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A parametric study of metal-to-metal contact flanges with optimised geometry for safe stress and no-leak conditions

M. Abid[a,*], D.H. Nash[b]

[a]Faculty of Mechanical Engineering, Ghulam Ishaq Khan Institute of Engineering Sciences and Technology, Topi, Pakistan
[b]Department of Mechanical Engineering, University of Strathclyde, Glasgow, UK

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Abstract

This paper presents the results of a parametric study of the behaviour of metal-to-metal contact flanges that have different surface profiles. Using a finite element analysis approach, the important stress values in the flange and bolts and flange rotation/displacement have been obtained for variations in flange thickness, bolt pre-stress and taper angle (different surface profiles) whilst maintaining other leading flange dimensions (hub length and hub thickness) constant, when the vessel/flange component is subjected to internal pressure. In addition, results are compared for the flange geometry with no taper angle on the flange surface with the predictions obtained from the appropriate sections of the ASME, PD5500 and new European unfired pressure vessel standard EN 13445 Part 3. Based on the results of this study, the best flange dimensions are recommended for 'no leak' conditions from the joint.

Keywords: Flanges; Bolted joints; Metal-to-metal; Taper angle; Finite element analysis; Code comparisons; ASME; PD and CEN codes

1. Introduction

Full-face metal-to-metal bolted flange joints are often used in high integrity situations and also when it is desirable to install a compact arrangement of pipe and flange to minimise space. Although a self-sealing gasket, in the form of an 'O' ring, is used to avoid leakage at low pressures, it can be, 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 type 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 levels 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, EN 13445 [3], provide a less detailed design approach for these flanges, which ignores the influence of the shell. In addition, all such codes do not provide information regarding maximum bolt spacing, minimum flange thickness and exact flange surface profile, namely the taper angle on the flange surface for a 'no leak' condition. No leak condition in this context means 'zero leak' from the flange joint i.e. zero displacement at the inside diameter of the mating faces.

A parametric study was performed by Spence et al. [4] for large diameter metal-to-metal non-gasketed flanges with variation of hub thickness, flange thickness and hub length without consideration of the taper angle on the flange surface, using finite element analysis (FEA) and analytical approaches from standard design codes. In the present paper, a parametric study of non-gasketed metal-to-metal flange of a size 4 in. nominal bore, 900# class is performed for the variation in flange thickness, bolt pre-stress and taper angle whilst keeping hub length and hub thickness constant. The major international codes, ASME, PD5500 and CEN do not apply to these variations in their flange design approach due to the inability to take account of the taper angle on the flange surface. Similarly, the results of other studies such as Meck's formula [5] for the maximum bolt spacing for flange sealing cannot be applied to non-gasketed metal-to-metal flanges, due to its limitation of being only applicable to flanges with no taper angle on the main flange mating surface.

* Corresponding author. Fax: +92-938-71889.
E-mail address: abid@giki.edu.pk (M. Abid).

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Nomenclature

A	outside diameter of the flange
B	inside diameter of the flange
C	bolt circle diameter
g_0	wall thickness of basic shell/pipe
g_1	taper-hub thickness
h	taper-hub length
t	flange thickness.
ID	inside diameter
PCD	pitch circle diameter
PT	positive taper angle (convex surface profile)
NT	negative taper angle (concave surface profile)
NOT	no taper angle (flat surface profile)
θ	taper angle on flange surface
PS	pres-stress

The flexibility of the flange is a function of the thickness and the joint height is defined as the combined thickness of both flanges in assembly. The required value of minimum joint height in order to avoid flange rotation or displacement at the inside diameter for no leak condition whilst minimising bolt fatigue due to flange rotation has been determined to be an important factor in this study. This is calculated to be at least six times the bolt diameter used in the joint and agrees with the statement by Webjorn [6,7], whereas codes as ASME, PD, CEN do not recommend a minimum flange thickness for the no leak condition. In addition, standard design codes do not provide any information regarding flange rotation or axial flange displacement at the inside diameter.

Lewis et al. [8] and Fessler et al. [9] performed an experimental study for three different flange joints with positive, negative and zero taper angles on the mating flange surfaces for small sizes and low pressure application flanges made of different materials. They concluded that when a small positive taper angle was machined onto the faces of the flange, then the passage becomes convergent and leakage can be significantly reduced when compared to a similar flange with a small negative taper angle.

2. Aims and scope

The present work examines the effect noted by Lewis and Fessler in addition to the stress behaviour of metal-to-metal contact joint, varying different flange parameters using both finite element analysis and code methods (ASME Appendix-Y, PD5500 working form 16 and new European code). These predictions are therefore compared and discussed and appropriate results for these studies are presented graphically. The model used has the following features: three different flange surface profiles (positive, negative and no-taper angle), six flange thicknesses (10–35 mm) and four different values of bolt pre-stress (382, 472, 512 and 640 N/mm^2). In all cases the maximum values of the longitudinal (axial) bending stress, the radial bending stress and the tangential (circumferential) bending stress, axial flange displacement, flange rotation, effect of pre-stress applied and bolt stress variation were determined, for internal pressure loading. The bending stress was isolated in these cases to provide a ready comparison with the bending stress given in the codes. A total of 18 different flange geometries were examined, and four bolt pre-stress values, making a total of 72 different cases.

3. Allowable stresses and flange joint configuration

3.1. Allowable stresses

The yield stress of the flange and shell material selected was 372 N/mm^2, giving a nominal design stress of 248 N/mm^2. For this flange joint, a high strength bolt of 10 mm diameter as per ISO 898, grade 8.8, with minimum yield strength of 640 N/mm^2 was calculated and selected.

3.2. Flange geometry

The flange configurations under examination are shown in Fig. 1a–c. The flange with dimensions; thickness = 10, 15, 20, 25, 30 and 35 mm and taper angle = 0, + 0.03, and − 0.03° were used (Table 1). 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. This was checked using Meck's formula [5] for flange sealing. The number of bolts required and their appropriate diameters were determined using the procedures set out in the codes for full face, taper-hub flanges. From code calculations, with the proof test pressure of 23 N/mm^2, it was found necessary to have 16 bolts of 10 mm diameter.

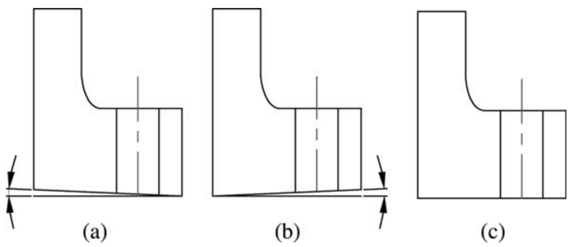

Fig. 1. Flanges with different surface profiles, (a) concave profile or negative taper angle, (b) convex profile or positive taper angle, (c) flat surface or without any taper on the Flange Face, Taper angle used during analysis was 0.03 °.

Table 1
Flange geometric parameters

Fixed parameters					Variable parameters		
Flange hub length (mm)	Pipe thickness (mm)	Flange hub thickness (mm)	Bolt circle diameter (mm)	Flange outside diameter (mm)	Flange inside diameter (mm)	Flange thickness (mm)	Flange surface taper angle (degree)
34	13.5	15.5	146	171	87.3	10,15,20,25,30,35	0, + 0.03, − 0.03

4. Finite element modelling

In the previous papers by Spence et al. [12] and Nash et al. [4], the viability was established of the approach of using a two-dimensional axisymmetric model, for what is essentially a three-dimensional component. Details of the parameters used in the finite element model and a quarter model developed are shown in Fig. 2a and b. Throughout the analysis the following material constants were used; Young's modulus, 203,395 N/mm^2 for flange and 204,000 N/mm^2 for bolt and Poisson's ratio, 0.3. The ANSYS, version 5.7, finite element code was employed throughout this work.

4.1. Element selection, constraints and mesh

The flange, taper-hub and shell were modelled using the standard two-dimensional (four noded) solid element, 'PLANE42' (ANSYS) with the axi-symmetric option. 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, 'CONTACT48', 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 'BEAM3' elements were employed, the bolts were modelled using 'PLANE42' elements. Seven of these 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. 3, where it is noted that there are seven elements across the vessel and flange thickness.

4.2. Bolt pre-stress and pressure loading

As indicated earlier, the ASME Code in Appendix Y makes the recommendation that a bolt pre-stress be applied before pressurisation of the component, and that the value of this load be equal to the bolt design stress. The value of this concept has been recognised in previous studies, for example in Webjörn, [6,10,11], and Nash et al. [4] and one of the present authors [13], in preventing leakage of the joint. In the present studies, the bolt design stress was set at 512 N/mm^2, which is 80% of the yield of the bolt material for better joint strength due to the fact that 'the higher the pre-stress the better the joint is'. In order to examine

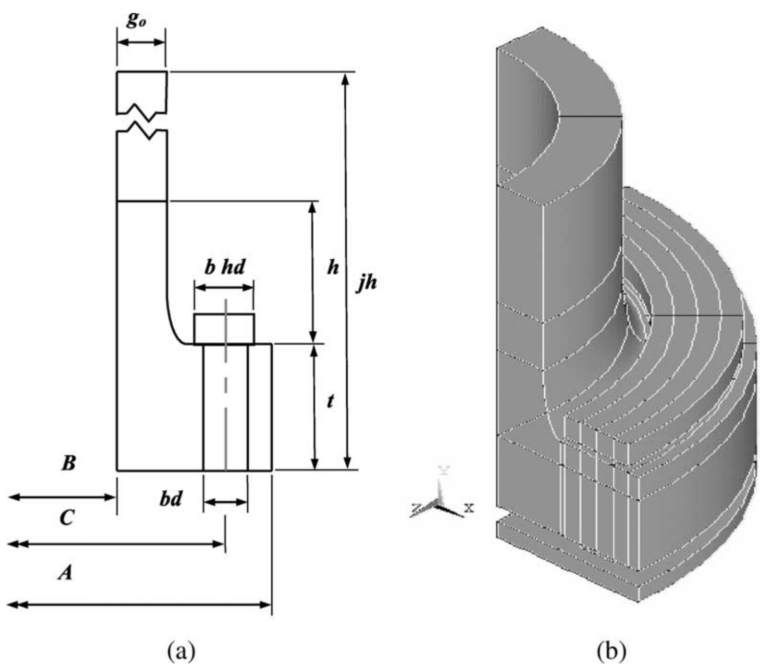

Fig. 2. (a) Parameters used in the finite element model, (b) schematic quarter model.

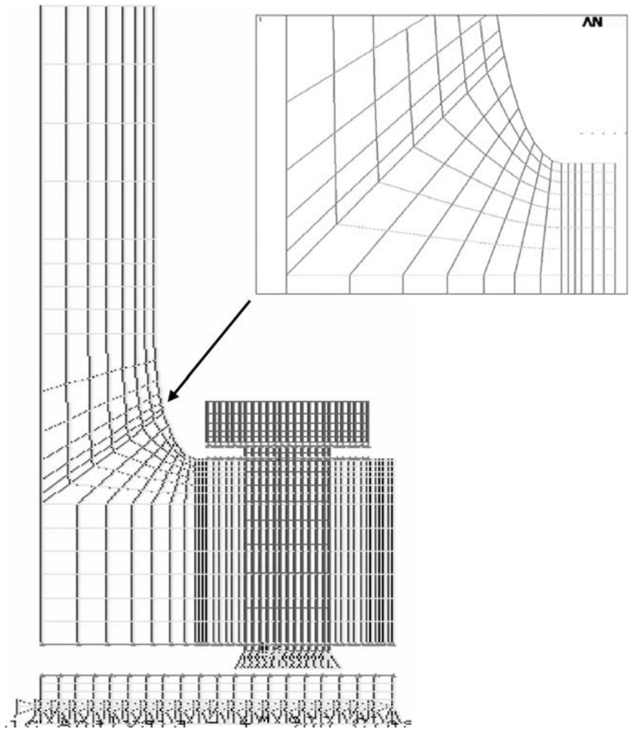

Fig. 3. Flange joint, element plot with applied boundary conditions and enlarged hub portion.

the influence of the pre-stress four values of the initial bolt stress were applied, viz, 382, 472, 512 and 640 N/mm^2, which are 60, 70, 80 and 100% of the yield stress of the bolt material. The effect of initial pre-stress on either side of the bolt design stress is applied to study its effect on stresses in flange and bolts in addition to joint opening. The procedure for achieving these initial stress values in the FEA was by assigning certain displacement values to the lower bolt surface, which was modified, until the required pre-stress was achieved. After pre-stress application, an internal pressure of 23 N/mm^2 (proof test pressure = 1.5 times design pressure) was applied which was the proof test pressure for flange size of 4 in., 900# class as per codes for all the cases.

5. Code predictions

In the case of the ASME, Appendix Y, the bending stress values in the three directions, longitudinal, radial and tangential, can be determined directly using a comprehensive analytical approach. The equation for the maximum longitudinal hub bending stress, given in the code, does not indicate the location of this stress, but rather only the magnitude. 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 the Appendix Y procedure, at the inside diameter of the flange and so is compared directly with the FEA predictions.

In the case of PD 5500:2000, for the metal-to-metal full faced flange, the equation given is a form which enables the flange thickness to be determined from the allowable stress. In essence the equation arises from a '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.

For the full faced metal-to-metal flanges, the new European pressure vessel code follows exactly the same approach as in PD 5500. Since these only provide radial bending stresses the comparisons with the FEA results for the PD and European codes are restricted to these values.

6. Parametric studies

6.1. Study-1: comparison of code calculated and FEA stress results (Flange thickness = 30 mm, Flange surface profile = no taper angle)

Flange stress results from FEA and code calculations are given in Table 2 and plotted in Fig. 4 for flange thickness of 30 mm without any taper angle on flange surface and with four bolt pre-stress values. Results are compared for the following two flange geometry conditions as;

1. satisfying all the requirements for the codes regarding bolt spacing, clearance from the edge and hub as well as the allowable stresses
2. the original dimensions of the flange used in the experimental work [13] but ignoring the taper angle on flange surface.

With reference to Table 2, the following may be concluded.

The ASME code provides predictions both for longitudinal and radial stresses whereas PD/CEN provides results, only for radial stresses at PCD.

1. *Longitudinal stress*: ASME and FEA predictions at the hub flange intersection show almost the same stress results.
2. *Radial stress*: At the inside diameter, stresses are found slightly higher than the FEA predictions. The ASME and the PD5500/CEN results are *conservative* when compared with the predictions from the FEA at the bolt circle diameter (PCD). However, results from PD5500/CEN are found higher than the ASME predictions.

Table 2
Flange stress as per FEA and codes

Stress	FEA (N/mm^2)	ASME (as per code) (N/mm^2)	ASME (original flange) (N/mm^2)	PD 5500/CEN (as per code) (N/mm^2)	PD 5500/CEN (original flange) (N/mm^2)
Longitudinal	28.47	36.03	29.36	–	–
Radial (ID)	11.00	18.77	15.28	–	–
Radial (PCD)	25.96	63.13	58.05	80.75	74.37
Tangential	44.52	16.74	13.44	–	–

3. *Tangential stress*: The ASME results are *non-conservative* when compared with the predictions from the FEA at the flange inside diameter.
4. *Effect of bolt pre-stress*: The magnitude of the longitudinal and tangential bending stress is almost independent of the pre-stress (see Fig. 4); however a negligible small increase in radial bending stress is observed with the increase in bolt stress.

6.2. Study-2: parametric FEA stress results (Flange thickness = 10–35 mm, Flange surface profile = positive, negative and no taper angles)

Longitudinal, radial and tangential bending stress from FEA, are plotted for the four bolt pre-stress values with positive, negative and no taper angle flange profiles in Fig. 5a–c.

6.2.1. Flange stresses

6.2.1.1. Longitudinal hub bending stress. The longitudinal hub stress decreased with an increase in flange thickness for all the three flange profiles and became constant above a flange thickness of 25 mm (Fig. 5a). For flange thicknesses up to 20 mm, the longitudinal stress is almost independent of the pre-stress applied in the bolts. The highest stress pattern was observed for the positive taper angle profile, whereas the lowest stress pattern was observed for the negative taper angle.

6.2.1.2. Flange radial stress at PCD. For an increased pre-stress, the radial bending stress at the bolt circle diameter (PCD) decreased for all the three flange surface profiles. The radial flange bending stress decreased almost to zero with increase in flange thickness with positive taper angle (Fig. 5b) and became constant for flange thickness of 25 mm and above. It was highest for the flange with negative taper angle and decreased with increase in flange thickness. The same pattern was observed for 'no taper angle' however the magnitude of the resulting stress was less than those of the negative tapered angle flange.

6.2.1.3. Flange tangential stress. For the negative and no taper angle flanges, the tangential stress decreased with increase in flange thickness and applied pre-stress, whereas it increased for the positive taper angle flange. Except for negative taper flange, stress was independent of the pre-stress (Fig. 5c).

6.2.2. Flange displacement and rotation

In an effort to quantify the possibility of leakage from the metal-to-metal contact flange joint, the longitudinal displacements at the inside diameter, that is the opening of the flange faces are plotted in Fig. 6. For all the three flange profiles, displacement decreased with an increased flange thickness. Displacement in positive taper angle flange was found almost independent of pre-stress applied and became almost zero for a flange thickness of 30 mm and above. Displacement decreased to 0.0162 mm for negative and to 0.0076 mm for zero taper angle flanges of maximum 35 mm thickness for maximum pre-stress of 100% (640 N/mm^2) of bolt yield stress. The maximum flange displacement is obvious for the flange with

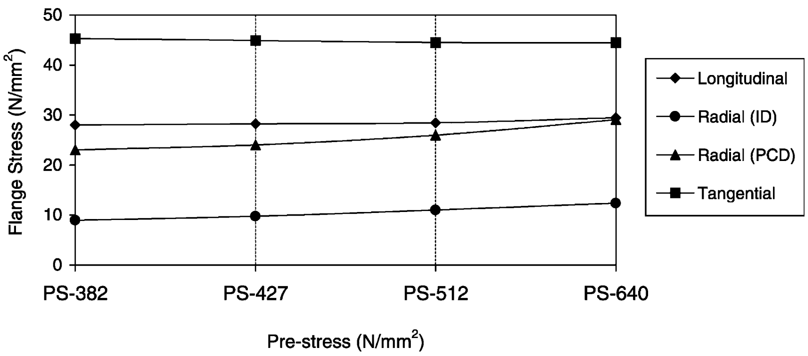

Fig. 4. Flange stresses at different pre-stress, (PS-Pre-stress).

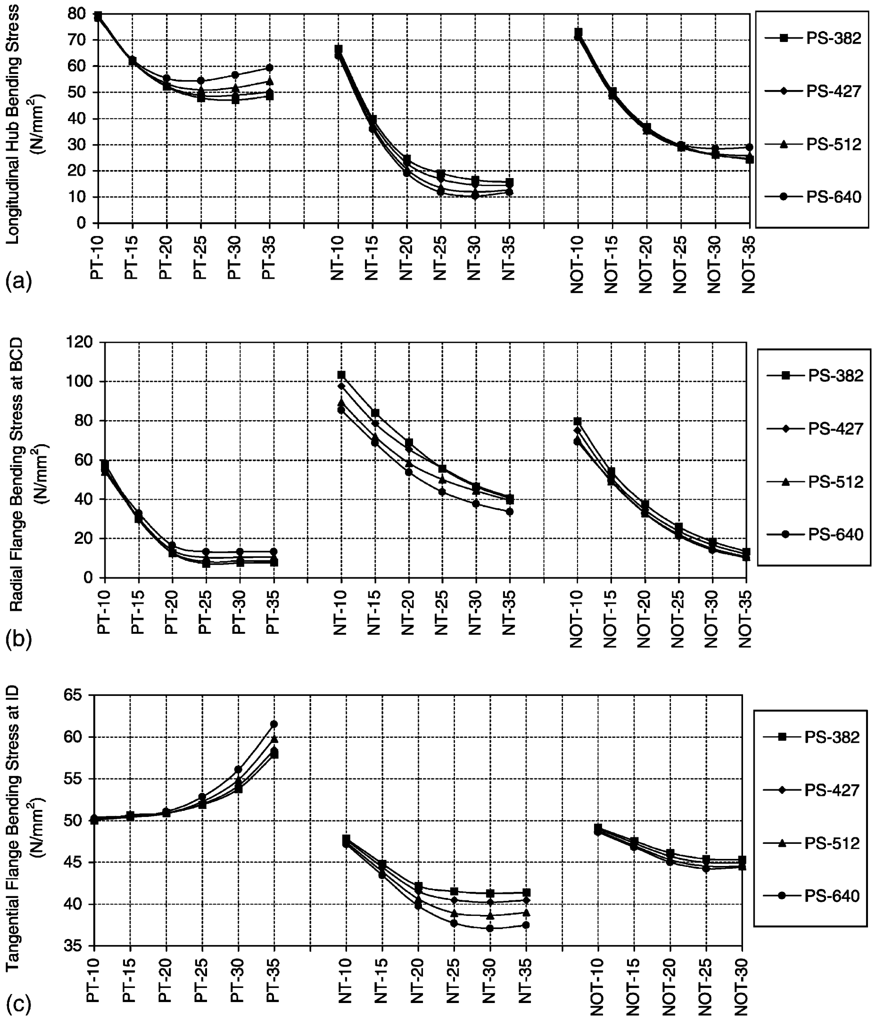

Fig. 5. Comparison of FEA stress results for Flange with positive, negative and No Taper angles and various Pre-stress applied for a range of flange thickness (10–35 mm) (a) maximum longitudinal Hub bending stress, (b) maximum Flange radial bending stress at PCD, (c) maximum tangential bending stress at ID.

negative taper angle even for the same pre-stress and pressure loading for higher flange thickness of 35 mm (Fig. 6). It is even more than the flange of thickness 15 mm with positive taper angle and close to the flange thickness of 20 mm with no taper angle.

Displacement comparison of flange design (thickness) determined using codes and FEA: Results of minimum flange thickness calculated using ASME, PD, CEN codes (15 mm) and FEA (30 mm) for flange displacement at the inside diameter with different flange surface profiles are

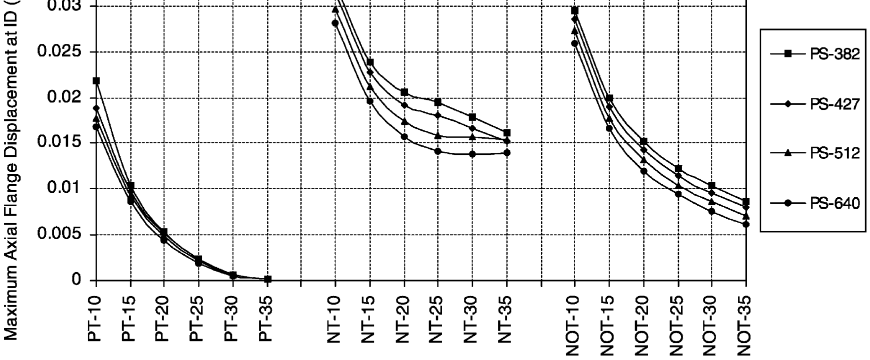

Fig. 6. Comparison of maximum axial Flange displacement at ID (from FEA) for Flange with positive, negative and No Taper angles and various Pre-stress applied for a range of flange thickness (10–35 mm).

Table 3
Comparison of flange displacement at inside diameter (ID) for flanges with positive, negative and no taper angles, with different pre-stress values

Pre-stress	Flange displacement at inside diameter					
PS (N/mm^2)	PT-15 (mm)	NT-15 (mm)	NOT-15 (mm)	PT-30 (mm)	NT-30 (mm)	NOT-30 (mm)
382	0.0103	0.0239	0.0200	0.0007	0.0179	0.0103
427	0.0098	0.0228	0.0190	0.0006	0.0167	0.0096
512	0.0092	0.0212	0.0178	0.0006	0.0157	0.0086
640	0.0086	0.0196	0.0167	0.0005	0.0162	0.0076

recorded in Table 3. Rotation for flange with zero taper angles is almost double and for negative taper angle is almost 3.5–4 times that of the flange with positive taper angle. The displacement of flange of thickness 15 mm by codes is about 15 times more than the FEA recommended flange thickness (30 mm). From results, it is concluded that good contact at the inside diameter with negligible small initial gap can only be obtained using positive taper angle at the flange surface.

6.3. Variation in bolt stresses under different pre-stress

6.3.1. Bolt stress and bolt bending

Bolt-bending behaviour based on stress variation on inside and outside node of the bolts is plotted in Fig. 7a–c during different pre-stress and operating (pressure load) conditions. Average bolt stress variation is maximum for the flange with negative taper angle and is almost independent of pre-stress for all the three flange profiles (Fig. 7a). Flange rotation and stress variation is obvious at the inside and outside nodes of bolts from Fig. 7b and c, even at a pre-stress of 100% and pressure loading. For flange with no taper angle, the difference decreased at higher pre-stress values. For flange with positive taper angle, a negligible small difference was noted only for the flange of 10 mm thickness which decreased to almost zero, due to little or no flange rotation.

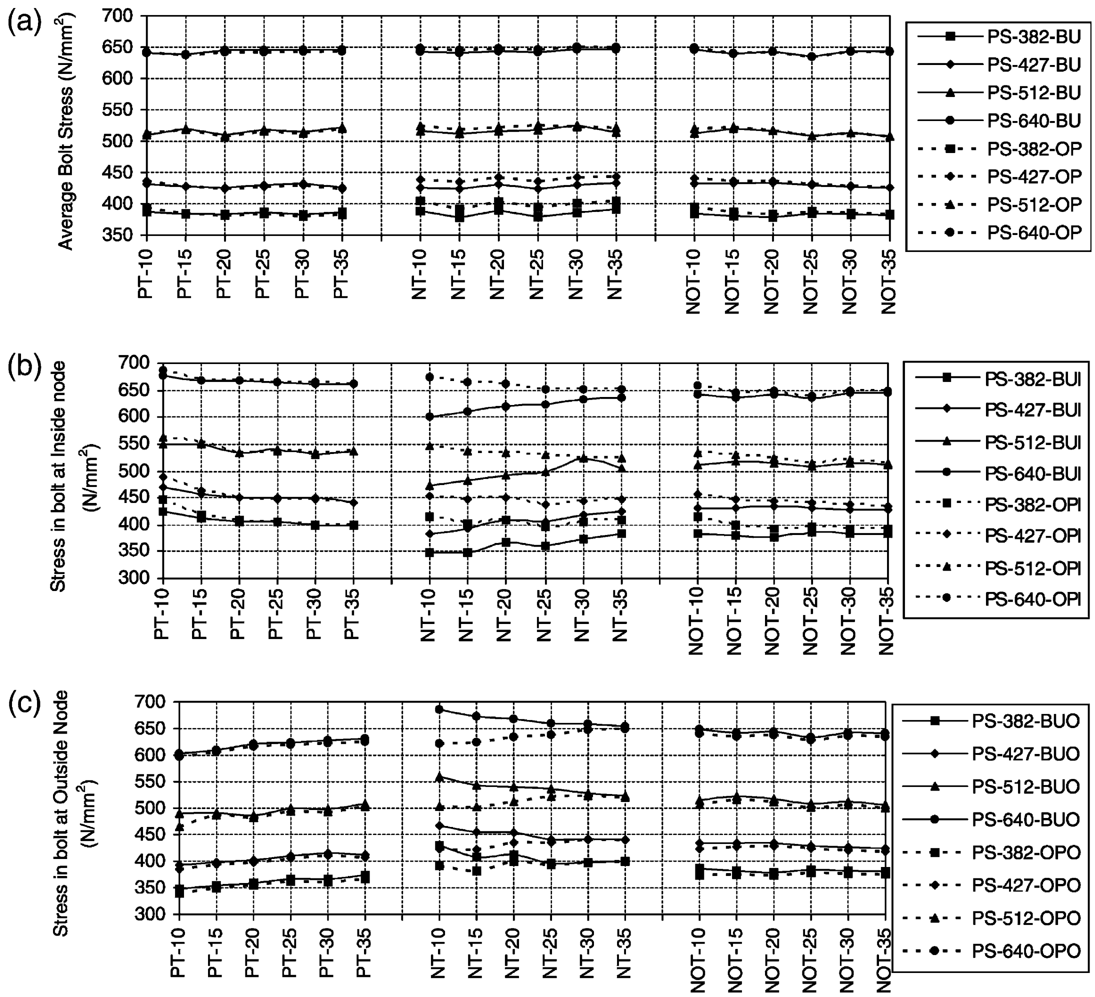

Fig. 7. Comparison of FEA results during pre-stress and pressure loading of for Flange with positive, negative and No Taper angles and various Pre-stress applied for a range of flange thickness (10–35 mm) (a) Avg. Bolt stress, (b) stress in bolt at inside node (c) stress in bolt at outside node.

7. Discussion

From the results, the maximum longitudinal bending stress is at the hub-flange intersection, which is in contradiction to the flange design analysis in the study by Nash et al. [4], where longitudinal stress was at the hub-pipe intersection. It is concluded that this is due to the better parametric selection, i.e. instead of sharp taper hub (portion between flange and hub) geometry, an elliptical portion is introduced between the hub and flange intersection. In addition, this flange design with original dimensions [13] and no taper angle on the flange face provides a better control for the flange bending stresses compared to the flange design using code requirements. Overall, the stress results calculated from FEA and codes are (20–40% of the yield stress of the flange material) within the allowable stress limits up to the proof test pressure, which concludes better joint strength. Therefore, a limit analysis may be performed to achieve a better joint load capacity as the same flange can be used for higher pressure ranges.

Knowing the comparatively small stress pattern, the next main concern is to achieve proper contact at the inside diameter and avoid flange displacement/rotation for proper joint sealing. Thickness of flange calculated using codes does not guarantee a 'no-leak' condition due to flange displacement under pressure loading; therefore the need is to determine the minimum required flange thickness. From parametric FEA studies, it is determined as at least six times the bolt diameter. Available codes do not provide any information other than for flat flange surface profiles. The machining of the exact flange profile with suitable magnitude of taper angle is determined essential to achieve proper contact at inside diameter of flange and static mode of load in the joint and is concluded for the flange with positive taper angle on flange surface. Regarding bolt pre-stress, no information is provided by any of the available codes except ASME code. However, it is determined that the use of high strength bolts with higher bolt pre-stress has provides a beneficial stiffness to the flange and reduces the opening of the flange face on the inside diameter. From bolt stress variation results, it is concluded that bolt fatigue strength is highest for the flange with positive taper angle and lowest for the flange with negative taper angle.

8. Conclusions

To achieve 'no-leak' condition from a flange joint, a flange with positive taper angle, elliptical hub, and thickness at least six times the bolt diameter, using high strength bolts with minimum pre-stress of 80% of the yield stress of bolts is advisable.

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