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STRESS ANALYSIS AND SEALING PERFORMANCE EVALUATION OF BOLTED PIPE FLANGE CONNECTIONS WITH SMALLER AND LARGER NOMINAL DIAMETER UNDER REPEATED TEMPERATURE CHANGES

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ABSTRACT

Pipe flange connections with gaskets in chemical plants, electric power plants and other industrial plants are usually exposed to elevated internal pressure with cyclic thermal condition. It is important to investigate the sealing performance of pipe connections under long term severe thermal exposure swings to ensure operational safety. In this study, the effects of cyclic thermal conditions on the sealing performance and mechanical characteristics in larger and smaller nominal diameter of pipe flange connection are examined using FEM calculations. Helium gas leakage is predicted using the contact gasket stress obtained from the FEM results. On other hand, the leakage tests using the smaller nominal diameter of pipe flange connection were conducted to measure the amount of helium gas leakage and to compare with the predicted amount of gas leakage. As the results, the contact gasket stress distributions were changed dramatically under cyclic thermal condition and elevated internal pressure. In the pipe flange connections with smaller nominal diameter, the contact gasket stress was the smallest in the restart condition. On other hand, the minimum contact gasket stress in the pipe flange connection with larger nominal diameter was depending on the materials of connection. In the pipe flange connection with larger nominal diameter, the contact gasket stress distributed and changed in the radial direction due to the flange rotation. A fairly good agreement was found between the experimental leakage result and predicted leakage results.

INTRODUCTION

Pipe flange connections with gaskets in chemical plants, electric power plants and other plants are usually exposed to internal pressure under repeated thermal conditions. Even a plant operated at a lower temperature (e.g. room temperature) pipe flange connections are exposed to a change in ambient temperature. In order to avoid an unexpected leakage, thermo-mechanical analysis is required since the change in the mechanical characteristics and physical properties of the gasket affect the stress state in the pipe flange connections. The properties of the gasket are especially important in evaluating the sealing performance of pipe flange connections due to the relatively large change in properties compared to that of steel. For example, the thermal expansion coefficient of a gasket is usually much larger than that of steel.

In general, the increment of temperature doesn't always improve the sealing performance of pipe flange connections and hot bolting is often required. The other factor that should be considered in analyzing the sealing performance at elevated temperatures is the nominal diameter of flange connection and deformation behavior, such as stress-strain curve. It is well known that the characteristics of larger nominal diameter of pipe flange are completely different from that of smaller nominal diameter of pipe flange connection. Although the volume of the gasket increases as increasing the temperature due to thermal expansion, the contact gasket stress wouldn't improve because the elastic modulus of the gasket will decrease. It is necessary to examine the factors for optimum design method.

In many studies the sealing performance of the pipe flange connections has been investigated and evaluated by analyzing the contact stress at the interfaces, a variation of the axial bolt force and creep behavior of gaskets [1]-[10]. These studies evaluated the sealing performance of the pipe flange connections under an elevated temperature and the difficulty of detail evaluation of the connections due to the uncertain behavior of the gasket under thermal conditions was revealed.

In this paper, in order to make a further evaluation and the effect of the nominal diameter of pipe flange connection, thermal stress of bolted flange connections subjected to repeated thermal conduction and internal pressure is analyzed using Finite Element Method (FEM) calculations. In the experiments, the amount of helium gas leakage in the connection was measured.

NOMENCLATURE

P: internal pressure [MPa]
F_f: bolt preload [N]
F_c: force eliminated from the contact surface near a bolt [N]
F_t: increment in axial bolt force per bolt [N]
F: axial bolt force [N]
W: axial tensile force due to internal pressure ($=\pi b^2 P/4$) [MPa]
σ_{zm}: initial target contact gasket stress [MPa]
σ_z: contact gasket stress [MPa]
r: distance from pipe center [mm]
a_3: the gasket inner radius [mm]
b_3: the gasket outer radius [mm]
b:the pipe inner radius [mm]
L_S: fundamental leakage rate [Pa・m^3/s]
L: the leak rate [Pa・m^3/s]
N: number of bolts
t: time for leakage test [s]
V: Volume of internal jacket and pipe for test setup [mm^3]
W': total axial tensile force due to internal pressure ($=\pi a_3^{\ 2} P$) [MPa]
ρ : Density of Helium gas [kg/m^3]

FEM CALCULATION

Figure 1 shows a pipe flange connection with a gasket fastened by N bolts and nuts with a bolt preload F_f. When the assembly is subjected to an internal pressure P, an increment in axial bolt force F_t occurs in the bolts and the contact force F_c (per bolt) is eliminated, that is, the total axial force W'/N (per bolt) due to the internal pressure equals the sum of F_t and F_c. To find out the values of F_t and F_c is statically indeterminate problem. The actual gasket stress must be predicted exactly when the internal pressure is applied to the connection. Furthermore, the pipe flange connection is subjected to repeated thermal changes.

The gasket stress distributions were calculated using FEM calculations. Figure 2 shows an example of mesh divisions used in FEM calculations. Taking into account the symmetry of the connection, one-eighth of the connections were analyzed. The numbers of the elements and the nodes employed were 63000 (2 inch Class 600), 123912 (20 inch Class 300) and 69667 (2 inch), 137389 (20 inch), respectively. The name of the FEM code is ANSYS.

Figure 3 shows the calculation sequence (Steps) for pipe flange connection. Step.1: All bolts are tightened until bolt preload F_f. Step.2: Internal pressure P applied to inner surface of pipe flange and axial tensile force W due to internal pressure P applied at the end of pipe. Step.3: Pipe flange connection is heated up to 300 degree Celsius. Step.4: Internal pressure is relieved of inner surface of pipe flange connection. Step.5: The temperature of pipe flange connection is reduced to room temperature (20 degree Celsius). Step.6: Internal pressure is applied to inner surface of pipe flange connection again. Step.7: pipe flange connection is heated up to 300 degree Celsius again. Three cases are analyzed ((i) Pipe flange and bolts are made of stainless steel. (ii)Pipe flange is made of stainless steel. Bolts are made of carbon steel. (iii) Pipe flange is made of carbon steel. Bolts are made of stainless steel.) Table 1 shows the material property.

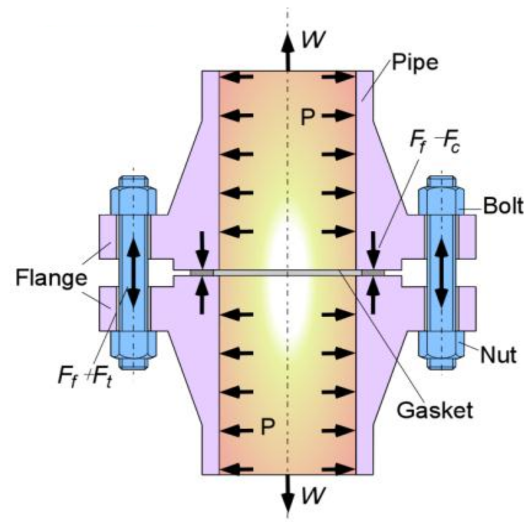


Fig. 1 A pipe flange connection with spiral wound gasket subjected to internal pressure under elevated temperature

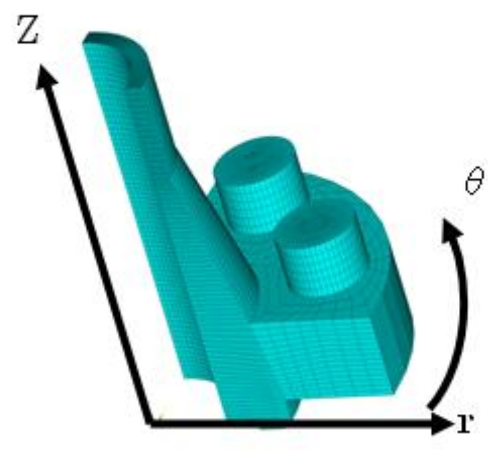


Fig. 2 An example of mesh divisions using FEM calculations

Table.1 Material thermal property at room temperature

At 25 degree Celsius	Carbon steel	Stainless steel
Thermal expansion coefficient	1.09×10^{-5} 1/K	1.64×10^{-5} 1/K
Young's modulus	201.8GPa	195.0GPa
Poisson's ratio	0.31	0.30

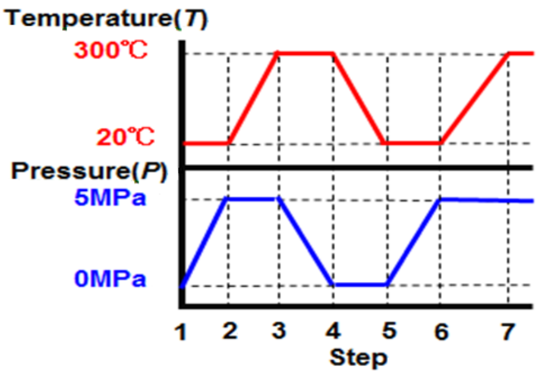


Fig. 3 Calculation and test sequence for pipe flange connection under repeated thermal condition and internal pressure

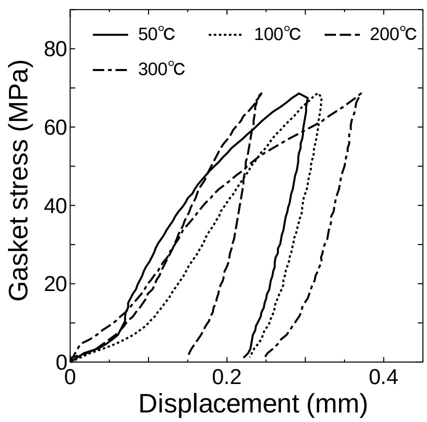


Fig. 4 Stress-displacement curves of spiral wound gasket under elevated temperature

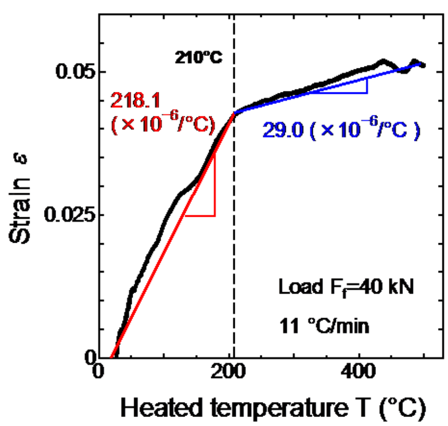


Fig. 5 Thermal expansion of the spiral wound gasket as a function of temperature

Figure 4 shows the stress-displacement curves of the spiral wound gasket under the elevated temperature used in the FEM calculations. Figure 5 shows the thermal expansion coefficient of spiral wound gasket[18].

LEAKAGE TEST

Experiments were carried out for measuring the amount of gas leakage in an actual pipe flange connection subjected to cyclic internal pressure P and thermal conditions. Figure 6 shows the experimental setup. An internal fluid was heated and internal pressure was applied to the connection using an electric heater and helium gas. Then, the temperature distribution of the connection was measured with thermocouples attached to pipe flange. The magnitude of the applied internal pressure was measured by a pressure transducer. Two strain gauges were attached to the shank of the hollows and were calibrated prior to the measurements. The bolt preload F_f was controlled by the attached strain gauges. The outputs were recorded by a recorder and through dynamic amplifiers. The gas leakage was measured from the variation of the pressure. The bolt preload is chosen as the initial average gasket stress 100MPa.

Figure 7 shows the dimensions of pipe flange and a spiral wound gasket (filler: graphite) used in the leakage test. The nominal diameters of pipe flanges used are 2 inch. The pipe flange and the bolts are made of stainless steel SUS304 (Japan Industrial Standard: JIS). The inner and outer rings are made of stainless steel SUS304 (JIS). The nominal diameter of bolts used is M20 (JIS) and the size of the gasket is ASME/ANSI Class 600, 300 [11].

LEAKAGE TEST RESULT

Figure 8 shows the results of the leakage test for the pipe flange connection under repeated thermal condition and internal pressure. The ordinate shows measured leakage rate (Pa・m^3/s). The abscissa shows the step number. It is found that the leakage rate was increased as the temperature was increased. The leakage rate in Step 3 was minimum value. The leakage rate in Step 3 was smaller than that in Step 7. However, there was no significant difference between Step 3 and 7. The red dotted line shows the predicted result using the gasket stress distributions obtained from FEM calculation. A fairly good agreement is seen between the experimental result and the predicted result. Using the same FEM models, the effects of the repeated temperature changes and internal pressure on the pipe flange connections are discussed.

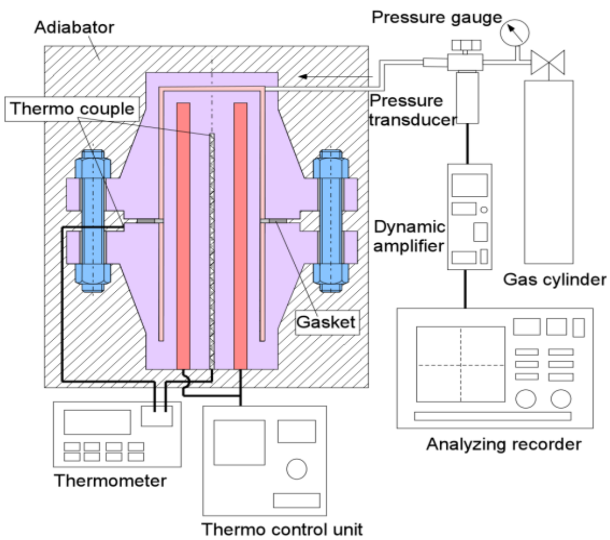


Fig. 6 Schematic of experimental setup for helium gas leakage test

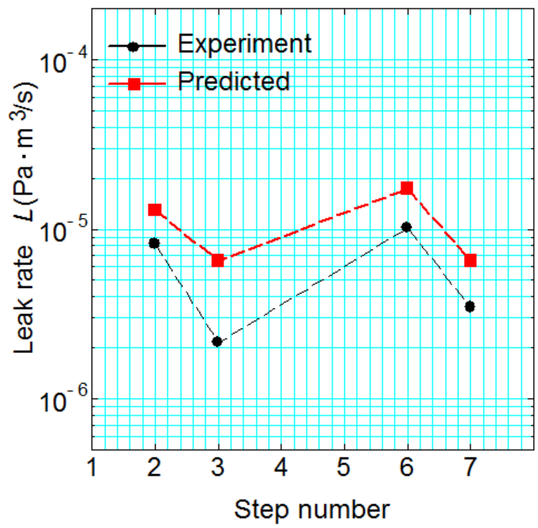
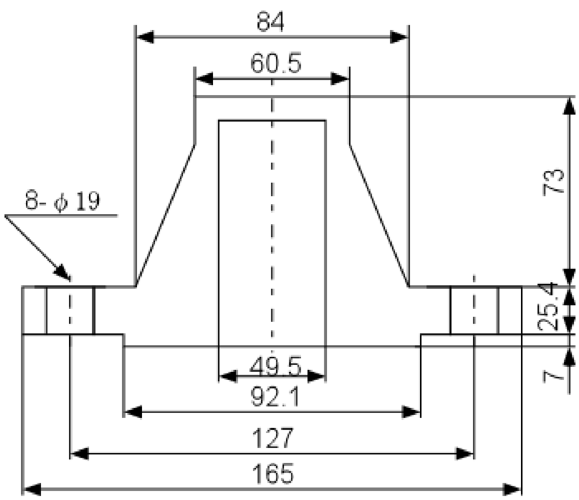
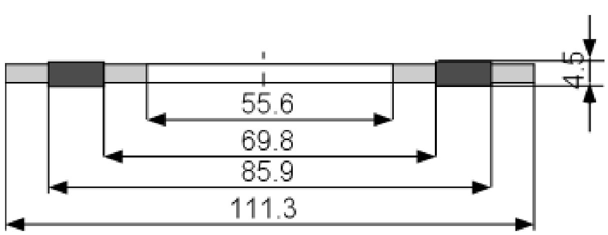


Fig. 8 Comparison of measured leak rates and predicted leak rates from FEM analysis



(a) Pipe flange (2 inch Class600)

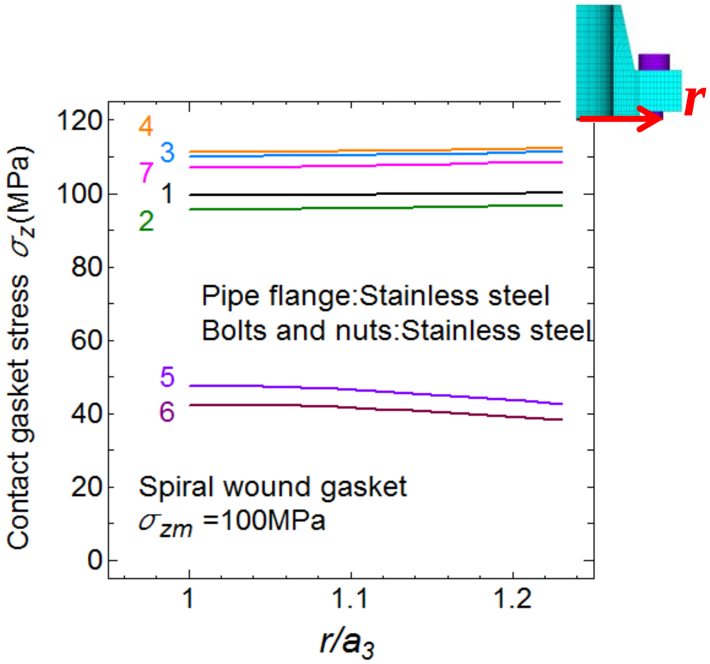


(b) Spiral wound gasket (Filler: graphite)

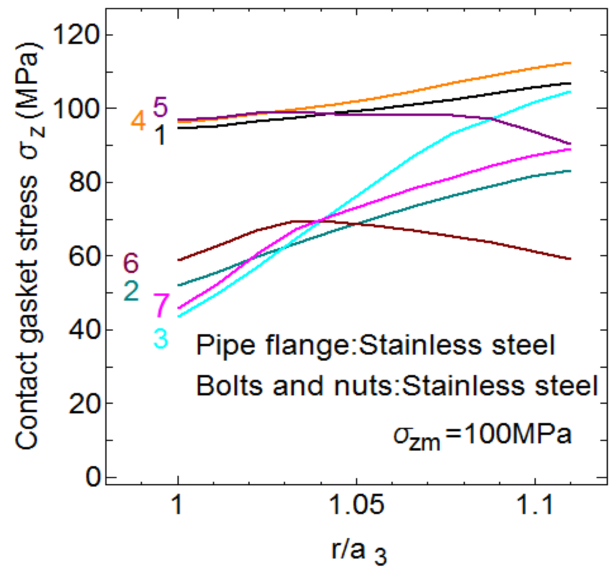
Fig. 7 Dimension of pipe flange and spiral wound gasket used in leakage test (Unit: mm)

FEM CALCULATION RESULTS

Figure 9 shows the contact gasket stress distributions in radial direction of pipe flange connection under repeated internal pressure and thermal condition in each step. The ordinate shows contact gasket stress. The abscissa shows the ratio of the distance from the pipe center and the gasket inner radius. Bolt preloads are 24.6kN (2 inch) and 188.8kN (20 inch) , that is, average contact gasket stress is 100MPa at initial clamping state. Applied internal pressure is 5MPa. Figure 9(a) shows the contact gasket stress distributions in radial direction in the case of smaller nominal diameter pipe flange (2 inch). In Step 1(initial clamping state), the contact gasket stress is uniformly almost 100MPa. In Step 2, when the pipe flange connection is subjected to an internal pressure *P*, the contact gasket stress is reduced. In Step 3, when the pipe flange connection is heated, the contact gasket stress is increased due to the large difference of thermal expansion coefficients between the spiral wound gasket and pipe flanges. In Step 4, the contact gasket stress is increased due to the internal pressure set to 0. In Step 5, when the temperature of pipe flange connection is reduced to room temperature, the contact gasket stress is decreased eminently. The Young's modulus of the gasket in section is much larger than that in loading section. In Step 6, the contact gasket stress is decreased more due to internal pressure. In Step 7, when the pipe flange connection is heated, the contact gasket stress is increased. However, the contact gasket stress in Step 7 is smaller than that in Step 4. Figure 9(b) shows the contact gasket stress distributions in radial direction in the case of larger nominal diameter pipe flange (20 inch). After the internal pressure is applied to the connection, the thermal load is applied (Step 3). The slope of gasket stress distribution in step 3 is increased due to the thermal load. It is consider the reason why the pipe flange deformation (flange rotation) is larger as the flange temperature

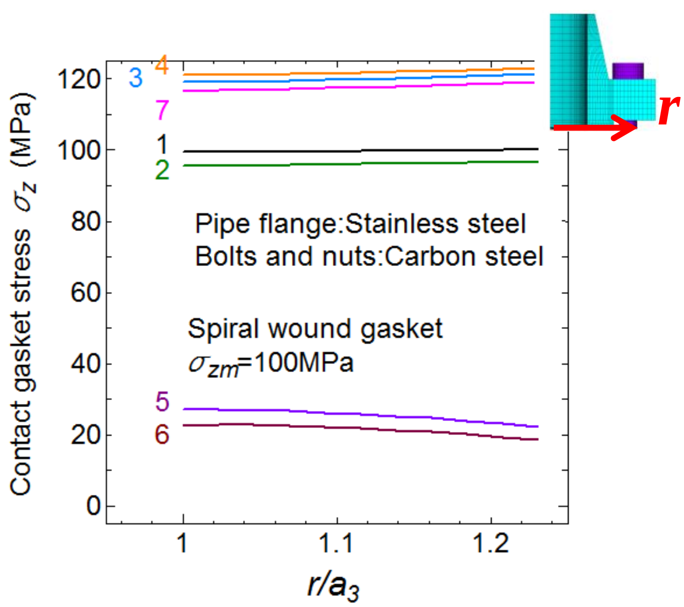


(a) smaller nominal diameter pipe flange (2 inch)

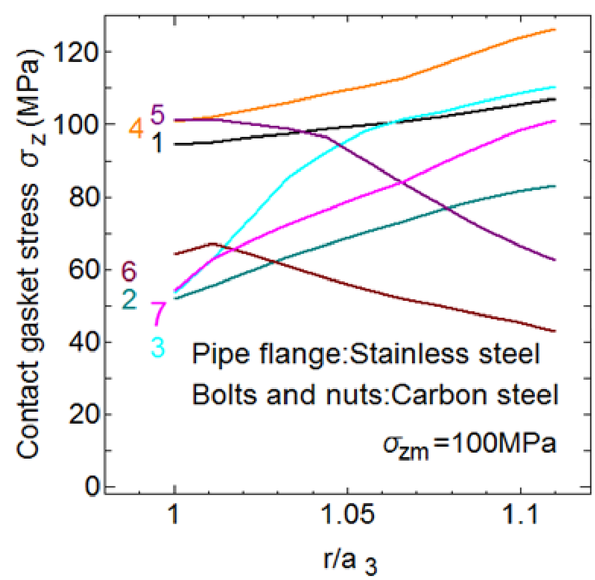


(b) larger nominal diameter pipe flange (20 inch)

Fig. 9 Contact gasket stress distributions in pipe flange connection (in case of pipe flange and bolt made of stainless steel)

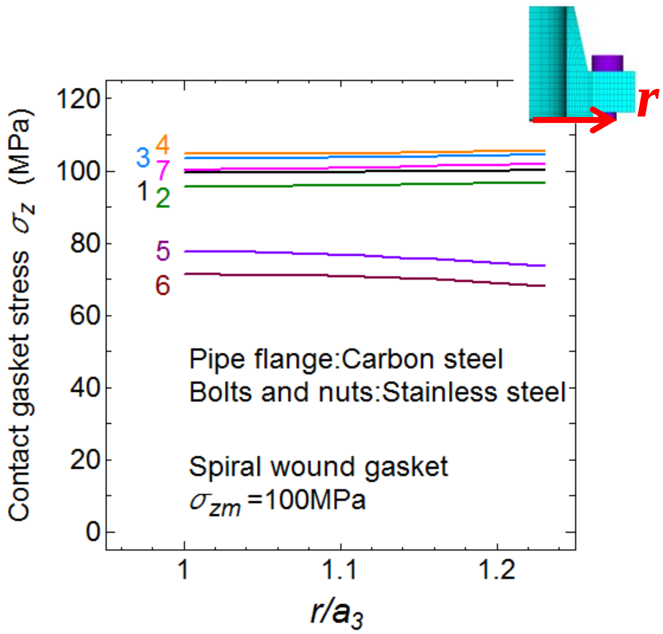


(a) smaller nominal diameter pipe flange (2 inch)

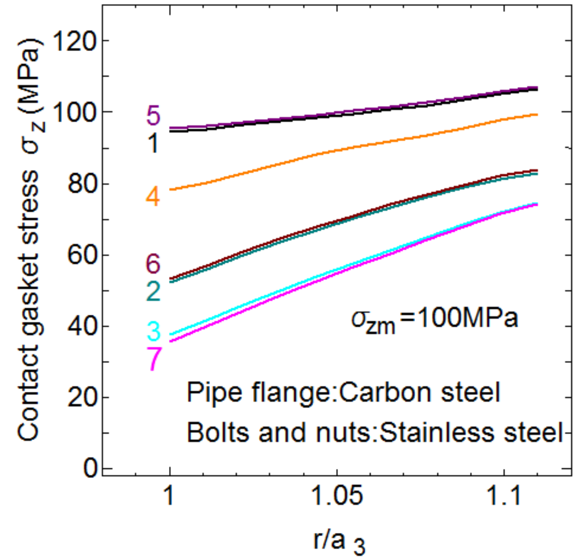


(b) larger nominal diameter pipe flange (20 inch)

Fig. 10 Contact gasket stress distributions in pipe flange connection (in case of pipe flange made of stainless steel and bolts made of carbon steel)



(a) smaller nominal diameter pipe flange (2 inch)



(b) larger nominal diameter pipe flange (20 inch)

Fig. 11 Contact gasket stress distributions in pipe flange connection
(in case of pipe flange made of carbon steel and bolts made of stainless steel)

increases. The flange rotation makes the gasket stress distribution more distributed. Thus, it is consider that the thermal stress, mainly, occurred at outside area. The contact gasket stress at the outside area is changed dramatically.

Figure 10 shows the contact gasket stress distributions in pipe flange connection (in case of using a pipe flange connection made of stainless steel and bolts made of carbon steel). Figure 10(a) shows in the case of the smaller nominal diameter pipe flange (2 inch). Figure 10(b) shows in case of the larger nominal diameter pipe flange (20 inch). In this case, the thermal expansion coefficient of bolt and nut is smaller than that of the pipe flange. It is found that the contact gasket stress changes in Fig. 10(a) show the same tendency as that in Fig. 9(a). However, the maximum value of contact gasket stress in Fig. 10(a) is larger than that in Fig. 9(a) and the minimum value of contact gasket stress in Fig. 10(a) is smaller than that in Fig. 9(a). It is considered that the thermal expansion coefficient of carbon steel is lower than that of stainless steel. Moreover, the larger the contact gasket stress is, the larger the Young's modulus of gasket in unloading section is. On other hand, it is seen the same tendency between the Fig. 10(b) and Fig. 9(b). The maximum value and minimum value of contact gasket in Fig. 10(b) is larger and smaller compared with that in Fig. 9(b). This reason is the same in the case of smaller nominal diameter pipe flange. In the larger nominal diameter, it is also seen that the effective area of thermal stress due to the thermal load is increased.

Figure 11 shows the contact gasket stress distributions in pipe flange connection (in case of using pipe flange made of carbon steel and bolt made of stainless steel). Figure 11(a) in the case of the smaller nominal diameter pipe flange (2 inch). Figure 11(b) shows in case of the larger nominal diameter pipe flange (20 inch). In Fig. 11(a), the same tendency is seen as Fig. 10(a). It is considered that the thermal expansion coefficient of spiral wound gasket is much larger than that of pipe flange and bolts. It is found that it is quite important to decide the materials of gasket, pipe flange and gasket when those are used under repeated thermal condition and internal pressure. In Fig. 11(b), the minimum contact gasket stress is at Step 7. After the thermal load, the contact gasket stress is decreased. From this result, the flange deformation due to the thermal load is larger than the thermal expansion of spiral wound gasket. The leakage from the larger nominal pipe flange connection is to be maximum value. In the design of pipe flange connection, the flange deformation due to the thermal load has to be into account even if the same material in pipe flange is used taken.

CONCLUSIONS

This paper was addressed the stress analysis and sealing performance of pipe flange connections under cyclic thermal condition and elevated internal pressure. The contact stress distribution of a pipe flange connections was obtained by elasto-plastic finite element analysis taking into account the hysteresis and non-linearity of the gaskets. The experiments were also conducted to measure the amount of gas leakage. The following results were obtained.

1) Pipe flange connection was analyzed using FEM calculation. It is found that the contact gasket stress in the case of resume

operations is the smallest in the case of smaller nominal diameter.

2) In the case of larger nominal diameter pipe flange, the thermal stress, mainly, occurs at the outside area of gasket due to the flange rotation.

3) The same materials are used between the smaller nominal diameter and larger nominal diameter. However, the different tendencies on the gasket stress distributions are seen. In the design the larger nominal diameter, the flange deformation due to the thermal load has to be taken into account.

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