THERMAL STRESS ANALYSIS OF THE PARTING PLANE FLANGE JOINT OF AN INDUSTRIAL STEAM TURBINE

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A typical 14 MW steam turbine casing is analysed using a range of flange parameters and heating conditions with three-dimensional and two-dimensional models, in order to find the contact pressures and thermal stresses during transient thermal operations. Thermal stresses are also calculated with simple analytical formulae which are available in literature. Two-dimensional and analytical results are compared with three-dimensional results, and some correction factors are suggested. These correction factors can be used in day to day design work to obtain quick and simple solutions. Contact behaviour of the parting plane flange under transient thermal loads is studied using 'gap' elements available in the application software 'NISA'.

1 INTRODUCTION

For easier assembly and maintenance, the outer casings of industrial steam turbines are usually split at a horizontal parting plane and joined by flanged and bolted joints. The flanged joint is subjected to thermal stresses during transient operations i.e., start-up, shut-down, and load changes and requires the joint width and flange height to be a minimum to reduce thermal stresses. On the other hand, for effective sealing, the flange dimensions are usually substantial compared to the casing thickness.

1.1 Notation

- E Young's Modulus
- α Coefficient of linear thermal expansion
- μ Poisson's ratio
- T Temperature
- σ Stress
- ϕ_f Non-dimensional factor

2 REVIEW OF EARLIER WORK

Although stress analysis has become a requirement in pressure vessel design codes, from the point of view of stress analysis and contact pressure, sealing design is not given sufficient attention. There are no design criteria for sealing design, e.g., thermal transients, not even in the ASME boiler and pressure vessel code (**1**)†. Since the 1960s, a flange has generally been considered as a rigid beam in computational methods. However, the real situation is actually much more complicated, because the bolt loading may change with internal pressure and temperature; thus the problem is actually a thermo-elastic-contact problem.

Simplified equations for day to day design work are given by Troupel (**2**) to calculate the bolt force required to maintain the minimum contact pressure to avoid steam leakage during steady state. These calculations do not give the distribution of contact pressure on the parting plane and do not consider the effect of thermal transients.

Photoelastic tests have been made on two- and three-dimensional models of steam turbine casing flanges for various combinations of flange parameters e.g., depth, width, bolt position, and bolt force to steam force ratio (**3**) (**4**). Design curves have been produced which show the relation between these flange parameters. Comparisons have been made between the experimentally determined stress distributions and distributions calculated by various methods. However, the effect of thermal transients is not considered in this analysis.

Manson (**5**) suggested a simple formula for calculating thermal stress in a beam with a non-uniform temperature gradient which can be modified as

$$\sigma_z = \frac{E\alpha}{(1-\mu)}(T_{av} - T) \tag{1}$$

where T is the temperature at any point within the material and T_{av} is the weighted average of temperatures.

A non-dimensional factor (**6**) has been given in VGB R105M to relate thermal stresses directly to fluid temperature (T_f) and initial metal temperature (T_0).

$$\sigma_z = \frac{E\alpha}{(1-\mu)}(T_f - T_0)\theta_f \tag{2}$$

where θ_f is a non-dimensional factor derived from experimental and theoretical investigations by assuming the flange to be a beam.

However, in all the above cases, the contact stress problem and thermal stress problems are decoupled, whereas the actual problem is a thermo-elastic-contact problem.

Jaidi and others (**7**) studied the sealing behaviour of a nuclear pressure vessel under transient thermal loads. The measured values of temperatures, stresses, and dis-

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† *References are given in the Appendix*

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tortions are compared to axi-symmetric finite element results.

On the whole, limited information is available on high temperature sealing in pressure vessels, particularly vessels with non-circular flanges. Therefore, the present work is aimed at establishing the validity of simple methods to calculate thermal and contact stresses during transient operations with sufficient accuracy.

3 PROBLEM DESCRIPTION

In the present analysis a typical 14 MW steam turbine casing is investigated. The upper and lower parts of the casing are joined by parting plane flanges and a sealing force is applied by tightening the parting plane bolts. These bolts are to be re-tightened frequently because of the relaxation in the bolt force due to creep.

The phenomena of heat transfer between steam and the casing metal, which determines the temperature distribution within the casing, is mainly governed by the convective heat transfer coefficients. The convective heat transfer coefficients for the present problem are calculated based on the data available (**8**).

The calculated heat transfer coefficients are based on certain assumptions like hollow cylinder, uniform steam velocity etc., which may not give exact values when applied to turbine casings. Hence the value of heat transfer coefficient is also taken as a parameter for study of flanged joint behaviour. The other parameters taken are flange width and flange height. A total of twelve load cases are studied by varying heat transfer coefficients three times and flange parameters twice as shown in Tables 1(a) and 1(b).

4 FINITE ELEMENT ANALYSIS

For the present investigations, a finite element application software NISA is used, which has graphic facilities in addition to heat transfer and stress analysis capabilities including 'gap' analysis. The casing is descritized using eight-noded brick elements. By considering the symmetry conditions inherent in the structure, only a quarter of the casing is taken for analysis with appropriate boundary conditions. A typical three-dimensional model consists of 849 elements and 1414 nodes (shown in Fig. 1).

The behaviour of the parting plane joint is such that only compressive stress in a direction normal to the contact surface can exist on the parting plane. If the bolt force is not sufficient to create compressive stress, the joint opens rather than tensile stress being developed. This is a kind of geometric non-linearity and can be solved by using the 'gap' element available in 'NISA'. The 'gap' element is a line element with two nodes: the element output gives the force transmitted through the element or the amount of opening. The contact pressure (σ_y) on each solid element surface lying on the parting plane can be calculated by adding the contributions of compressive force transmitted by each 'gap' element corresponding to the solid element, and dividing by the surface area.

For all the above load cases, the following steps are used:

(a) determination of transient temperature distributions;
(b) three-dimensional thermal stress analysis;
(c) contact pressure distribution analysis at joint interface using 'gap' elements.

To check the accuracy of the results, one load case is analysed with another mesh and the results are compared. The second mesh consists of 233 elements (20 noded brick) and 1580 nodes. The values of temperatures and stresses obtained are within 5 percent deviation, which can be treated as a good convergence considering the size of the problem and the complicated boundary conditions.

5 RESULTS AND DISCUSSION

The temperatures at each node for each time step for all the load cases are obtained by the heat transfer analysis using NISA. Typical three-dimensional temperature contours are given in Fig. 2. A typical temperature distribu-

Table 1(a). Flange widths and heights for different load cases

Load case	1	2	3	4	5	6	7	8	9	10	11	12
Width (m)	0.26	0.26	0.26	0.23	0.23	0.23	0.26	0.26	0.26	0.23	0.23	0.23
Height (m)	0.12	0.12	0.12	0.12	0.12	0.12	0.15	0.15	0.15	0.15	0.15	0.15

Table 1(b). HTC, temperature and pressures at different zones for each load case

Load case	Parameter	Zone (as marked in Fig. 1)																		
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1, 4, 7, 10	HTC	0.02	0.03	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.04	0.04	0.04	0.03
	Temp.	150	200	240	260	270	280	280	280	280	280	280	280	280	280	280	260	240	200	160
	Pr.	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.01	0.01	0.01	0.01
2, 5, 8, 11	HTC	0.02	0.03	0.04	0.04	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.04	0.04	0.04	0.03
	Temp.	150	200	240	260	270	280	280	280	280	280	280	280	280	280	280	260	240	200	160
	Pr.	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.01	0.01	0.01	0.01
3, 6, 9, 12	HTC	0.02	0.03	0.04	0.04	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.04	0.04	0.04	0.03
	Temp.	150	200	240	260	270	280	280	280	280	280	280	280	280	280	280	260	240	200	160
	Pr.	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.01	0.01	0.01	0.01

Units: HTC = $W/Cm^2/C$
Steam temp. = Deg. C
Pressure = $N/m^2 \times 10^{-8}$.

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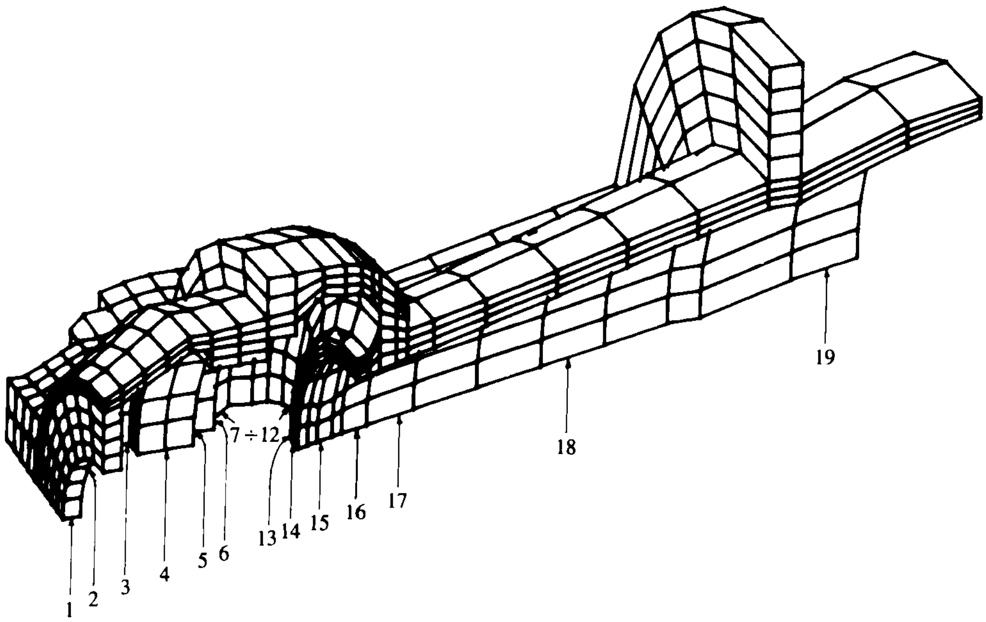

Fig. 1. Three-dimensional finite element model of the steam turbine casing

tion along the parting plane on a transverse cross-section at the steam inlet zone (zone 11 in Fig. 1), at different time steps, obtained using three-dimensional analysis is shown in Fig. 3. The overall stress contours of σ_{eq} obtained from a typical load case of three-dimensional analysis are presented in Fig. 4. It can be seen that the maximum values are at the steam inlet zone on the inner surface near the parting plane flange (zone 11 in Fig. 1). Three-dimensional thermal stress fields within the casing for all load cases are obtained. The values of different stress components and von Mises equivalent stress (σ_{eq}) for different load cases at the maximum stress location in the stream inlet zone are presented in Table 2.

The ratios of the results at location (11) (Fig. 1) obtained by three-dimensional analysis and simple formulae (1) and (2) are given in Fig. 5 and Fig. 6 for various heat transfer coefficients and flange dimensions. The temperature distribution required for equation (1) is obtained by a two-dimensional finite element heat transfer analysis carried out by taking a transverse cross-section at the steam inlet zone as shown in Fig. 7. The boundary conditions are given in Table 1(b). It can be seen from Fig. 5 that the simple formula given by Manson (5) along with two-dimensional finite element heat transfer analysis gives reasonably accurate results for the range of parameters considered in this analysis. But there is a significant difference between the results obtained by the three-dimensional analysis and the VGB formula (2) particularly for low heat transfer coefficients and wide flanges.

In order to compare the results obtained by simple two-dimensional (2D) plane strain analysis with three-dimensional (3D) results, a typical cross-section at the steam inlet zone shown in Fig. 7 is analysed both for

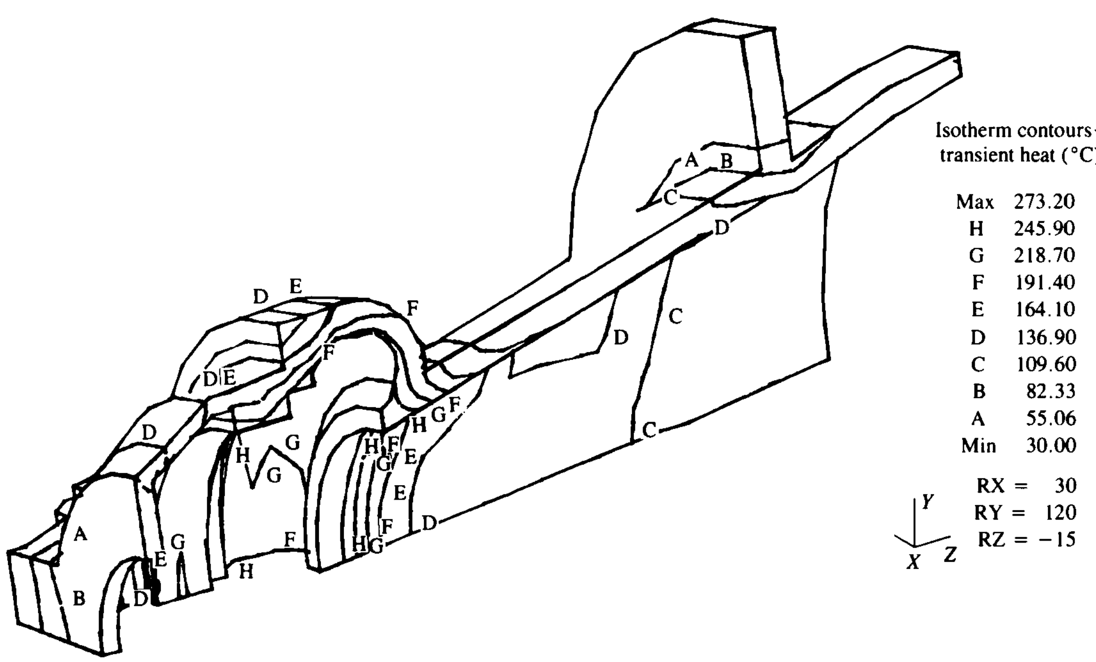

Fig. 2. Typical temperature contours in the casing during thermal transients

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Table 2. Comparison of three-dimensional and two-dimensional stresses

Load case	Three-dimensional analysis results (n/M^2 × 10^{-8})				Two-dimensional analysis results (n/M^2 × 10^{-8})			
	σ_x	σ_y	σ_z	σ_{eq}	σ_x	σ_y	σ_z	σ_{eq}
1	−2.014	−1.461	−2.519	2.627	−0.541	−1.560	−5.242	4.306
2	−2.818	−2.208	−3.221	3.309	−0.699	−1.806	−5.994	4.859
3	−3.718	−3.193	−3.964	3.912	−0.924	−2.205	−6.641	5.241
4	−1.893	−1.480	−2.812	2.822	−0.355	−0.509	−5.083	4.655
5	−2.656	−2.307	−3.426	3.474	−0.543	−0.979	−5.788	5.040
6	−3.545	−3.429	−3.916	4.008	−0.750	−1.526	−6.483	5.388
7	−2.100	−1.595	−2.550	2.683	−0.427	−0.908	−4.920	4.314
8	−2.943	−2.740	−3.265	3.285	−0.669	−1.556	−5.780	4.738
9	−3.444	−3.838	−3.853	3.940	−0.877	−2.156	−6.600	5.209
10	−1.970	−1.636	−2.809	2.882	−0.394	−0.775	−5.052	4.480
11	−2.771	−2.604	−3.494	3.530	−0.574	−1.257	−5.849	4.969
12	−3.572	−3.659	−4.031	4.084	−0.786	−1.854	−6.579	5.342

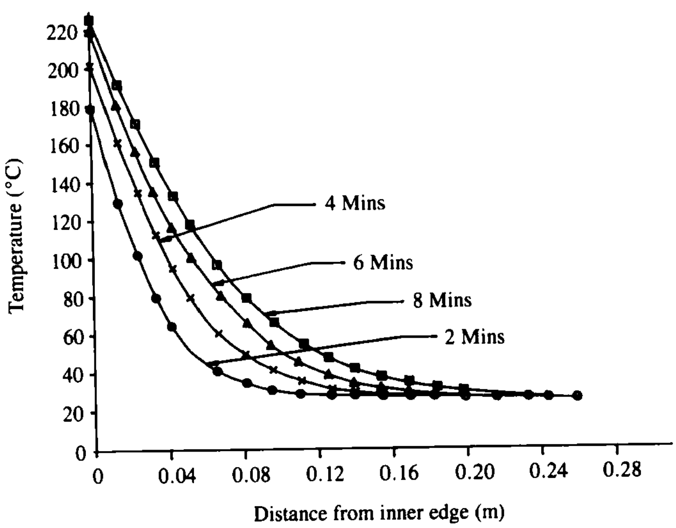

Fig. 3. Typical temperature variation across the parting plane flange during thermal transients

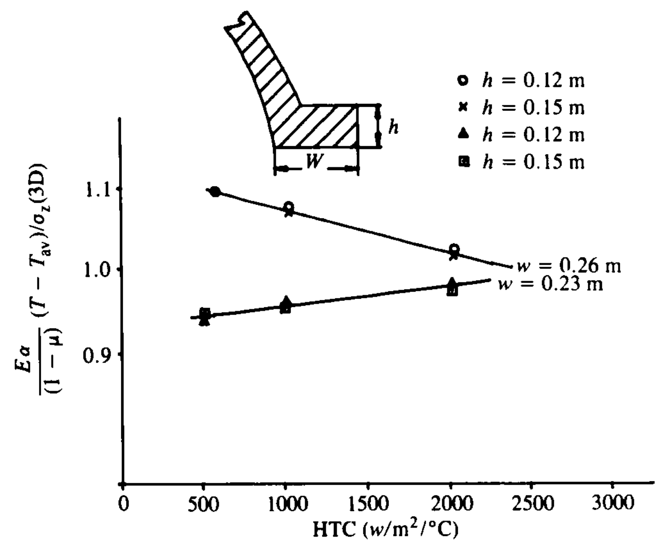

Fig. 5. Ratios of the σ_z stresses obtained from equation (1) and three-dimensional finite element analysis

temperature distribution and stress analysis. The load cases and boundary conditions are shown in Table 1(a) and Table 1(b). The ratios of $(\sigma_z)2D/(\sigma_z)3D$ are plotted as w.r.t heat transfer coefficients in Fig. 8. It can be seen from this figure that the ratio decreases with increasing heat transfer coefficients. During severe heating conditions, temperature change can penetrate only through a short distance. Hence the major bulk of the material which is at a lower temperature does not allow the heated region at the inner surface to expand. Therefore,

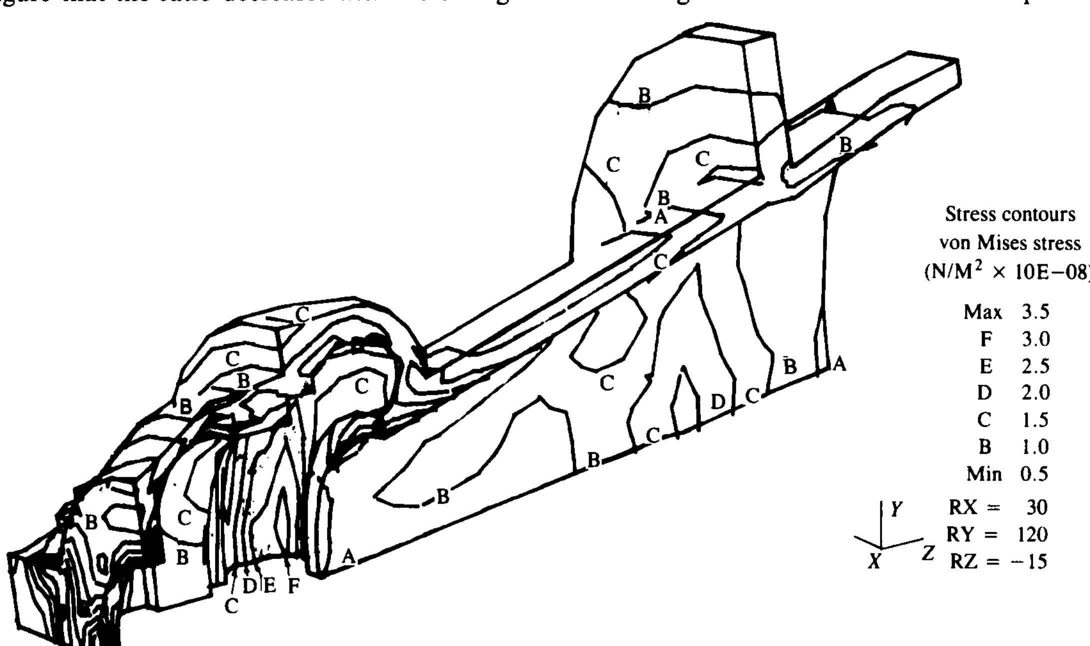

Fig. 4. Von Mises stress contours obtained from three-dimensional thermal stress analysis. (Stress in kg/cm^2)

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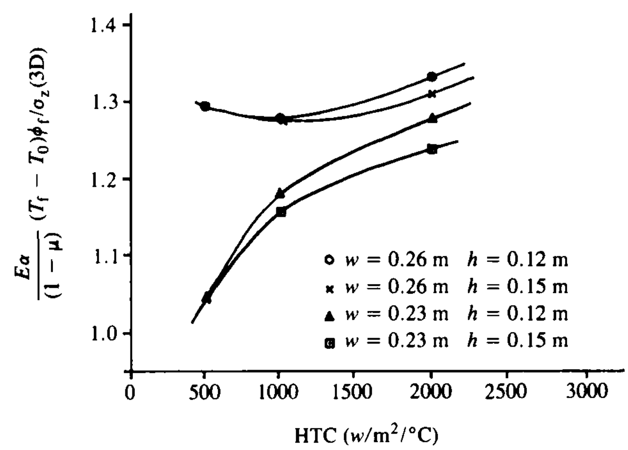

Fig. 6. Ratios of the σ_z stresses obtained from equation (2) and three-dimensional finite element analysis

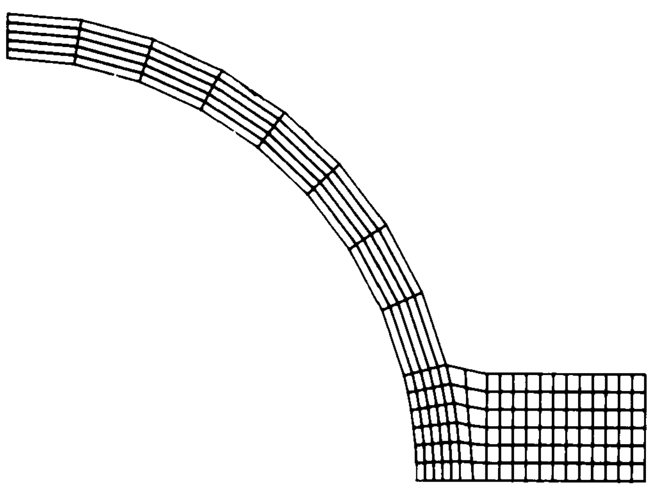

Fig. 7. Two-dimensional finite element model of the casing

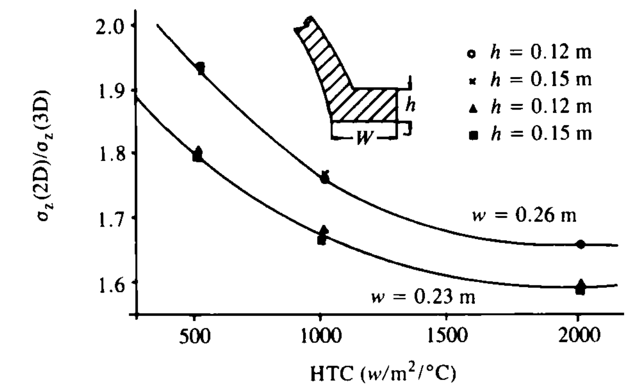

Fig. 8. Comparison of three-dimensional analysis and two-dimensional analysis results. σ_z stresses

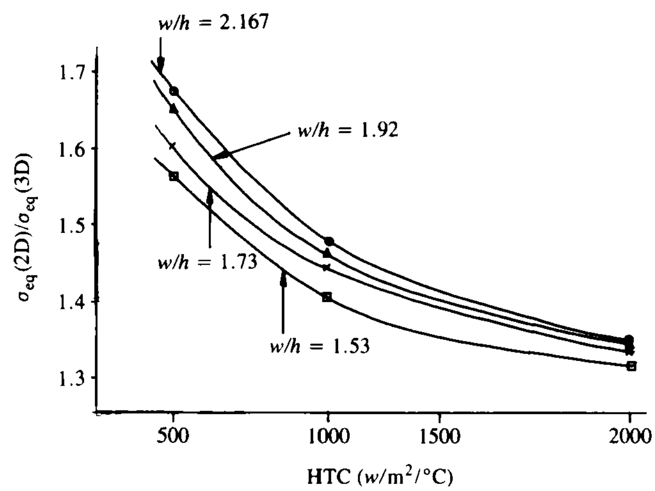

Fig. 9. Comparison of three-dimensional analysis and two-dimensional analysis results. σ_{eq} stresses

the stresses tend to be similar to plane strain condition and the ratio of σ_z 2D to σ_z 3D would be nearer unity. Variation of $(\sigma_{eq})2D/(\sigma_{eq})3D$ values with heat transfer coefficients is shown in Fig. 9.

Typical contact pressure distributions on the parting plane during transient thermal conditions are shown in Fig. 10. It can be seen that the contact stresses are high at the inner edge. To compare the contact pressure distribution on the parting plane with and without 'gap' elements, the σ_y values on the parting plane for a typical load case are presented in Fig. 11 and Fig. 12 for three-dimensional and two-dimensional cases, respectively. It can be seen from these figures that the joint opens near the middle zone of the parting plane. The zone is small in size and hence the difference between the contact stresses in the area of contact with and without 'gap' elements is small. However, to estimate the size of the opening, 'gap' elements should be considered in the analysis. In addi-

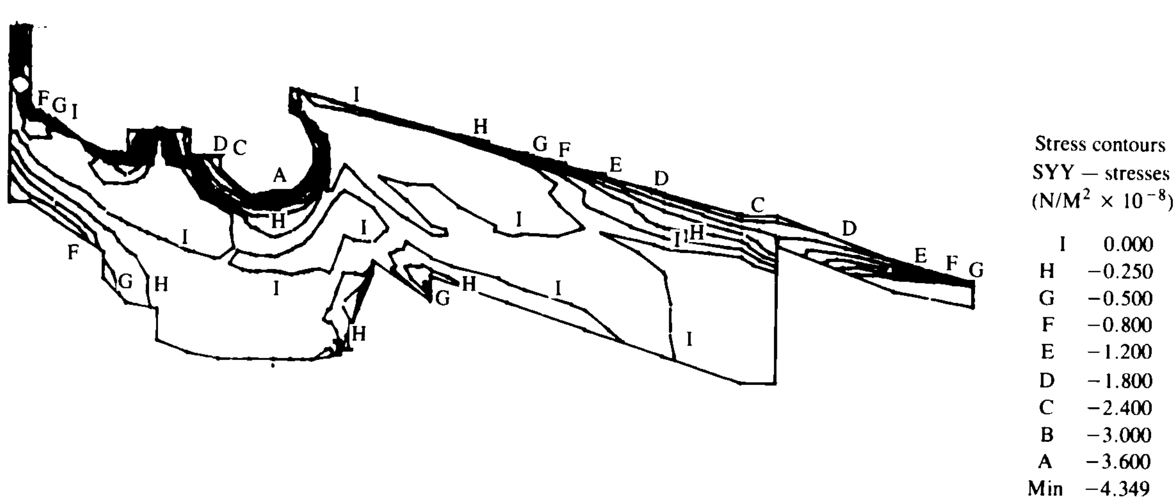

Fig. 10. Contact pressure distribution on the parting plane during transient thermal conditions

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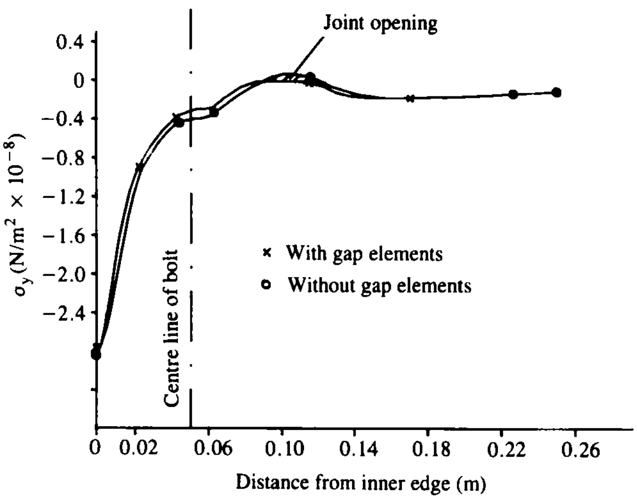

Fig. 11. Comparison of contact pressure distribution obtained with and without 'gap elements' – two-dimensional analysis

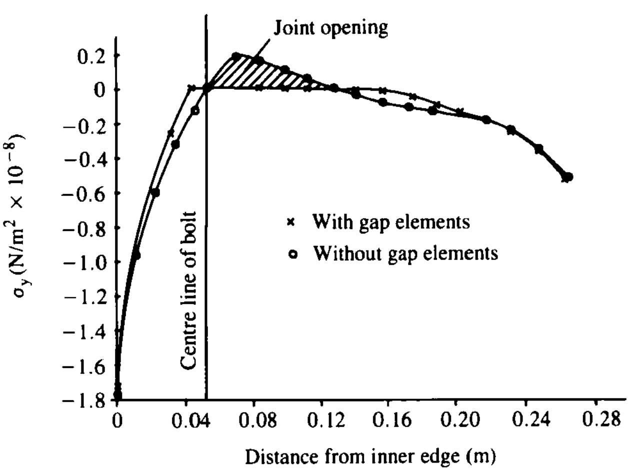

Fig. 12. Comparison of contact pressure distribution obtained with and without 'gap elements' – three-dimensional analysis

tion it can be seen that there is a significant difference in the contact pressure at the inner edge, between two- and three-dimensional results. Therefore, a three-dimensional analysis is required for contact pressure estimation.

6 CONCLUSIONS

The two- and three-dimensional analyses implemented above lead to the following recommendations.

(1) Combining formula (1) and two-dimensional heat transfer analysis can give reasonably accurate results for the range of parameters considered in this analysis. For any other parameters, the validity of the above formula should be proved.
(2) Two-dimensional plane strain analysis gives higher stresses and, therefore, cannot be used. However, if the parameters are within the present range and a two-dimensional analysis is carried out, correction factors suggested in this paper may be applied.
(3) If the normal stress on the parting plane is tensile, analysis with 'gap' elements has to be carried out, in order to have a clear picture of the contact stress distribution. There is a significant difference between the contact pressures at the inner edge obtained by two- and three-dimensional analysis. Therefore, a three-dimensional analysis is necessary for contact pressure estimation.

APPENDIX 1

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