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Avoiding sealing failure of flanged connection for tubes made of dissimilar materials subjected to elevated temperature

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Abstract Carbonic composite materials and ceramics appear to be excellent structural materials for parts subjected to very high temperatures in molten salt reactors (MSRs), in which the reactor core outlet temperature is normally above 700 °C. Because of the high temperature, there are major challenges in the sealing of flanged connections for tubes made of alloys and nonmetallic materials. In this study, an improved method for sealing bolted flange connections for tubes made of dissimilar materials at high temperature is analyzed. The study focuses on the compensation mechanism for the difference in thermal expansion between the bolts and the flanges. An angle is introduced for the sealing surface in the flanged connection to provide effective sealing. The arctangent of the angle is the ratio of the thickness between the theoretical core of the sealing surface and the outside end face of the flange to the distances between the axis of the flanged joint and the theoretical core of the sealing surface of the flange; the sealing surface of the flange, which is made of the same material as the fastening assemblies, faces the fastening assemblies. To ensure effective sealing, the frictional coefficient between the two sealing surfaces should not exceed the tangent of the angle. This result does not agree well with the solution given by previous researchers. Further, in the modified flanged connection, the compression of each bolt in the clamped condition is increased to maintain the compaction force unchanged without increasing the number of bolts on the flanged joint.

Keywords Sealing · Composite flanged connections · Thermal expansion · Mechanical analysis · Molten salt reactors

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1 Introduction

Because of the superior heat transport capability, molten salts are being used for several decades in industrial process heat transfer and thermal storage applications. As the molten salts can be heated up to 1000 °C at atmospheric pressure and can still maintain thermal stability, they can be used as excellent coolants for molten salt reactors (MSRs), where the reactor outlet temperature (ROT) will be above 700 °C [1–3]. However, the retention of mechanical properties at high temperatures is a big challenge for structural materials [4–10]. For metals or alloys, the mechanical strength decreases significantly with the increase in temperature [11]. Relatively, carbonic composite materials, as well as ceramics, generally possess very good physical, chemical, and mechanical properties at high temperatures, and they can be used as high-temperature structural materials for the MSRs. However, there are major challenges in the sealing of flanged connections, which are very common in the pressure vessels and piping systems of nuclear reactors, especially in the case of flanged connections for tubes made of dissimilar materials.

Thus, it is necessary to develop a reliable sealing method for connection of metallic and ceramic materials [12].

Bolted flange connections are widely used in structures where the easy removal and maintenance of structural components are required. Bolt pre-tightening force is one of the important factors influencing the sealing performance of flanged connections. If the bolt pretension is insufficient, the residual stress in the gasket would be too low to seal effectively; if the preload is excessive, the residual stress would be so large that the sealing ring may collapse easily, and lose its resilience, and thus will be unable to prevent leakage caused by changes in temperature or pressure. Therefore, the sealing performance can be improved by slightly increasing the preload of the bolt to increase the residual stress in the gasket. A new method has been developed to improve the reliability of sealing of flanged connections for tubes made of similar materials under high-temperature conditions in MSRs [13]. However, in the case of flanged connections for tubes with dissimilar materials, because of the difference in thermal expansion of the flanges and bolts at elevated temperatures, the amount of rebound of the sealing ring may be insufficient to compensate for the separation of the sealing surfaces. Therefore, loosening occurs, or the sealing ring yields and loses its resilience, resulting in sealing failure. Caccese et al. [14] analyzed a hybrid composite-to-metal bolted connection for plates subjected to flexural loading using experimental investigation and finite element method. Kabche et al. [15] experimentally investigated the structural response of a panel assembly with hybrid composite-to-metal bolted joint under uniform pressure loading. Caccese et al. [16] analyzed the influence of stress relaxation on clamping force in hybrid composite-to-metal bolted joints. All these studies have considered bolted joints for plates with composite structure at room temperature. These results are useful for bolted flange connections operating at room temperature, but are not applicable for high-temperature conditions.

To overcome the above problems, in this study, a method of sealing bolted flange connections for tubes made of dissimilar materials working at high temperature was developed based on mechanical analysis. The compromise mechanism and the prerequisites for the difference in thermal expansion levels of the composite flanges in high-temperature conditions are discussed in detail. The influence of the flanged joint on the tensile stress in the connecting bolts is further analyzed.

2 Composite bolted flange assembly configuration

The widely used conventional bolted flange joint is shown in Fig. 1a. It consists of a male flange, a female flange, bolts, nuts, and flat washers. The sealing surface is a horizontal plane that extends perpendicular to the central axis. A sealing ring or gasket is normally included. An innovative bolted flange assembly concept, first proposed by Lessmann [17], is shown in Fig. 1b, in which the significant modification is that the sealing surface forms a cone angle (α) with respect to a plane perpendicular to the central axis. This concept is further developed and discussed in detail in this study. Schülke [18] considered that the angle (α) shall be at least equal to arctan μ_0 and can vary up to 45°, wherein μ_0 is the coefficient of friction between the sealing material and the adjacent material which includes the theoretical core of the cone. However, the result of this study is not in agreement with this concept; this is discussed below in detail.

3 Analysis and discussion

To study the process of avoiding sealing failure, theoretical analysis of the mechanical coupling of the modified flange assembly is conducted. Based on the analytical solution, the compensation mechanism and the prerequisites for the difference in thermal expansion values of the composite flanges subjected to elevated temperature are discussed in detail.

3.1 Mechanism analysis

A typical arrangement of the modified bolted flange joint without seals or gaskets is shown in Fig. 1b, in which the thermal expansion coefficients (TECs) of the female flange (FF) and male flange (MF) are k_1 and k_2, respectively, where $k_1 \neq k_2$. The material of the fastening assemblies, including bolts, nuts, and flat washers, is the same as that of the FF or MF. When the bolted flange joint is assembled at room temperature (T_0), with the interference force in the sealing surface created by compression, the size relationship can be expressed as follows.

$$L = t_1 + t_2, \tag{1}$$

where t_1 and t_2 represent the thickness between the theoretical core of the sealing surface and the outside end face of the FF and MF, respectively, and L is the length of the bolt between the two outside end faces of the FF and MF.

Consider the case when the sealing surface is perpendicular to the axis of the flange, i.e., $\alpha = 0$, as shown in Fig. 1a. When the bolted flange assembly operates at an elevated temperature T_1 ($T_1 > T_0$), because of thermal expansion, the thickness t_1 between the theoretical core of the sealing surface and the outside end face of the FF changes to t_1', given by

$$t_1' = t_1 + t_1 \cdot k_1 \cdot (T_1 - T_0). \tag{2}$$

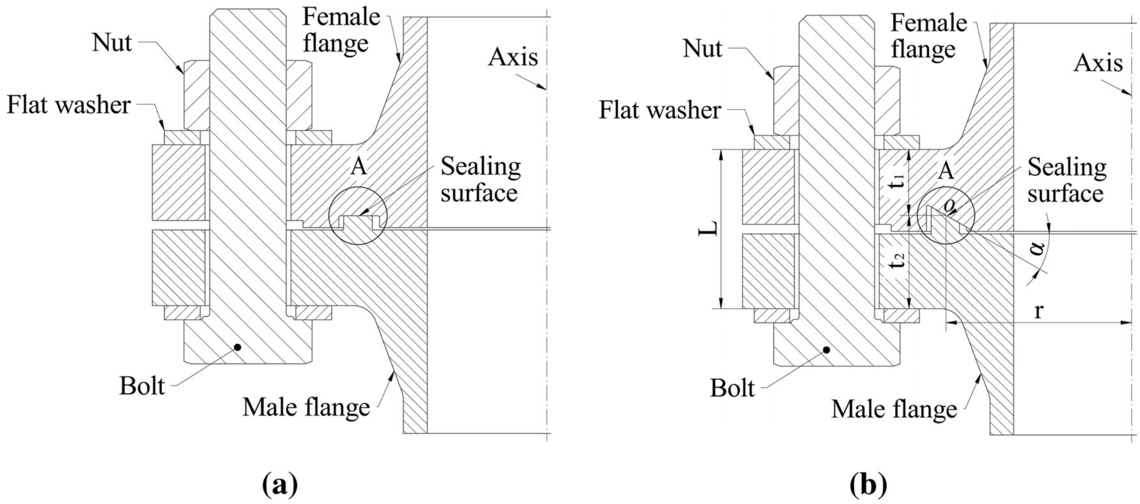

Fig. 1 Cross section of bolted flange connections. **a** Conventional and **b** modified

Similarly, the thickness t_2' is given by

$$t_2' = t_2 + t_2 \cdot k_2 \cdot (T_1 - T_0). \tag{3}$$

The length L changes to L', given by

$$L' = L + L \cdot k_1 \cdot (T_1 - T_0). \tag{4}$$

To ensure that the flanged connection still has a good sealing performance at elevated temperatures, the following relationship should be fulfilled.

$$L' = t_1' + t_2'. \tag{5}$$

Substituting Eqs. (2)–(4) into Eq. (5), we get

$$t_2 \cdot k_1 = t_2 \cdot k_2. \tag{6}$$

It can be seen that $k_1 = k_2$, which is contradictory to the previous assumption of $k_1 \neq k_2$. It can be concluded that the flanged connection would fail to maintain effective sealing at high temperatures without other support measures such as the use of spring washers or disk springs [19].

At the elevated temperature T_1, the distances r_1' and r_2' between the axis of the flanged joint and the theoretical core of the sealing surface of the FF and MF, respectively, are given by

$$r_1' = r + r \cdot k_1 \cdot (T_1 - T_0), \tag{7}$$

$$r_2' = r + r \cdot k_2 \cdot (T_1 - T_0), \tag{8}$$

where r is the distance between the theoretical core of the sealing surface and the axis of the flanged couple at room temperature. Because $k_1 \neq k_2$, it can be safely deduced that $r_1' \neq r_2'$, which indicates the presence of relative radial displacement between the theoretical cores of the sealing surfaces of the FF and MF. This displacement can be expressed by the following equation.

$$\Delta r = |r_1' - r_2'| = r \cdot (T_1 - T_0) \cdot |k_1 - k_2|. \tag{9}$$

(1) When the fastening assemblies are made of the same material as the FF: If $0 < \alpha < \frac{\pi}{2}$, as shown in Fig. 1b, with the sealing surfaces of the two flanges not separated, when $k_1 > k_2$, the change in thickness, Δt, between the two outside end faces of the FF and MF is given by

$$\Delta t = +\Delta r \cdot \tan\alpha. \tag{10}$$

When $k_1 < k_2$, Δt becomes

$$\Delta t = -\Delta r \cdot \tan\alpha. \tag{11}$$

The following size relationship would be required to ensure a good sealing performance in the flanged connection at high temperatures.

$$L' = t_1' + t_2' + \Delta t \tag{12}$$

Substituting Eqs. (7)–(11) into Eq. (12), we get

$$\tan\alpha = \frac{t_2}{r}. \tag{13}$$

Consequently,

$$\alpha = \arctan\left(\frac{t_2}{r}\right). \tag{14}$$

(2) When the fastening assemblies are made of the same material as the MF: If $-\frac{\pi}{2} < \alpha < 0$, the following result may be safely obtained as in the previous case.

$$\alpha = -\arctan\left(\frac{t_1}{r}\right). \tag{15}$$

From Eqs. (14) and (15), it is evident that the angle (α) is determined only by the location of the sealing surface in the flanges, and that it is independent of temperature and thermal expansion coefficient of the material. The arctangent of α (the angle between the sealing surface and inside end face of the flange) is the ratio of the thickness (t) to the radius (r); t is the thickness between the theoretical core of the sealing surface and the outside end face of the flange, whose material is different from that of the fastening assemblies; r is the radius of the circle on which the center line of the sealing surface is located. It follows that

$$\alpha = \arctan\left(\frac{t}{r}\right), \tag{16}$$

wherein the sealing surface of the flange, whose material is the same as that of the fastening assemblies, faces the fastening assemblies.

3.2 Discussions

When the flanged joint referred above is used in an elevated temperature condition, because of the difference in radial expansion of the flanged couple with different TECs, a relative radial displacement will be produced between the sealing surfaces of the flanged couple. When the sealing surface is not perpendicular to the axis of the flange, a relative axial displacement will also be passively developed because of induced stress. Because of the presence of the angle (α) stated above, the axial displacement may compensate for the difference in the axial thermal expansion between the bolts and the flanges, as given in Eq. (12), and consequently avoid leakage in the flanged connection caused by either separation of the sealing surfaces or overstressing the material on the sealing surfaces.

As mentioned above, to ensure that a tiny slippage can occur on the sealing surfaces of the flanged couple, as shown in Fig. 2, the following condition may not be unnecessary.

$$F\sin\alpha \geq F\cos\alpha \cdot \mu_0, \tag{17}$$

$$\mu_0 \leq \tan\alpha, \tag{18}$$

where F is the force caused by thermal expansion of the tubes bonded with the flanged connection assembly, and μ_0 is the frictional coefficient between the two sealing surfaces (shown in Fig. 2).

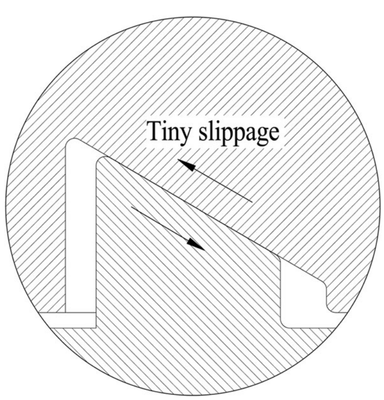

Fig. 2 Flanged joint without sealing ring

When arctan $\mu_0 \leq \alpha < \pi/4$, as considered in the study by Schülke [18], instead of Eq. (16), the tiny slippage may be ensured to occur, nevertheless, not to avoid either separation or overstressing at elevated temperature.

The above method of sealing using lapped surfaces was chosen as the research object to conduct theoretical analysis. In practical applications, O-rings are considered as the most suitable seals for this innovative flanged connection (Fig. 3), especially hollow metal O-rings for high-temperature cases and rubber O-rings for very-low-temperature cases. Metallic flexible graphite spiral wound gasket may be another promising candidate for the seal. Under these conditions, the difference in radial expansion of the flanged couple may result in some subtle deformation of the sealing ring instead of tiny slippage mentioned above, and therefore, the condition $\mu_0 \leq \tan\alpha$ may be not necessary.

There are some major benefits of the above-mentioned flanged connection. When a bolted flange joint is assembled, as shown in Fig. 4, an interference force (F_1) is created on the sealing surfaces by bolt compression (F_b). This compression, which is also referred as interference, squeeze, or nip, is normally perpendicular to the sealing face. If n is the number of the bolts in the flanged connection, considering the symmetry of the flanges and balance of forces and moments, the following equation can be given.

$$n \cdot F_b = F_2 = F_1/\cos\alpha \tag{19}$$

Further,

$$F_b = \frac{F_1}{n \cdot \cos\alpha}. \tag{20}$$

When $\alpha = 0$, when the sealing surface is of conventional type, $\cos\alpha = 1$; consequently,

$$F_b = \frac{F_1}{n}. \tag{21}$$

When $0 < |\alpha| < \frac{\pi}{2}$, when the sealing surface is of modified type introduced above, $0 < \cos\alpha < 1$; consequently,

$$F_b' = \frac{F_1}{n} \cdot \frac{1}{\cos\alpha} > F_b. \tag{22}$$

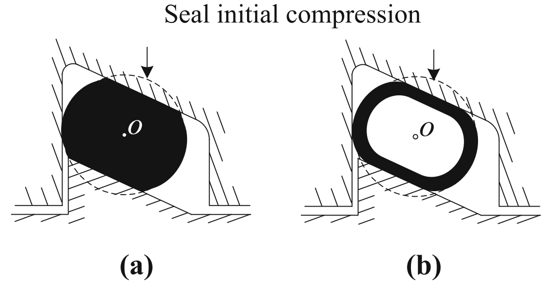

Fig. 3 Two options of the most suitable sealing used in the proposed flanged joint: **a** O-ring and **b** hollow metal O-ring

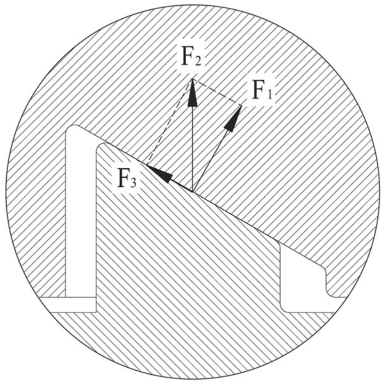

Fig. 4 Loading characteristics

From Eqs. (21) and (22), it can be seen that if the sealing surface is not perpendicular to the axis of the flange, larger compression would be necessary on each bolt to ensure that the same compression force prevails as in the conventional case. In other words, the modified flanged connection may need higher compression on each bolt to maintain the compression force unchanged without increasing the number of bolts on the flanged joint.

3.3 Cases studies

As an example, consider a steel pipe of PN10 welding neck flange (designated as PN) with DN50 connection [20] with a ceramics one as shown above. In this case, t is 18 mm, and r is roughly 40 mm; accordingly, the cone angle (α) should be designed with $\arctan(18/40)$, i.e., approximately 24.2°. Evidently, the frictional coefficient μ_0 between the two sealing surfaces should not exceed 0.45; this performance is possible with most materials [21–23]. The relationship of cone angle (α) with nominal diameter (DN) of the flanged connections is shown in Fig. 5. As can be seen in the figure, the angle (α) derived from Eq. (16) decreases with the increase in DN, showing a

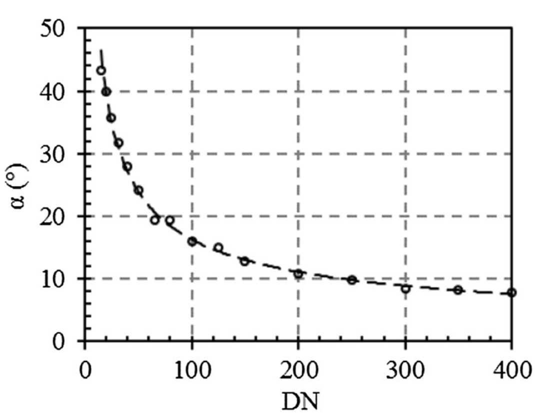

Fig. 5 Cone angle (α) versus nominal diameter (DN) of the flanges

typical power function relationship. A larger flange nominal diameter requires a smaller cone angle (α), and hence a lower friction coefficient (μ_0) derived from Eq. (18) is required.

4 Conclusion

Because of the difference in thermal expansion of the flanges and bolts in flanged connections for tubes with different materials in MSRs, the rebound amount of the sealing ring may be insufficient to compensate for the separation of the sealing surfaces; hence, loosening can occur, or the sealing ring can yield and lose resilience, resulting in sealing failure. In the flanged assembly, a certain acute angle (α) is formed between the sealing surface and the base plane perpendicular to the central axis. The arctangent of α equals the ratio of the thickness (t) to the radius (r), wherein the sealing surface of the flange, whose material is the same as that of the fastening assemblies, faces the fastening assemblies. To ensure effective sealing, the frictional coefficient (μ_0) between the two sealing surfaces should not exceed $\tan\alpha$. Moreover, in the flanged assembly, without changing the residual stress in the gasket, the pre-tightening force borne by the bolts is increased without increasing the number of bolts.

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