Prediction of Time-Correlated Leakage Rates of Bolted Flanged Connections by Considering the Maximum Gasket Contact Stress

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The leakage rates of bolted flanged connections change with service time due to the gasket contact stress relaxation and degradation of gasket material. The nonuniform distribution of gasket contact stress has significant influence on the leakage behavior of the connections. In order to predict the time-correlated leakage rates accurately, both appropriate approaches to dealing with the gasket residual contact stress and a corresponding time-correlated leakage model of sealing elements are needed. In this paper, an analytical model has been developed to determine the maximum contact stress at gasket outer diameter in consideration of creeps of flanges, bolts and gasket coupled to the axial deformation compatibility equation. The analysis was verified against the finite element numerical simulation considering the nonlinear behavior of gasket and material creep. The analytical results of the maximum contact stress of gasket and its change over time compared well with those obtained by finite element method. A time-correlated leakage model of nonmetallic gasket sealing connections based on the porous medium theory was established, in which the effects of the gasket material degradation and contact stress relaxation on the sealing performance were taken into consideration. Furthermore, a leakage rate prediction method was proposed. Some long-term sealing performance tests were performed on two types of gaskets to obtain the coefficients in the leakage model. The leakage rate prediction method proposed in this paper was also validated against the experimental data presented by other researchers, and the errors between the predicted values and the experimental results were within 18%. [DOI: 10.1115/1.4004818]

Keywords: leakage rate, leakage model, bolted flanged connection, time-correlation, gasket contact stress

1 Introduction

Bolted flanged connections are commonly used in pressure vessels and piping. The failure of sealing system is mainly caused by the leakage of the connections. An accurate prediction of the leakage rates is the basis of the design and tightness assessment of bolted flanged connections. Actually, the leakage rate is not only correlated with working conditions but also with service time. With the extension of service time, the gasket material degrades, the connection relaxes, and the sealing performance declines significantly.

In real bolted flanged connections, the gasket contact stress declines gradually because of the material creep and stress relaxation of the connection, which make the tightness of the sealing system decrease [1]. The gasket contact stress at the outer diameter is usually higher due to flange deflection and this increases the tightness of the connection to some extent [2,3]. It has been found that the leakage rate depends not only on gasket effective contact width but also more on the maximum contact stress of gasket than on its average value [4]. Some research on the creep relaxation of bolted flanged connections and its effect on gasket contact stress change and tightness of the connections has been conducted [1] and [5–7], but little research work on the influence of the nonuniform gasket stress distribution on time-correlated leakage behavior of the connections has been reported.

The prediction method of leakage rates by considering the maximum gasket contact stress has also not been found up to now. Several leakage models were presented in the literature, such as flat circular plate model, triangle groove model, and porous medium model. In the porous medium model, most nonmetallic gaskets and compound gaskets of metallic and nonmetallic materials are assumed to be made of porous media. This model can be used to describe nonmetallic gasket leakage behavior satisfactorily [8,9]. A laminar-molecular flow model was used to predict gasket leakage rates considering the leakage paths as parallel straight capillaries of uniform length by Marchand et al. [10]. Jolly and Marchand used the intrinsic permeability k_v and a characteristic Knudsen number K_n' of the gasket to predict the leakage rates of different gas mediums through the gaskets considered to be made of porous medium [11]. The time dependent leakage characteristic of nonmetallic gaskets was investigated and a leakage model was proposed by Gu et al. [12]. Most existing leakage models are only suitable for calculating the leakage rates at room temperature, in which the effect of the service time, gasket contact stress relaxation, its nonuniform distribution and the degradation of gasket materials on the sealing performance of the connections were not taken into consideration.

In this paper, an analytical model has been developed to determine the maximum gasket contact stress with material creep of each element in the connection coupled to the axial deformation compatibility equation. The analysis was verified against the finite element (FE) numerical simulation. A time-correlated leakage rate prediction model of nonmetallic gasket sealing connections based on the porous medium theory was established, in which the effects of the material degradation and contact stress relaxation process of gasket on the sealing performance were taken into

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consideration. Some long-term sealing performance tests were performed on two types of gaskets; namely, flexible graphite (FG) gasket and compressed nonasbestos fiber (CNA) gasket. The leakage rate prediction method proposed in this paper was also validated against the experimental data presented by other researchers.

2 Analysis of the Maximum Gasket Contact Stress

2.1 Analytical Method to Determine the Maximum Gasket Contact Stress. The tightness of bolted flanged connection is largely related to gasket residual contact stress. In a real connection, the distribution of gasket contact stress in the radial direction is nonuniform due to the flange deflection, and the leakage rate depends more on the maximum gasket contact stress than on its average value.

The bolted flanged connection is a statically indeterminate structure. The flexibility interaction analysis in the axial direction is the key to determine the final remaining load in terms of the initial one. The correct modeling of the gasket is an important factor to obtaining reliable analytical results. The model for gasket deformation analysis is shown in Fig. 1.

The decrease of gasket compressive stress and the degradation of gasket material at elevated temperature result in the leakage of bolted flanged connections eventually. The realization of initial seal of flanged connections depends largely on gasket compressive performance, while good gasket resilient performance and sealing behavior are particularly expected in order to decrease the separation of contacting surfaces and to eliminate the leakage of the bolted flanged connections under operating conditions. The relationships between gasket contact stress and deformation are given by Gu and Zhu [13] as follows:

Compressive performance formula is

$$S^i = (A_C - B_C T^f)\left(D^i_g\right)^{N_C} \tag{1}$$

Resilient performance formula is

$$\frac{S^f}{S^i} = A_S + B_S\left[\frac{D^f_g}{D^i_g}\right]^{(A_T + B_T T^f)} \tag{2}$$

where S^i and S^f are the gasket seating stress and average residual contact stress on the entire gasket area, respectively, D^i_g and D^f_g are the corresponding gasket compressive deformations, T is the temperature, and A_C, B_C, N_C, A_S, B_S, A_T, and B_T are the regression coefficients.

The total axial deformation of gasket $D_g(r)$ at different radial positions r consists of two parts

$$D_g(r) = D^f_g(r) + D^c_g \tag{3}$$

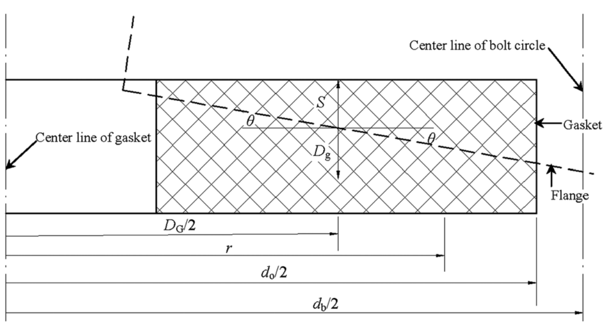


Fig. 1 Model for gasket deformation analysis

where $D^f_g(r)$ is the gasket axial deformation calculated by Eq. (1) or (2), and D^c_g is the creep deformation of gasket that is given by Zhang et al. [14] as follows:

$$D^c_g = D^i_g(B_R + C_R T)\ln(t) \tag{4}$$

where B_R and C_R are the regression coefficients and t is a service time.

Equation (3) gives the axial deformation of the gasket at its outer diameter as follows:

$$D_g\left(\frac{d_o}{2}\right) = D^f_g\left(\frac{d_o}{2}\right) + D^c_g \tag{5}$$

where d_o is the gasket outside diameter.

The relationship between the deformation $D_g(r)$ at any radial position r and the average gasket deformation D_g can be described as

$$D_g(r) = D_g + (2r - D_G)\tan\theta \tag{6}$$

where D_g is the average gasket deformation, D_G is the diameter where gasket average contact stress acts, and θ is the flange rotation.

The flange rotation under seating condition is given by Feng et al. [15] as follows:

$$\theta^i = \frac{(1-\mu^2)(1+\mu)^3}{E_f g_1^3\sqrt{D_{if}/g_0}\,L'}MV_1 \tag{7}$$

The flange rotation under final operating condition is

$$\theta^f = \frac{(1-\mu^2)(1+\mu)^3}{E_f g_1^3\sqrt{D_{if}/g_0}\,L'}MV_1 + \frac{C_1 D_{if}^2\gamma p}{t_f\left(t_f^2 + C_2 g_e \gamma\right)} \tag{8}$$

where μ is the Poisson's ratio, D_{if} is the flange inside diameter, E_f is the elasticity modulus of flange material, g_0 is the thickness of the small end of cone neck, g_1 is the thickness of the large end of cone neck, M is the flange moment, p is the pressure of medium, t_f is the thickness of flange, and C_1, C_2, L', and V_1 are the dimensionless coefficients that can be obtained according to Ref. [16].

The displacement $D^i_g(\frac{d_o}{2})$ at gasket outside diameter under seating condition is obtained from Eq. (6) as follows:

$$D^i_g\left(\frac{d_o}{2}\right) = D^i_g + (d_o - D_G)\tan\theta^i \tag{9}$$

Similarly, the displacement $D^f_g(\frac{d_o}{2})$ at gasket outside diameter under operating condition is given by

$$D^f_g\left(\frac{d_o}{2}\right) = D^f_g + (d_o - D_G)\tan\theta^f \tag{10}$$

A relationship exists between the gasket residual contact stress and the initial bolt-up force and this is established by considering the axial displacement compatibility at any radial position [7,17]. In consideration of the creep of bolts and flanges, and their thermal expansion in axial direction, the deformation compatibility equation of bolted flanged connection at the bolt circle can be expressed as follows:

$$\begin{aligned} &D^i_g\left(\frac{d_o}{2}\right) - \left[D^f_g\left(\frac{d_o}{2}\right) + D^i_g\left(B_R + C_R T^f\right)\ln(t)\right] \\ &+(d_b - d_o)\left(\tan\theta^i - \tan\theta^f\right) = \left[q^f_b W^f - q^i_b W^i + w^c_b\right. \\ &\left.+\alpha^f_b l_0\left(T^f - T^i\right)\right] + 2\left(D^f_f + w^c_f - D^i_f\right) + 2\Delta t_f \end{aligned} \tag{11}$$

where q_b is the elastic coefficient of bolt, W is the bolt load, d_b is the central circle diameter of bolt, w_b^c and w_f^c are the axial creep deformations of bolt and flange, respectively, which can be obtained according to Ref. [6], α_b^f is the linear expansion coefficient of bolt material under operating temperature, l_0 is the initial length of bolt, D_f is the flange deformation at bolt circle, and Δt_f is the axial expansion deformation of flange at bolt circle, which can be calculated according to Ref. [18].

It can be seen from Eq. (11) that the deformation of gasket is correlated with the material of each element and working conditions of bolted flanged connection. The maximum gasket contact stress at different service time $S_{\max}(t)$ can be obtained by solving the transcendental equations.

2.2 FE Numerical Simulation. Because the gasket contact stress cannot be measured accurately by testing, a three-dimensional (3D) FE model of bolted flanged connection considering the nonlinear behavior of gasket and material creep was constructed to verify the maximum time-correlated gasket contact stress obtained by analytical method.

A bolted flanged connection, which is made of a pair of ANSI B16.47 CLASS300, NPS30 welding neck flanges and tightened with 28 bolts of NPS1-3/4, operates under the pressure of 3 MPa and at the temperature of 813 K. The materials of flange and bolt are 10CrMo910 and 25Cr2MoVA, respectively. FG gasket is used as the sealing element. The connection was simplified to a cyclic symmetric model, and a part of the connection containing a bolt was selected for the study and the 3D FE model was established using the software ANSYS, as shown in Fig. 2.

A 3D 20-node solid element was used to model the materials of flange and bolt. A gasket interface element was used to model gasket material and the nonlinear behavior of FG gasket was described by Ref. [13] as follows:

$$S^i = 36.18\left(D^i\right)^{1.912} \tag{12}$$

$$\frac{S^f}{S^i} = -0.208 + 1.114\left(\frac{D^f}{D^i}\right)^{8.028} \tag{13}$$

At the temperature of 813 K, the creep of flange and bolt materials was simulated according to the time hardening creep constitutive equations provided by Xuan et al. [19] and Zheng et al. [20] as follows:

$$\text{10CrMo910}: \dot{\varepsilon}_c = 5.85E-10\sigma^{2.1} \times t^{-0.61} \tag{14}$$

$$\text{25Cr2MoVA}: \dot{\varepsilon}_c = 4.03E-11\sigma^{2.731} \times t^{-0.332} \tag{15}$$

where $\dot{\varepsilon}_c$ is the creep rate and σ is the stress.

According to Ref. [13], the regression coefficients in Eq. (4) are $B_R = 2.84 \times 10^{-3}$ and $C_R = 1.93 \times 10^{-5}$. The creep function of gasket has been modified into a creep equation suitable for the ANSYS 6.1 [21] according to the creep experimental data of FG gasket presented in Ref. [13]. In addition, since the gasket interface element of ANSYS does not have the creep option, a plane plate of half the thickness of the gasket modeled with a volume element is placed in the middle of the gasket to handle the creep effect of gasket [4]. Because of the deformation compatibility of the bolted flanged connection, the gasket stress will change when the creep of the plate takes place.

3 Establishment of Time-Correlated Leakage Model of Nonmetallic Gasket by Considering the Gasket Contact Stress Relaxation Process

3.1 Establishment of Time-Correlated Leakage Model of Nonmetallic Gasket. The leakage rate of nonmetallic gasket depends not only on the maximum gasket contact stress but also on the properties of gasket material.

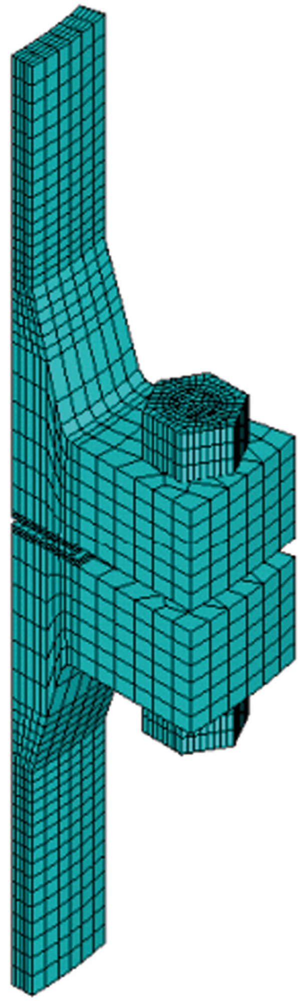


Fig. 2 3D FE model of bolted flanged connection

The degradation of nonmetallic gasket materials is a physicochemical process under the condition of both heat and oxygen. The reaction rate is related to the temperature, and it can be expressed by Arrhenius equation

$$K = A\exp(-U/RT) = A\exp(-B/T) \tag{16}$$

where U is the reaction activation energy, R is the gas constant, A is a constant, and $B = U/R$.

For gaskets used at elevated temperatures, their creep relaxation increases; gasket contact stress and resilience performance decrease, and the number and size of leakage paths increase with service time. According to Maxwellian attenuation model, the degradation of gasket performances can be described by the following empirical formula:

$$P_t = C\exp(-Kt^{\alpha})P_0 \tag{17}$$

where P_0 and P_t are gasket performances at time $t = 0$ and t, respectively, and α and C are constants.

Most nonmetallic gaskets can be assumed to be made of porous media material. The flow of gas through porous media includes molecular transfer and convection transfer. The total flow rate L of gases through porous media is the sum of the laminar flow rate and the molecular flow rate. The formula was given by Gu et al. [22] as follows:

$$L = \sum_{i=1}^{k}\left(\frac{\pi r_i^4}{8\eta c l_m} p_m + \frac{4 r_i^3}{3 c l_m}\sqrt{\frac{2\pi R T}{M'}}\right)(p - p_1) \quad (18)$$

where p and p_1 are the pressures at the entrance and exit of leakage paths, respectively, p_m is the average pressure, $p_m = (p + p_1)/2$, l_m is the average length of leakage paths, c is a bending coefficient of leakage paths, η is the dynamic viscosity of gas, r_i is the radius of any leakage path, k is the total number of leakage paths, and M' is the gas molecular weight.

The radius r and number of leakage path k are characterized by ε as follows:

$$\varepsilon = f(k, r) \quad (19)$$

There exists the following relationship among the number k, radius r_i of leakage paths, and the maximum gasket contact stress

$$k = f_1\left(S_{\max}(t)^{-a_1}\right); \quad r_i = f_2\left(S_{\max}(t)^{-a_2}\right) \quad (20)$$

where a_1 and a_2 are the constants.

Then, ε can be expressed as

$$\varepsilon = C \exp(K t^{\alpha}) f(S_{\max}(t)^{-n}) \quad (21)$$

According to Eqs. (18)–(20), the total leakage rate of gas through gaskets L can be calculated by the following formula:

$$L = \left\{A_L \frac{1}{l\eta} \exp\left[B_L \exp\left(-\frac{C_L}{T}\right) t^{\alpha}\right] \left(\frac{S_{\max}(t)}{S^i}\right)^{-n_L} p_m + A_B \frac{1}{l}\sqrt{\frac{T}{M'}} \exp\left[B_M \exp\left(-\frac{C_M}{T}\right) t^{\beta}\right] \left(\frac{S_{\max}(t)}{S^i}\right)^{-n_M} p_m\right\} \times (p - p_1) \quad (22)$$

where l is the effective sealing width of gasket, A_L, A_M, B_L, B_M, C_L, C_M, α, β, n_L, and n_M are the regression coefficients, which can be obtained from experimental data.

When the gasket contact stress is smaller and the medium pressure is higher, the leakage is predominantly laminar flow and, in this case, the influence of molecular flow can be ignored. Thus, Eq. (22) is simplified to

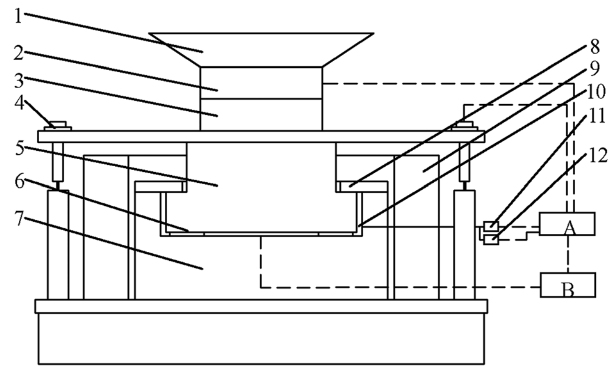
$$L = A_L \frac{1}{l\eta} \exp\left[B_L \exp\left(-\frac{C_L}{T}\right) t^{\alpha}\right] \left(\frac{S_{\max}(t)}{S^i}\right)^{-n_L} p_m (p - p_1) \quad (23)$$

which expresses the relationship among the leakage rate L, the maximum gasket contact stress $S_{\max}$, medium pressure p, operating temperature T, and service time t.

3.2 Experimental Verification. In order to validate the established leakage model and obtain the coefficients in the leakage model, some long-term sealing performance tests were performed on FG and CNA gaskets. These two kinds of gaskets can be considered to be made of porous media.

3.2.1 Test Apparatus. Tests were performed on a testing machine for gasket comprehensive performances at elevated temperatures, as shown in Fig. 3.

The machine consists of a rack, two test rigid flanges, a loading system, a heating system, a medium supplying system, a leakage rate measuring system, a controlling system, and a data acquisition system. There exists a low pressure chamber between the gasket and flanges. According to the state equation of ideal gas, the leakage rate can be obtained by measuring the change of the pressure and temperature of test medium in the chamber over time and the volume of the chamber. All test parameters are measured with high precision sensors, such as gasket compressive stress, gasket deformation, test temperature, test medium pressure, and pressure



1. Rack 2. Force sensor 3. Thermal insulation board 4. Displacement sensor
5. Upper flange 6. Tested gasket 7. Lower flange 8. Platen 9. Electric furnace
10. Leak detection chamber 11. Temperature sensor 12. Low pressure sensor
A. data acquisition system B. medium supplying system

Fig. 3 Schematic diagram of test apparatus

and temperature of the leakage fluid accumulated in the leak detection chamber. The measurement range of the leakage rate is 10^{-6}–0.1 $\mathrm{Pa{\cdot}m^3{\cdot}s^{-1}}$, and the minimal measurable leakage rate is 10^{-6} $\mathrm{Pa{\cdot}m^3{\cdot}s^{-1}}$.

3.2.2 Test Procedure. The test parameters are listed in Table 1. Two groups of samples for each kind of gasket were selected for testing. For each kind of gasket, three different combinations of test pressure, temperature, and gasket seating stress were determined according to uniform design method with three levels for each parameter. At each stress level, the test medium was conducted at certain time. At each test point, the leakage rate was monitored until a stable value was obtained.

4 Leakage Rate Prediction

According to Eq. (23) and the regression coefficients obtained by regression of experimental data for each type of gasket, the leakage rates of gas leaking through FG and CNA gaskets of different sizes can be predicted under various working conditions.

Obviously, the leakage rate calculated by Eq. (23) is correlated with the material and size of gaskets. It can be assumed that the leakage paths in nonmetallic gaskets are distributed uniformly in the circumferential direction. The larger the gasket perimeter, the more the leakage paths are. Therefore, when the leakage rate is calculated for the gaskets with different sizes, a correction of Eq. (23) is necessary; the leakage rate L becomes

$$L = A_L \frac{1}{l\eta} \exp\left[B_L \exp\left(-\frac{C_L}{T}\right) t^{\alpha}\right] \left(\frac{S_{\max}(t)}{S^i}\right)^{-n_L} \times p_m (p_1 - p_2) \frac{D_2}{D_1} \quad (24)$$

where D_1 and D_2 are the outer diameters of the tested gasket and the gasket of which the leakage rate will be predicted, respectively.

Table 1 Test parameters

Test sample	Test medium	Medium pressure p (MPa)	T(K)	S^i (MPa)	t(h)
FG gasket ($\varphi 91 \times \varphi 61 \times 1.5$ mm)	99.9% nitrogen	3	813	40	1, 20, 35, 50, 65, 80, 100, 120, 140, 160, 180, 200
		4	373	60	
		6	273	20	
CNA gasket ($\varphi 109 \times \varphi 61 \times 2$ mm)		3	473	40	
		4	298	60	
		6	373	20	

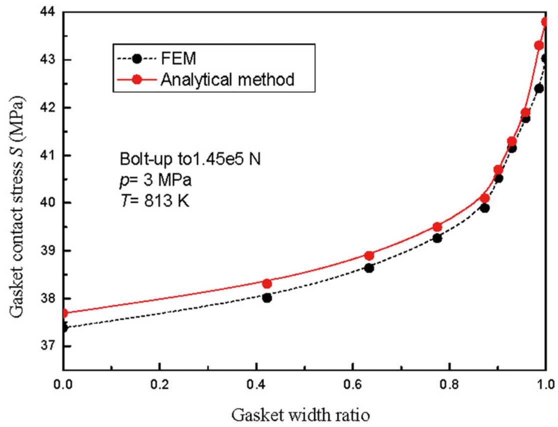


Fig. 4 Radial distribution of gasket contact stress for FG gasket

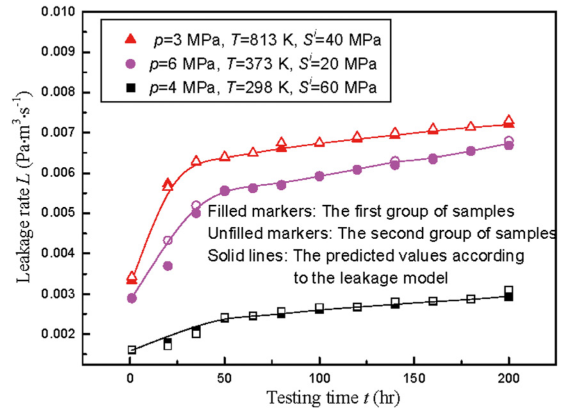


Fig. 6 Relationship between L and t under different experimental conditions for FG gaskets

5 Results and Discussion

In order to verify the analytical method for determining the maximum gasket contact stress, the gasket contact stress at different radial positions, and the maximum and average gasket contact stresses at different service time were calculated by both analytical method and finite element method (FEM) under the same conditions as those described in the FE model.

When different radial positions, r, are chosen, the gasket contact stress at these position can be calculated by solving the corresponding deformation compatibility equation. So, the gasket contact stress at any radial position can be obtained. Figure 4 illustrates the radial distribution of gasket contact stress. The contact stress increases from inside to outside in radial direction. At the outer diameter of gasket, the contact stress is higher due to flange deflection and this can be beneficial for some kinds of gasket provided the crush limit is not reached. The results obtained by the analytical method compares well with those calculated by the finite element method.

Figure 5 shows the relationship between the average and the maximum gasket contact stresses and operating time obtained by analytical and FE methods under two different bolt loads. The gasket contact stress decreases with the operating time. In the first 50 h, the stress decreases noticeably because of the higher rate of creep and relaxation of the connection. After that, the stress decreases slowly. It can be seen that the difference between $S^f(t)$ and $S_{\max}(t)$ is bigger with a higher pretightening load of bolt. The results obtained by the proposed analytical method compare well with those obtained by FEM. Therefore, the results of the maximum gasket contact stress from analytical method could be used to predict the leakage rates of bolted flanged connections.

Some long-term sealing performance tests were performed on FG and CNA gaskets on the test apparatus shown in Fig. 3. The relationships between leakage rate L and service time t for these two types of gaskets are shown in Figs. 6 and 7. The leakage rates obtained under the experimental conditions are indicated by filled and unfilled markers. It can be seen that, at the early stage of the tests, the leakage rates increase noticeably with testing time because of the higher degradation rate of gasket material and notable decrease of the gasket contact stress. After that, the leakage rates increase slowly. At lower test temperatures, the leakage rates increase slowly with time, while they change very noticeably at higher test temperatures, which indicate that the test temperature has a significant effect on gasket sealing performance, the gasket material becomes stiff, brittle, and more porous. It can be seen that the sealing performance of FG gasket is better than that of CNA gasket at elevated temperature.

The coefficients A_L, B_L, C_L, n_L, and α in time-correlated leakage model can be obtained by regression analysis of the experimental data, and they are listed in Table 2.

By substituting working parameters such as p, T, S^i, and $S^f(t)$ into Eq. (23) after obtaining the regression coefficients in the time-correlated leakage model, the leakage rates of FG and CNA gaskets under the given working conditions were predicted, as indicated by solid lines in Figs. 6 and 7. It can be seen from Figs. 6 and 7 that the variation of leakage rate with testing time follows well the curves expressed by Eq. (23), as indicated by the high

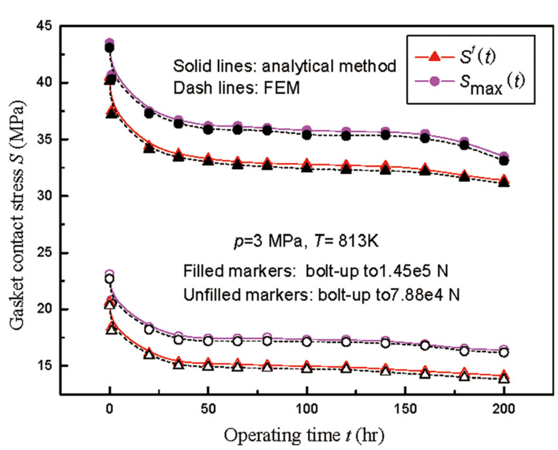


Fig. 5 Comparison between $S^f(t)$ and $S_{\max}(t)$ for FG gasket

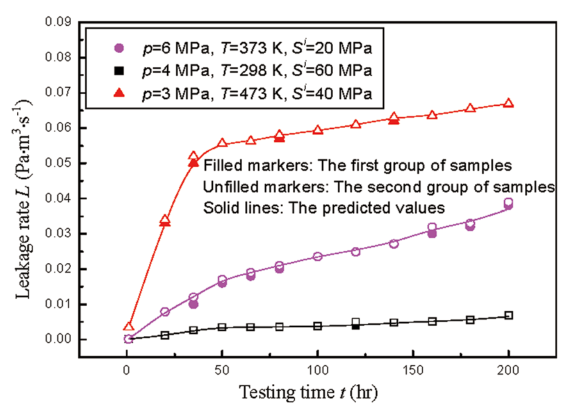


Fig. 7 Relationship between L and t under different experimental conditions for CNA gaskets

Table 2 Regression coefficients in Eq. (23)

Gasket	A_L	B_L	C_L	n_L	α	Correlation coefficient R
FG	4.724×10^{-8}	19.54	212.8	−220.7	0.02500	0.9540
CNA	6.440×10^{9}	−37.76	−54.05	−11.31	−0.008046	0.9770

value of the correlation coefficient R. Therefore, it can be concluded that the proposed Eq. (23) well models the time-correlated leakage behavior of the tested gaskets, and the errors between the prediction values and the experimental results are within 17%.

In order to validate the leakage rate prediction method proposed in this paper, some predictions were made according to Eq. (24). The outer diameter D_2 of the gasket in this connection is 124 mm. Figure 8 presents the comparison between experimental leakage rates on a PN10, DN300 welding neck flange presented in Ref. [23] and calculated leakage rates using the established model for FG gaskets under the working condition that $p = 2.5$ MPa, $T = 673$ K, and $S^i = 30$ MPa. It can be found from Fig. 8 that the leakage rates predicted based on the maximum gasket contact stress $S_{max}(t)$ are closer to the results presented in Ref. [23] than those calculated based on the average gasket contact stress. The maximum error between the predicted values based on $S_{max}(t)$ and the experimental results is about 18%. The essential reason for the error of prediction is largely due to the different leak detection methods used in Ref. [23]; they had larger error than in the present work.

Figure 9 illustrates the comparison between experimental leakage rates on a rigid dummy flange presented in Ref. [24] and calculated leakage rates using the established model for CNA gaskets under the working condition that $p = 3.5$ MPa, $T = 473$ K, and $S^i = 35$ MPa. The outer diameter D_2 of the gasket in this case is 109 mm. It can be seen that the predicted results based on $S_{max}(t)$ and $S^f(t)$ are both closer to the experimental results for the short term. But with the extension of service time, the leakage rates increases due to the combined effect of gasket material degradation and contact stress relaxation. The maximum error between the predicted values based on $S_{max}(t)$ and the experimental results is about 12%. It is more reasonable to predict the leakage rates of bolted flanged connections in long-term service according to the maximum gasket contact stress than the average gasket contact stress.

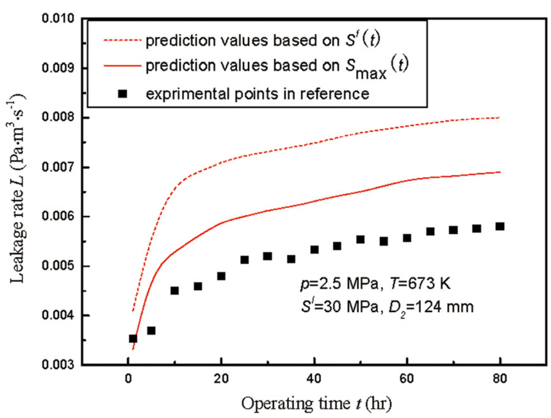


Fig. 8 Comparison between experimental results presented in Ref. [23] and calculated leakage rates using the established model for FG gaskets

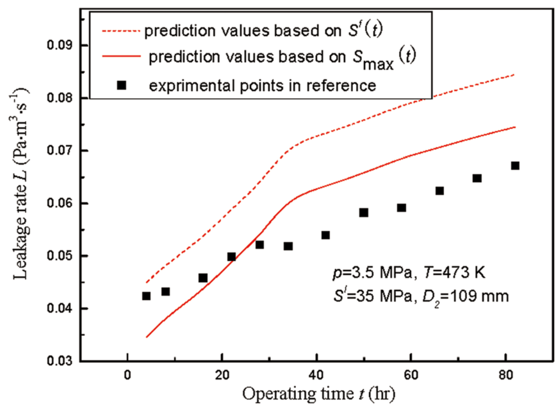


Fig. 9 Comparison between experimental results presented in Ref. [24] and calculated leakage rates using the established model for CNA gaskets

6 Conclusion

An analytical model has been developed to determine the maximum contact stress at gasket outer diameter in which the material creep of each element in the connection was coupled to the axial deformation compatibility equation. The analysis was verified against the finite element numerical simulation considering the nonlinear behavior of gasket and material creep. The difference between $S^f(t)$ and $S_{max}(t)$ is bigger with a higher pretightening load of bolt than that with a lower load. The results from FEM agree well with those obtained by the analytical method. Therefore, the results of the maximum gasket contact stress from analytical method can be used to predict the leakage rates of bolted flanged connections.

A time-correlated leakage model of nonmetallic gasket sealing connections based on the porous medium theory was established, in which the effect of the material degradation and contact stress relaxation process of gasket on the sealing performance were taken into consideration. Some long-term sealing performance tests were performed on FG and CNA gaskets, and the coefficients in leakage model for two types of gaskets were obtained by regression analysis of the experimental data. At the beginning stage of the tests, the leakage rates are generally small but increase noticeably with increase in testing time because of the slight degradation degree but considerable degradation rate of gasket material and significant decrease of gasket contact stress. After that, the leakage rates increase slowly. At lower test temperatures, the leakage rates increase slowly with testing time, while they change very noticeably at higher test temperatures, which indicate that the test temperature has a significant effect on gasket sealing performance.

A corresponding leakage rate prediction method for FG and CNA gaskets with different sizes and used under various working conditions was put forward. It is more reasonable to predict the leakage rates of bolted flanged connections in long-term service according to the maximum gasket contact stress than the average gasket contact stress. The leakage rate prediction method proposed in this paper was also validated against the experimental data presented by other researchers and shown to produce relatively accurate predictions.

The present work was fulfilled on the welding neck flange connection adopted FG and CNA gaskets as sealing elements. For other types of flanges, such as slip-on welding flanges and lapped flanges, further research on both the mechanical and sealing behavior should be undertaken. The additional tests should also be conducted before predicting leakage rate of other types of gaskets. Furthermore, a parameter study is required for a whole range of sizes of flanges in order to validate the effectiveness of the proposed model.

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Nomenclature

α, β = regression coefficients
θ = flange rotation
μ = Poisson's ratio
η = dynamic viscosity of gas
ε = a function of k and r
$\dot{\varepsilon}$ = creep rate
$\overset{c}{\sigma}$ = stress
α_b^f = linear expansion coefficient of bolt
Δt_f = axial expansion deformation of flange at bolt circle
w_b^c = axial creep deformation of bolt
w_f^c = axial creep deformation of flange at bolt circle
a_1, a_2, A = constants
A_C, A_L, A_M, A_S, A_T = regression coefficients
B_R, C_R = gasket creep coefficients
B_C, B_L, B_M, B_S, B_T = regression coefficients
c = bending coefficient of leakage paths
C = constant
C_1, C_2 = sealing performance coefficients
C_L, C_M = regression coefficients
d_b = central circle diameter of bolt
d_i = inside diameter of gasket
d_o = outside diameter of gasket
D_{if} = flange inside diameter
D_f = flange deformations at bolt circle
D_g = gasket compressive deformation
D_g^c = creep deformation of gasket
$D_g(r)$ = gasket axial deformation at different radial positions
E_f = elasticity modulus of flange material
g_0 = thickness of small flange cone neck
g_1 = thickness of larger flange cone neck
k = total number of leakage paths
K = reaction rate
L = effective gasket width
l_0 = initial length of bolt
l_m = average length of leakage paths
L = leakage rate
L' = coefficient
M = flange moment
M' = gas molecular weight
n_L, n_M = regression coefficients
P_0 = gasket performance at time $t = 0$
P_t = gasket performance at time t
p_1 = pressure at the entrance of leakage paths
p_2 = pressure at the exit of leakage paths
p_m = average pressure
q_b = elastic coefficients of bolt
r = radius position
R = gas constant
S = gasket contact stress
S^i = gasket seating stress
S^f = average residual contact stress on the entire gasket area
S_{max} = the maximum gasket contact stress on gasket contact width
$S_g(r)$ = gasket contact stress at radius r at a given time
t = service time
t_f = flange thickness
T = temperature
U = reaction activation energy
V_1 = coefficient
W = bolt load

Superscripts

$_i$ = initial seating condition
f = final operating condition

Acronyms

FE = finite element
FEM = finite element method
FG gasket = flexible graphite gasket
CNA gasket = compressed nonasbestos fiber gasket

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