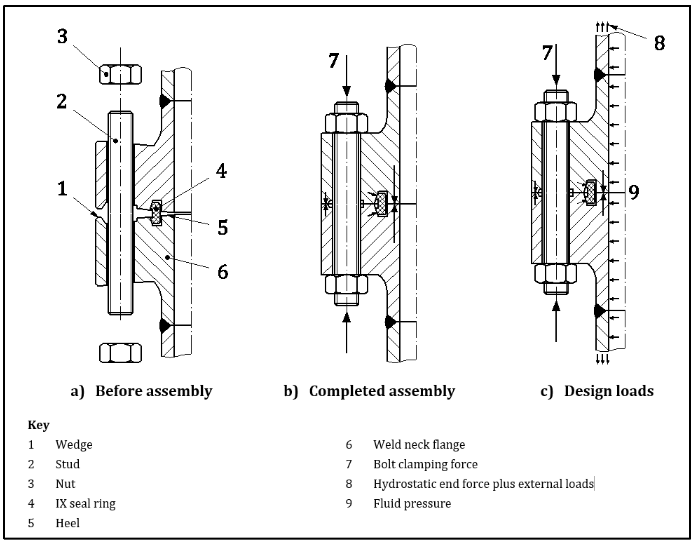

FIGURE 5: CF assembly stages, contact areas definition (after ISO 27509 [7])

Bolt Stresses & Pretension

The target bolt pre-tension level for CF connection is 75% of bolt yield. The long-term residual bolt pre-tension is assumed as equal to 70% to account for uncertainties in pre-loading procedures and long-term relaxation (related with thread crushing and/or embedding, coating compression, windup relaxation, lubrication compression and "squeeze-out" and other effects). By using high pre-tension together with pre-energized flange ring by warping, the static connection is assured for high design loads. The bolt force is stable (see FIGURE 6) and even for the highest allowable design load condition there is usually no more than 10% difference from the initial pre-tension value at design temperature or 5% for room temperature following ISO 27509 [7]. Therefore, by assuring the static connection, the highest bolt-loading is available as no major bolt force variation is introduced during the flange life (not fatigue sensitive).

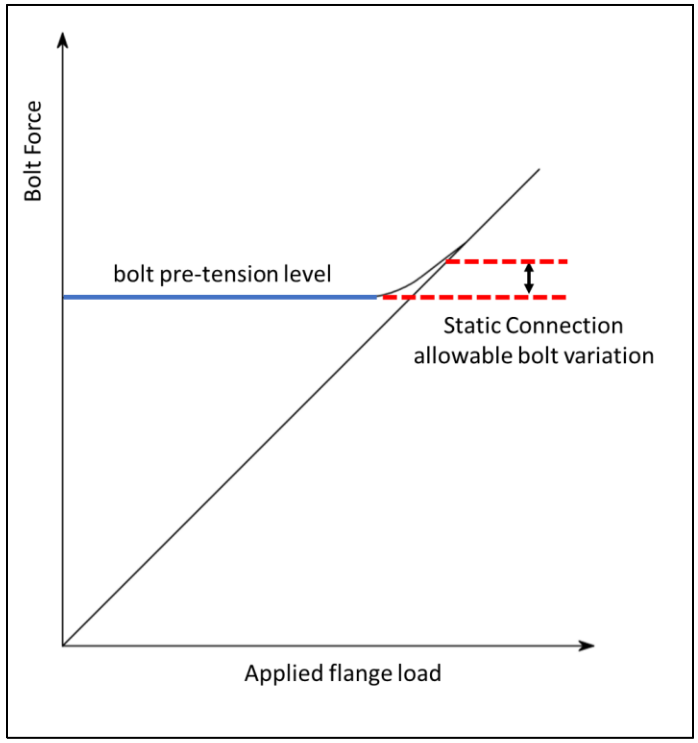

FIGURE 6: Bolted joint diagram, static condition criterion.

For the API 6A [2] conventional flanges, the allowable tensile stress in the closure bolting is $S_A=0.83S_y$, where S_A is the maximum membrane stress in the bolting for all loads and S_y is the minimum specified yield strength of bolting. Bolt stress is calculated by dividing the tension in the bolt by the root area. There is no requirement to bolt bending stresses. Bolt pre-tension level is set for 50% bolt material yield strength and is lower than for CF type.

Static Connection

The CF connection was designed and tested at first for risers. As the fatigue resistance and reliability is highly demanded for such applications, CF was designed specifically for it. The key to success was to use the seal ring (not a gasket) and pre-energize the whole flange by warping. By using the seal ring instead of a gasket, the loads are transferred directly from flange to flange and tightness is not affected by the loading condition until flanges are separated.

To protect CF flanges from separation, the flange face is not flat, but machined with a specific angle. The angle itself is related with flange stiffness and plastic capacity limit. The idea behind is to use the flange themselves to store the pretension energy by warping them. During assembly the flange is warped by a specific angle (related with flange face angles), which transfer the loads trough the contact on the flange faces and as such the whole flange is pre-energized. The flange to flange contact on the bore diameter is maintained for all test and design load conditions. In that way the connection is static and can be treated as a pipe discontinuity for fatigue calculations. The connection remains static for all load cases which are below the bolt pre-tension level. After crossing the flange pre-energized limit by higher than pre-tension level loads, flange is no longer static (see FIGURE 6).

To minimize SCF the elliptical transition is used between the pipe (neck) and flange ring. As the result, the connection is insensitive to cycling load and the connection fatigue life is limited by the adjoining weld.

FEA COMPARISION

Two SPO CF flanges were chosen to compare to the equivalent API 6A [2] convectional flanges. As the most common pressure rating for subsea equipment is now going towards 15ksi, this will be the pressure rating for the flanges being studied or analyzed.

Based on the API 6A [2] requirements 75ksi yield strength material is set for the flanges. Grade 22 material is chosen. The same flange material is used for SPO CFs for better comparison, even if 65ksi material can be used. To have consistence for all models the same pipe dimensions are used for API 6A [2] conventional and SPO CFs. Pipe dimensions are based on API 6A [2] dimension tables. Even if SPO CF was designed for higher temperature, for the comparison 250°F (121°C) will be used and flange performance will be analyzed by FEA. The temperature distribution is obtained from the thermal analysis. The boundary conditions are defined in the way as it was described in API 6AF2 [4]. The pipe in the analysis is fully isolated (adiabatic conditions on the outer pipe surface). All external flange surfaces have fixed 30°F (-1°C) temperature. All internal surfaces (pip/flange bore) have fixed 250°F (121°C) temperature (see FIGURE 7).

For the API flanges two types of analysis will be performed. The first one is fully elastic for checking the stresses level against the allowable values. In that way the structural integrity is checked with safety factor of 1.5 (which is used for allowable stress calculation). Other analysis will use full elastic plastic material formulation. The isotropic strain hardening up to UTS value is used. This analysis is aimed to check the functionality/tightness of the flange. As a criterion the contact pressure between flange and gasket need to be greater than twice the inside pressure following guidance from ISO 13628-7 [27] (Annex H).

For SPO CF three types of analysis will be conducted. Functionality/tightness will be checked with the same method and analysis steps as for conventional API 6A [2] flanges. The other analyses will be used to check the capacity of the flange by using elastic ideal plastic material properties and using safety factor on the loads directly. The last load at which analysis is still stable, divided by safety factor 1.5 is the maximum allowable load for the flange. The third additional analysis will be similar to the second one, but with full elastic plastic material properties (isotropic strain hardening up to UTS value). For that analysis an additional criterion will be introduced to limit the maximum total strain to 5%. All results will be discussed and compared in next sections.

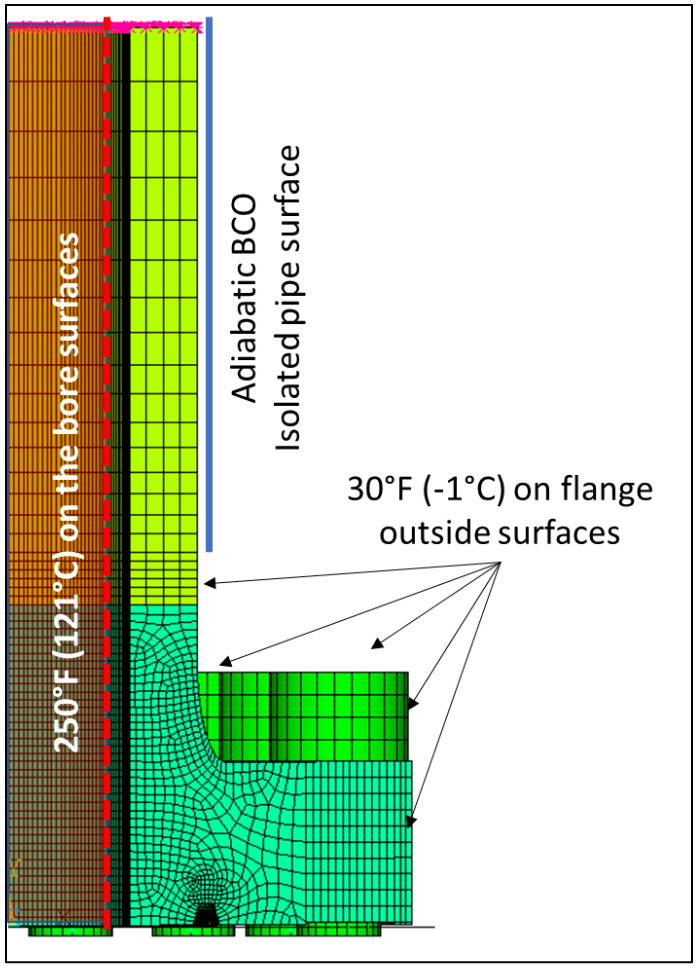

FIGURE 7: Thermal boundary conditions for FEA (only 90° section of 180° the model of 10” SPO CF WN 15K HXL-308 flange is shown).

Reference data and hand calculations

The allowable stress and leakage criteria, which is defined in PRAC 88-21 were used to generate the load rating charts in API Technical report 6AF [3] and 6AF2 [4] type load rating charts (see FIGURE 8 and 9). The applied loads were bolt preload, tension, bore pressure and bending moment. Most of the time the leakage criterion is a driving one. In the origin work (API Technical report 6AF and 6AF2), leakage is assumed to occur when the net reaction force is equal to zero at the tension side of the groove (without gasket). This is a conservative assumption in sense that neglects the pressure energized effect of the gasket [29]. This explains why the leakage based rating load are usually lower than the stress based rating loads. For API 6AF2 [4], the rating was published separately for the leakage and stress criteria for clarification.

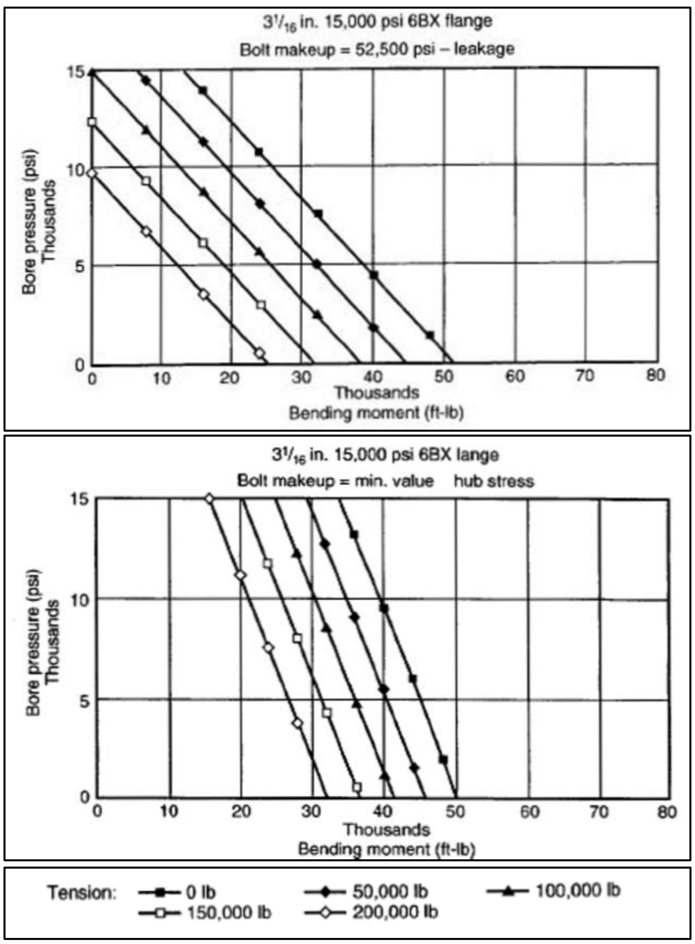

FIGURE 8: Allowable loads, 3 1/16in 15000psi Type 6BX flange (after API 6AF2)

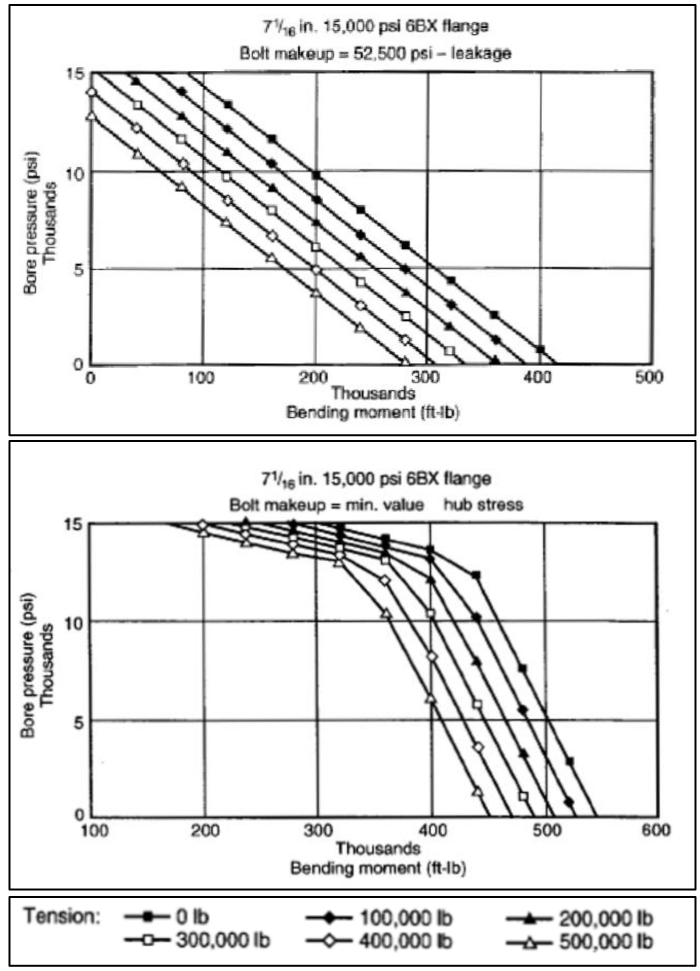

FIGURE 9: Allowable loads, 7 1/16in 15000psi Type 6BX flange (after API 6AF2)

For SPO CF the maximum allowable external loads for design pressure and temperature are calculated based on ISO-27509 [7] standard. Results for design pressure and temperature are presented in the form of a force diagram (Axial Force to Bending Moment). It is also possible to present these results in the 6AF [3] / 6AF2 [4] rating charts style, as shown in FIGURE 10 and FIGURE 11. The SPO CF flange allowable loads are governed by structural integrity (capacity). It should be highlighted that the pipe capacity is not taken into account. It is possible to generate charts for any temperature, loads and flange material. For the 4in SPO CF WN 15K HXM-132 the pipe is the weakest component and based on hand calculations for uniform 250°F (121°C) temperature. Leakage is never the problem due to static connection behavior and pressure energized seal ring design and will be also checked by FEA. The same criterion will be used for tightness check as for API 6A [2] conventional flange (based on ISO 13628-7 [27], Annex H).

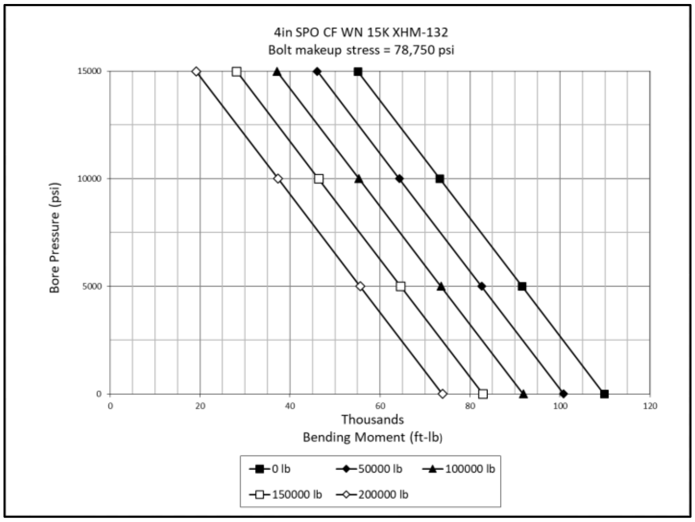

FIGURE 10: Hand calculation results for 4” SPO CF WN 15K HXM-132 (equivalent to conventional API 3 1/16 in 15000psi 6Bx flange)

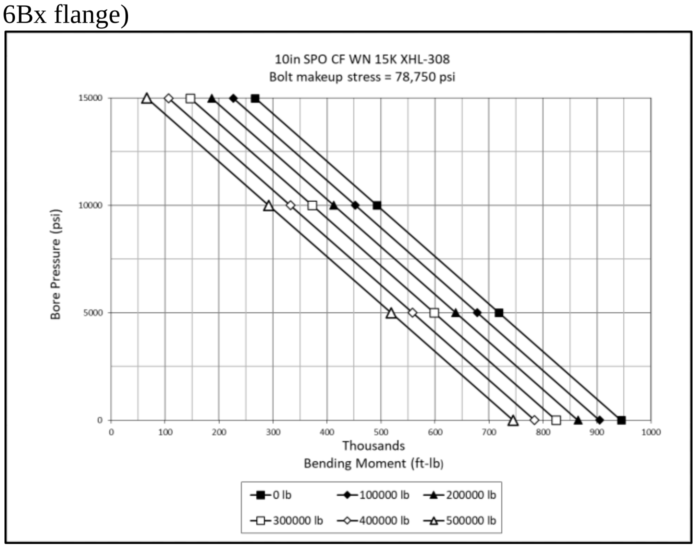

FIGURE 11: Hand calculation results for 10” SPO CF WN 15K HXL-308 (equivalent to conventional API 7 1/16 in 15000psi 6Bx flange)

FEA modeling

For the analysis two different software were used. ABAQUS for API flanges and ANSYS for SPO CFs. API models were created by using imported CAD model and interaction with the GUI of the program. For SPO CFs all aspects of modeling (setting, pre and post processing) were done by APDL scripts and commands.

The same general boundary conditions were used for all models. The 3D 180deg model concept is used. It consists half symmetric flange with half symmetric gasket / seal ring and bolts. Bolts were modeled as per guidelines in API 6AF2 [4]. The head of the bolts were dimensioned using face to face dimension for heavy hexagonal head nuts. The length of the extended hub above the flanged connection was chosen based on the minimum length required to prevent boundary conditions at the end affecting the results in the flange. A minimum six nodes through the pipe wall thickness rule was used to get accurate stress output. The contacts between bolts and corresponding bolt holes and between bolt heads and flange were modeled using frictional contacts with friction coefficient of 0.08 for API 6A [2] flanges and 0.12 for SPO CFs.

A remote point (master node) was created at the center of the top face (the end of the pipe section extension) to apply flange loads. A surface based MPC constraint was given between the master node and the top face of the pipe extension. Bending moment is applied as distributed load via master node and MPC. The same is used for the external axial tension and pressure endcap force. Pressure is applied for API and SPO CFs up to the outer gasket / seal ring contact point with the flange ring (sealing diameter). The bolt pretension is applied by pretension elements and the values given correspond to API 6A [2], Annex D, Table D.2 for API flanges and 70% of bolt material yield for SPO CFs long term conditions.

Different material modeling was also used. The elastic material properties were used for stress analysis for API flanges. Ideal plastic material was used for structural capacity in relation to hand calc. prediction for SPO CF design. And finally, full plastic material description (with strain hardening up to UTS), was used for close to real capacity modeling and functionality check. The stress-strain curve material model was following ASME VIII, div.2, Annex 3-D [25].

API FEA

In the linear elastic analysis, the stress criterion adopted was as per ASME Section VIII Div. 2 [25], which has been followed in API 6AF2 [4]. It employs the concept of stress intensity. Since in this analysis the flanges were subjected to combined loads such as pressure load, thermal load, tension, bolt make up loads and bending moment, the allowable stress values were as follows:

Allowable Membrane Component		Allowable Membrane and Bending Component	
Flange Sections	Hub Sections	Flange Sections	Hub Sections
1.5*Sm	1.5*Sm	3.0*Sm	3.0*Sm

TABLE 1: Allowable stresses for Flange and Hub sections

Stress linearization was carried out in this analysis and the stress classification lines (SCL's) were taken at exact locations as mentioned in API 6AF2 [4]. It means the stresses were extracted at the specified locations mentioned in the paper and compared against the allowable stresses. During the analysis, 4 different loading paths with constant 0 ksi, 5ksi, 10 ksi and 15ksi pressure were used to get results which can be compared with API 6AF2 [4] graphs. Three different levels of additional external tension load (0lb, 100000 lb and 200000 lb) were considered.

The comparison of the results from the FEA to API 6AF2 [4] is very similar. The bolt stresses of 83% did not govern for both 3 1/16-15K and 7 1/16-15K API flanges. That means the flange/hub stress is the one driving the structural capacity for these sizes. It is therefore concluded that the bolts will not approach their limiting criterion under the load conditions.

For elastic-plastic analysis, the material was based on full elastic plastic curve with strain hardening up to UTS value. As a criterion the contact pressure between flange and gasket needs to be greater than twice the inside pressure following guidance from ISO 13628-7 [27]. The maximum contact pressure around the circumference is always greater than the requirement at the structural capacity limit for both 3 1/16-15K and 7 1/16-15K. This means that the leakage rating load is higher than the stress based rating load with this criterion. That explains the conservative assumption by API 6AF2 [4] on leakage criterion of reaction force without the gasket included. It neglects the gasket's ability to work as a pressure energized seal.

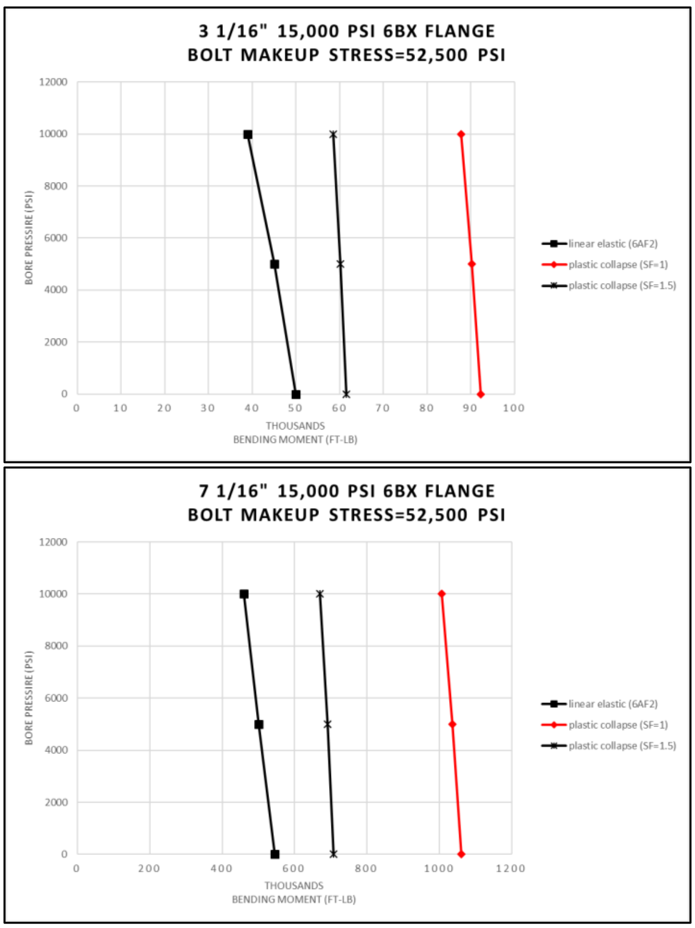

FIGURE 12: Structural capacity compared with linear elastic and elastic plastic analysis (safety factor used on loads)

Results for full elastic plastic, and linear elastic analysis can be seen on FIGURE 12. In addition, the collapse limit is represented by solid red line for comparison (a safety factor equal to 1.0 used on the loads). Using elastic plastic analysis can optimize the flange under combined loading. Even with a Safety Factor of 1.5, the capacity is still higher than the linear-elastic analysis.

The collapsing pressure and bending moment values are also extracted from the FEA results and presented below on the TABLE 2. The collapsing loads are listed for SF=1.0 and in relation to full elastic plastic model which is closed to the real material.

	Collapse BM Thousand ft-lb	Collapse P ksi
3 1/16-15K API flange	92.3	47.9
7 1/16-15K API flange	1061	47.7

TABLE 2: FEA results - plastic collapse BM and P for API flange.

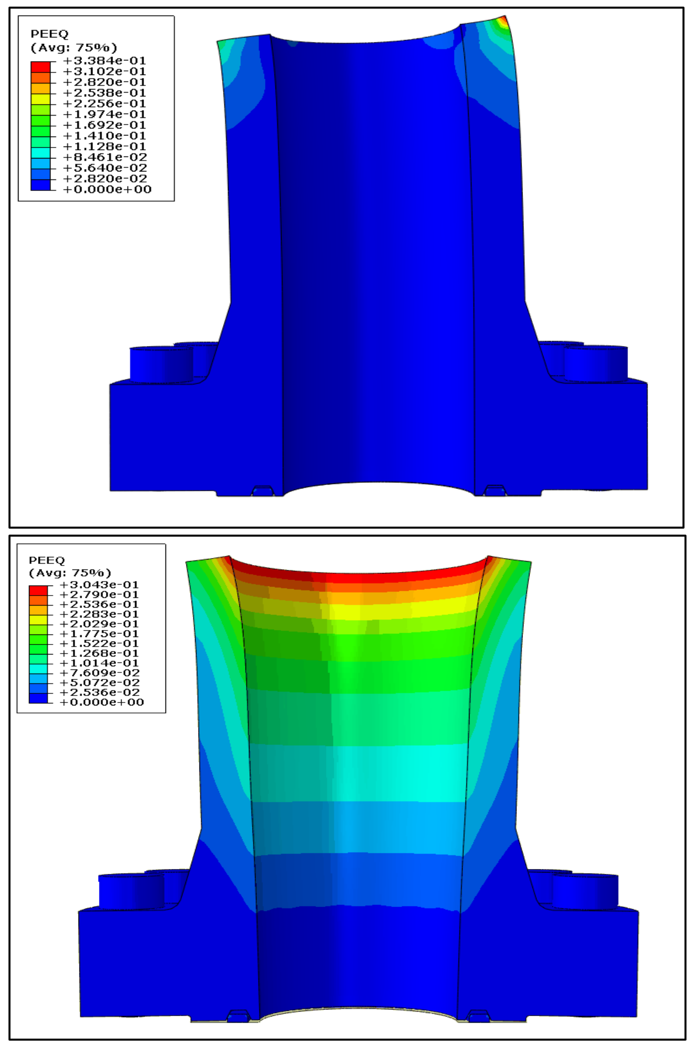

FIGURE 13: Strains plot - last converged solutions for BM (upper) and P only (lower) for 7 1/16-15K API flange

For these two API flanges, the pipe appears to be the weakest component and collapses first. This can be seen on the by the strain figures where the highest strains are building up on the pipe prolongation not in the flange body. FIGURE 13 shows strains for last converged solutions (before collapse) for BM and P only FEA for 7 1/16-15K API flange. The same behavior was observed for 3 1/16-15K as well.

In API 6AF2 [4], the leakage capacity is a critical failure criterion for API flange with net reaction force equal to zero at gasket as an assumption. A BX gasket can be shown empirically to seal in a flanged connection with minimal bolt makeup stress. Using criterion of the contact pressure between flange and gasket to be greater than 2x of inside pressure might be mis-leading. To determine suitability criteria for metal gasket seating and leak tightness should be proposed, especially for 20K working pressure. It can help to increase the capacity of the API BX flange.

The contact pressure for the BX 156 gasket in 7 1/16-15K API flange where the highest raise face separation was observed is presented on FIGURE 14 for last sub step before collapse for pure internal pressure and pure bending moment only load. On the results the uniform high contact pressure band can be seen on the all seal ring circumference. For the pressure only, the maximum value of 1157 MPa (167.9ksi) is much higher than the criterion (47.7ksi x2). Even for the case where pure bending moment was applied (no pressure energizing effect) the minimum contact pressure is about 207MPa (30ksi) at the tension side of flange.

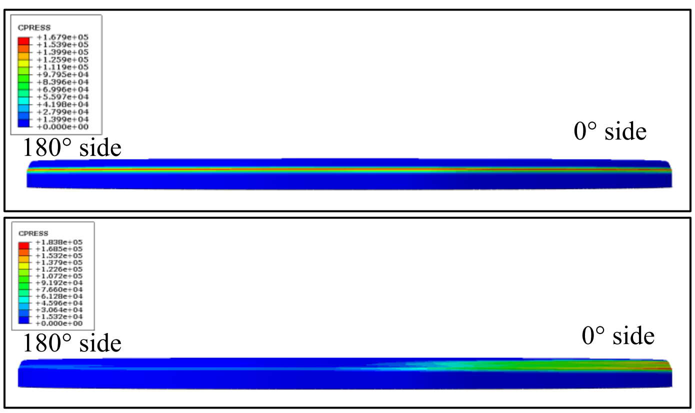

FIGURE 14: Contact pressure on the 7 1/16-15K API flange BX gasket for collapsing Pressure load (upper) and collapsing BM (lower), results in psi

SPO CF FEA

At the beginning the thermal steady state distribution is calculated for SPO CFs in the way described before (see also FIGURE 7). This will be used after for all other analysis. The thermal distribution for 4" SPO CF is shown on FIGURE 15. The 10" SPO CF results are similar.

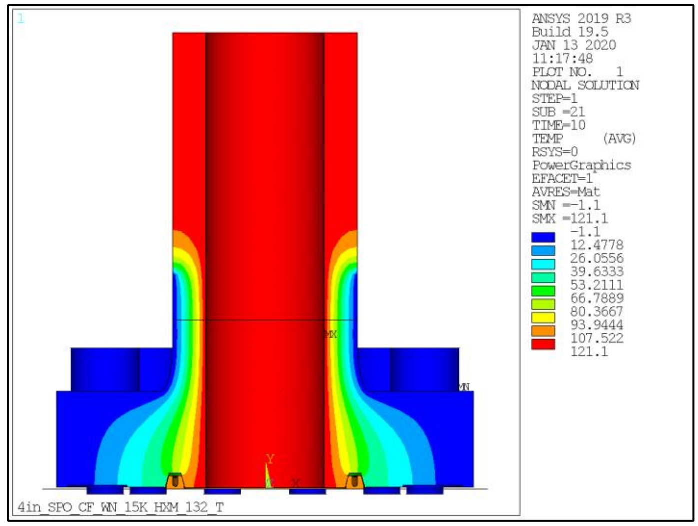

FIGURE 15: Steady state thermal distribution for 4" SPO CF 15K HXM-132, results in °C.

During the analysis 4 different loading paths with constant 0 ksi, 5ksi, 10 ksi and 15ksi pressure were used to get results which can be compared with hand calc. prediction graphs (FIGURE 10 and 11). The loading paths are described graphicly on FIGURE 16 by red dash lines. All of them were used for 3 different levels of additional external tension load (0lb, 100000 lb and 200000 lb). Over that, one additional loading case, with pressure load only was made (also see FIGURE 16, blue dash line).

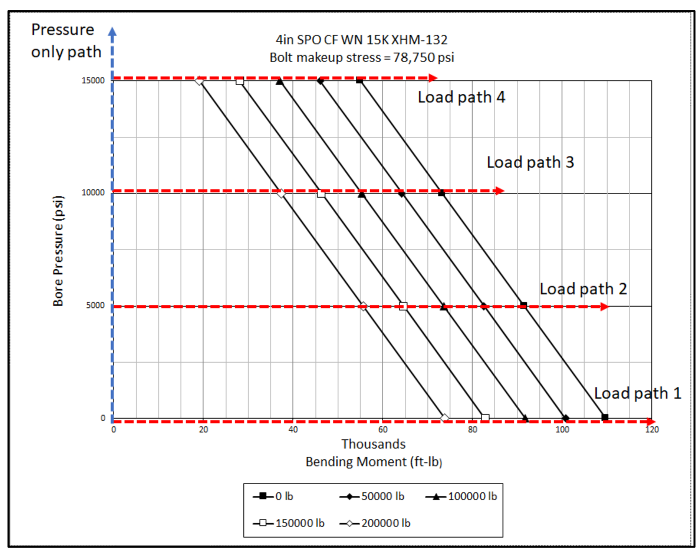

FIGURE 16: Loading paths graphical description for SPO CF FEAs based on the 4" SPO CF 15K HXM-132 example.

Two different analysis types were done for each of those loading paths for structural capacity evaluation. The difference being the material model used for the analysis. For first one, the material was based on full elastic plastic curve with strain hardening up to UTS value. The second one was based on ideal plastic material (up to SMYS value). For the capacity results to compare with hand calculation prediction loads were multiplied by a safety factor of 1.5. For the elastic plastic results strains were additionally limited to 5%. Results for fully plastic, ideal plastic (dash lines) and hand calculation estimation (solid black line) can be seen on FIGURE 17. In addition, the collapse limit is represented by a solid red line for comparison (safety factor equal to 1.0 used on the loads).

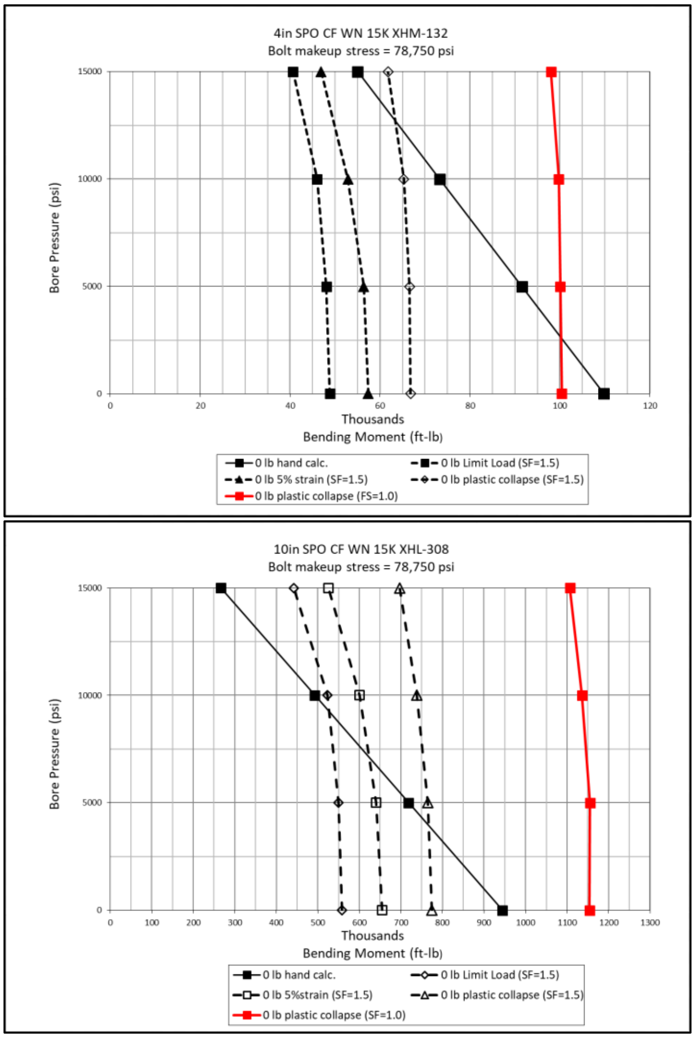

FIGURE 17: Hand calc. loads prediction to the different FEA analysis types (different safety factors used on loads is marked on the legend)

On the FIGURE 18 the ideal plastic results are drawn for all 3 different additional external tension loads. The collapsing pressure and bending moment values are also extracted from the FEA results and presented below on the TABLE 2. The collapsing loads are listed for SF=1.0 and in relation to full elastic plastic model which is close to the real material.

	Collapse BM Thousand ft-lb	Collapse P ksi
4” SPO CF 15K HXM-132	100.4	41.9
10” SPO CF 15K HXL-308	1154.4	40.5

TABLE 2: FEA results - plastic collapse BM and P for SPO CFs.

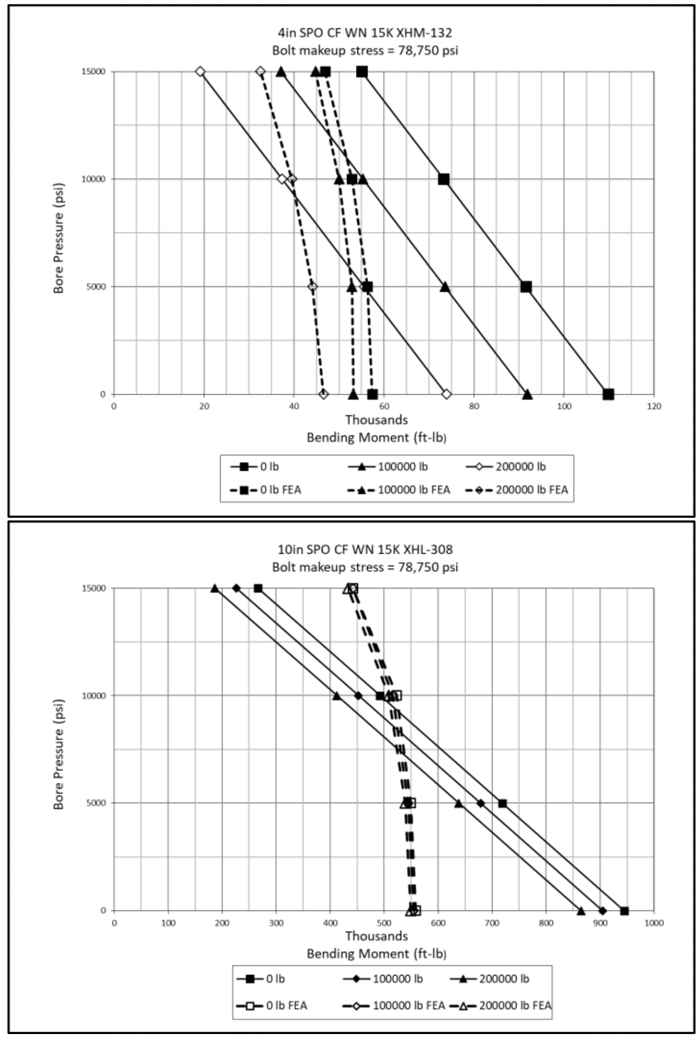

FIGURE 18: Hand calc. prediction to the Limit Load FEA results (1.5 safety factor used on loads)

For all FEA the pipe appears to be the weakest component and collapses first. This can be seen on the strain result, where the highest strains are building up on the pipe prolongation not in the flange body. FIGURE 19 shows strains for last converged solutions (before collapse) for BM and P only FEA for 10” SPO CF WN 15K HXL-308 size. The same behavior was observed for 4” size as well.

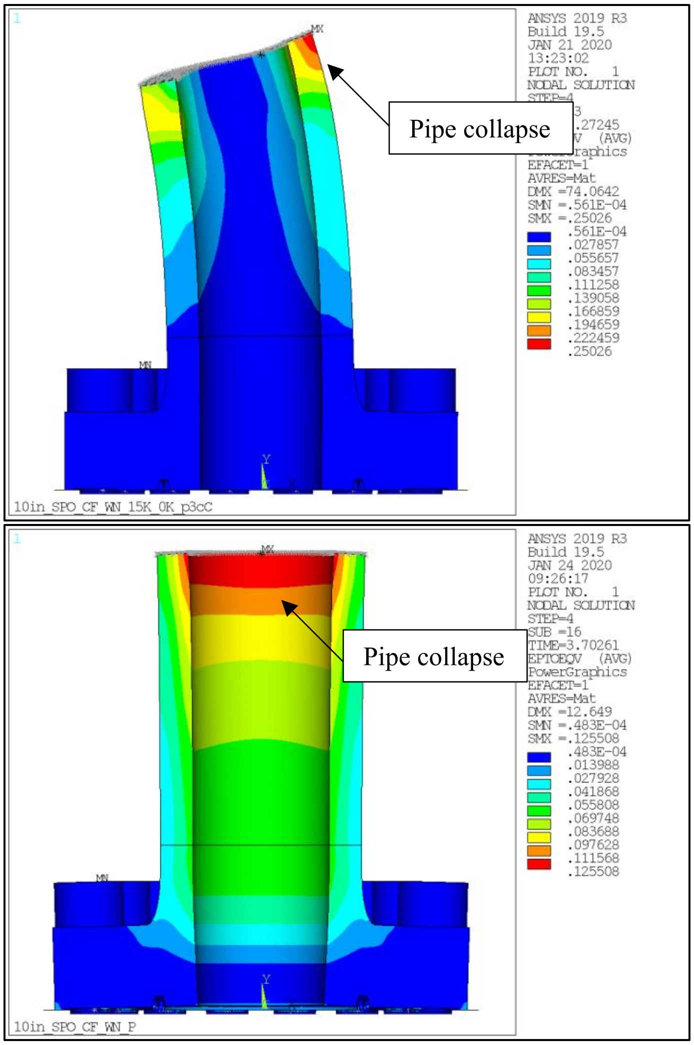

FIGURE 19: Strains plot - last converged solutions for BM only (upper) and P only (lower) for 10" SPO CF WN 15K HXL-308.

The structural capacity was checked as this is critical failure mode for SPO CF design. As the SPO CF is a static connection and the seal ring is self and pressure energized the tightness requirements are always met for maximum design loads. The fact that based on FEA the seal ring was tight for all analysis up to the structural collapse confirm it. The contact pressure for the seal ring for 10" size where the highest flange separation was observed is presented on FIGURE 20 for last sub step before collapse for P and BM only load. On the results the uniform high contact pressure band can be seen on the all seal ring circumference. For the P only results the maximum value of 1512MPa (219ksi) is much over the criterion (>40.5ksi x 2.0). Even for the case where no P load was applied (no pressure energizing effect) the contact pressure is sound and the constant bond of 341MPa (49ksi) is present.

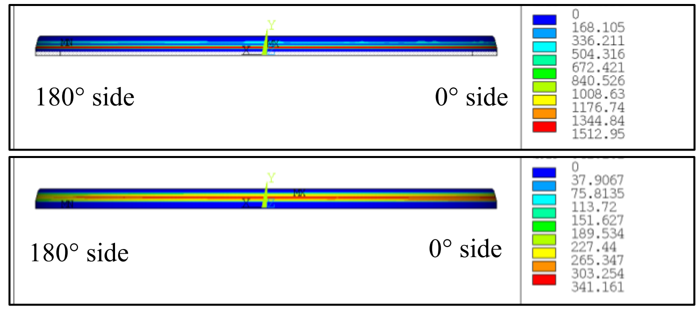

FIGURE 20: Contact pressure on the 10" SPO CF WN 15K HXL-308 seal ring for collapsing Pressure load (upper) and collapsing BM (lower), results in MPa.

The seal ring tightness is also an effect of the static behavior of the SPO CF. Even for the maximum allowable BM load application, the bolt force variation for the 0° and 180° sections is below 10% range (see FIGURE 21).

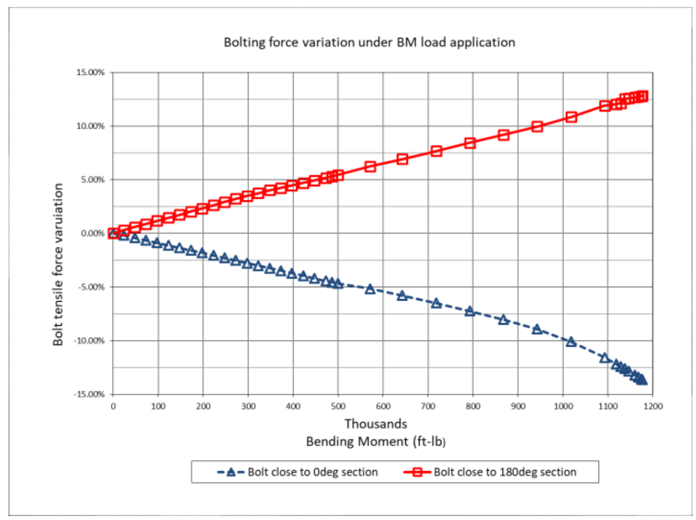

FIGURE 21: Contact Bolt force variation during BM load application for 10" SPO CF WN 15K HXL-308.

FEA RESULTS DISCUSSION

The FEA results show, that for both sizes, the pipe is the weakest part of the connection. The API conventional and SPO CF are stronger than the pipe for the sizes chosen for comparison. It can be seen on the FEA results, that strains build up in the pipe region and causing the instability for the analysis and leads to the collapse.

API charts have been already delivered as a result of FEA. Therefore, no big difference can be seen between them, even when more accurate (higher mesh density, and gasket included) model is used. The allowable loads charts presented in API 6AF and 6AF2 [4] are confirmed. As the pressure energizing effect is significant based on FEA, previous simplified approach (and leakage charts) is confirmed to be conservative for API design.

For SPO CF it should be highlighted, that the SPO CF flanges are proven to be tight up to the structural collapse of the pipe. As the flange is stronger than the pipe, pipe collapse is seen in the elastic plastic analysis before the flange capacity is reached. Therefore, it is not possible to confirm by FEA the flange capacity predicted by the hand calculations. It should be highlighted, that in ISO- 27509 [7] capacity calculations the pipe capacity is not taken for account.

For SPO CF hand calculation conservatism is seen vividly. Based on hand calculations the weakest component for 10” SPO CF WN 15K HXL-308 flange should be a flange ring. On the hand calculations conservative assumption is used and define uniform temperature across all connection. In FEA check, thermal analysis was made following API 6AF2 [4] procedure. Based on the results, only pipe is exposed to 250°F (121°C) elevated temperature, when most of the flange section and bolts has around 30°F to 104°F (-1°C to 40°C). This is the reason, why based on FEA results, flange is stronger than the pipe.

Hand calculations are using elastic ideal plastic material model. It can be seen, that even for low ductility related with F22 material grade, it is still a conservative approach. Despite the fact, that the pipe is the weakest component, allowable loads guided by the full elastic plastic analysis (FEA results) are higher than ones from ideal plastic analysis. That can be linked with UTS/SMYS F22 material ratio (95ksi/75ksi=1.3). In the result, it can be seen clearly that the Safety Factor related with elastic plastic FEA (which are closer to real material behavior) is much higher than 1.5 used in elastic ideal results (and hand calc.).

From the sealing perspective it can be seen, that the gasket design (API conventional flange) is affected by the BM load application. For the seal ring (SPO CF) the influence is minor. Looking on the FIGURE 14 (API results) the 0° side is much different in contact pressure pattern than 180° side. On the other hand, on FIGURE 20 (SPO CF results), the 0° side is not so far in value and pattern from the 180° one.

From the numerical perspective it can be seen, that ABAQUS (API results) allows to go much farther with strains than ANSYS (SPO CF results) for elastic plastic analysis. The same material formulation was used for both software’s as well as BCOs in both models. In both cases, the pipe is the same and pipe is the collapsing component, but for ABAQUS slightly higher value in stains and pressure was obtained.

CONCLUSIONS

The SPO CF design is still not so common in use as API 6A [2] conventional flanges specially on the other than European market. The following paper shows the difference in the design methodology and checks the performance by FEA in comparison to conventional API 6A [2] design. The comparisons between ASME B16.5 flanges to the SPO CF design was already discussed in the past (see [6] and [13]).

The API flange sizes, especially for weld neck type are limited in API 6A [2]. SPO CF on the contrary has a wide range for all pressure classes. In addition, for pressure classes below 15K much saving in the weight can be seen for SPO CF design (up to 70%).

The SPO CF design is based on standard pipe sizes (based on ASME B36.10) and is easy to adapt to any not standard size. The flange capacity can be conservatively calculated based on the simple equations and adopted to any piping size, flange dimensions and safety factors. In other way, the custom-made, special flange version, can be designed for any condition and configuration requested. The only attention needs to be made, that SPO CF hand calculations are not taking for account pipe capacity, however in worst case “flange is stronger than the pipe” will be as the result and this is acceptable.

The SPO CFs were designed based on weaker material and in relation to higher design temperature than API 6A [2] design. As the result, the allowable loads for SPO CF are much higher for the API 6A [2] related materials and temperature range.

The higher temperature allowed for SPO CF (350°F for 20K rating flanges) can help the pipe designers, as often together with higher pressure, the temperature follows higher values.

In case of material requirement, SPO CF allowed to have 65ksi material for 15K and 20K pressure classes and in that way avoid the welding problems (like in case of F22 75ksi API 15K flange welded to the X65 pipe).

The API 6AF2 [4] charts were confirmed to be accurate for API 6A [2] flanges and for the thermal distribution proposed (isolated pipe and flange cooled to 30°F on the outside surface). Based on the FEA results it can be seen, that the charts values can be guided by the pipe dimensions rather than flange ring capacities. For the example sizes used the flange ring (API and SPO CF) is stronger than the pipe. The other conclusion based on the FEA result is that the gasket solution is affected by the BM load and pressure energizing effect has a positive and strong influence on the tightness.

The SPO CF hand calculations according to ISO-27509 [7] are conservative based on FEA results. The strong point for SPO CFs is a sealing performance and confirmation that tightness is not influenced by the external loads (especially BM type). This is a result of the self- energize and pressure energize effects related with seal ring design. It is also related with static behavior of the connection. High tightness observed based on the FEA results is in line with functionality test results regarding fugitive emission and reliability evaluation ([16] to [18]).

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