TWO DIMENSIONAL FINITE ELEMENT ANALYSIS FOR LARGE DIAMETER STEEL FLANGES

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ABSTRACT

This paper presents numerical results obtained using Finite Element Analysis (FEA) in studying large diameter welded neck steel flanges under different loading conditions. Obtained FEA results show the effect of the clamping pressure, internal pressure, axial end load, temperature effect, gasket elasticity modulus on the contact pressure between the gasket and the steel flange. As expected clamping pressure is a determinate factor for the sealing condition. Gasket material is another primary factor in designing flanged joints.

Keywords: *Flange, Gasket, Sealing, Finite Element Analysis*

الملخص

تقدم هذه الورقة النتائج العددية المستقاة من تحليل العناصر المحددة لدراسة شفة كبيرة القطر ملحمة العنق تحت تأثير أحمال مختلفة. بينت النتائج تأثير كلا من ضغط المسك و الضغط الداخلي و الحمل الأفقي الأخير وتأثير الحرارة ومعامل المرونة للحشية على ضغط التلامس بين الحشية والشفة الصلبة. كما هو متوقع يمثل ضغط المسك عاملا رئيسيا لحالة السداد. كذلك تعتبر مادة الحشية عاملا هاما في تصميم الوصلات ذات الشفة.

1. INTRODUCTION

Finite Element Analysis (FEA) is a powerful approximation numerical technique used to solve engineering problems. FEA has been used to model flanged joint by many researchers, see for example [Sawa et al., 1991; Brown et al., 2001; Nagata, et al., 2001]. Two dimensional as well as three dimensional models have been used mainly to study the mechanics of bolted flanges. Investigated parameters include internal pressure [Sawa et al., 2001], high temperature application [Brown, et al., 2001], nominal pipe diameter [Sawa et al., 1991], and gasket geometry [Sawa and Ogata, 2001]. Different finite element packages were used including ABAQUS [Brown, et al., 2001] and ANSYS 5.6 [Nagata, et al., 2001].

Analysis of flange bolted joint goes back to the early work of Waters et al. [1949] who reformed the basis of bolted flange analysis. However, bolted flange analysis, see for example Lake and Boyd [1957], including the current ASME code [ASME, 1998] accounts for stresses in the flange including the rim and hub stresses due to applied load. In recent years bolted joints are designed based on tightness criteria and failure of other components, i.e. gasket collapse or bolt failure [Nagata, et al., 2001].

In all of the previous studies, nominal pipe diameters were small and the largest reported value is 24 inches. Saudi Aramco was pioneer in using large diameter steel flanges for steel pipelines and piping systems. In fact Saudi Aramco started using large diameter steel flanges (Nominal Pipe Size (NPS) 26 to 60 inches) before ASME developed their large diameter steel flanges ASME B16.47 for NPS 26 through NPS 60 in 1990 [ASME, 1996].

This paper addresses the stress analysis of flanged joint made up of the flange and the gasket for large diameter steel flanges. Stress analysis is directed toward the contact pressure and the gap condition between the steel flange and the gasket using two dimensional finite element analyses. Parametric studies include studying the effect of clamping pressure, internal pressure, axial end pressure, temperature and gasket material on the contact pressure.

2. FINITE ELEMENT MODELING

The bolted joint is modeled as simplified two dimensional (2D) axisymmetric case. The line of symmetry is taken with respect to the mid plane of the gasket while the axisymmetric line is taken with respect to the pipe axis. The two dimensional axisymmetric quarter of the bolted joint is shown in Figures 1 and 2. Note the gasket underneath the hub of the flange shown clearly in Figure 2. Although the bolt hole is showing in Figure 1, as first order approximation, the model is taken to be away from the bolt in the middle between two successful bolts. Such approximation is common and acceptable for flange and gasket analysis purposes [Nagata et al., 2001]. However, if one needs to study stresses on bolt or the contact stresses between the flange and the bolt, he has to incorporate the bolts in his 2D model.

Three dimensional (3D) analysis is needed when it comes to model issues such as; the unevenness of bolt tightening, the effective gripping area of the bolt, waviness of the gasket, and other assembly problems such as lack of coaxially of the system, radial misalignment, angular misalignment and so on.

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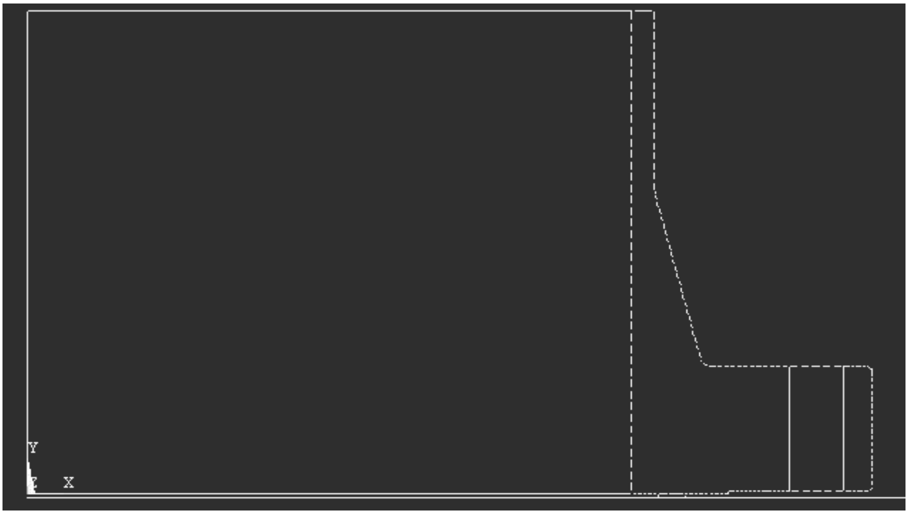

Figure 1: The Right Top Symmetric Quarter of the Flange Joint.

The materials used in this 2D analysis are given in Table 1 for the flange and the gasket. Although the spiral wound gasket is composite material made of asbestos fibers and a binder, it is modeled as homogenous isotropic material in the literature, see for example [Sawa and Ogata, 2001]. Thus, gasket is model as two dimensional isotropic homogenous entity. However, failure analysis of the gasket needs to look at it as a composite material using bimaterials concept or classical laminate plate theory. The material used for the flange is ASTM A105 with 250MPa yield strength.

Dimensions of the bolted joint are given in Table 2. The modeled flange connection is 48 inches NPS (1,219mm), 300 class with ¾ inches (19.05 mm) thickness. The flange is assumed to be perfectly welded to the pipe. For assumed API 5L Grade 42 pipe the maximum permissible internal pressure is 66 bar based on yielding criteria. Figure 3 shows the finite element mesh of the joint. All solid elements assumed to be PLANE43 with four nodes and two degrees of freedom per node. These elements were treated as axisymmetric elements with respect to pipe axis.

Table 1: Material Properties of the Joint.

Part	Materials	Elasticity Modulus (GPa)	Poisson's Ratio	Yield Strength (MPa)	Thermal Expansion Coefficient (m/m/C^o)
Flange	Steel	200	0.3	250	15e-6
Gasket	Asbestos	5	0.33	**	**

Note the very fine mesh at the gasket and the fine mesh at the stress concentration areas at the extreme ends of the rim, the rim-hub joint and the raised face fillet. A contact pair is used between the gasket and the flange with the gasket as the contact surface. Table 3 gives the loading conditions used in the parametric study.

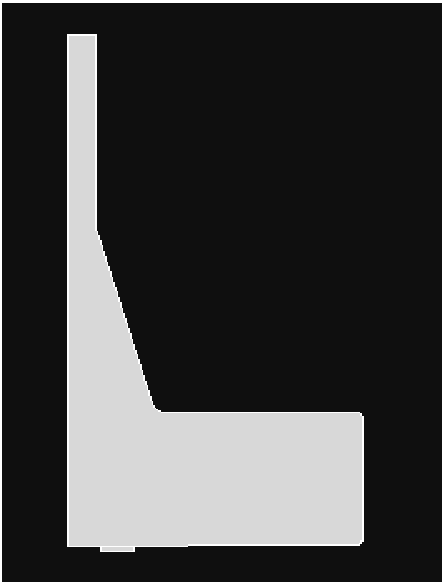

Figure 2: The Quarter Axisymmetric 2D Model

Table 2: Dimensions of the Joint Parts.

Part	Item	Unit	
Flange	Bore	mm	1143
	Bolt Circle Diameter	mm	1534
	Raised Face Diameter	mm	1372
	Flange Outer Diameter	mm	1651
	Raised Face	mm	1.588
	Smaller Outer Hub Diameter	mm	1227
	Larger Outer Hub Diameter	mm	1245
	Flange Thickness	mm	130.2
	Total Flange Length	mm	311.2
	Pipe/Shell thickness	mm	19.05
Gasket	Gasket Outer Diameter	mm	1286
	Gasket Inner Diameter	mm	1235
	Gasket Thickness	mm	4.45
	Gasket Width	mm	25.4

The applied boundary conditions consist of internal pressure (P) and internal temperature (T) both applied to the inner surface of the flange. The clamping force is replaced by uniform distributed compressive stress (σ_c) and axial end load is represented by σ_a as shown in Figure 4. Contact surface is assumed between the flange and the gasket and the displacement of the gasket is restricted in the vertical direction.

Values given with strike in Table 3 are the default values. Unless otherwise stated, when changing a variable like internal pressure, other parameters are held constant to the strike value. The clamping pressure is varied from a minimum value of 20 MPa to a maximum value of 50 MPa., value less than 20MPa resulted in unstable system due to lack of the equilibrium at 50MPa axial pressure. Clamping pressure is assumed to be uniform throughout the effective clamping area and obtained by using 40 bolts. Clamping pressure of 20 MPa is corresponding to 12.5% of the proof strength for ASTM 193 B7 bolt materials. Internal pressures are varied from zero applied pressure to value below yielding of the internal surfaces of the flange. The axial stress is the end effect due to the presence of cover to the piping system like blind flange. Temperature of the transported fluid sensed at the inner surface of the flange is changed from room temperature to a high value of 500^{o}C.

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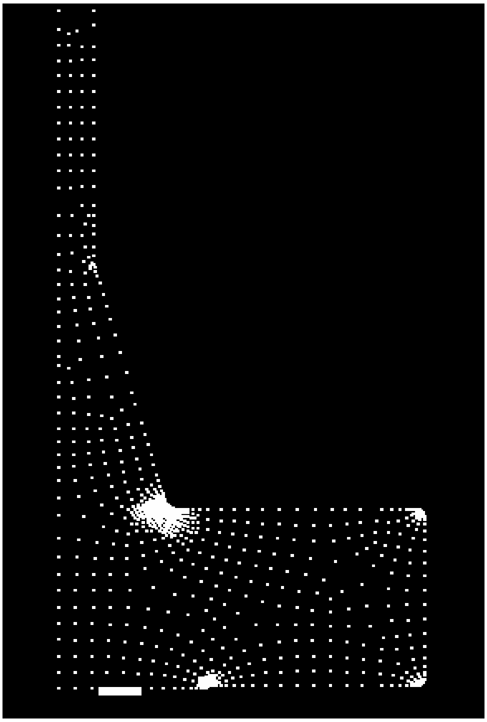

Figure 3: The Finite Element Mesh Shown by Nodes

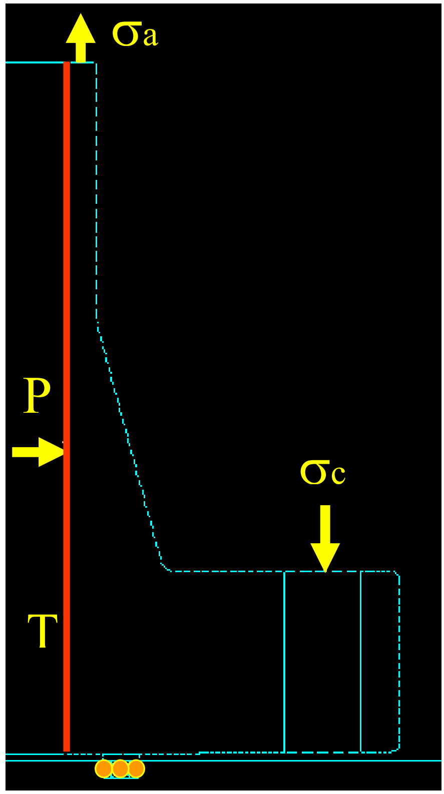

Figure 4: Applied Loadings and Boundary Conditions

The gasket modulus of elasticity is changed from low value for polymer and asbestos to high values for aluminum and steel. Vertical displacement is restricted at the lower surface of the gasket. Nonlinear FEA solution is assumed with elastic perfectly plastic material response for the flange.

3. RESULTS AND DISCUSSIONS

Obtained numerical results are shown in Figures 5-12. Figure 5 shows the equivalent Von-Mises stress under default loading conditions, i.e. internal pressure, IP=3MPa, clamping pressure, CP=30 MPA , axial pressure, AP=50 MPa, temperature, T=25 C^o and gasket modulus of elasticity, E_g=5GPa. The minimum stress value is 2.4 MPa and the maximum one is 254 MPa which is the yield limit.

Maximum value occurs at stress concentration area which is a localized value. The top side of the flange sees stress value in the range of 114 to 170 MPa; such value is acceptable for the given loading condition. The rim of the flange undergoes less stress due to its bulkiness when compared to the tube shell, while inner areas in the rim are exposed to less stress. Looking at the gasket, one can see the gradual change in the stress from left to right. The right fibers of the gasket are exposed to more stress due to flange rotation in the clockwise direction. This rotation caused mainly by clamping forces and to less extents the internal pressure. Stress distributions at the gasket contact surface are shown in Figures 8-12.

Table 3: The Parametric Study Values.

Clamping Pressure, CP (MPa)	Axial Pressure, AP (MPa)	Internal Pressure, IP (MPa)	Internal Temperature, T (C^o)	Gasket Modulus of Elasticity, E_g (GPa)
20	0	0	25*	1
25	10	1	50	2
30*	20	2	100	5*
35	30	3*	200	10
40	40	4	300	80
50	50*	5	500	200

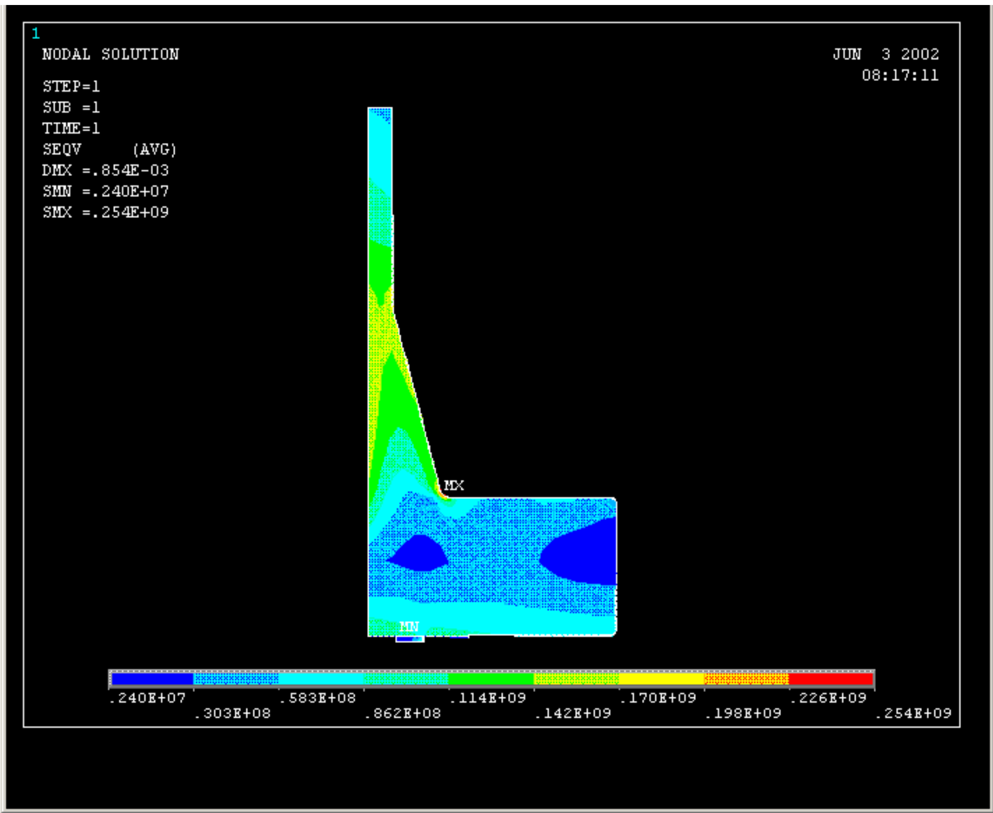

Figure 5: Equivalent Von-Mises Stress for the Flange-Gasket Connection under Default Loading Conditions (see Table 3).

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View of the Von-Mises stress contours at the gasket is shown in Figure 6. The assumed modulus of elasticity in this image is 80GPa corresponding to aluminum value. Such assumption results in higher stress at the right side of the gasket but with no contact pressure on most of the gasket. Details of the contact pressure at different gasket materials will be shown later. Another image of Von-Mises stress is shown in Figure 7 under 40 MPa clamping pressure while other defaults loads are the same. Yielding of the rim-hub joint area is very clear and the overall stress distribution is the same as before except the high intensity due to high clamping pressure.

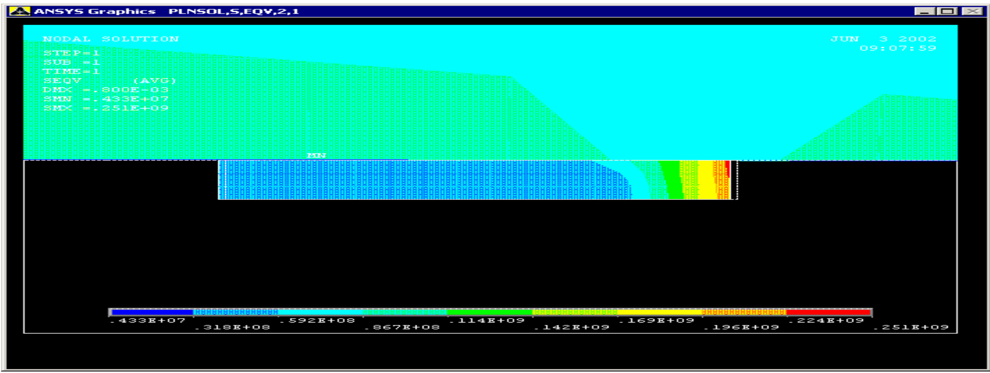

Figure 6: Von-Mises Stress at the Gasket for Gasket with 80GPa Elasticity Modulus.

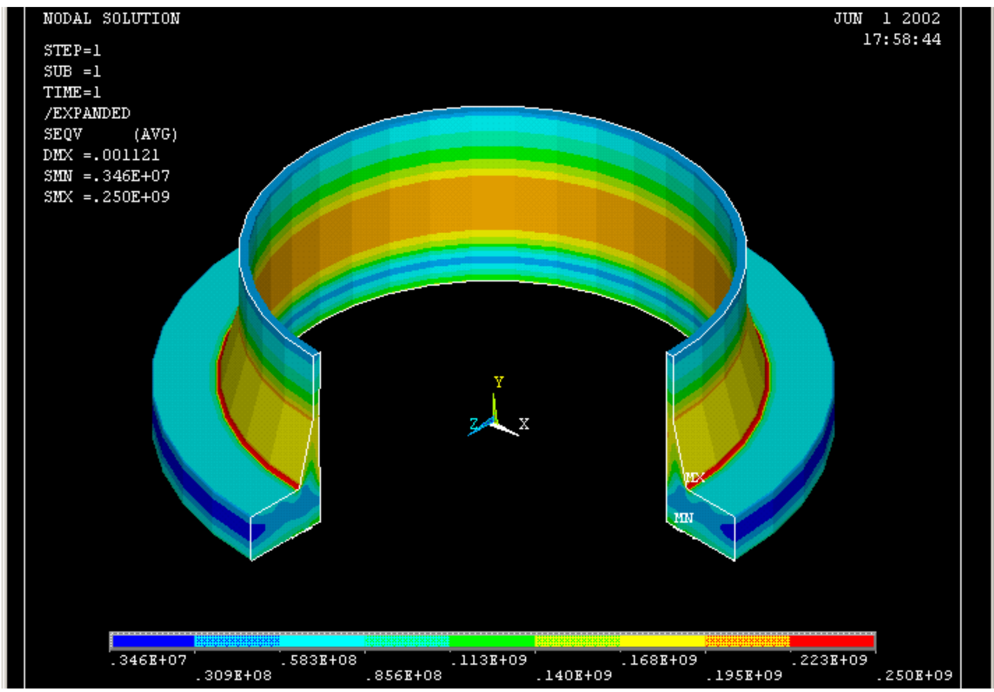

Figure 7: Equivalent Von-Mises Stress for the Flange-Gasket Connection under 40 MPa Clamping Pressure, All other loadings are the Default Loads.

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3.1 Effect of clamping pressure

Clamping pressure can be varied by bolt tightening or loosening. The limiting factors for such a variable is the bolt strength and bending and bearing stresses of the flange. The direct applied clamping pressure is assumed to be uniformly distributed throughout the annular area passing by bolt circle diameter at a width equals to bolt diameter. Due to the rigid-body rotation of the flange, the contact stress is maximum at the right hand side of the gasket and gradually decreases from right to left. Results of clamping pressure effect on the contact pressure are shown in Figure 8. The vertical axis gives the contact pressure at the gasket surface in MPa while gasket width is given along the x-axis. As expected, by increasing the clamping pressure, the contact stress increases too and the gap area decreases. At 20 MPa clamping pressure, corresponding to 12.5% of the bolt proof strength, gap cover 60% of the gasket area and the contact pressure varies from zero to 47 MPa. By increasing the clamping pressure to 50MPa, or 31% of the proof strength, contact pressure jumps to maximum value of 241MPa and the gap reduces to 15% of the contact area.

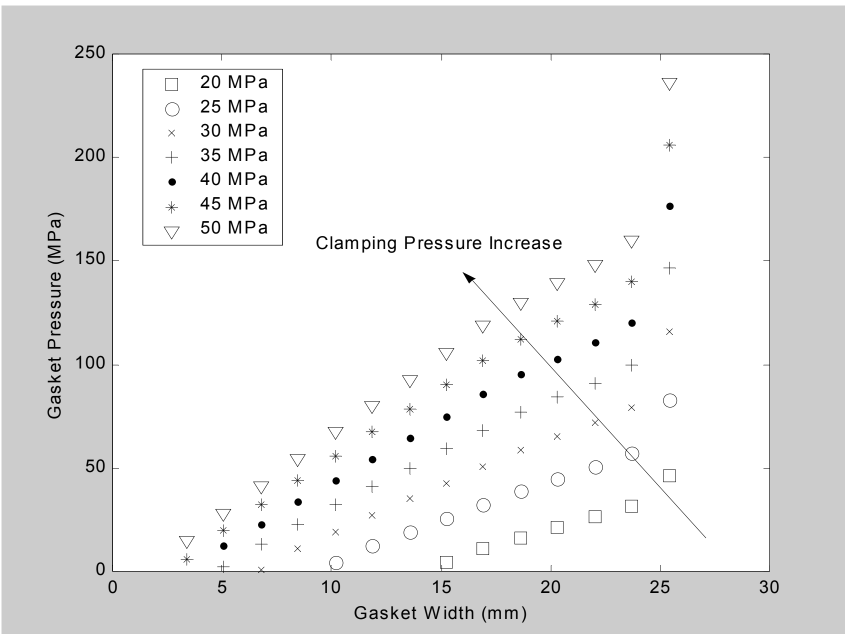

Figure 8: Effect of Clamping Pressure on Gasket Contact Pressure.

3.2 Effect of axial stress

Axial stress results from the presence of nearby cover to the piping system. The cover can be a blind flange, or a t-section. Axial or longitudinal stress is a direct function of the internal pressure, however, axial stress is separated here, to simulate the condition of having short (with nearby blind flange) in-plant piping system from long 100s km length pipelines. Leakage of bolted joint due to axial load or external load is a typical problem in piping engineering.

Figure 9 shows the contact pressure verse the gasket width at various axial stresses. Axial stress of 10MPa will results from internal pressure of 627kPa (6.2 bar) with 100% end cover efficiency. As expected, the increase in the longitudinal stress will reduce the contact pressure, increase the gap, start leakage and loss of the bolted joint efficiency.

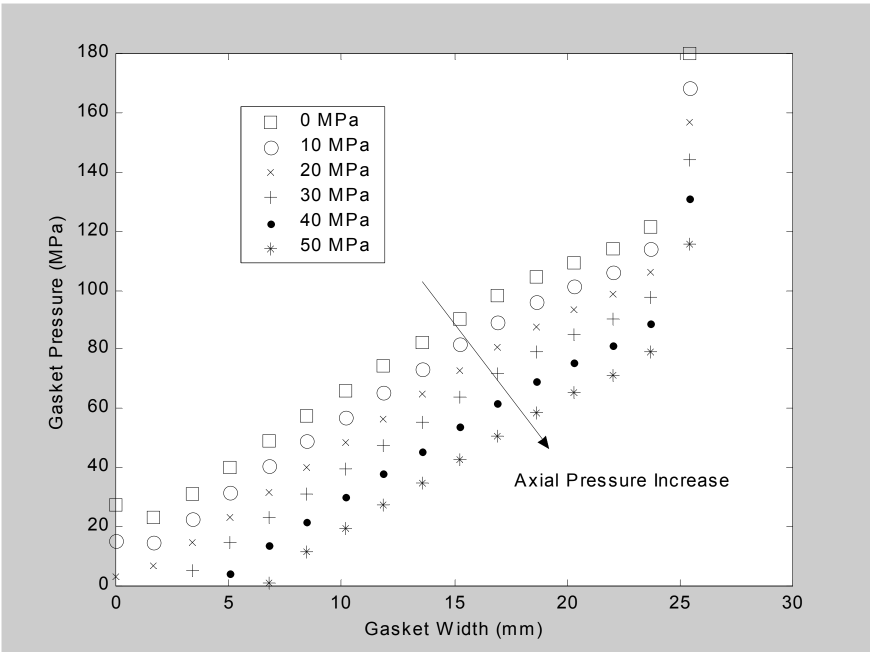

Figure 9: Effect of Axial Stress on the Gasket Contact Pressure.

3.3 Effect of gasket stiffness

It is very well known in flanged joint that gasket has to be soft and durable. In Figure 10 the gasket modulus of elasticity is changed while fixing all other variables to the default values. For the sake of easy comparison, gasket contact pressure is plotted in log scale against linear gasket width. Gasket modulus E_g is changed for low value of 1 GPa, corresponding to soft industrial rubber or polymer, to steel. Gap starts at E_g=5GPa, corresponding to asbestos, and reach maximum values at E_g=80 and 200GPa, corresponding to aluminum and steel, respectively. The increase in the contact pressure at high gasket stiffness values is obtained at high price of loosing the contact, i.e. gap expansion. The uniformity of the gasket contact pressure is obtained by softening the gasket as the case of E_g=1 GPa and E_g=2 GPa. Also, high stiffness gasket results in low or even zero contact pressure, see the curve corresponding to E_g=5GPa. Linking leakage occurrence to low contact pressure is not an easy task because analytically contact pressure will be there and in abounded value far more than the internal pressure. Leakage has to be linked with real experimentation setup.

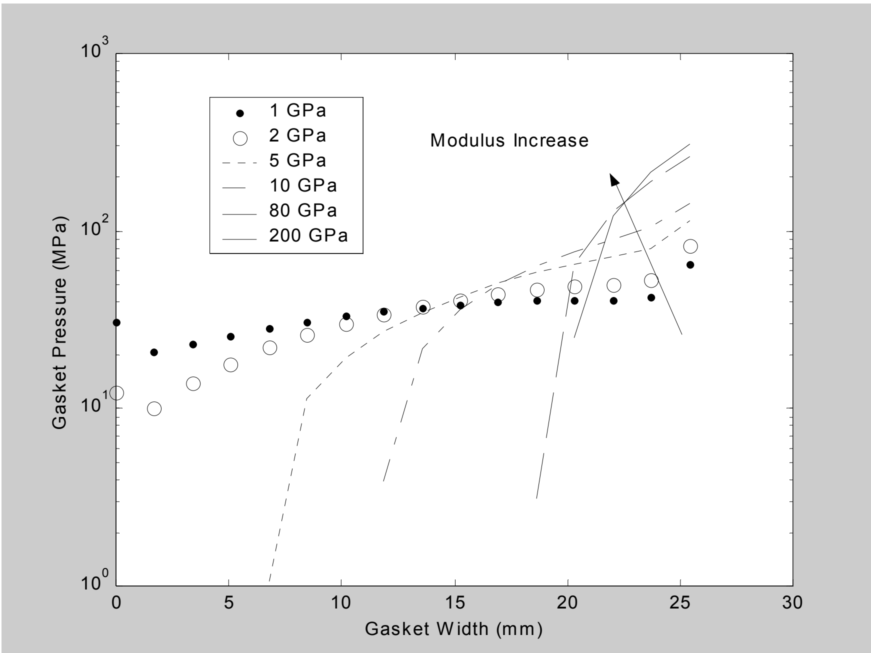

Figure 10: Effect of Gasket Stiffness on the Gasket Contact Pressure.

3.4 Effect of internal pressure and temperature

Effect of internal pressure, at constant longitudinal stress, is shown in Figure 11. The contact pressure between the gasket and the flange is plotted verse gasket width for different internal pressure values. One can see negligible effect for the increase in the internal pressure on the contact pressure. Internal pressure increase tends to rotate the flange clockwise at slow rate (stiff rotation). Thus slight increase in the contact pressure at the outer part of the gasket and slight reduction in the contact pressure at the inner side. Similar things can be said to the temperature effect shown in Figure 12. Increase in temperature cause rotation of the flange due to different thermal expansion rate of the flange as the thickness changes in the vertical direction.

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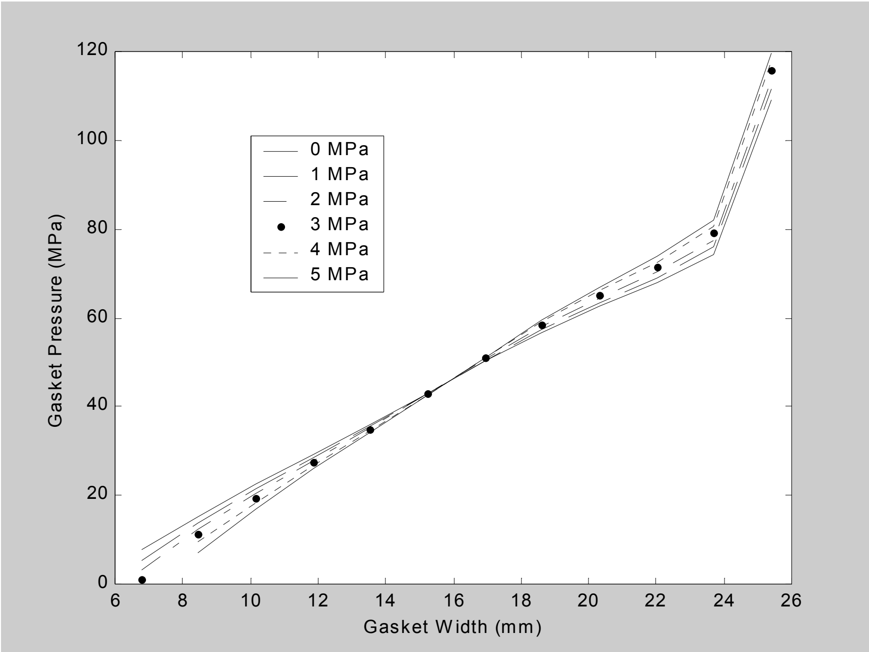

Figure 11: Effect of Internal Pressure on the Gasket Contact Pressure while Neglecting End Effect

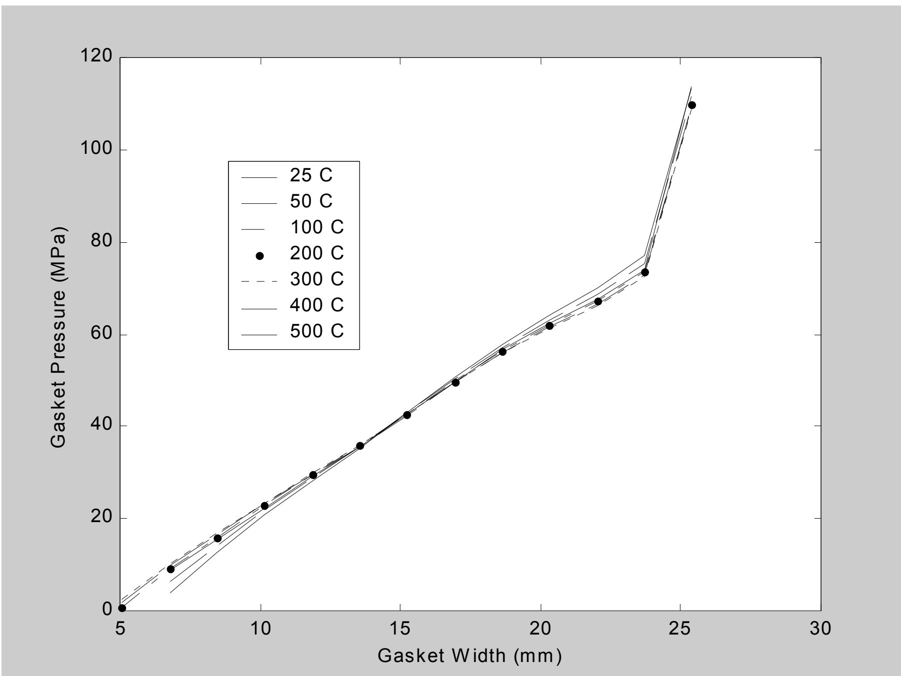

Figure 12: Effect of Fluid Temperature on the Gasket Contact Pressure

4. CONCLUSIONS AND RECOMMENDATIONS

This paper investigated the operational parameters affecting the flange-gasket assembly for large diameter steel flanges. Clamping pressure needs to be carefully selected to get proper sealing of the flange-gasket assembly. Increasing the clamping pressure will result in better contact pressure but at the cost of higher flange stress. Gasket has to be made of soft material with low modulus of elasticity to ensure better sealing of the assembly. Axial end load may result in gasket leakage if the clamping pressure is not sufficient. Internal pressure and temperature were found to be less important to the flange-gasket assembly.

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