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A parametric study of metal-to-metal full face taper-hub flanges

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Abstract

The paper presents the results of a parametric study, using finite element analysis, of the behaviour of full face metal-to-metal taper-hub flanges. The important stress values in the flange have been obtained for a range of flange thickness, taper-hub thickness and length, when the shell/flange component is subject to internal pressure. The influence of the pre-stress in the bolts is examined. The results obtained have been compared with the predictions from the appropriate sections of the ASME, BS and the new European Unfired Pressure Vessel Standard (Draft BS: *pr*EN 13445). © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Taper-hub flanges; Finite element analysis; Code comparisons; ASME code

1. Introduction

Full face metal-to-metal flanges are used in high pressure situations and also when it is desirable to install a compact arrangement of pipe and flange. In association with this the taper-hub provides a transition piece for ‘smoothing’ the stresses from the flange to the pipe or shell wall. Although a self-sealing gasket, in the form of an ‘O’ ring, is used to avoid leakage at low pressures, it can, and generally is ignored in the stress analysis of the flange. It is worth noting that although the primary design requirement is to seal the flange connection there are situations where knowledge of the stresses may be required. To this end a comprehensive design approach for this component is given in Appendix Y of ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, [1]. In this it is suggested that the pre-stress in the bolts should be made equal to their operating design stress. The corresponding British Standard, PD 5500:2000, [2], and the new European Code, Draft BS: *pr*EN 13445, [3], provide a less detailed design approach for these flanges, which ignore the influence of the shell.

In an earlier study by Spence et al., [4] on the metal-to-metal non-gasketed flange design, an examination was conducted to obtain the most appropriate finite element model. Following this, further minor refinements have been carried out. This refined finite element analysis (FEA) model was used in the present study, in which a range of taper-hub flange sizes were considered together with three different values of bolt pre-stress. In all cases the maximum values of the longitudinal (axial) bending stress, the radial bending stress and the tangential (circumferential) bending stress in the flange were determined, when the shell (or pipe) was subject to an internal pressure. The bending stress was isolated in these cases to provide a ready comparison with the bending stress given in the Codes. A total of 125 different flange geometries were examined, and three bolt pre-stress values, making a total of 375 different cases. The way, in which the stresses behave, as the flange parameters are varied, have been determined by both the FEA and by the Codes. These predictions are compared and since they follow a similar pattern it is possible to isolate the cases in the mid-range, which are typical of all the cases. The appropriate results for these are presented graphically.

2. Allowable stresses and the component geometry

2.1. Allowable stresses

The yield stress of the flange and shell material was assumed to be 300 N/mm^2, giving a nominal design stress of the flange material of $(2/3) \times 300 = 200$ N/mm^2. Using the membrane stress shell equation, with the mean vessel diameter, and the above geometry, a design pressure of approximately 3.95 N/mm^2 (39.5 bar) was achieved. This approach is generally accepted for all international design codes. This value was applied in all cases. The allowable bolt stress, or operating bolt design stress, was taken throughout as 300 N/mm^2.

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Nomenclature

A	outside diameter of the flange
B	inside diameter of the flange
C	bolt circle diameter
g_o	wall thickness of basic shell/pipe
g_1	taper-hub thickness (Fig. 1)
h	taper-hub length (Fig. 1)
t	flange thickness.

2.2. Flange geometry

For these investigations a value of internal diameter, B, of the shell (or pipe), was fixed at 500 mm., and the shell wall (or pipe) remote from the flange, g_o, was taken as 5 mm, for all cases. A range of different values of flange thickness, taper-hub thickness and length, were considered. The bolt circle diameter, C, and the outside diameter of the flange, A, were prescribed by satisfying the rules given in TEMA Table D-5, [5], which gives bolt locations in relation to the taper-hub and the outer flange diameter, for different diameter of bolts. The 'bolt spacing' requirements round the bolt circle, bolt centre to bolt centre, were set, by the authors, at a value of three times the bolt diameter. The number of bolts required and their appropriate diameters, to satisfy the codes, were determined using the procedures set out in the codes for full face, taper-hub flanges.

The codes used were the ASME, Section VIII, Division 1, Appendix Y, [1], the PD 5500:2000, [2], and the draft of the new European Code, *pr*EN 13445, [3]. Only those flanges that predicted stresses which were within the allowable code limits, and which satisfied all the other requirements, were selected for the FEA. It was found, in general, from code calculations, with the design pressure of 3.95 N/mm^2, that it was necessary to have 24 bolts of 22 mm diameter. For this bolt diameter it was recommended, in TEMA, [5], that the hole in the flange be of 25 mm diameter. For these bolts, the minimum hub clearance and the minimum edge clearance, from the bolt centre, was taken from TEMA, [5], as 32 and 23 mm, respectively, in all the flange geometries examined. These are shown in Fig. 1.

Table 1
Flange geometry

Thickness t (mm)	30	35	40	45	50
Hub thickness g_1 (mm)	5.0	7.5	10.0	12.5	15.0
Hub length h (mm)	20	30	40	50	60

Using the above criterion it was found that the five different flange thickness values, t, the five different taper-hub thickness values, g_1 and the five different taper-hub lengths, h, shown in Table 1, totally satisfied the codes. To aid the transition between the flange and the hub, a 5 mm radius was used between the taper-hub and the flange. The dimensions and leading parameters are shown on Fig. 1. A total of 125 different geometries were examined.

3. Finite element modelling

In a previous paper, Spence et al. [4], the viability was

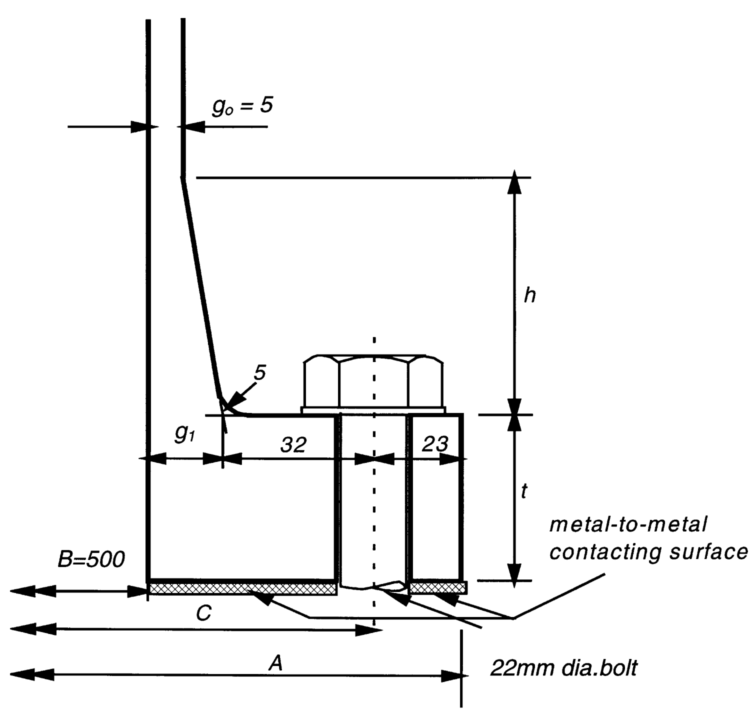

Fig. 1. Details of metal-to-metal flange for parametric study (dimensions in mm).

established of using a two-dimensional axisymmetric model, for what is essentially a three-dimensional component. In neglecting the holes in the flange, and the presence of individual bolts round the flange, it is assumed that the system can be considered as a simple axisymmetric applied effect. This may be thought of in the form of a continuous bolt ring located at the bolt centre and running round the circumference of the bolt circle.

Throughout the analysis the following material constants were used; Young's modulus, 207 000 N/mm^2 and Poisson's ratio, 0.3. The ANSYS, version 5.5, finite element programme was employed throughout this work.

3.1. Element types used

As previously, the main flange, taper-hub and shell were modelled using the standard two-dimensional (four-noded) solid element, 'Plane 42' (ANSYS) with the axisymmetric option activated. At the contact zones where the two flange metal-to-metal surfaces meet and also at the nut-washer top flange surface, a two-dimensional (three-noded) node-to-surface contact element, 'Contact 48', was employed — this assuming zero friction. The procedure used to handle these elements was similar to that detailed and used earlier [4]. In order to provide a more accurate modelling of the bolts than used previously, where two-noded beam elements 'Beam 3' were employed, the bolts were modelled using 'Plane 42' elements. Of these elements 10 elements were used across the bolt width, graded to provide a finer mesh at the sides of the bolts; this enabled the distribution of the pre-stress across the width of the bolt to be examined with some accuracy. The layout of the elements is shown in Fig. 2, where it is noted that there are 10 elements across the vessel and flange thickness.

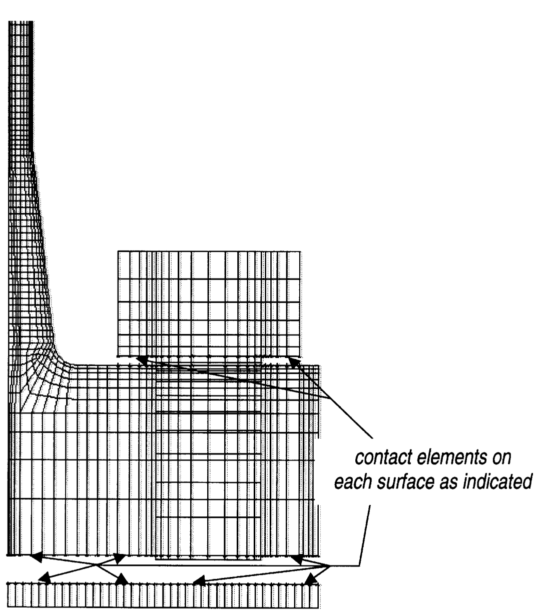

Fig. 2. Finite element mesh.

3.2. Initial bolt stress

As indicated above, the ASME, Appendix Y, Code, [1], makes the recommendation that a bolt pre-stress be applied before pressurisation of the component, and that the value of this be equal to the bolt design stress. The value of this concept has been recognised in previous studies, for example by Webjörn, [6], and Spence et al., [4], in preventing leakage of the joint. In the present studies the bolt design stress was set at 300 N/mm^2 (see Section 2.1). To examine the influence of the pre-stress on the stresses in the flange, three values of the initial bolt stress were examined, viz. 200, 300 and 400 N/mm^2, that is values on either side of the ASME suggested value. The procedure for achieving these initial stress values in the FEA, was by assigning certain displacement values to the lower bolt surface, which were optimised, until the required pre-stress was achieved.

4. Results from the FEA

4.1. Maximum bending stress results

All the 375 cases, for the 125 different flange geometries, have been studied, but it is not possible to reproduce all of the results here. Fortunately the pattern and trends are consistent so that a reduced set of results can convey a good sense of the flange behaviour. For this presentation, the influence of the flange thickness was obtained with a hub thickness, g_1, of 10 mm and the hub length of 40 mm. The influence of the hub thickness was obtained with a flange

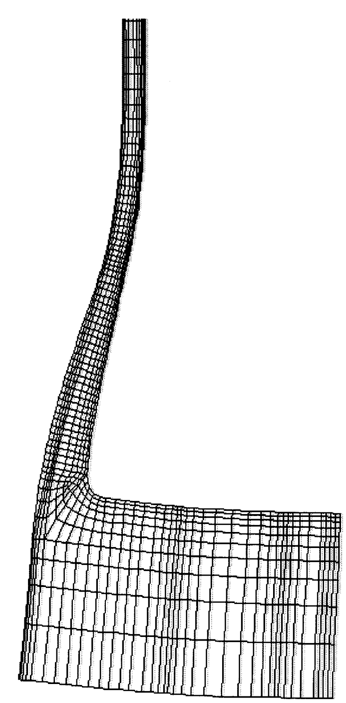

Fig. 3. The displaced shape.

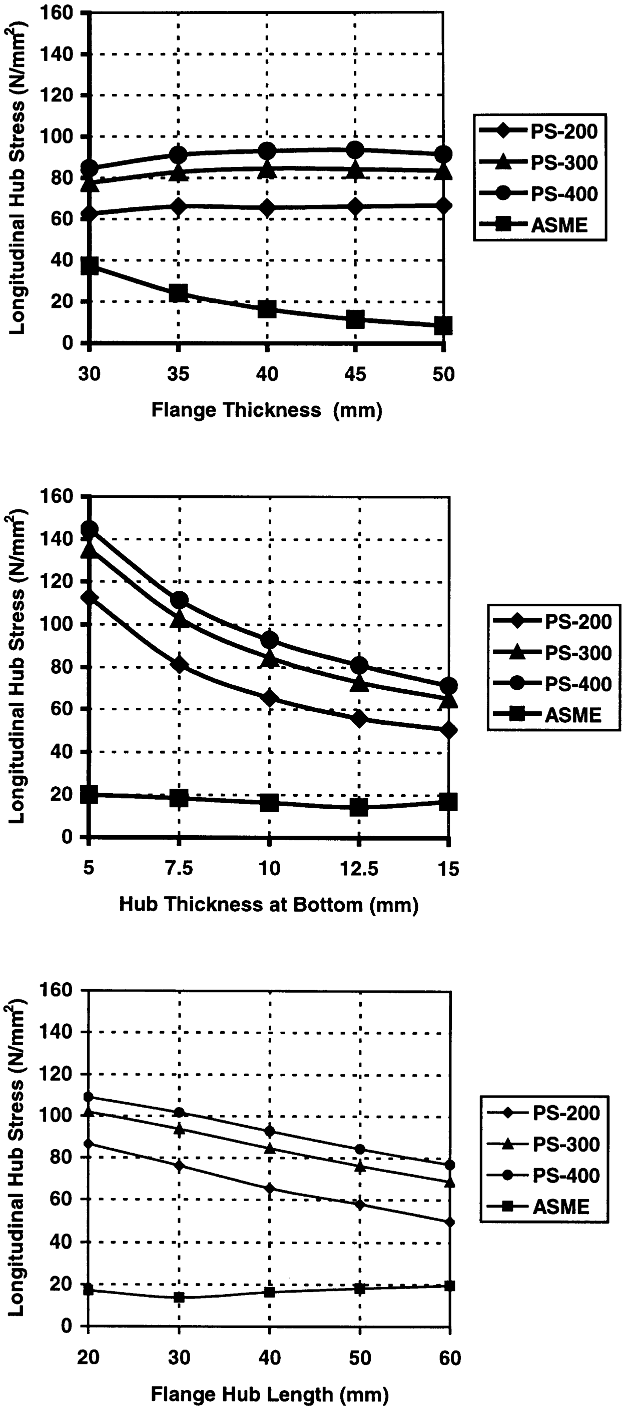

Fig. 4. Comparison of maximum longitudinal hub bending stress (from FEA and ASME code) for a range of flange thickness, flange hub thickness (g_1) and flange hub length.

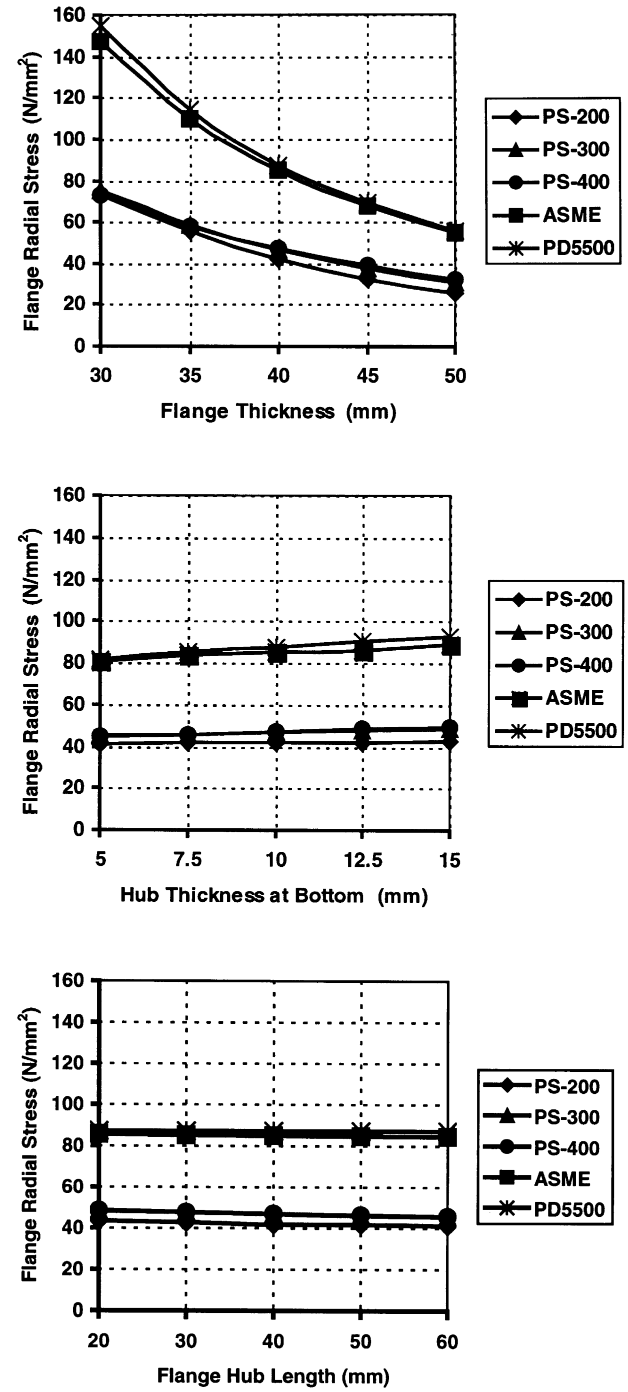

Fig. 5. Comparison of maximum flange radial bending stress (from FEA, ASME and PD5500/CEN codes) for a range of flange thickness, flange hub thickness (g_1) and flange hub length.

thickness of 40 mm and hub length of 40 mm. and finally, the influence of the hub length, was obtained with a flange thickness of 40 mm and the hub thickness of 10 mm. In the FEA work, the maximum bending stresses in the three directions, longitudinal, radial and tangential were considered. In the case of the longitudinal bending stresses, the maximum occurred at the large end of the taper-hub, along the line of the top of the flange. The maximum radial bending stress occurred at the bolt circle diameter, and the maximum tangential bending stress occurred at the inside surface of the flange. These three stress values, obtained from the FEA, are plotted along with the appropriate Code values for the three pre-stress bolt values, 200, 300 and

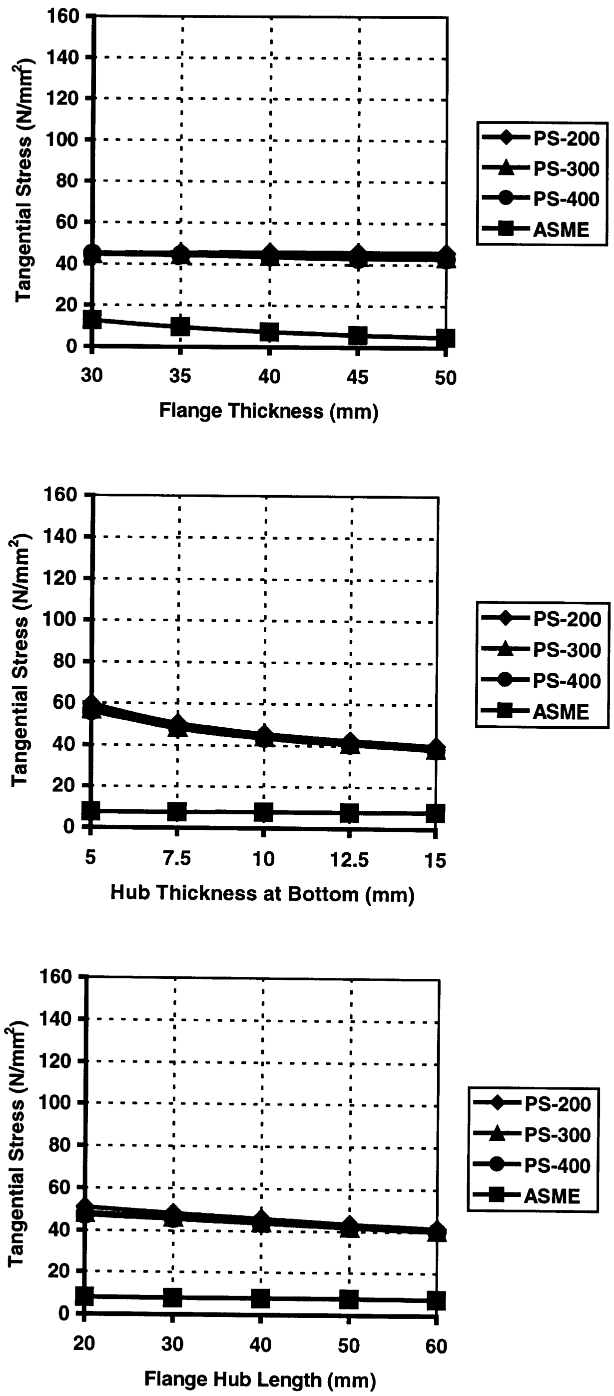

Fig. 6. Comparison of maximum flange tangential bending stress (from FEA and ASME code) for a range of flange thickness, flange hub thickness (g_1) and flange hub length.

400 N/mm^2, in Figs. 4–6. These are denoted as PS-200, PS-300 and PS-400, respectively, on the legends on each figure.

4.2. Displacements of the flange

The displaced shape, of a typical flange, when subject to the internal pressure load, is shown in Fig. 3. In an effort to quantify the possibility of leakage from such a flange, the longitudinal displacements at the inside surface, that is the opening of the flange faces on one side, are recorded in Table 2. The results are for the flange with dimensions; thickness = 40 mm, taper-hub length = 40 mm and hub thickness = 10 mm. As previously it is subject to an internal pressure of 3.95 N/mm^2 and the three cases of bolt pre-stress are considered.

Table 2
Longitudinal opening (on one side) of flange face at inside surface when an internal pressure ($p = 3.95$ N/mm^2) and bolt pre-stress are applied

Bolt pre-stress (N/mm^2)	Longitudinal displacement (mm)
200	0.06836
300	0.04052
400	0.02807

5. Code predictions

In the case of the ASME, Appendix Y, [1], the bending stress values in the three directions, longitudinal, radial and tangential, can be determined directly using the comprehensive analytical approach. The equation for the maximum longitudinal hub bending stress, given in the Code, does not indicate the location of this stress. In the case of the radial flange bending stress, this can either be determined at the bolt circle or at the inside diameter. It was found that the radial bending stresses at the bolt circle were always greater and therefore, these are plotted. The tangential flange bending stress was determined from Appendix Y, at the inside diameter of the flange and so is compared directly with the FEA predictions.

In the case of the PD 5500:2000, [2], for the metal-to-metal full face flange, the equation given is cast in the form which enables the flange thickness to be determined from the allowable stress. In essence the equation arises from the ‘ring bending’ analysis. It thus assumes that the maximum stress is the radial bending stress in the flange and limits this stress to the allowable stress. The approach does not consider the influence, or the existence, of the taper-hub, nor does it enable the longitudinal stress to be determined. The treatment assumes that the applied bending can be obtained in a ‘statically determinate’ manner from the applied forces. Of course, the longitudinal stress in the hub could be determined outwith the Code, using a cylinder ‘edge bending’ calculation. This is discussed later in Section 7. For the full face metal-to-metal flanges, the new European Pressure Vessel Code [3], follows exactly the same approach as is contained in the PD 5500. Since these only provide radial bending stresses the comparisons with the FEA results for the BS and European Codes, are restricted to these values.

6. Discussion of the results

6.1. Longitudinal hub bending stress

With reference to Fig. 4, the following may be concluded:

1. The ASME values are *unconservative* (i.e. unsafe, since they predict lower stresses which would, of course, lead to a false sense of security) when compared with the predictions from the FEA.
2. The ASME values reduce as the flange thickness is increased, but are not influenced by the different hub thickness or hub lengths.
3. The FEA predictions show that, for these flanges, as the bolt pre-stress (PS) increases from 200 to 400 N/mm^2 so the magnitude of the longitudinal bending stress increases.
4. The magnitude of the FEA predictions is almost independent of the flange thickness, but decrease significantly as the values of the hub thickness and the hub lengths increase.

6.2. Flange radial stress

With reference to Fig. 5, the following may be concluded:

1. The ASME and the PD 5500 codes give almost identical results, which are *conservative* (i.e. safe, since they predict higher stresses) when compared with the predictions from the FEA.
2. The ASME and PD 5500 values reduce as the flange thickness is increased, but are not unduly influenced by the different hub thickness or hub lengths.
3. The FEA predictions show that, for these flanges, bolt pre-stress is not significant.
4. The magnitude of the FEA predictions is almost independent of the hub thickness and lengths, but reduces as the flange thickness is increased.

6.3. Flange tangential stress

With reference to Fig. 6, the following may be concluded:

1. The ASME results are *unconservative* (i.e. unsafe) when compared with the predictions from the FEA.
2. The ASME values reduce slightly as the flange thickness is increased, but are not influenced by the different hub thickness or hub lengths.
3. The FEA predictions show that, for these flanges, bolt pre-stress is not significant.
4. The magnitude of the FEA predictions are almost independent of the flange thickness, but they reduce slightly as the hub thickness and length increase.

6.4. Flange displacements

It is clear from Table 2 that the bolt pre-stress provides a beneficial stiffness to the flange and reduces the opening of the flange face on the inside surface. The behaviour of the flange opening and the bolt pre-stress is almost linear.

7. Conclusions

The non-conservative nature of the longitudinal bending stress in the hub and the tangential bending stress in the flange, as predicted by the ASME Code, is significant, and would justify further analytical study since the reasons for the differences are not obvious. It is, however, worth noting that all the FEA stress values are within the allowable stresses used in the Codes.

The longitudinal bending stress situation is not dealt with at all in the BS standard. However, an indication of the order of this stress at the junction of the vessel and flange can be determined using the cylinder edge bending solutions. One simple procedure would be to consider the flange as a ring area, partially restraining the radial movement of the vessel and totally restraining the rotation, as caused by the internal pressure (3.95 N/mm^2). The assumption of zero rotation avoids the necessity of considering the flange rotation and thus simplifies the analysis; it is probably also a realistic assumption for the high bolt pre-load case. It was found that for a hub thickness of 10 mm this analysis gives longitudinal bending stresses of the same order as the FEA, but considerably higher than FEA for a 5 mm hub and less than the FEA for a 15 mm hub. It would appear that the edge bending approach oversimplifies the problem.

Flange leakage can be considered to be due to the flange face rotation. Indicative values of the longitudinal displacement at the inside surface are given in Table 2, where it can be seen that increased bolt pre-stress reduces this displacement and hence reduces the flange rotation. The use of high values of the bolt pre-stress in reducing flange leakage is therefore important, although the benefit of this has to be measured against the higher values of the longitudinal hub stress.

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