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# COMPACT FLANGED CONNECTIONS FOR HIGH TEMPERATURE APPLICATIONS

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## ABSTRACT

The Steelproducts Offshore Compact Flange System (SPO CFS) has proven to be an exceptionally good flange design for the oil and gas industry with service temperatures normally ranging from -100°C to +250°C. High reliability, small size and low weight are properties the offshore industry has appreciated.

The design relies on a high bolt pre-tension in order to obtain the double sealing capability and the static behavior. For limited temperatures, the high pre-tension can be applied without any risk of loosing the pre-tension when the operating temperature is reached.

For high temperatures, the temperature dependent material properties in flange and bolt need to be carefully evaluated and taken into account when designing the connection.

Finite element analysis simulating all relevant phases from flange make-up to process start up and shut down have been performed in order to study flange behavior such as bolt tension, flange stresses, and seal contact.

Relatively simple analytical equations have been used in order to predict the flange behavior and hence been basis for choosing bolting material, prestress and flange face angle.

For process industry dealing with temperatures up to 720°C, it is now possible to use compact flanges. The use of compact flanged connection will first of all increase the reliability of the flanged connection, reducing the need for maintenance.

## INTRODUCTION

The SPO CFS has been on the market since 1989 and the SPO CFS standard was completed in 1996. The SPO CFS has been used in the offshore industry especially in critical applications such as risers with extreme environmental loading, high-pressure gas systems, and subsea installations.

In the on-shore based industry, the use of SPO compact flanges has been very limited and with maximum operational temperatures of around 300°C. However, during the year 2001 two different companies inquiring for compact flanges for very high temperatures (600°C and 720°C) approached SPO.

To be able to meet the requirements from the onshore industry, and specially the process industry, the flanged connections have to solve the problem that often arises in such industries. The main problem being that flange leakage occurs when the process is re-started after a shut down.

The paper describes the necessary steps that need to be taken when using the compact flange design on extreme temperature applications.

At this stage, only the SPO CFS with metallic elastic seal ring has been studied. The Finite Element Analyses and calculation examples are for a 6" CL600 flange.

This paper does not consider the problem of creep-relaxation of the bolts in detail.

## NOMENCLATURE

| | |
|---|---|
| $A_B$ | bolt area |
| $A_F$ | average stressed area of flange per number of bolts |
| $B$ | flange bore |
| $D_{W3}$ | flange outer diameter |
| $E_{B0}$ | Young's modulus of bolt at initial temperature |
| $E_{BI}$ | Young's modulus of bolt at final temperature |
| $E_{F0}$ | Young's modulus of flange at initial temperature |
| $E_{FI}$ | Young's modulus of flange at final temperature |
| $F_{B0}$ | initial bolt load |
| $F_{BI}$ | final bolt load |

$K$ bolt hole diameter
$\Delta L$ bolt elongation
$L$ bolt clamp length
$n_B$ number of bolts
$\Delta t$ temperature difference from initial to final state
$\alpha_B$ bolt material mean thermal coefficient of expansion from $t_0$ to $t_I$
$\alpha_F$ flange material mean thermal coefficient of expansion from $t_0$ to $t_I$
$\sigma$ bolt pre-stress
$\varepsilon_{BO}$ initial bolt strain
$\varepsilon_{BI}$ final bolt strain
$\varepsilon_{F0}$ initial flange strain
$\varepsilon_{FI}$ final flange strain

## THE SPO CFS DESIGN

The principle of SPO compact flanges has been described for instance in Ref. 1 and Ref. 2. The flange has a beveled face where the area between the bore and the seal groove is called the flange heel, and on the toe of the flange there is a wedge, see Figure 1.

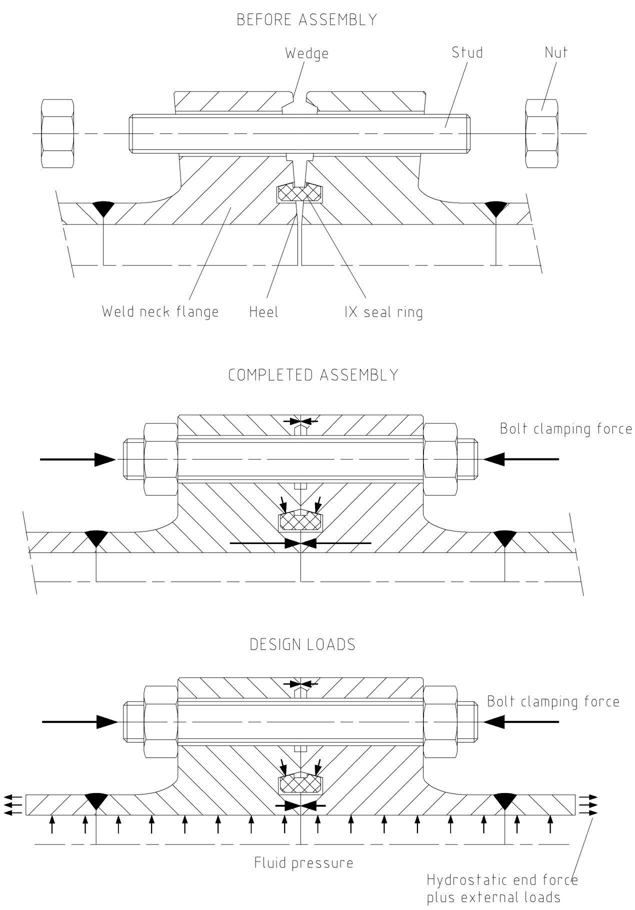


**Figure 1 How the SPO CFS works**

When the two flanges are assembled, they first make contact at the seal ring, then at the bore. As the pre-tension increases, the contact pressure at the heel rapidly increases, while the pressure at the seal ring is almost constant because of radial deformation in the ring. When correct pre-tension is achieved, the flanges will be closed at the wedge around the outer rim of the flanges. The bolts and sealing surfaces are then completely protected from environmental influence, hence avoiding any corrosion on seal faces and bolts.

In assembly condition the flanges are in face to face contact with each other, making a completely static joint with no relative movement between the flange halves. The connection is insensitive to cyclic loads in the sense that it will always be the adjoining weld that governs fatigue life of the joint. In order to achieve this static behavior the SPO CFS design relies on a relatively high pre-tension in the bolts compared to standard ASME flanges. Standard specified residual bolt pre-tension for SPO compact flanges is 70 percent of bolt yield for bolt to specifications A320 L7 and A193 B7. The two seals are independent, and they will both be effective for all load combinations below the structural capacity of the connection.

## THE CHALLENGES OF HIGH TEMPERATURE COMPACT FLANGES

At temperatures up to 720°C we meet several challenges since the material properties change drastically from ambient temperature (assembly and test conditions) to the extreme temperatures (operating conditions).

The most important and governing material properties are yield strength, Young's modulus and thermal expansion for flange material, bolt material and for seal ring material. Other physical phenomenon that need to be considered are bolt relaxation and thermal gradients.

One of the main goals of the design work, was to see if the standard SPO CFS main dimensions were suitable for the high temperature application without having to make any changes. In practice, this meant that the flange and bolt dimensions were not among the parameters that could be altered.

Given a certain flange material, which generally is governed by the selected pipe material, the parameters that are possible to vary are:

- Bolt material on basis of strength, thermal expansion, Young's modulus and relaxation
- Bolt pre-tension level
- Flange face angle
- Seal ring material on basis of thermal expansion, Young's modulus and relaxation

### Thermal expansion

An important material property for high temperature applications is the thermal expansion. It is important to select a bolt material with thermal expansion that matches the flange.

Due to the compact design of the flanges, the elongation in the bolts to achieve the correct pre-tension is small. For instance taking a 6" CL600 flange. It uses ⅝"-UNC bolts and the flange thickness is 36mm giving a conservative clamp length of 72mm if washers are not used. The bolt elongation is according to the equation below

$$\Delta L = \frac{\sigma \cdot L}{E} = \frac{0.7 \cdot 725 \cdot 72}{210000} mm = 0.17 mm \qquad (1)$$

With such a small bolt elongation to achieve the pre-load it is easy to understand that if there are great difference in expansion coefficients, the pre-stress may be lost, or on the other hand the bolt or flange may start yielding due to increased pre-load

We have the two situations that may occur:

1. Greater expansion in the bolts compared to flange
2. Greater expansion in the flange compared to the bolts

## Greater expansion in the bolts

The SPO CFS design has two independent seals, the heel seal and the seal ring. To ensure sealing at the heel, a certain contact pressure has to be maintained at the heel during operation.

While bolt pre-tension introduces heel force, internal pressure and external loads reduces it. In addition we have to study the thermal expansion in the bolts to reduce further loss in heel force. If the bolts expand a certain amount more than the flange, the bolt pre-tension may be lost at temperature.

## Greater expansion in the flange

If the thermal expansion of the flange is larger than in the bolts, another problem might arise. Increased pressure at the flange heel combined with reduced yield strength may cause permanent deformations at the heel. In extreme cases the deformations can subsequently cause the metallic IX seal ring to touch the bottom of the ring groove. The IX seal ring may then not be pressure tight any longer.

When the bolts are pre-loaded to 70 % of yield at room temperature, yielding in the bolts at higher temperatures has to be considered. A combination of thermal stresses and reduced yield limit in the bolts may lead to unwanted yielding in the bolts.

## Young's modulus

Increasing temperatures normally leads to a reduction in Young's modulus of the material. Most flanges are assembled at room temperature, and the bolts are given a specified elongation to achieve the correct pre-tension. As the temperature rises, the modulus of elasticity decreases, hence the initial pre-tension will be reduced. This may be found directly from Hooke's law, which states a linear relationship between stress and strain. The SPO CFS has relatively slender bolts compared to standard flanges and the Young's modulus dependency is less pronounced.

## Level of pre-tension

Given the bolt dimensions from SPO CFS standard, the level of pre-tension is selected such that the total pre-tension force is greater than the applied loading, typically minimum two times the end cap force from the design pressure.

The flange face angle is then calculated based on this total pre-load. The minimum specified pre-load for flange assembly will in addition take into account the expected bolt relaxation which has to be guessed based on bolt material and temperature. The specified bolt pre-load must be checked against the maximum load the flange can withstand before axial yielding occurs in the flange ring, especially important when flange material has low yield strength at the maximum temperature.

## Estimate of residual pre-tension at temperature

Bolt pre-load is important in order to maintain the static condition and hence the heel seal function. When selecting bolts and initial pre-tension it is important to have some simple equation to evaluate the choice of bolts and pre-load level for given operational conditions and the given flange and pipe material. Here an attempt has been made to predict the bolt load at temperature, it is assumed that the joint is made up at installation temperature by the initial elongation, ΔL of the bolts. Only uniform temperature in flange and bolt is included.

The initial bolt force $F_{B0}$ can be written as:

$$F_{B0} = A_B \cdot \varepsilon_{B0} \cdot E_{B0} = -A_F \cdot \varepsilon_{F0} \cdot E_{F0} \qquad (2)$$

The equation above assumes that the strain is uniform in the whole concerned stressed area. The subsequent load conditions

$$F_{BI} = A_B \cdot \varepsilon_{BI} \cdot E_{BI} = -A_F \cdot \varepsilon_{FI} \cdot E_{FI} \qquad (3)$$

The total elastic strain in the assembly condition is

$$\begin{aligned} \varepsilon_0 = \varepsilon_{B0} - \varepsilon_{F0} &= \frac{F_{B0}}{A_B \cdot E_{B0}} - \left( -\frac{F_{B0}}{A_F \cdot E_{F0}} \right) \\ &= F_{B0} \cdot \left( \frac{1}{A_B \cdot E_{B0}} + \frac{1}{A_F \cdot E_{F0}} \right) \end{aligned} \qquad (4)$$

where $A_F$ is the estimated effective clamped area for each bolt of the flange taken as:

$$A_F = \frac{\pi}{n_B \cdot 4} \cdot \left( Dw3^2 - B^2 \right) - \frac{\pi}{4} \cdot K^2 \qquad (5)$$

The total elastic strain at a subsequent condition is

$$\begin{aligned} \varepsilon_I = \varepsilon_{BI} - \varepsilon_{FI} &= \frac{F_{BI}}{A_B \cdot E_{BI}} - \left( -\frac{F_{BI}}{A_F \cdot E_{FI}} \right) \\ &= F_{BI} \cdot \left( \frac{1}{A_B \cdot E_{BI}} + \frac{1}{A_F \cdot E_{FI}} \right) \end{aligned} \qquad (6)$$

The elastic thermal strain at temperature $t_I$ is then

$$\varepsilon_I = \varepsilon_0 + \Delta t \left( \alpha_F - \alpha_B \right) \qquad (7)$$

and the bolt force at temperature $t_I$ is given as

$$F_{BI} = F_{B0} \cdot \left( \frac{\dfrac{1}{A_B \cdot E_{B0}} + \dfrac{1}{A_F \cdot E_{F0}}}{\dfrac{1}{A_B \cdot E_{BI}} + \dfrac{1}{A_F \cdot E_{FI}}} \right) + \frac{\Delta t(\alpha_F - \alpha_B)}{\dfrac{1}{A_B \cdot E_{BI}} + \dfrac{1}{A_F \cdot E_{FI}}} \quad (8)$$

The following can be observed from the expression:

1. The bolt force will normally be reduced with increased temperature with equal thermal expansion in bolt and flange due to the drop in E with increased temperature.

2. Higher thermal expansion in the bolts than in the flange will reduce the pre-load with increasing temperature.

## FINITE ELEMENT ANALYSIS - MATERIALS AND MODELLING

To study the effects of extreme temperatures in detail, several Finite Element Analyses were performed on a 6" SPO CFS CL600. The design temperature for the flange is 720°C, and the design pressures in the analyses have been taken as the listed pressure rating in ASME B16.5 (Ref. 3) for the materials used. Pipe and flange material has been A182 F347H, which is an austenitic chromium stabilized chromium nickel steel. The material is developed specially for high temperatures, the yield strength as function of temperature is plotted in Figure 2. The material is ASME B16.5 material group 2.5

The table below gives some key data for the selected flanged connection:

| | |
|---|---|
| Flange identification: | 6" SPO CFS WN CL600 sch.40. |
| Flange outer diameter: | Ø 258 mm |
| Flange thickness: | 36 mm |
| Bolt circle diameter: | 226.8 mm |
| Bolt hole diameter: | Ø 18mm |
| Bolt configuration: | 12 off ⅝"-UNC |
| Seal ring: | IX6 steel ring, here in Inconel 625 |
| End cap force for ASME rating pressure at 20°C:<br>End cap at 500°C:<br>End cap at 720°C: | $F_{end}$= 173 kN<br>$F_{end}$= 125 kN<br>$F_{end}$= 17.3 kN |

**Table 1 Flange details**

It is not easy to find bolt materials that can be used at such high temperatures. The bolt material selection was based on a maximum use temperature of at least 650°C, which is 90% of 720°C. The basis for 90% of fluid temperature is ASME B31.3 (Ref. 4) para 301.3.2. However the analyses were run with a homogeneous temperature at 720°C to simplify the analysis. Three bolt materials were analyzed; A286 (A453-660 S66286), which is an austenitic iron chromium nickel alloy, and may be used up to 700°C. The second bolt material was Nimonic 80A (UNS 07080) which, is a nickel chromium alloy and may be used up to 815°C. The third bolt material was DIN 17240 X8CrNiMoBNb1616 (Wk 1.4986) which is a hot rolled and age hardened bolt material which can be used up to 650°C, this material is commonly used in European process industry piping systems. The A286 is a bolt material, which has thermal expansion similar to austenitic stainless steels, while the 80A has considerably smaller thermal expansion. Bolts were pre-tensioned to 70% of yield strength at ambient, which is the same principle as for standard SPO flanges for moderate temperatures.

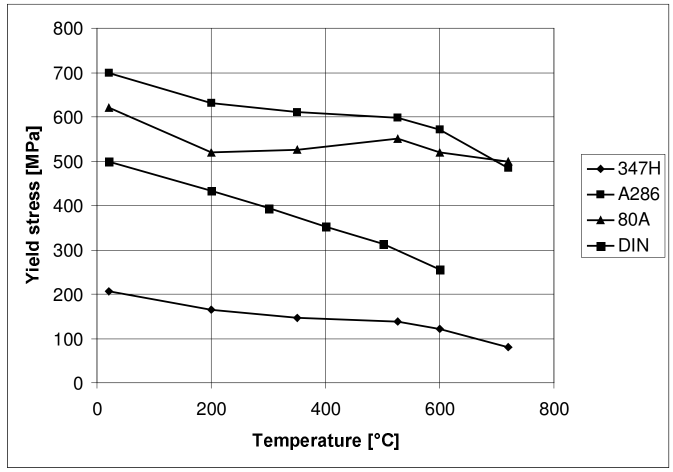


**Figure 2 Yield strength as function of temperature**

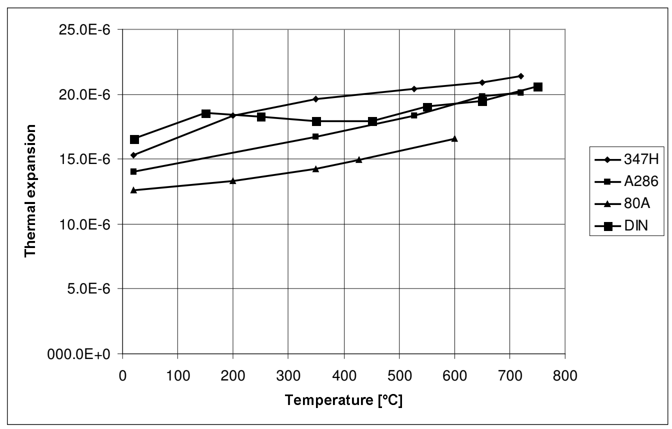


**Figure 3 Instantaneous thermal expansion coefficient $K^{-1}$ as function of temperature**

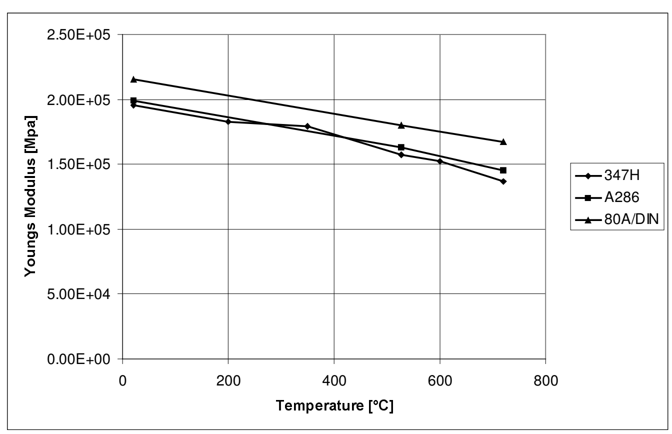


**Figure 4Young's Modulus as function of temperature**

The seal ring material has in all analysis been Inconel 625. The pressure energizing effect that will occur if the seal ring is subjected to internal pressure has not been modeled in the analysis.

| Analysis | Material | Face angle | Bolt material | Analysis temp. and pressures |
|---|---|---|---|---|
| A1 | | | | amb. + 99.3 bar |
| A2 | | | A286 | 500°C + 66.9bar |
| A3 | | | | 720°C + 9.3bar |
| A4 | | | Nimonic 80A | amb. + 99.3 bar |
| A5 | A182 F347H | 0.46° | | 500°C + 66.9bar |
| A6 | | | | 720°C + 9.3bar |
| A7 | | | X8 CrNiMoBNb 16 16 | amb. + 99.3 bar |
| A8 | | | | 500°C + 66.9bar |
| A9 | | | | 720°C + 9.3bar |
| B1 | | | | amb. + 99.3 bar |
| B2 | | | A286 | 500°C + 66.9bar |
| B3 | | | | 720°C + 9.3bar |
| B4 | | | Nimonic 80A | amb. + 99.3 bar |
| B5 | A182 F347H | 0.32° | | 500°C + 66.9bar |
| B6 | | | | 720°C + 9.3bar |
| B7 | | | X8 CrNiMoBNb 16 16 | amb. + 99.3 bar |
| B8 | | | | 500°C + 66.9bar |
| B9 | | | | 720°C + 9.3bar |

**Table 2 FE analysis overview.**

Table 2 gives an overview of all analyses that were run. In addition to the material combinations, a variable describing the flange face angle were studied in order to see if standard SPO CFS face angles could be used, or if the angles should be calculated based on a lower bolt pre-load.

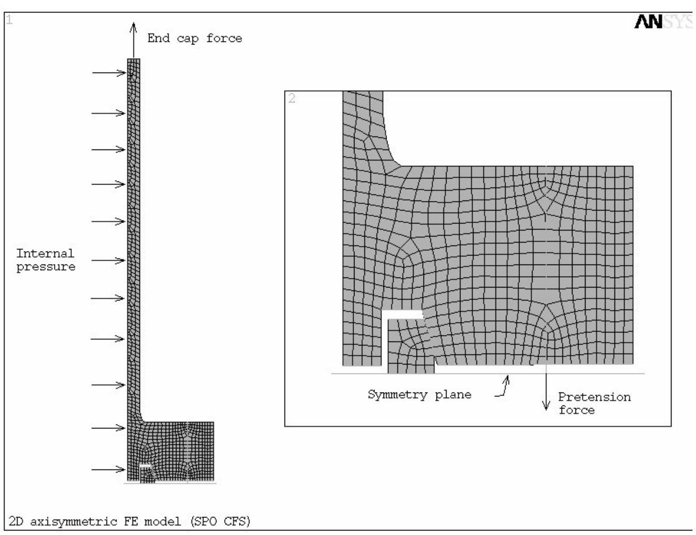


**Figure 5 FEA model showing boundary conditions**

The finite element analyses were 2D axi-symmetric analyses in which the material in the area where the bolts are located has reduced stiffness in order to account for the bolt holes. The FEA software ANSYS was used taking into account non-linearities such as gap/contact and non-linear material properties. Where available, the material properties were taken from ASME B31.3 and ASME II, but some data were taken from the material producer's data sheet.

The analyses were run with the pseudo time steps as explained in Table 3. Within each time step the load is incrementally applied. The temperature was assumed to be uniform for the whole flange and bolt.

| Time | Load application |
|---|---|
| 0-1 | Pre-loading of bolts |
| 1-2 | Internal pressure radially |
| 2-3 | End cap force |
| 3-4 | Increase temperature to max |
| 4-5 | Decrease temp to ambient |
| 5-6 | Remove end cap |
| 6-7 | Remove internal pressure |
| 7-8 | Internal pressure radially |
| 8-9 | End cap force |
| 9-10 | Increase temperature to max |
| 10-11 | Decrease temp to ambient |
| 11-12 | Remove end cap |
| 12-13 | Remove internal pressure |

**Table 3 Analysis load steps.**

## FINITE ELEMENT ANALYSIS - RESULTS

The results from the analyses have been plotted showing the results explained in Table 4 as function of analysis time. The important times being time 1 representing pre-load only, time 4 which is first occurrence of maximum pressure and temperature, time 7 which is unloaded (pre-load only), time 10 which is second occurrence of maximum pressure and temperature and at last time 13 which again is the unloaded case.

| Label | Explanation |
|---|---|
| Seal | Total contact force in Newtons between seal ring and groove. |
| wedge | Total contact force in Newtons between flanges at the outer wedge. |
| mid | Total contact force in Newtons between flanges in area between ring groove and bolt holes. |
| heel | Total contact force in Newtons between flanges at the heel. |
| bolt | Total force in Newtons in bolts. |

**Table 4 Description of plotted results.**

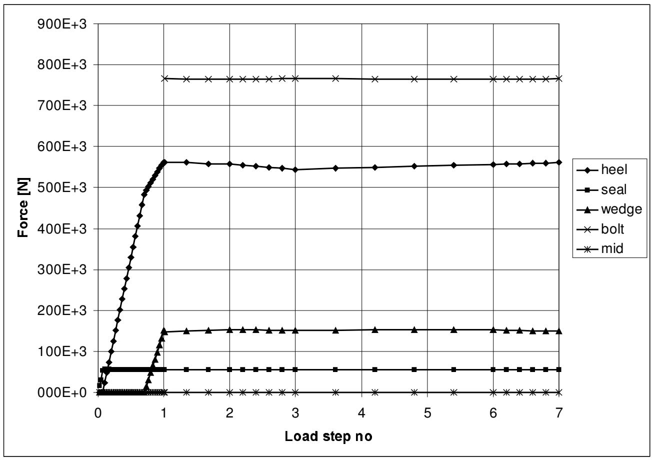


**Figure 6** ANALYSIS A1 - 20°C - A286

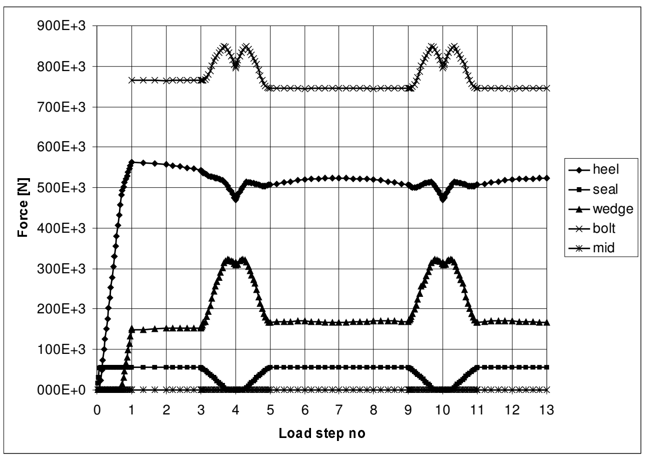


**Figure 7** ANALYSIS A2 - 500°C - A286

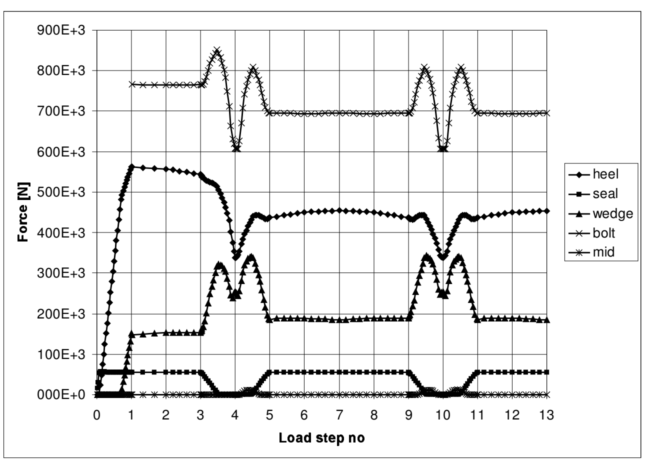


**Figure 8** ANALYSIS A3 - 720°C - A286

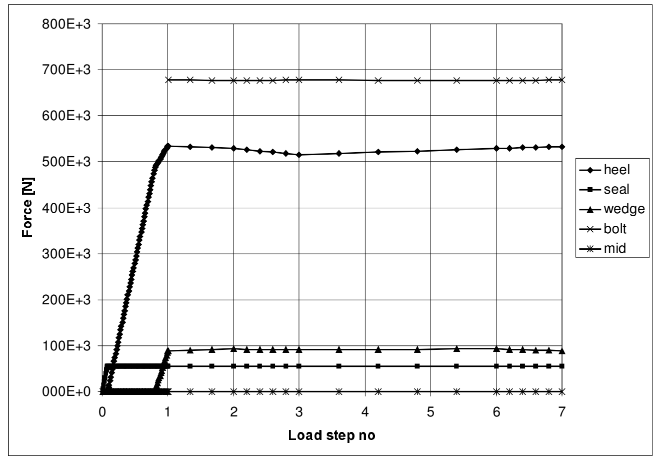


**Figure 9** ANALYSIS A4 - 20°C - 80A

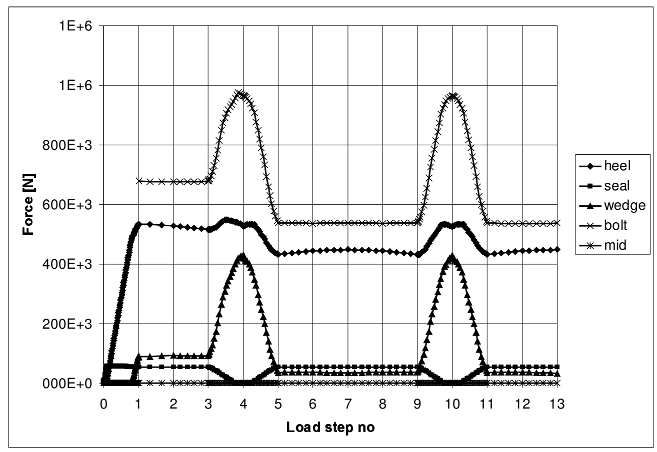


**Figure 10** ANALYSIS A5 - 500°C - 80A

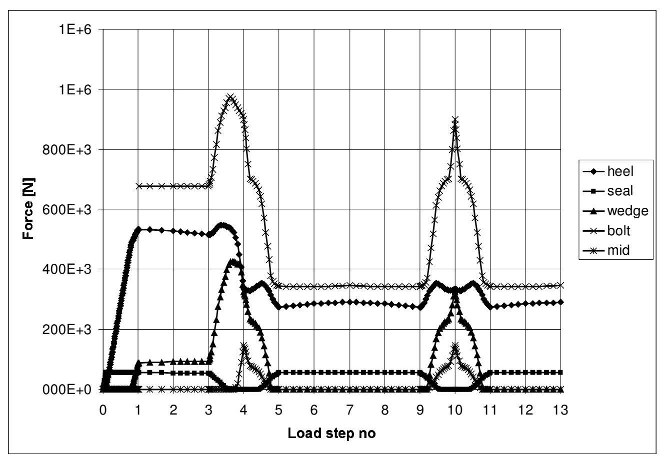


**Figure 11** ANALYSIS A6 - 720°C - 80A

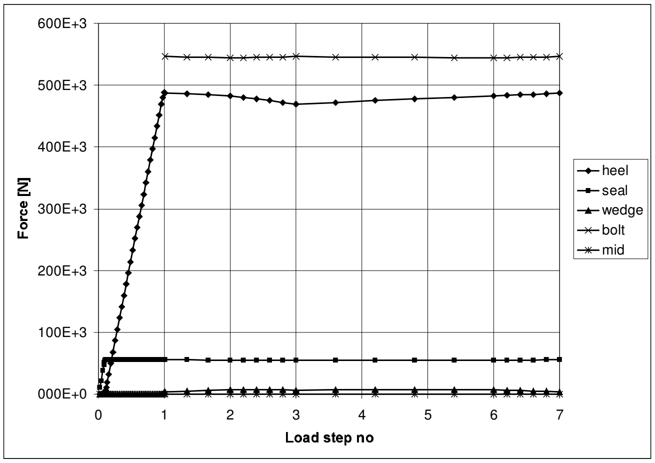


**Figure 12 ANALYSIS A7 - 20°C - DIN**

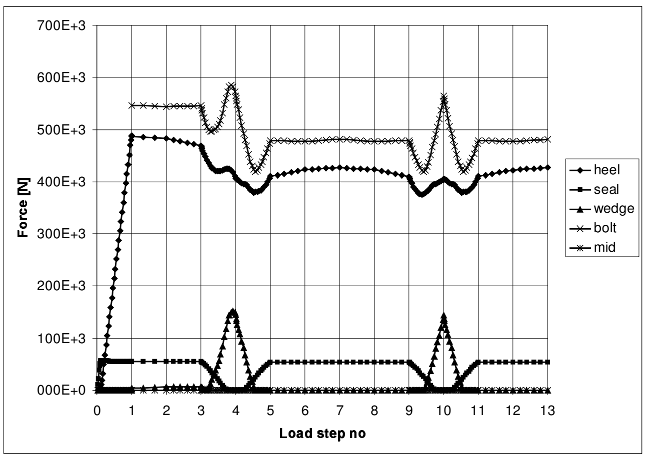


**Figure 13 ANALYSIS A8 - 500°C - DIN**

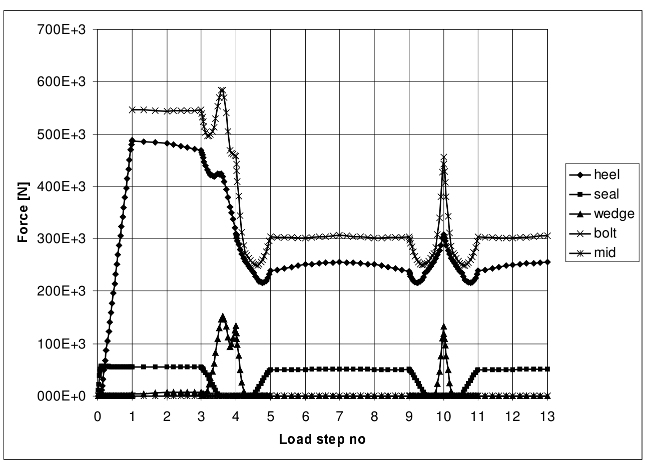


**Figure 14 ANALYSIS A9 - 720°C - DIN**

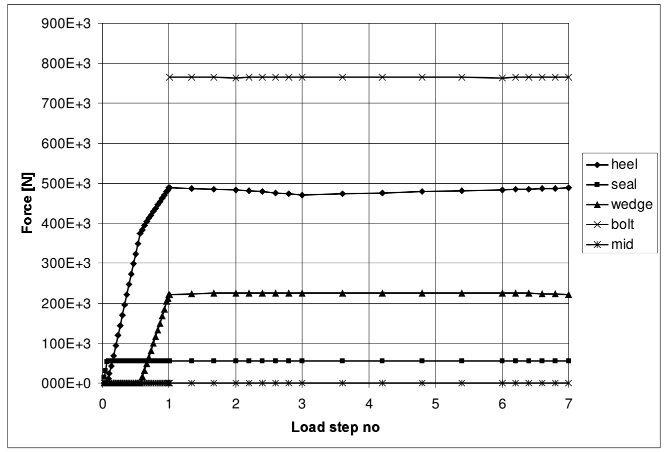


**Figure 15 ANALYSIS B1 - 20°C - A286**

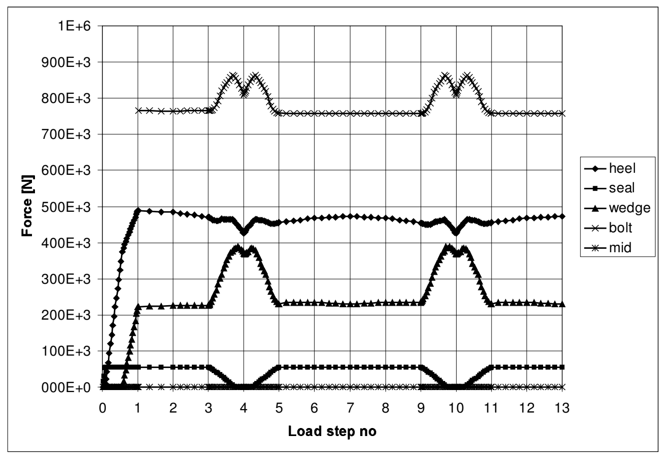


**Figure 16 ANALYSIS B2 - 500°C - A286**

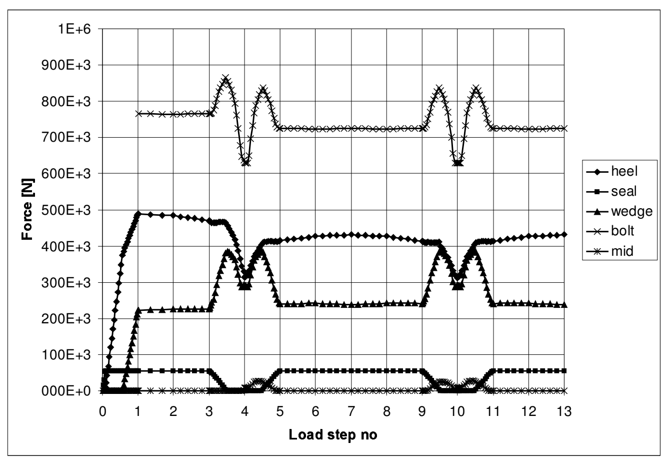


**Figure 17 ANALYSIS B3 - 720°C - A286**

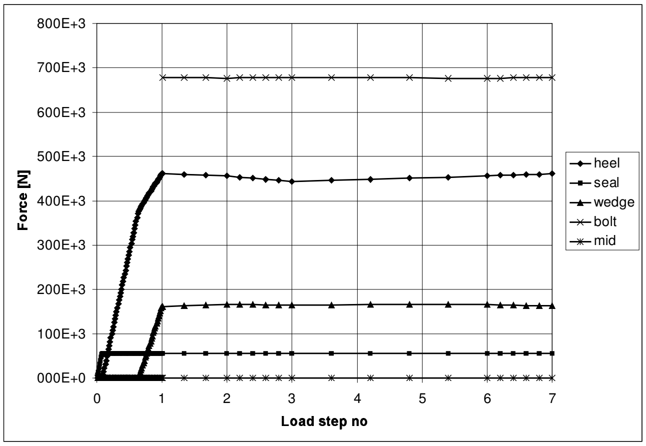


**Figure 18 ANALYSIS B4 - 20°C - 80A**

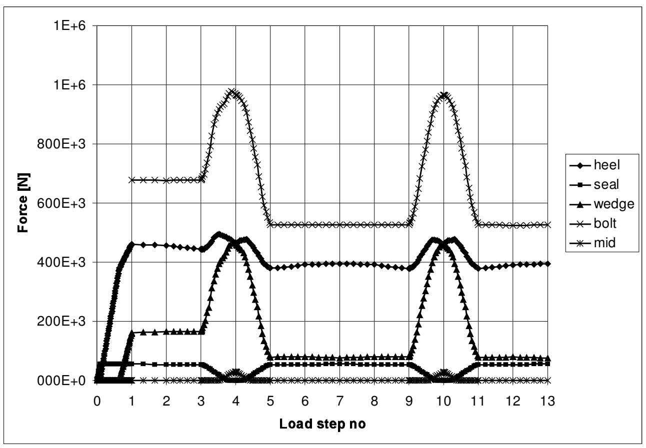


**Figure 19 ANALYSIS B5 - 500°C - 80A**

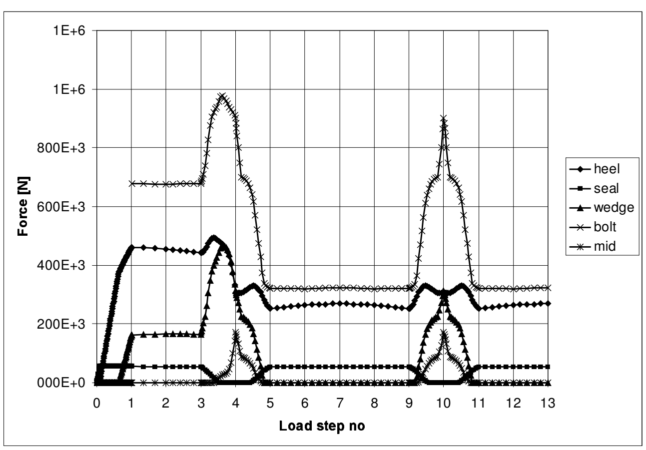


**Figure 20 ANALYSIS B6 - 720°C - 80A**

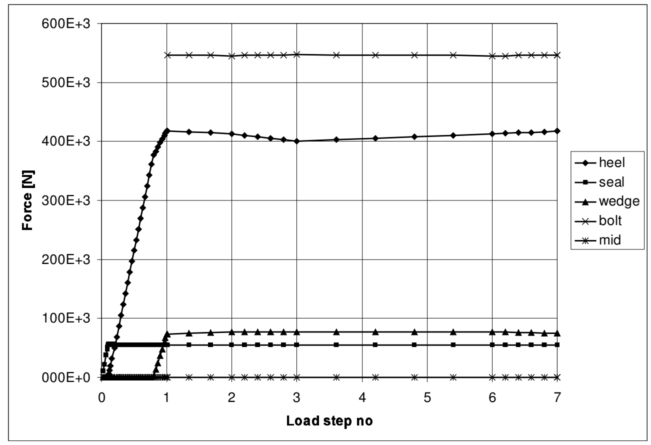


**Figure 21 ANALYSIS B7 - 20°C - DIN**

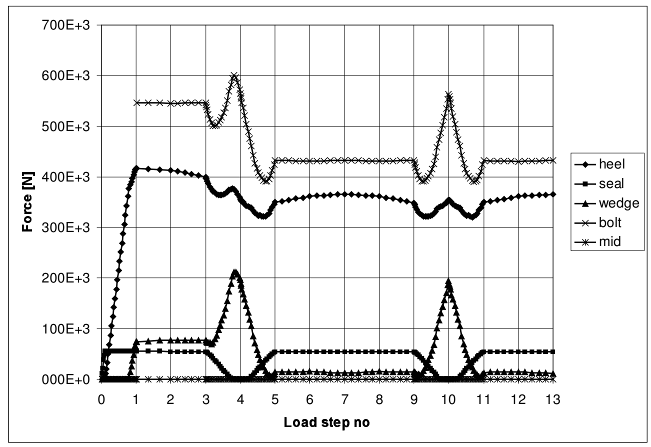


**Figure 22 ANALYSIS B8- 500°C - DIN**

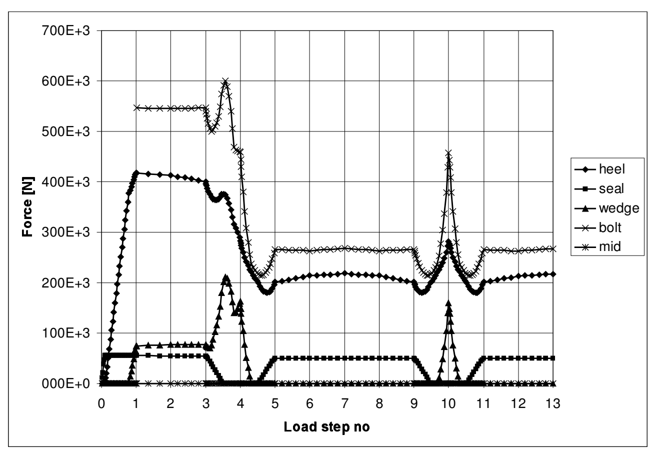


**Figure 23 ANALYSIS B9 - 720°C - DIN**

## Choice of bolt material

From the bolt forces in analysis A4, A5 and A6, one can see that the results of using Nimonic 80A bolts for flanges in austenitic materials, is that the smaller thermal expansion of the bolts give drastic increase in bolt stress. Looking at analysis A6, significant yielding of the bolts is observed. It is also seen that the bolt yielding has led to loss of pre-tension in the bolts when the temperature is reduced to ambient. The residual bolt pre-load is about 50% of the initial value and it is not sufficient to keep the flange closed at the outer rim (wedge contact force is zero). It is evident that the 80A bolts do not match very well with austenitic flanges, and cannot be used. The DIN bolts are not strong enough to close the flange with standard face angle. They work a lot better with the reduced face angle. However the low yield strength at 720°C give significant yielding of the bolts (analysis B9). At 500°C it looks a lot better, but still the contact on the outer rim during a process stop is marginal and there is not much bolt relaxation that can take place before the contact is lost. The A286 bolts show the best performance. The design with the standard face angle will work, but the reduced face angle show an even better performance. The residual bolt tension after first temperature cycle is higher and the bolt stress range for a temperature cycle is smaller than for the two other bolt materials. The flange heel contact and outer rim contact are maintained throughout the whole load cycle.

## Face angles

The SPO standard face angles are used in analyses A while reduced face angles are represented by analyses B. It is seen that in analyses A1-A3 which represent a connection with bolts that suits well with the flange material, and the bolts are relatively strong compared to the flange, then it is possible to use a flange with a standard face angle. Looking at the DIN bolts in analyses A7 to A9, one can see that the bolts have not sufficient pre-load capacity to close the flange properly, however in analysis B7 to B9, the flange is closed after pre-load due to the smaller face angle.

From analyses B2 and B3 it is seen that the total pre-load level at around 600 - 800 kN is large compared to the end cap forces of 17kN at 720°C and 125kN at 500°C. The static condition will therefore be maintained even if the bolt pre-load relax to 50% of the initial value and with total equivalent load in the order of 2 times the end cap force.

## Seal ring material

It is seen from all analyses performed that the Inconel seal ring will loose contact on the seal ring when the temperature reaches about 300°C due to the smaller thermal expansion in the Inconel 625 compared with the 347H austenitic flange material. It should be noted that the pressure energizing effect on the seal ring is not included in the analyses. However, the seal ring material in a more suitable material should be selected in order to ensure contact in the ring, e.g. grade 660 material similar to the A286 bolts should be suitable.

## SIMPLIFIED CALCULATION OF BOLT PRE-LOAD.

Using the equations outlined, the total bolt load at 500°C and 720°C was calculated for the three bolt materials. The results are tabulated and compared with the actual values from the FEA.

| Bolt material and temp. | Total bolt pre-load from equation (8). | Bolt load from FEA analyses B load step 4. | Total bolt plastic capacity |
|---|---|---|---|
| A286 at 500°C | 749 kN | 810 kN | 920 kN |
| A286 at 720°C | 800 kN | 630 kN | 760 kN |
| 80A at 500°C | 1200 kN | 966 kN | 860 kN |
| 80A at 720°C | 1300 kN | 900 kN | 780 kN |
| DIN at 500°C | 540 kN | 564 kN | 490 kN |
| DIN at 720°C | 580 kN | 458 kN | NA |

**Table 5 Total bolt pre-load at temperature**

The analytical equations predict correctly that the 80A bolts do not match with the flange material 347H showing bolt pre-load values far in excess of the ideal plastic bolt capacity. For the other bolt materials the accuracy is fairly well. The uncertainty in the actual values and the correlation between the instantaneous thermal coefficient and the mean coefficients may be some of the reason for the discrepancy in the calculated bolt pre-load at temperature.

## CONCLUSIONS

Using materials in the bolt and flange with similar thermal expansion coefficients reduces the danger of introducing thermal stresses due to dissimilar expansion between the bolt and flange. It is possible to access this problem using the relatively simple equations presented. At least the equations will give an indication whether the bolt material matches the flange material. However, the results are sensitive to material properties, which are not always easily available, especially at extreme temperatures. To find a more accurate behaviour, FEA has to be performed modelling all temperature dependent material properties.

The flange face angle should be calculated based on minimum bolt pre-load necessary to maintain the static behaviour for all load conditions. However, a greater bolt pre-load should be specified in order to account for relaxation in the bolts. When bolts are properly selected, the high specified pre-load in the bolt will not lead to stress levels that are damaging to the flange and hence its sealing capability.

To minimise the influence from reduced Young's modulus and from thermal expansion, one should try to make the bolts as slender as possible. This makes the elongation of the bolt due to thermal effects small compared to initial elongation from pre-loading and minimises loss in heel force due to the geometric design of the flange.

The seal ring material should be selected as a material with similar thermal expansion as the flange material. It is also important that the seal ring material has sufficient strength and relaxation resistance at maximum temperature.

## FIELD EXPERIENCE.

The field experience so far is very good. There is no reported leakage or any other problem during operation of the plants. On one of the process plants (cracking of ethylendicloride) the process was suddenly stopped due to power failure. When restarting the process a standard gasketed flange in the same system was leaking while all the compact flanges were pressure tight.

## REFERENCES

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