Thermal Analysis for the Sealing Test on a RPV Model

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INTRODUCTION

Up to now there are no sealing design criteria for Class I nuclear pressure vessels. For this reason we have undertaken some tests and analyses on this aspect.At present there are various simplified methods for sealing analysis. A typical one is considering the relative rotation of the flange taken as a stiff ring and assuming an arbitrary support point between the upper and the lower flanges. This method obviously depends to a certain extent on experience. The essence of analytical methodology study lies in acquiring the safety margin of simplified analysis through detailed calculation, which is based on considering the elasto-plastic contact of the flange mating surface and the bolt loading change with internal pressure and temperature (Qu 1987). It should be noted that owing to the nonlinearity of sealing problem the thermal effect and the internal pressure effect can not be simply combined without any justification.

The tough points in thermal analysis lie in that it is rather difficult to precisely obtain the thermophysical parameters, to determine the heat resistence at the contact boundaries and the bolt root position and to calculate the bolt temperature lag, due to the complexity of the heat transfer across the gap between the bolt and the bolt hole. So it is of significance to perform thermal test as well as thermal analysis. Information about high temperature sealing test is very limited,probably because of the cost of such tests. Spaas (1977) reported a thermal test on a 1:4 pressure vessel model of a 50 MWe BWR, mainly to observe the effect of bolt lag under heatup condition. In that case the temperature effect and the pressure effect were observed independently and then linearly combined. So the test was not a coordinated one.

The authors of this paper presented at ICPVT-6 a report (Qu et al, 1988) on the thermal sealing test of a 1m ID PWR pressure vessel model (1:4). The test was performed under three conditions: heatup, temperature maintaining and cool-down. This paper provides some results on the relevant thermal analysis on the basis of the above test.

THERMAL ANALYSIS

The dimensions of the test vessel and the relevant material parameters are given in the paper mentioned above. 3D thermal analysis is performed on a slice model (6°) which is cut in accordance with the symmetric condition on the bolt position of the vessel (see Fig. 1). Hexahedron elements, each with 12 nodes, are adopted to form the FEM mesh, with 80% nodes and 65% elements gathering on the bolt and bolt hole area. The heat transfer across the area between the bolt and the bolt hole is considered to be dual-directional, the convection following the Newton's cooling law and the radiation being simplified in engineering way as follows:

$$h_o = h_c + h_r \quad ; \quad q_w = h_o \ (T_w - T_f)$$

here q_w is the density of heat flow; h_o the general heat transfer coefficient; t_w the metal temperature and t_f the media temperature; subscripts c and r represent convective and radiative conditions respectively. h_c has only minor changes with the variation of temperature difference and in high temperature and large temperature difference condition h_r takes the major role of h_o, at 300^oK 50% of the h_o. Thus taking $h_r = h_c$, $h_o = 2h_c$.

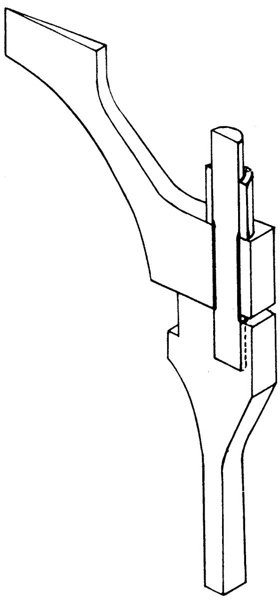


Fig. 1 Analytical model

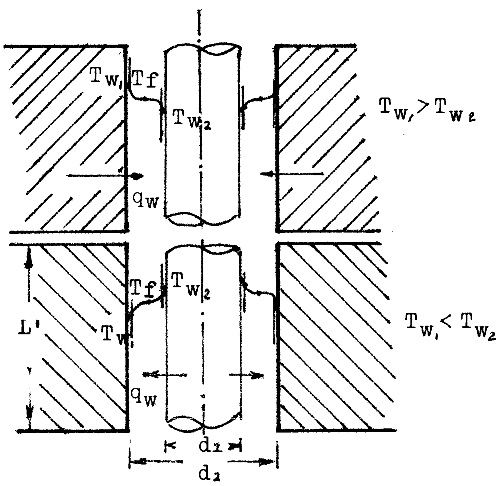


Fig. 2 Heat transfer between the bolt and the bolt hole

The heat transfer of the bolt hole is treated as shown in Fig. 2. The subscripts 1 and 2 represent the flange and the bolt respectively. For the narrow passageway between the concentric cylinders d_1 and d_2 and the heat transfer per unit length W/m, we have the following:

$$q_1 = \frac{2\pi K_e \ (T_{w_1} - T_{w_2})}{\ln \ (d_2/d_1 \ )} = h_c L' \quad ; \quad K_e \ / \ K = f(G_r, P_r)$$

Here K_e is the equivalent heat transfer coefficient, G_r and P_r the Grashof number and the Prandtl number respectively:

$$G_r = gL^3 \beta \Delta T \ / \nu^2 = 16.105 \quad ; \quad P_r = \nu / a = C_p \mu / K = 0.684$$

The characteristic length is taken as $L = (d_1 - d_2)/2$ and $K_e/K = 0.11(G_r, P_r)^{0.2}$. So $q = 2.59$ W/m, and $h = 1.57\times10^{-3}$ W/cm.

Based on the test measurements and the heat transfer calculation the average temperatures $\Delta \overline{T}_{ji}$ on different sections of the vessel v_j at different time intervals $\Delta \tau_i$ can be obtained and then the convective coefficient $\alpha_i(T)$ on the isothermal surface can be formulated through numerical integral as follows:

$$\alpha_i(T) = (\rho_1 V_1 C_{p1} \Delta T_{1i} + \sum_{j=1}^{n} \rho_2 C_{p2} v_j \Delta T_{ji}) / \Delta \tau_i F (T_{wi} - T_{fi})$$

In the formula, F is the heat transfer surface (area); T_W the temperature of the water; T_f the surface temperature of the vessel; the subscripts 1 and 2 respectively indicate the media and the vessel. At different temperatures of 80°C, 240°C and 300°C, the values of α are 0.201, 0.241 and 0.176 (W/cm^2 °C).

The engaging surfaces at the root of the stud are treated as one body or partially connected in transient calculation for comparison, since the one-body assumption would give non-conservative effects of the bolt temperature lag in the cooldown process. The convective and equivalent radiative heat transfer conditions, are considered for the gap between the bolt and the bolt hole and the result is compared with that of no heat trasfer condition. Two boundary conditions, i.e., given temperatures or perfect insulation on the outer wall of the vessel are taken for comparison. Comparing with some finite difference methods the above treatments show some improvements in thermal analysis for bolt and flange, and, furthermore, make preparations for the further contact analysis and the incremental analysis of the bolt loading.

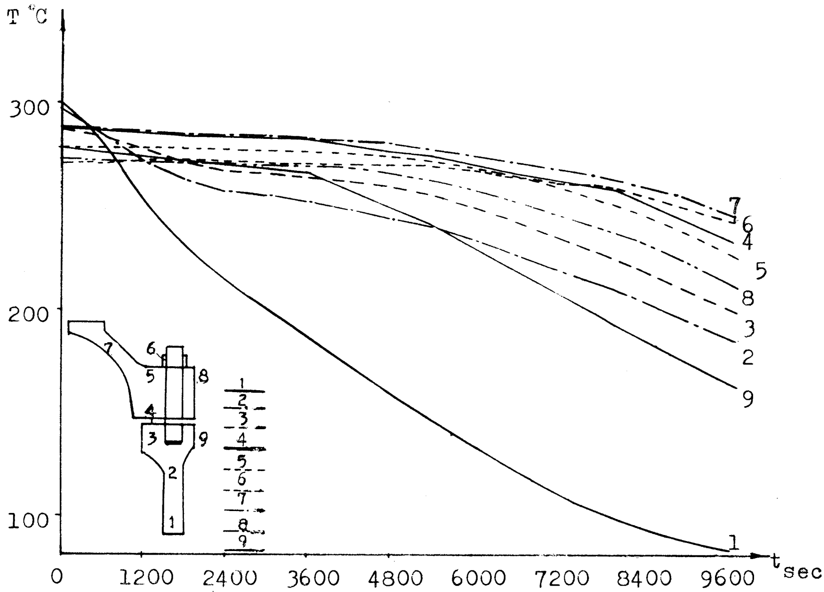


Fig. 3 Axial distribution of transient temperature field

MAIN RESULTS

Fig. 3 is the calculated transient temperature field in axial direction for the bolt and the vessel in the cool-down process. The maximum temperature diffence is as high as 140°C, which is a noticeable value because at this point flange rotation and deformation reach the highest values, the sealing margin reaches its ebb and leakages may usually occur due to insufficient spring-back of sealing components.

The local temperature field of the bolt and bolt hole is shown in Fig. 4-a, b and c, which reflect the complexity of the bolt temperature lag. The lag appears both in the circumferential and the axial direction, and exibits

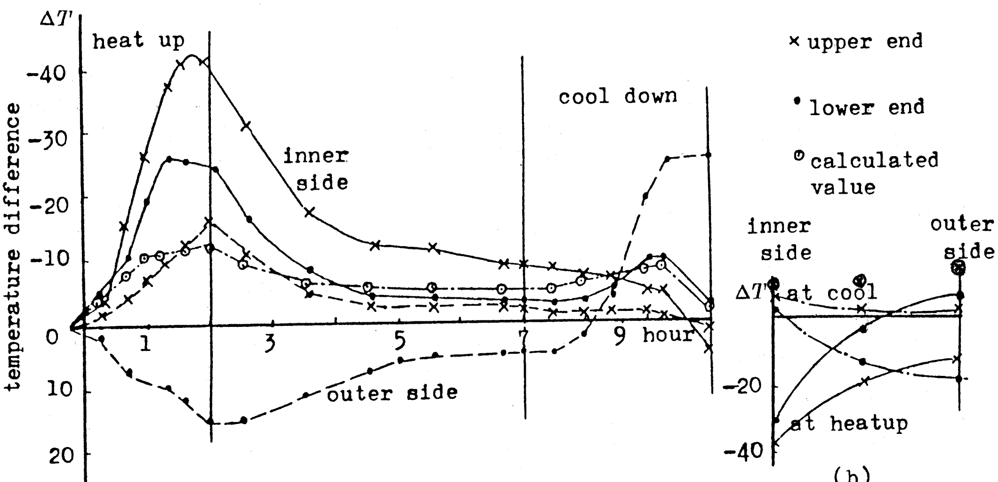
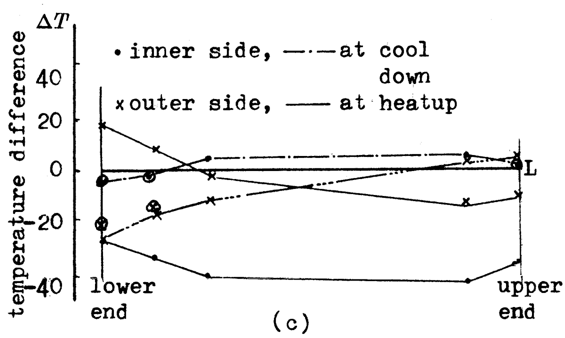


Fig. 4 (a)

(a) Transient distribution of bolt lag
(b) Circumferential distribution of bolt lag
(c) Axial distribution of bolt lag



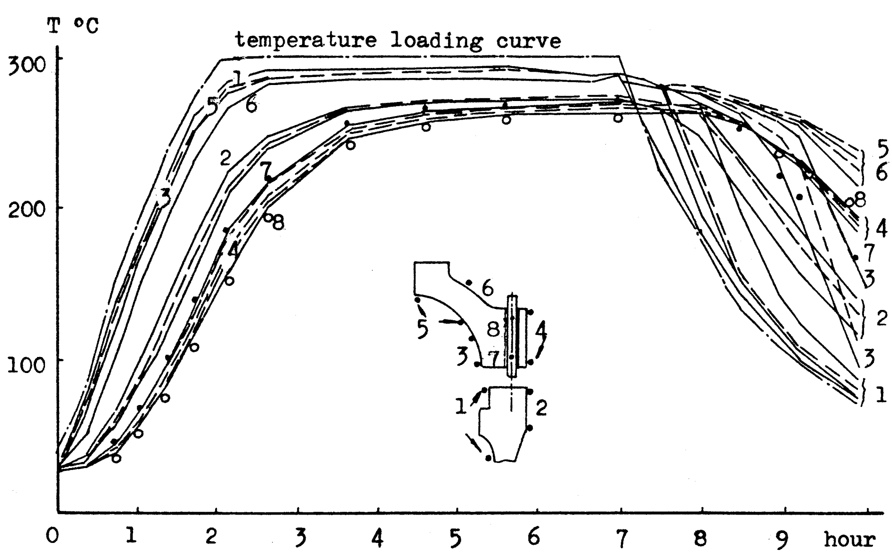


Fig. 5 Global temperature distribution

different behavior in the heatup and the cool-down process. In the heatup the lag on the inner side of the bolt is higher than that on the outer (as high as 37°C); while during the cool-down the situation is just the opposite. For the axial distribution, in the beginning the curve is linear and then, at the last stage, the value at the middle part is somewhat higher than that at the upper end. Generally the calculated values lie in between those at the inner and outer sides of the bolt, reflecting the average temperature difference (see Fig. 5).

The comparison for the two kinds of boundary conditions, i.e., given temperatures or insulation on the outer wall of the vessel, shows that the difference is apparent and it affects the steady temperature field of the bolt at last stage. There is no doubt that boundary conditions affect considerably the results of the calculation, especially that for the steady state. In Spaas's paper (1977), only the insulation condition was taken on the outer wall of the vessel and the effect of natural convection of the air was neglected. This is acceptable only for the heatup situation.

Owing to the existence of the bolt hole there are nonaxisymmetrically distributed temperatures in the circumferential direction and with increasing transient time the nonaxisymmetry increses. The maximum temperature difference is about 10°C.

When convective condition in the gap between the bolt and the bolt hole is considered, circumferential distribution of the bolt temperature appears. Comparing with the results of the insulation boundary, the temperature at the upper and middle parts of the bolt inner side decreases by 7°C to 16°C, while those at the corresponding positions of the bolt hole, by 2°C to 3°C and the temperature difference drops about 10°C. The test results indicate that there is an obvious circumferential temperature distribution at the bolt hole. For the bolt, however, the temperatures are basically in axial distribution and the circumferential one is not so distinct. The calculated temperature decreasing rate is lower than the test value, this shows that only considering the bottom part of the bolt to be in contact with the flange will give over-conservative results.

Many variation rates obtained from the calculation agree well with the test results. For example, the calculated maximum temperature of the bolt appears at 30 mintes during the cooldown after keeping temperature for 5 hours and the temperature rise is within 1°C, while in the test the maximum temperature occurs at 18 minutes and the temperature rise within 0.5°C; the calculated shows that after 115 minutes of cool-down the temperatures at the upper end of the bolt inner side begin to be higher than those on the bolt hole and the maximum temperature difference is 10°C, while in the test the corresponding values are 120 minutes and 7°C respectively.

CONCLUSIONS

The transient bolt temperature lag distributions given by this paper provide a foundation for the simplified treatment in analysis. Both cases of taking the engaging surfaces at the root of the stud as partially connected and as one body are analysed, the former gives over conservative results.

The axial temperature difference in the cool-down process can be taken as a sigh which indicates the minimal sealing margin. Thus the axial temperature difference is a characteristic value.

There is a distinct difference between the insulated and given temperature boundary of the outer wall of the vessel. The effect of insulation layer is not negligible, especially for bolt temperature calculation.

It is still rather difficult to make the calculation agree satisfactorily in all aspectes with the test, because it is too difficult to obtain precise thermophysical parameters.The approach presented here only shows the simlarity in the general tendencies. And it is still necessary to make improvements.

APPENDIX

Fig. 6 and Table give the dimensions of the test model and the relevant material parameters respectively.

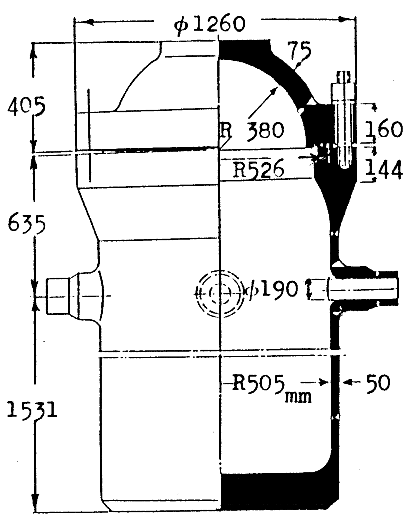


Fig. 6 . Test vessel

	Temp.	Vessel 18MnMoNb	Bolt 18CrNiWA
E	50°	20.80	20.01
	300°	19.23	16.29
ν	50°	0.318	0.26
	300°	0.295	0.28
Cp	50°	45.98	48.49
	300°	61.86	52.25
K	50°	45.98	23.83
	300°	35.11	25.92
α	50°	12.20	11.7
	300°	13.25	13.6

Here E: Young's modulus 10^4 MN/m^2
ν: Poisson's ratio
Cp: Specific heat 10^{-2} J/Kg °C
K: Coeff. of thermal conductance 10^{-3} J/mm sec °C
α: Coeff. of thermal expansion 10^{-6} mm/mm °C

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