Behaviour of flange joints in Steam Generator under thermal loads

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Abstract. Pressure vessels such as steam generators are subjected to high temperature, in addition to high pressure during the operating condition. Flanges and bolts are made up of different materials whose coefficient of thermal expansion varies. Usually, thermal expansion in bolts is greater than that of flanges. At elevated temperatures bolts expand more than that of flanges, resulting in decrease of compression in connected members achieved during assembly stage, which in turn decreases the contact stress in gasket. This can lead to leakage of internal fluid. The loss in gasket contact stress due to differential thermal expansion can be nullified by using sleeves of higher thermal expansion between the flange-nut and flange-bolt head interfaces. At higher temperatures sleeves expand more than bolts and flanges, pushing the flanges closer towards each other, thus decreasing gap created due to differential thermal expansion. The behaviour of gasketed blind flange joint with and without sleeves is analysed and the performances are compared under thermal loads. The non-linear behaviour of gaskets is included by specifying the loading and unloading characteristics with hysteresis.

1 Introduction

Flanged joints with gasket play a vital role in sealing the leakage phenomenon, in pressure vessels. Sealing of these joints at high temperatures is very important in aerospace, chemical and nuclear reactor applications. The flange joints are generally designed using analytical formulae and calculations based on a number of assumptions. ASME codes are based on many assumptions and empirical relations. Thus, these codes are not completely successful in designing a leak proof joint. Suyuan et al. [1] carried out experiments on sealing performance of high temperature gas-cooled reactor (HTR-10) using metallic O-ring and welded Ω-ring for sealing and compared their performance. Flange joints with gasket can be considered as springs connected in series and the overall spring constant of the joint is given by Eq. (1).

\[ \frac{1}{K_j} = \frac{1}{K_g} + \frac{1}{K_n} + \frac{1}{K_h} \]  

(1)

In many applications gaskets materials are soft and its stiffness is low compared to that of flange material. Hence, the gasket material behavior will dominate the elastic behavior of other members in the joint. Minor changes in the flange stiffness will not have any effect on the overall joint behavior. Because of lower stiffness of gasket material, external load can cause a large change in deformation of gasket than in flanges and bolts. Bouzid et al. [2] presented an analytical model based on evaluation of radial and axial movement of flange and its effect on gasket stress at elevated temperatures. Abid et al. [3] preformed finite element analysis on bolted joints considering the effect of temperature of internal fluid by considering the gasket as linearly elastic material.

The influence of including the non-linear behaviour of gasket material under loading and unloading with hysteresis was highlighted by Murali Krishna et al. [4]. The significance of gasket material behavior in the sealing performance of the joint is emphasized without including the internal fluid temperature. Henson [5] proposed analytical approach for evaluating the bolt forces in flange joint design. Nomesh et al. [6] carried out a detailed finite element study on the variation of bolt stresses in the flange joints used in pressure vessels. The analysis is performed only with assembly and pressure loads, excluding the thermal loads. Vinod et al. [7] studied the effect of differential expansion on gap between the flanges in a steam generator by considering the gasket as linear metallic material.

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In the present work, the blind flanged joint with gasket is analysed by including the internal fluid temperature. The effect of thermal loads on flange joint is analysed by considering the nonlinear behaviour of gasket material under loading and unloading condition. The variation in gasket contact stress under different operating condition are determined. In order to overcome the reduction in gasket contact stress under thermal load, compensating sleeves of different material having higher thermal expansion coefficient is provided between the flange-bolt head and flange-nut interfaces. The behaviour of blind flange joint with compensating sleeves under different operating condition are examined and compared with the earlier case.

Fig. 1. Schematic of the flange joint (Dimension in mm)

2 Joint Configuration

Geometry

The blind flange joint with raised faces and compensating sleeves considered for the present study is shown in Fig.1. It consists of a weld-neck flange, a blind flange, compensating sleeves between flange and nut, spiral wound gasket and steel bolts with nuts. The joint is exposed to 500°C of steam flow inside and ambient temperature outside. Spiral wound gasket made of PTFE filler with stainless steel spiral ring is used. The inner and outer diameters of the gasket considered are 266 mm and 312 mm respectively and thickness is 5 mm.

2.2 Material properties

Flange and bolt materials are considered as ASTM A-105 and ASTM A-193 B7 respectively. The materials’ isotropic behavior of flange and bolts are characterized using Young’s modulus of elasticity and Poisson’s ratio. The thermal behavior of the materials is modeled by providing the thermal conductivity of respective materials. Spiral wound gasket consists of alternate layers of metal and filler material in the form of spiral ring. Metallic layer provides structural support; whereas filler material acts as sealing agent. This gasket material possesses non-linear response with hysteresis as shown in Fig. 2. It has different responses under loading and unloading conditions. The gasket material is characterized using one loading curve and two unloading curves. A special material model available in the commercial finite element software ANSYS is used, for characterizing the interface material behavior. This model interpolates the required properties of the gasket at the intermediate loading and unloading points. The thermal expansion coefficient of gasket material is taken 11.7x10^-6 m/m°C [8].

Fig. 2. Load response of gasket material [4]

2.3 Sleeves for compensating the expansion

Sleeves made of material with higher coefficient of thermal expansion are used between the nut-flange and bolt head-flange interface to compensate for the differential thermal expansion between bolts and flanges. The thermal expansion of sleeve material should be greater than that of bolt material. Based on the comparison with bolt material, stainless steel grade 316 is chosen for sleeve. The material behavior of sleeve used in analysis is provided in Table 1. Due to the temperature rise, these sleeves also expand along with flanges and bolts. Since thermal expansion of sleeves is more than that of bolts, expanded sleeves push flanges closer, compensating for extra thermal expansion of the bolts. Sleeves of 130mm and 85mm are used at weld flange and cover plate respectively. Finite element analysis has been carried on flange joint with the sleeves considering the non-linear behaviour of gasket to evaluate sealing performance using the loading properties shown in Fig.2.

3 Finite Element Modelling

A gasketed flange joint with eight bolts shown in Fig.1 is considered for the analysis. The 3D geometry of gasketed flange joint with 1/16th (22.5°) angular portion is considered for the analysis. This simplification is done making use of the symmetry in geometry and loading conditions. Finite element analysis is done for both blind flange joint with and without sleeves, to study the gasket contact stresses.


3.1 Meshing

3.1.1 Structural and thermal elements

As the study includes thermal and structural analysis, compatible eight noded thermal (SOLID 70) and structural (SOLID 185) elements are used [10]. The developed finite element model after convergence study shown in Fig. 3 is used for analysis.

Table 1. Material properties of blind flange joint members.

| Parameter                        | Flange (ASTM A-105) | Bolt and Nut (ASTM A193-B7) | Sleeves (SS 316) |
|----------------------------------|----------------------|-----------------------------|------------------|
| Young’s Modulus (GPa)            | 190                  | 175                         | 193              |
| Poisson’s Ratio                  | 0.3                  | 0.3                         | 0.3              |
| Thermal Conductivity (W/mK)      | 47                   | 39                          | 1                |
| Coefficient of thermal expansion (x10^-6 m/m°C) | 12.57               | 14.17                       | 18.63            |

3.1.2 Interface elements

INTER 195 is used for modeling gasket entity for both structural and thermal analysis. The interfacing elements are used between two entities with one element along the thickness direction. These elements require closure value for evaluating gasket stress and are calculated using the relative displacement between the top and bottom surfaces of the element. A mid-plane is created from the nodes of top and bottom flanges. The stress normal to this plane gives the gasket contact stress and this value is constant throughout the thickness.

3.1.3 Contact elements

Contact elements are used for modeling the contact between bolt and flange entities. Bolt and flange are made of different materials. To find the difference in deformations and thermal expansion of these parts, they have to be modeled as separate entities. These contact elements are used to transfer the loads from one entity to other. TARGE 170 and CONTA174 elements are used in conjugation with each other for simulating the contact between surfaces. For without sleeve condition, the flange surface is taken as target surface and bolt/nut surface is taken as contact surface. When sleeves are used, contact is also established between sleeve-nut and between sleeve-flange interfaces. Here, the sleeve surface is chosen as target surface and bolt head bottom surface is chosen as contact surface to establish bolt-sleeve contact. Similarly, the flange surface is chosen as target surface and sleeve bottom surface is chosen as contact surface for sleeve-flange contact.

3.1.4 Pretension elements

Pretension in bolts is simulated using PRETS 179 elements. This divides the elements in bolt using a specified plane into two unconnected group of elements. These two groups of elements are connected using pretension node, on which the calculated value of bolt pretension as per ASME code is applied. This node is connected to all the elements on the specified plane and the applied pretension is distributed to all the nodes on that plane. The pretension node pulls these groups together depending on the type of load given i.e., force or displacement.

3.2 Boundary Conditions

3.2.1 Thermal

The internal fluid temperature is applied along the inner walls of the flange joint assuming a steady state thermal condition. The temperature of the internal fluid is taken as 500 °C and that of ambient condition is taken as 25°C. The outer walls of the flange joint are subjected to heat transfer due to convection with convection coefficient of 11 W/m²°C. Thermal contact resistances near contact between flange and bolt are neglected and analysis is performed assuming they were connected.

3.2.2 Structure

The blind flange joint considered is fastened using eight bolts, based on which 22.5° segment of flange joint is considered for analysis. This segment of the flange joint is composed of one-half portion of the bolt and half of web region between two bolts as shown in Fig. 4. Symmetric displacement boundary condition is applied to both 0° and 22.5° surfaces in flanges, gasket, bolt-nut and sleeves. The flange joint is constrained along the longitudinal direction by fixing the bottom surface of the flange. The initial bolt pretension required to be applied to bolts during assembly condition is calculated based on Eq. (2) from ASME BPVC code. The bolt pretension for the gasket to be sealed is determined as 110 kN and this pretension is applied in terms of force. During
pressurizing condition, an internal pressure of 15 MPa is applied on the inner walls of the flanges.

\[ F_b = 0.785 G \pi^2 P + 2 \pi \beta m GP \]  

(2)

![Fig. 4. Angular segment of blind flange joint](image)

### 3.3 Analysis Methodology

The sequence of analysis performed is given below,

i. Thermal analysis is performed to get the temperature distribution in the flanged joint.

ii. Structural Analysis is performed with loading in three steps
   a. Pretension of 110kN is applied to achieve seating stress in the gasket
   b. Internal pressure of 15MPa is applied.
   c. Temperature distribution obtained from the thermal analysis (i) is applied as load at each node in order to obtain the deformation due to temperature distribution.

### 4 Results and Discussion

#### 4.1 Temperature distribution

The blind flange joint is subjected to internal fluid temperature of 500°C and the temperature distribution is obtained. Figures 5 and 6 shows the temperature distribution in the flanged joint for bolt without and with sleeves respectively.

![Fig. 5. Temperature distribution in Blind Flange joint](image)

Here, the temperature distribution varies from 500°C on the inner walls to 330°C (approx.) near the nuts on upper surface for without sleeve condition. For with sleeve, the temperature distribution varies from 500°C to 300°C (approx.). The drop in temperature near the nuts in blind flange side is more compared to that of flanges, because nuts have large surface areas exposed to convection at ambient condition. The nuts near the cover plate are farther from the hot fluid compared to that of nuts near lower flange. Hence the nuts near lower flange are at a higher temperature than nuts on the cover plate.

#### 4.2 Gasket Stress

The analysis is performed simulating the assembly and pressurization condition. During assembly condition, bolt preload alone is applied; whereas under pressurization condition, fluid loads both internal pressure and temperature distribution are applied. The gasket stresses obtained under different loading conditions are provided in Table 2. Bolt pretension tends to move the flanges closer, resulting in compressing the connected members in the joint like, flange and gasket. Since, the gasket material is least stiff, higher gasket compression is observed, leading to higher gasket stress [11]. Along the width of the gasket, the stress varies from outer to inner circumference, based on the proximity towards the bolt. The gasket contact stress varies from 89.1 MPa along the inner edge to 91.8 MPa along the outer edge. The variation in gasket stress is due to the variation in distribution of bolt pretension.

![Fig. 6. Temperature distribution in Blind Flange joint with sleeves](image)

Due to internal fluid pressure, the gasket stress in the inner side is reduced from 89.7 MPa to 53.5 MPa. Here, the variation in gasket stress is higher due to the effect of pressure loads. The pressure load tends to move the flanges apart from each other, causing the gasket to relax more towards the inner edge. Upon considering the temperature distribution due to internal fluid from thermal analysis, the gasket stress at the inner edge further reduced to 48 MPa. There is a considerable loss in gasket contact stress at outer diameter which is reduced to 28.3 MPa. This reduction is due to differential thermal expansion of flanges and bolts.

With the use of compensating sleeves between flange-bolt head and flange-nut interface, the sleeves
expand more due to thermal deformation. As discussed earlier, the expansion in sleeves is more compared to the expansion caused in bolts. This acts as additional external clamping force to keep the flange joint intact and increases the contact stress in gasket. Upon considering the thermal loads due to internal fluid in addition to assembly and pressure loads, the gasket stress along the inner edge increases to 93 MPa from 48 MPa for without sleeves case. Similarly, the gasket stress in the outer edge also increases to 70.7 MPa from 28.3 MPa for without sleeves case. This confirms the better performance of blind flange joint with compensating sleeves.

4.3 Bolt Stress

Upon applying the bolt pretension, the bolt member will be subjected to tensile loading and the other members of the joint will be subjected to compression. Tensile stresses exist in the bolt as the flanges tend to pull the ends apart. During pretension axial tensile stress is 86.3 MPa. With application of internal pressure is increased to 97.0 MPa. As internal pressure is applied, fluid tends to pull the flanges apart and in turn flanges pull the bolt ends resulting in increasing the axial tensile stress in the bolt. With temperature rise the bolt stress decreased to 88.7 MPa due to differential thermal expansion of components. Thus, the bolts are relieved of some of the tension existing in it. When compensating sleeves are used, the axial tensile stresses are almost same under bolt pretension and pressure load condition. But, there is a rise in bolt stress upon applying the thermal loads showing that bolts have become tighter due to thermal expansion of sleeves. The variation in average axial tensile stress in bolt under different conditions under both with and without sleeves are shown in Table 3.

Table 3. Average axial tensile stress in bolt.

| Loading Condition | Average axial tensile stress (MPa) |
|-------------------|-----------------------------------|
|                   | Joint without sleeves | Joint with sleeves |
| 1 Bolt Pretension 110 kN | 86.3 | 86.2 |
| 2 (1) + Internal fluid pressure 15 MPa | 97.0 | 96.3 |
| 3 (2) + Thermal load 500°C | 88.7 | 130.5 |

5 Conclusions

Due to hysteresis phenomenon, there is a loss in gasket contact stress due to reduction of loads by applying pressure. Also, when there is a temperature rise this stress decreased further due to differential thermal expansion of joint components. Sleeves were used for compensating this effect. The sealing behavior is verified using finite element analysis. Sleeves not only nullified the loss in gasket contact stress, but also increased it. Thus, the sleeves enhanced sealing behavior of flange joints by raising gasket contact stress at higher temperatures. However, the increase in gasket contact stress at higher temperature upon using compensating sleeves is accompanied with increase in axial tensile bolt stress. The designer has to include both the aspects in the design and trade-off should be met considering the influence of high temperature applications such as nuclear reactors.

Nomenclature

- \( b \) Effective gasket width, m
- \( E \) Young’s modulus of elasticity, N/m²
- \( F_b \) Bolt pretension, N
- \( G \) Gasket reaction diameter, m
- \( K \) Thermal conductivity, W/mK
- \( K_j \) Combined spring constant of the joint, N/m
- \( K_g \) Spring constant of gasket, N/m
- \( K_{fu} \) Spring constant of upper flanges, N/m
- \( K_{fl} \) Spring constant of lower flanges, N/m
- \( m \) Gasket constant
- \( P \) Internal fluid pressure, N/m²
- \( \alpha \) Coefficient of thermal expansion, m°C/m
- \( \nu \) Poisson’s ratio

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