Numerical Investigation of Safe Tightening For Flange In Oil Pipe Line

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Abstract- *Rise in the population is increasing day by day with the increase in population Transportation and energy needs in the society is also increasing to sufficiently provide fuel to satisfy the demand more and more oil pipelines is installed every day one of the important aspects in any pipeline is to connect the couplings between Pipes with Flanges in most of the cases the flange failure or rotation occurs with inappropriate tightening or more tightening the project will be done using different ratios of bolt Pretension loads and find out what kind of stresses is occurring in the flange that leads to flange failure, 5 tightening percentages are used in this project to see the optimal tightening vs gasket compression with occurring flange stress Gasket thickness of 1m and 2mm is assumed for full compression in Spiral wound gasket All the flanges designed in the project will comply with ASME B16.5 standard. 3d modeling is done using Solidworks and The simulation at multiple bolt loads is done in Ansys.*

Keywords- Tightening compression thickness, Bolt load , Flange, FEA

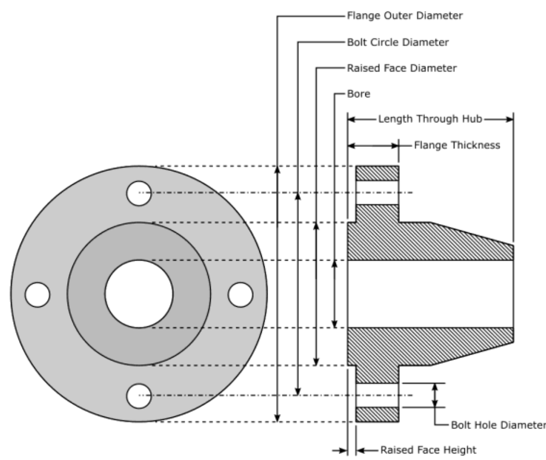
I. INTRODUCTION

Flanges are parted into two primary regions, the 'cutting edge', and the 'center point'.

- The flange cutting edge envelops the region where the bolts infiltrate through the flange and the flange face.
- The flange center is the region that obliges the line which connects to the flange.

To guarantee no leaking between the mating2 flanges happens, gaskets are utilized. It is feasible to mate two metal flanges together without the utilization of gaskets, yet fixing is troublesome and must be accomplished with uncommonly planned flanges.

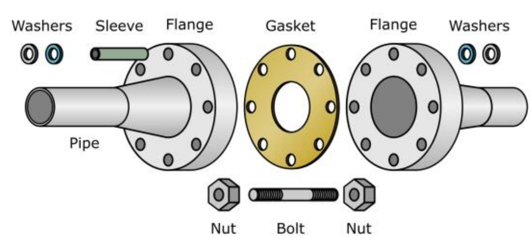
The end association determines how the flange is associated with its going with pipe (strung association or welded).



Flange Design

1.1 How Flange Work

A flange is created when two opposing surfaces are intentionally pressed together in order to create a leak tight seal. To obtain a seal, force must be applied and maintained to each of the opposing flange faces. As many flange faces have manufacturing imperfections (scratches, dents, pits etc.), it is necessary to put a softer material between the two mating sealing surfaces to obtain the seal; this softer material is the gasket.



Flange Assembly

1.2 Basic Flange Math

To see how flanges work, we should initially comprehend the idea of tension. Pressure is characterized as:

Pressure = Force/Area

P = F/A

Flanges seal since pressure is applied to the mating fixing surfaces; this tension is known as the 'gasket pressure' or 'fixing pressure'. The applied tension makes the two faces by the same token:

- Pulverize a gasket between the two mating faces.
- Press the two mating faces against one another.

In the gasket model, the gasket is distorted because of the strain applied; this disfigurement makes the gasket 'stream' into any surface flaws that might be available on either fixing face. Since the surface blemishes have been filled by the gasket material, leakage is at this point not conceivable.

The subsequent model accepts no gasket is available and that two flange faces are squeezed together. It is difficult to make a leak tight seal utilizing this technique, in spite of the fact that it is conceivable assuming the surfaces are all around machined and extremely spotless. The fixing pressure applied will frequently should be huge, as the flange surface might be produced from metal, which doesn't effortlessly misshape under tension (material and flange class subordinate). Metal to metal flange face fixing is costly and accordingly not normal.

To make the essential fixing pressure, the factors of power and region can be changed.

- Power alludes to the fixing force (shooting load) applied to the mating flange faces when the nuts on a flange get together are fixed. Power (F) relies on the force (T) applied, force grinding (K) and ostensible bolt measurement (D). The power depicted is classed as 'bolt assumption' or 'bolt preload', or 'bolt prestress', and is addressed by the situation F = T/(KD)
- Region alludes to the size of the fixing face region.

How much tension on the flange fixing faces relates to how much power applied while fixing the flange get together. Consequently it is feasible to control the strain by changing how much exertion that is applied while fixing the bolts during flange gathering.

The fixing region of a flange can't be just about as effectively changed as the power utilized during get together. A bigger fixing face requires more power to get a specific measure of tension, contrasted with while utilizing a more modest fixing face. The underneath model features this point, however without the utilization of units.

II. LITERATURE REVIEW

G. Mathan[1] led probes gasketed flange joint and broke down twisting burdens in flange joints through FEA and results were contrasted and test review.

Murali Krishna et.al [2] concentrated on the fixing execution of darted flange joints with gaskets utilizing limited component investigation; three layered limited component examination (FEA) of catapulted flange joint is done. Analysis shows that the dissemination of contact pressure has prevailing impact on fixing execution and gaskets assume a significant part in the fixing execution of shot flange joints.

M.Abid [3]discussed on parametric review of behaviour of flanges with various surface profiles. Stresses in flanges and bolts are acquired by differing flange thickness, tighten point and bolt prestress however keeping the flange aspects (center length and center point thickness) steady. He presumes that, to accomplish 'no-leak' condition from a flange joint, a flange having positive shape point, curved center, and thickness atleast multiple times the bolt measurement is required.

Vishwanath V. H [4] examined investigation of Bolted flange associations by executing the plan technique for gasketed blasted flanged associations according to ASME Boiler and Pressure Vessel Code and approved the outcomes with limited component examination programming ANSYS. Hub, outspread and distracting stresses are gotten by fluctuating flange thickness from 44.4 mm to 55 mm; bolt preloads are changed from 35 % of yield solidarity to 75% in sync of 5% and to acquire uniform pressure quantities of bolts are shifted from 6, 8 and 12 bolts.

Toshiyuki [5] talked about the qualities of a line flange association with a twisting injury gasket. The contact pressure conveyance and the new gasket constants are gotten by utilizing the axi-balanced hypothesis of flexibility and the limited component investigation considering a hysteresis and non-linearity of the gasket. The trials were likewise led to quantify a measure of gas leakage and a variety in hub bolt power, in outcomes it was viewed that contact pressure appropriation tends as uniform as Young's modulus of the gasket diminishes and the gasket thickness increments. stress dispersions in the line flange associations were examined by utilizing FEM.

Engines are generally utilized as main players as they give uniform movement and control of speed and bearing when contrasted with engines.[6] Flange mounted engine is an engine where flange is darted to divider on which entire engine is mounted. With the goal that whole weight of an engine follows up on flange. BhalePritish[6] examined with regards to static and dynamic examination of engine flange utilizing FEA. The accompanying outcomes were found,The most extreme burdens are created at the bearing area and greatest distortion happens toward the finish of the flange where bolt is to be applied to the engine assembly[6].

P. M. Desai [7] examined on plan, investigation and streamlining of body flange and cover flange by utilizing FEM approach and its approval by logical according to ASME Code. Mathematical Simulation methods are utilized for investigation of ring type flange. The consequence of mathematical reenactment conquers the limit of scientific methodology which is seen from the aftereffects of proposed model. The ideal Value of thickness of Cover Flange and Body Flange are 48 mm and 90 mm individually.

SauravRajgadia, et al [8] The investigation of present paper is to diminish the greatest shear pressure by picking an appropriate material implied for the flange coupling. Therefore, demonstrating of the flange coupling is completed in Solid works. What's more dissect in ansys work seat programming. Kondru

Nagendra Babu, et al [9] the current investigation of this paper is to decrease the anxieties that are following up on the coupling bolts by giving the equivalent strength on it. At any rate the knife segment the bolt is more modest in size than the strung piece of the bolt hence the strung part will take higher burdens which will brings about disappointment.

Somvir Singh, et al [10] now a days the each assembling industry has a test to build the solidarity to weight proportion of the part. In the current review the specialist objective is to diminish the heaviness of the flange coupling. What's more for this work the analyst picked the composite material of aluminum silicon carbide for the flange coupling since this material has lighter thickness. The reason for this paper is to decrease the heaviness of the flange coupling without influencing the presentation and furthermore to limit the expense of the material.

III. METHODOLOGY

3.0 Introduction

This Chapter deals with design and technical specification's considered for Flange design of the pipeline.

3.1 Description and Inputs

The specifications of the Pipe line rages from fluid flowing in the Range of 15 to 55 C and the Fluid pressure in the pipe line Ranges from 20 Bar to 60 Bar where the pretensions of the bolt varies depending on the tightening torque provided and gasket stiffness

Table 3.1 Specifications Required to Design a Flange is shown below

S.No	Parameter	Value	Unit
1	End Connection	Pipe	
2	Rate Pressure	20 to 60	Bar
3	Material	ASTM A182 grade F51	
4	Material Yield	275	MPa
5	Poisson's Ratio	0.3	
6	Pipe Diameter	50	mm
7	Pipe OD	84	mm
8	Factor of Safety	2	

The above table shows the technical specifications of the Piping system selected to design and simulate a selected flange , most of the companies rely on the ASME Code to select and validate the Flanges there are mainly 6 types

1. Weld Neck Flange.
2. Slip-on Flange.
3. Socket Weld Flange.
4. Lap Joint Flange.
5. Threaded Flange.
6. Blind Flange.

Out of all the Flanges Weld Neck flange is selected of the Pipe system because of Durability and ease of use Depending on the Standard sizes available in the ASME B16.5 a suitable size is selected to make a 3d model of the Flange.

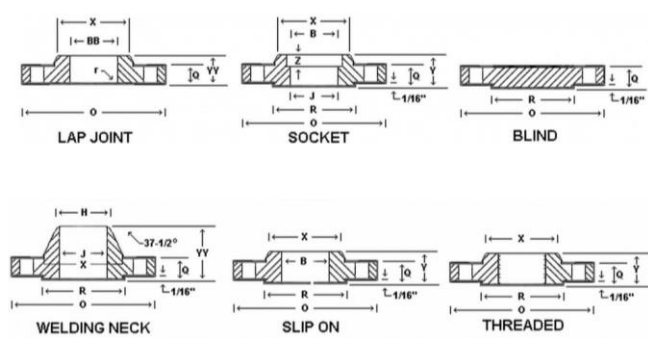


Fig 3.1 Flange Illustration

3.2 Leak tightness and Gasket selection

Leak tightness is a very important factor in any fluid transport mechanisms to avoid any catastrophic failure in the whole system select of gasket is very important to place between flanges, to avoid any leakage there are different types of flange faces and compatible gaskets available. Some of them are discussed below

Types of Flange Faces:

1. Flat Face.
2. Raised Face.
3. Lap Joint.
4. Ring Joint.
5. Male and Female.
6. Tongue and groove.

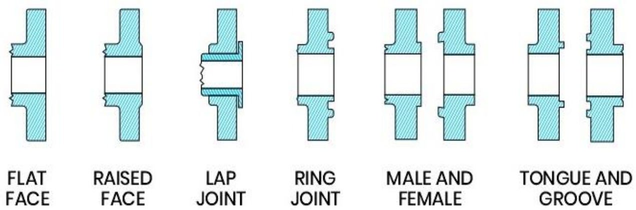


Fig 3.2 Flange Faces for gasket placement.

i. Flange Dimensions Finalised for Flange.

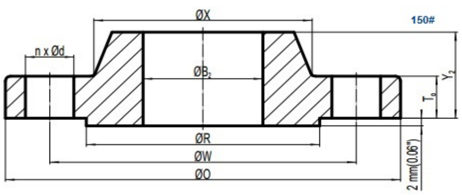


Fig 3.2 Flange Illustration for design

Table 3.2 Flange Specifications				
S.NO	**Parameter**	**Pipe side**	**Pump Side**	**Units**
1	NPS	2”	2”	
2	X	84	84	mm
3	W	127	127	mm
4	O	165	165	mm
5	To	25.4	25.4	mm
6	B2	61.9	61.9	mm
7	n	8	8	
8	d	19.05	19.05	mm
9	R	92.1	92.1	mm
10	do	15.875	15.875	mm
11	Lsrf	110	110	mm

Since the Standard dimension is selected from ASME B16.5 the Calculation for Thickness is necessary for selected material to find the minimum thickness required for both Hull side and system side flange following is the Formulae considered for calculation.

Material selection is one of the most important aspect in the flange design to rate the Pressure and temperature class the required specifications by the pipe line is 0 to 55 C and Pressure required is 60 bar to achieve safe design the pressure class selected is 100 Bar pressure.

➢ **Minimum Thickness Required for Flange.**

$$T_o = w\sqrt{\frac{3(3+\mu)P}{8f}}$$

Where **w**= Radius of PCD= 127 mm= 42 mm
μ= Poisson’s Ratio= 0.3
P= Pressure Acting= 0.612 Kgf/mm2
F= Allowable Stress= 22.9436 kgf/mm2
To=Thickness of Flange mm

$$T_o = \sqrt{\frac{3(3+0.3)0.612}{8\times 22.9436}} = \mathbf{14.9596\ mm.}$$

The selected flange thickness T_o is **25.4 mm** for Pump side flange where as the Minimum Thickness required for Factor of safety 2 with ASTM A182 grade F51 is **14.9596 mm** for a strength test of 60 Bar.

➢ **Hoop Stress Calculation for Shell**

From Hoop stress we can calculate the minimum thickness required by using the formula.

$$f = \frac{P\times D_{max}}{2\times t}$$

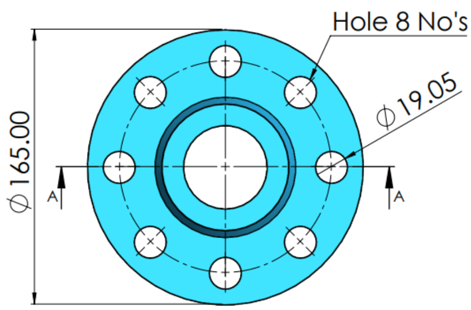
Where ,
$f = Allowable\ Stress = \mathbf{225\ MPa.}$
$P = Pressure\ Acting = 100\ Bar = \mathbf{6\ MPa.}$
$D_{max} = Maximum\ internal\ Diameter\ in\ the\ Shell = \mathbf{50\ mm.}$
$t_p = \frac{P\times D_{max}}{2\times f} = \frac{6\times 50}{2\times 225} = \mathbf{0.66667\ mm.}$
So the minimum required Shell thickness of the valve $t_p = \mathbf{0.66667\ mm.}$

Therefore the Shell outer diameter required is **X**= **50+0.66667** = **50.66667 mm < 84 mm.**

In general the Thickness of this kind of Pipe line is mostly localised depending on the Shape and Structure required at different sections of the Pipe maximum Diameter assumed in the shell is 50 mm therefore the outer shell diameter required at this section is 50.6667mm to maintain minimum stress at **60 bar** Pressure with a factor of safety of 2.

3.3 Modelling of the Flange using Solid works.

Dimensions catered from the table 3.2 is given the dimension which is used to model a 3d model for the simulation later to be done in ansys there are few steps to be followed in the software , Initially a sketch is drawn with the dimension designated as ‘O’ in table 3.2 the outer diameter of the flange to extrude the Outer dia with thickness 25.2 mm.



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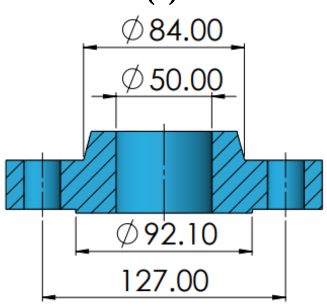
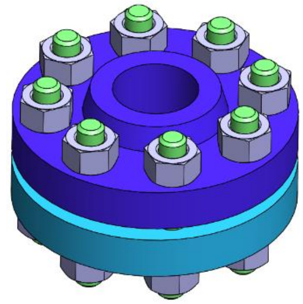


(b)

Fig 3.3 Flange Dimensions Designated and Designed.

3.4 Assembly and construction of two flanges

The Flange designed is assembled using solid works assembly module in which each part is mated together using mates option initially two flanges is made concentric using concentric command in between two bores of the two flanges assembled there after bolt holes has been mad concentric to align the two flanges later to tighten using bolts.



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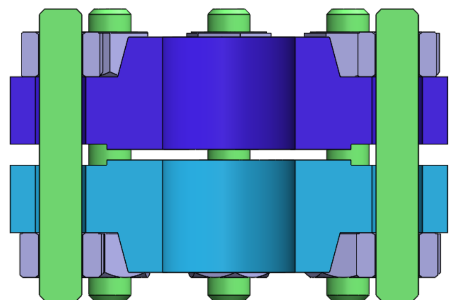

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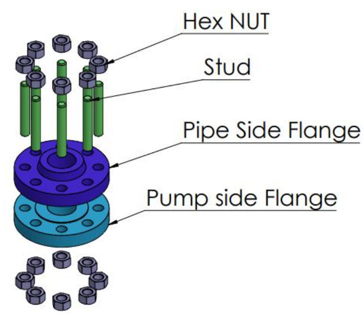


(c)

Fig 3.4 Flange Assembly
(a) Isometric View (b) Section View (c) Exploded View

3.4 Gasket Selection and Bolt Load Calculation.

The Gasket is the most important part in the flange assembly where a spiral wound gasket is selected to place in between the flanges the inner diameter of the gasket is 69 mm and the outer diameter of the gasket is 85.9 mm.

Bolt load is the Very important specification to effectively tightens the flanges.

$$Area\ of\ the\ Gasket = \pi \times (r_1{}^2 - r_2{}^2) = \pi \times \left(\left(\frac{85.9}{2}\right)^2 - \left(\frac{69}{2}\right)^2\right) = 2056.023\ mm^2.$$

Maximum Stress Possible for Gasket Compression S=300 MPa.
Bolt Diameter =M20.
Bolt Radius=9.515 mm.

$Root\ Area\ A_r = \pi \times r^2 = \pi \times 9.515^2 = 284.4247\ mm^2.$

$Total\ Gasket\ Load\ L_g = S \times A_r = 300 \times 2056.023 = 616806.9498\ N$

No. Of bolts=8.

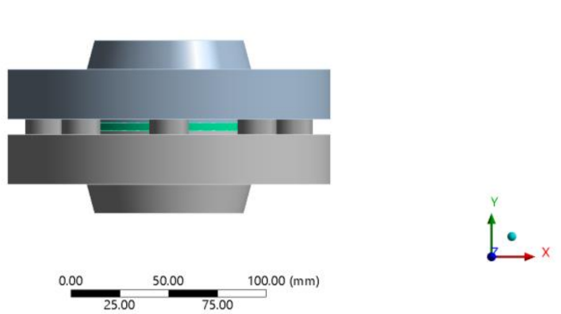
$$Force\ on\ Each\ Bolt = \frac{616806.9498}{8} = 77100.86873\ N$$

Table 3.4 Bolt Load at different Tightening Factor.

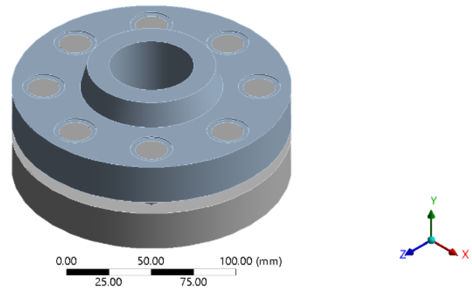
Considered Stress	Total Gasket Load	Bolt Load	Tightening %
300	616806.9498	77100.86873	100
270	555126.2549	69390.78186	90
240	493445.5599	61680.69498	80
210	431764.8649	53970.60811	70
180	370084.1699	46260.52124	60
150	308403.4749	38550.43436	50
120	246722.7799	30840.34749	40
90	185042.085	23130.26062	30
60	123361.39	15420.17375	20
30	61680.69498	7710.086873	10

3.4 Ansys Methodology

A 3D model is designed using solid works and is converted into workable file format STEP, Then imported to ansys workbench module design modeller to do pre processing step Where all the Cylindrical faces is merged together to get smooth meshing studs and bolts are suppressed and the materials is assigned to each individual part as specified. A frictional contact is defined in between gasket and Pump Flange and Pipe flange using Ansys Connections. To mimic the bolt joint a beam connection with appropriate bolt radius is defined as the connection.



(a)



(b)

Fig 3.5 Imported CAD geometry and Beam Connections.
(a) Imported CAD model (b) Beam Connections.

3.5 Beam Connections

Nuts and Bolts are suppressed in the Model to modify the FEA model to simplify the CAD model instead of nuts and bolts a face to face beam connection is given with a beam radius of 10mm material for bolts considered is structural steel. And a rigid type behaviour is assumed at the bolts beam connections.

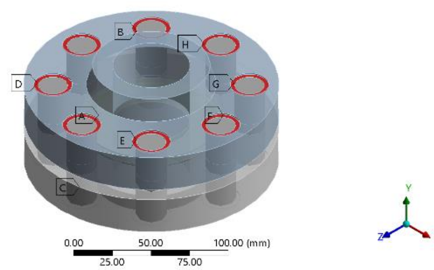


Fig 3.6 Beam Connections selected

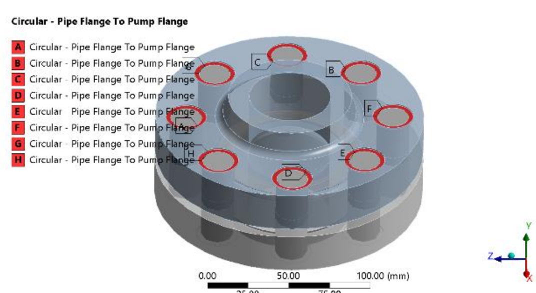


Fig 3.7 Beam Connections with designations.

3.6 Meshing

Meshing is done using Ansys workbench an automatic method is used to elaborate the element depending on the shape and size Local element sizing is done with 5 mm elements.

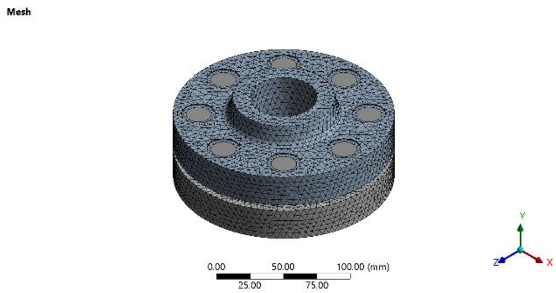


Fig 3.8 Meshing of the flanges.

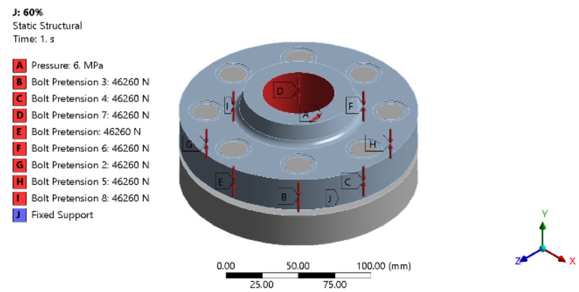


Fig 4.1 Boundary Conditions applied

IV. RESULTS

4.0 Introduction

This chapter represents the Results obtained from static structural analysis done for different Flange configurations at different tightening percentage from the literature is is known that the minimum required stress for leakproofness is 50 Mpa but the manual tightening is a non linear factor to ensure safe stress at different tightening percentages for a gasket thickness of 1mm and 2mm.

Table 4.1 Case Details

S.NO	**Case**
1	100% Tightening
2	90% Tightening
3	80% Tightening
4	70% Tightening
5	60% Tightening

Boundary conditions.

Bolt Pretensions is given with respect to tightening percentage between flanges at 8 bolt location a pressures of 6 MPa is applied to gasket and flow passage faces as shown in the given picture below

S.No	Tightening %	Bolt Load (N)	Pressure (MPa)
1	100	77100.86873	6
2	90	69390.78186	6
3	80	61680.69498	6
4	70	53970.60811	6
5	60	46260.52124	6

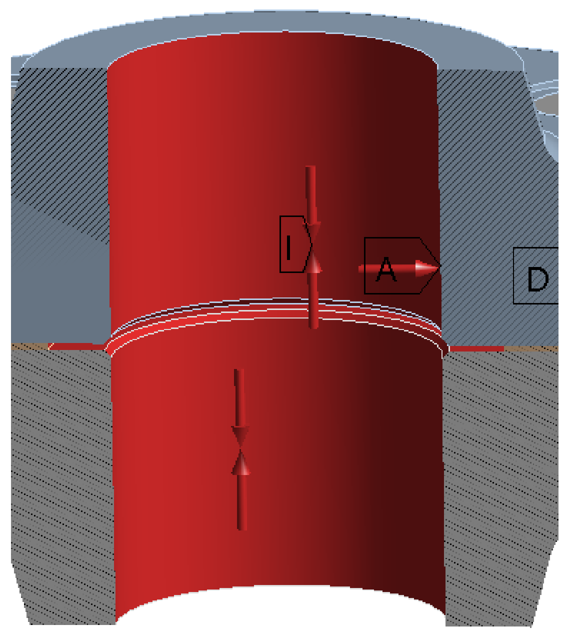
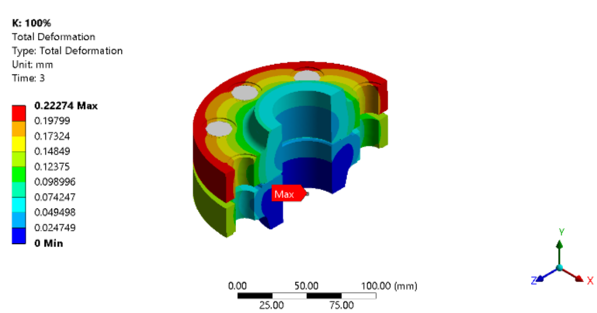
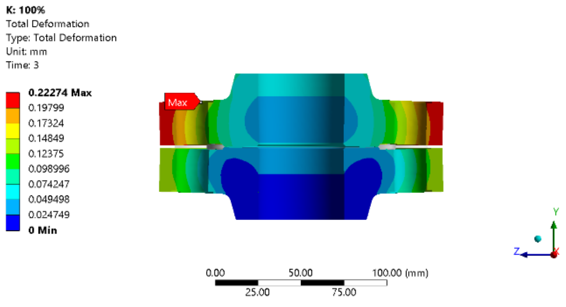


Fig 4.2 Pressure applied at Gasket area

Case 1 Case at 100% Bolt Load

The Bolt load is calculated for 100% tightening is 77100.86873 N





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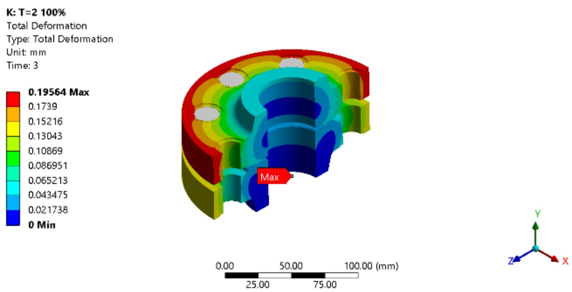
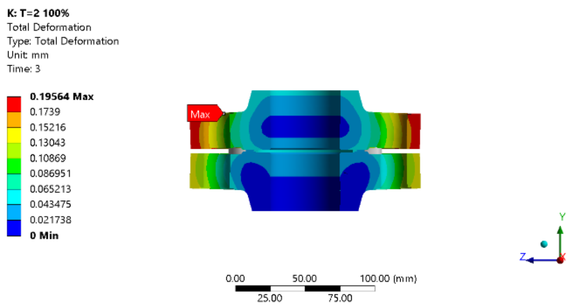
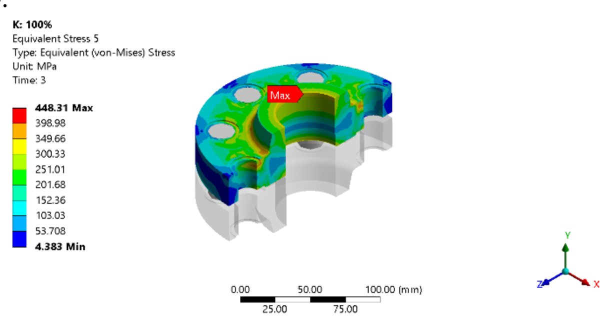




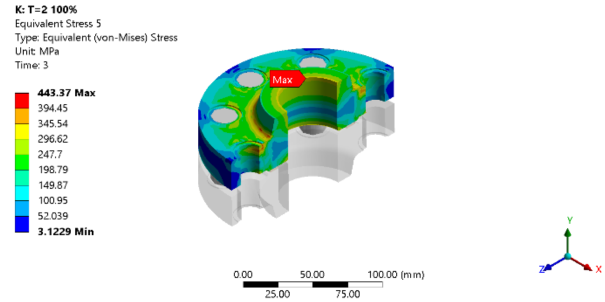
(b)

Fig 4.3 Deformation for 100% Bolt load.
(a) Deformation of 100% bolt load with 1mm gasket thickness. (b) Deformation of 100% bolt load with 2mm gasket thickness.

The above figure Represents the deformation of flanges simulated with 100% bolt load with 77100.86873 N pretension applied on beam joints as shown in Fig 3.6 for all 8 bolts and the steel gasket of 1 mm is placed in between pump side flange and pipe side flange fig 4.3 (a) represents 0.22274 mm deformation achieved with 1mm steel gasket fig 4.3 (b) represents 0.19564 mm deformation with 2mm steel gasket where red color indicates maximum deformation near flange OD.



(a)



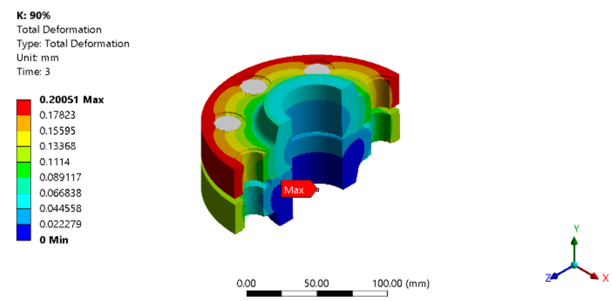
(b)

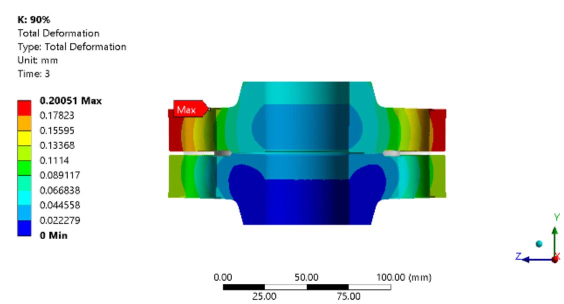
Fig 4.4 Stress generated for 100 % bolt load.
(a) Stress for 100% bolt load with 1mm gasket thickness. (b) Stress for 100% bolt load with 2mm gasket thickness.

The above figure represents the stress generated on the flange due to applied bolt load and pressure the flange with 1mm thickness have maximum stress generated on the edge at pipe connection as shown in the fig (a) which is 448.31 MPa with 2mm flange thickness the stress generated is 443.37 MPa.

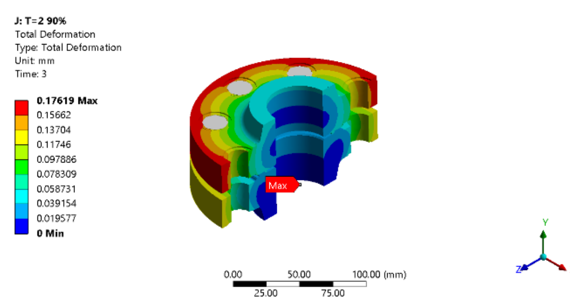
Case 2 Case at 90% Bolt Load

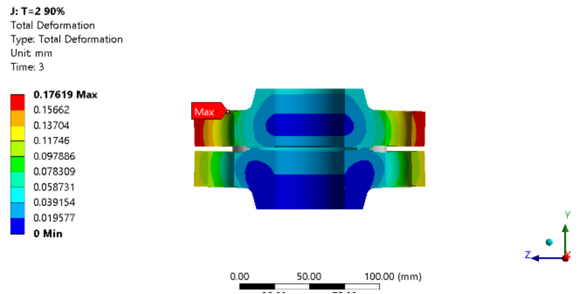
The Bolt load is calculated for 90% tightening is 69390.78186 N





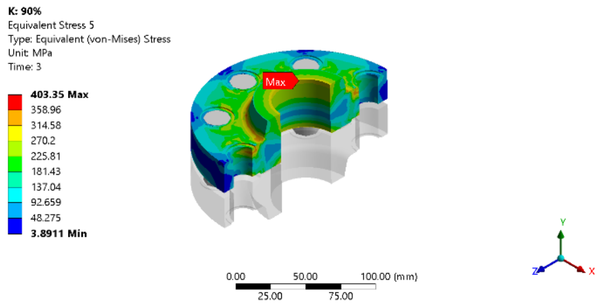
(a)



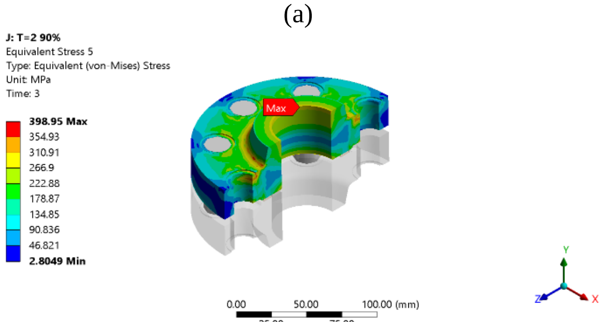


(b)Fig 4.5 Deformation for 90% Bolt load.
(b) Deformation of 90% bolt load with 1mm gasket thickness. (b) Deformation of 90% bolt load with 2mm gasket thickness.

The above figure Represents the deformation of flanges simulated with 90% bolt load with 69390.78186 N pretension applied on beam joints as shown in Fig 3.6 for all 8 bolts and the steel gasket of 1 mm is placed in between pump side flange and pipe side flange fig 4.5 (a) represents 0.20051 mm deformation achieved with 1mm steel gasket fig 4.5 (b) represents 0.17651 mm deformation with 2mm steel gasket where red color indicates maximum deformation near flange OD.



(a)



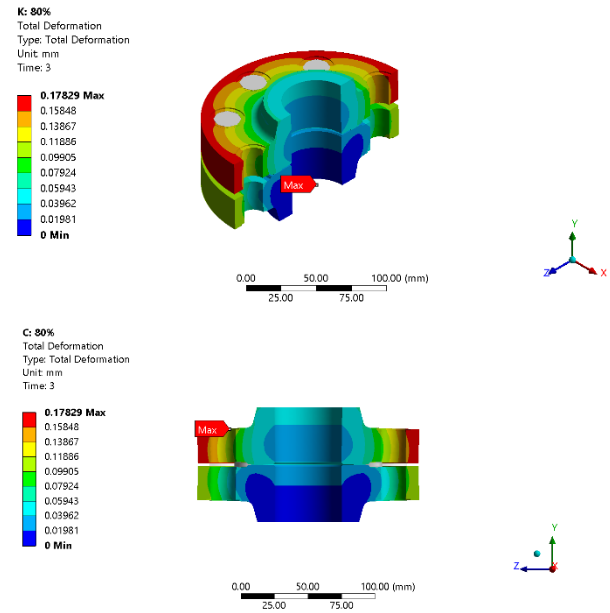
(b)

Fig 4.6 Stress generated for 90 % bolt load.
(b) Stress for 90% bolt load with 1mm gasket thickness. (b) Stress for 90% bolt load with 2mm gasket thickness.

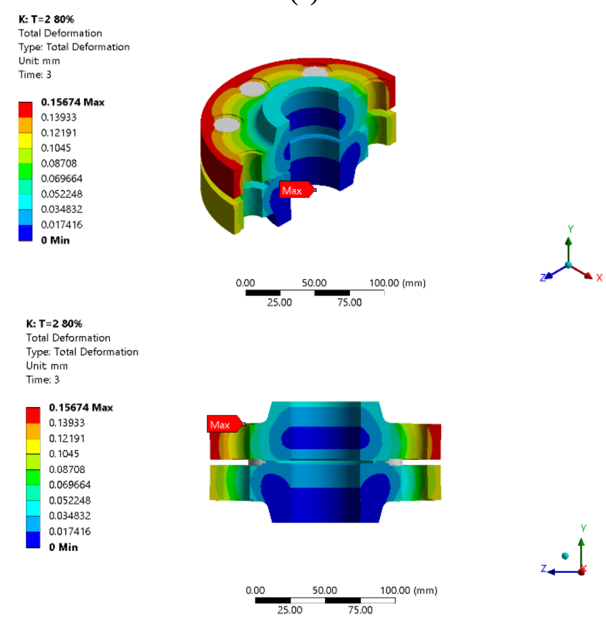
The above figure represents the stress generated on the flange due to applied bolt load and pressure the flange with 1mm thickness have maximum stress generated on the edge at pipe connection as shown in the fig (a) which is 403.35 MPa with 2mm flange thickness the stress generated is 398.95 MPa.

Case 3 Case at 80% Bolt Load

The Bolt load is calculated for 80% tightening is 61680.69498 N



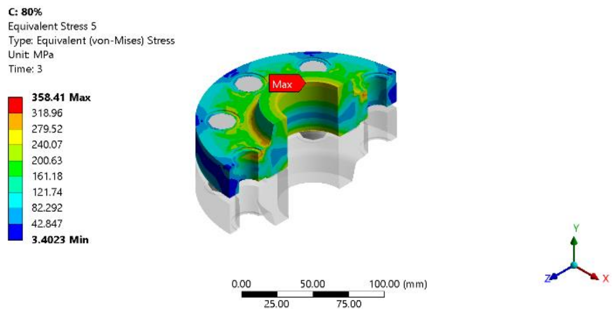
(a)



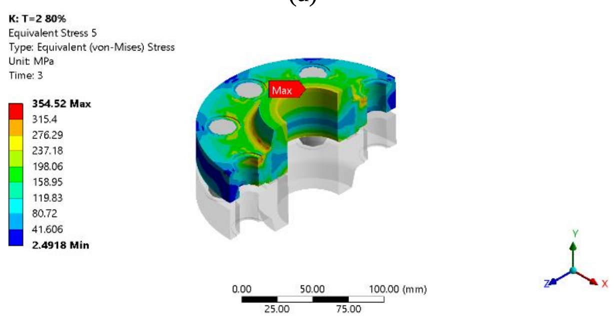
(b)

Fig 4.7 Deformation for 80% Bolt load.
(c) Deformation of 80% bolt load with 1mm gasket thickness. (b) Deformation of 80% bolt load with 2mm gasket thickness.

The above figure Represents the deformation of flanges simulated with 80% bolt load with 61680.69498 N pretension applied on beam joints as shown in Fig 3.6 for all 8 bolts and the steel gasket of 1 mm is placed in between pump side flange and pipe side flange fig 4.7 (a) represents 0.17829 mm deformation achieved with 1mm steel gasket fig 4.7 (b) represents 0.15674 mm deformation with 2mm steel gasket where red color indicates maximum deformation near flange OD.



(a)



(b)

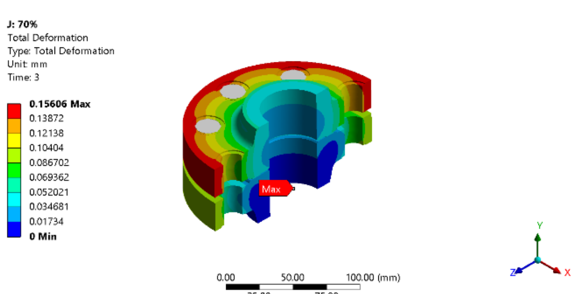
Fig 4.8 Stress generated for 80 % bolt load.

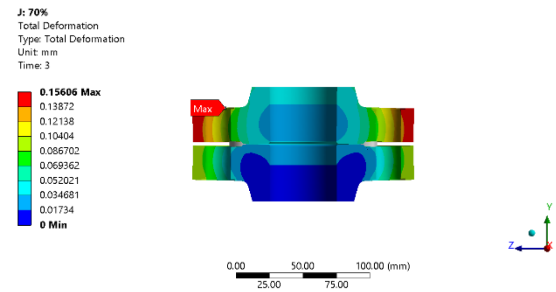
(c) Stress for 100% bolt load with 1mm gasket thickness. (b) Stress for 80% bolt load with 2mm gasket thickness.

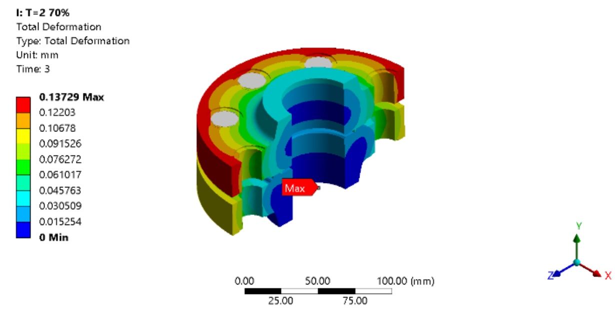
The above figure represents the stress generated on the flange due to applied bolt load and pressure the flange with 1mm thickness have maximum stress generated on the edge at pipe connection as shown in the fig (a) which is 358.41 MPa with 2mm flange thickness the stress generated is 354.52 MPa.

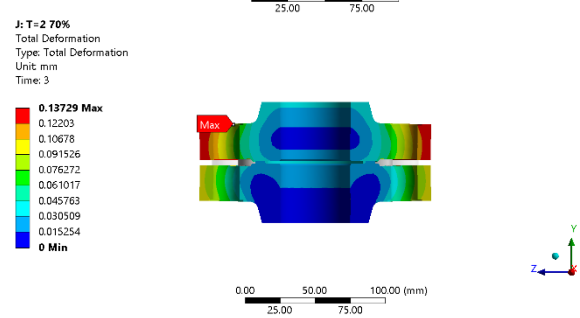
Case 4 Case at 70% Bolt Load

The Bolt load is calculated for 70% tightening is 53970.60811 N







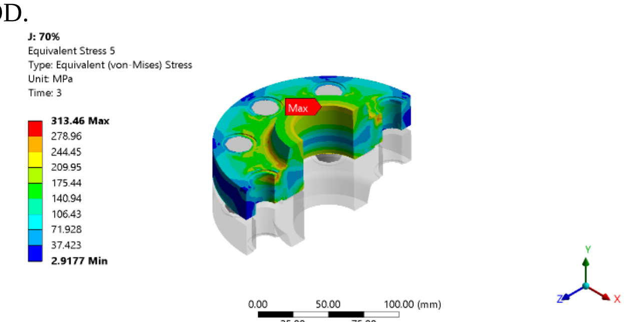


(b)

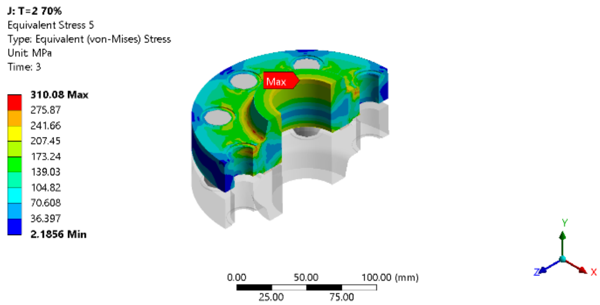
Fig 4.9 Deformation for 70% Bolt load.

(d) Deformation of 70% bolt load with 1mm gasket thickness. (b) Deformation of 70% bolt load with 2mm gasket thickness.

The above figure Represents the deformation of flanges simulated with 70% bolt load with 53970.60811 N pretension applied on beam joints as shown in Fig 3.6 for all 8 bolts and the steel gasket of 1 mm is placed in between pump side flange and pipe side flange fig 4.9 (a) represents 0.15606 mm deformation achieved with 1mm steel gasket fig 4.9 (b) represents 0.13729 mm deformation with 2mm steel gasket where red color indicates maximum deformation near flange OD.



(a)



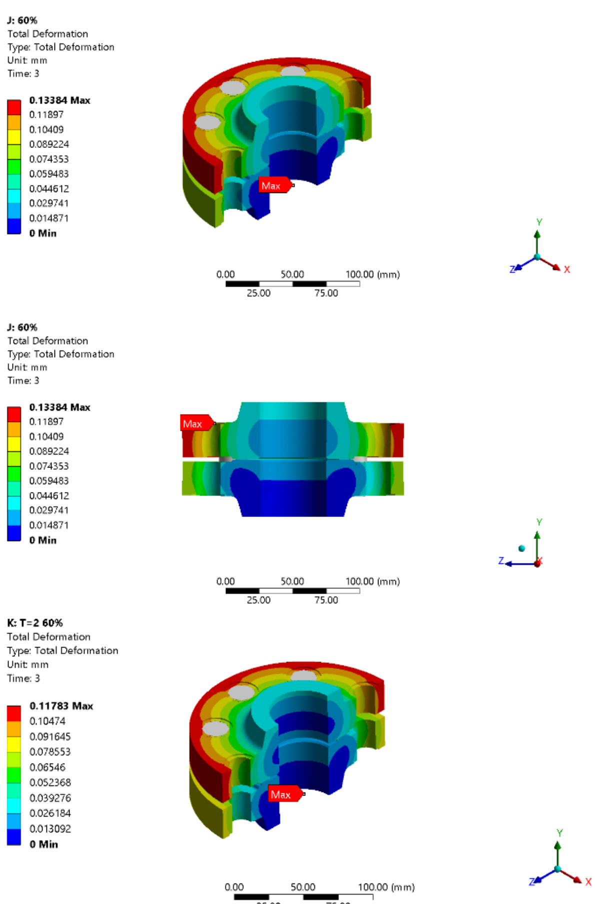
(b)

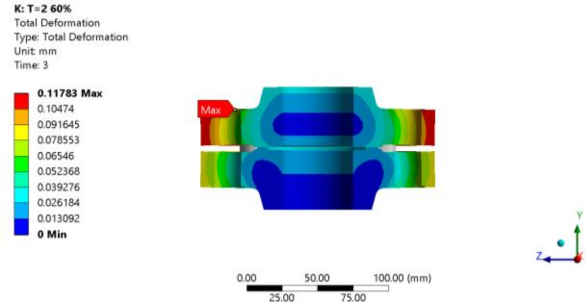
Fig 4.10 Stress generated for 70 % bolt load.
(d) Stress for 70% bolt load with 1mm gasket thickness. (b) Stress for 70% bolt load with 2mm gasket thickness.

The above figure represents the stress generated on the flange due to applied bolt load and pressure the flange with 1mm thickness have maximum stress generated on the edge at pipe connection as shown in the fig (a) which is 331.46 MPa with 2mm flange thickness the stress generated is 310.08 MPa.

Case 5 Case at 60% Bolt Load

The Bolt load is calculated for 70% tightening is 46260.52124 N

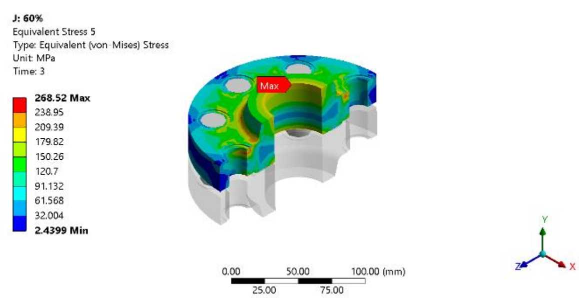




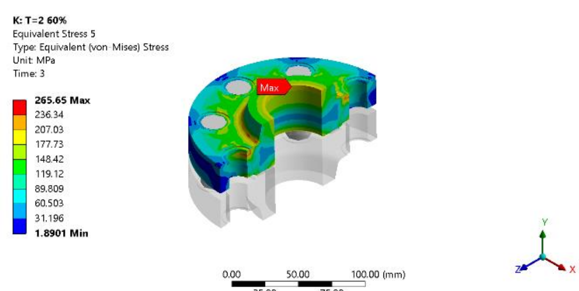
(b)

Fig 4.11 Deformation for 60% Bolt load.
(e) Deformation of 60% bolt load with 1mm gasket thickness. (b) Deformation of 60% bolt load with 2mm gasket thickness.

The above figure Represents the deformation of flanges simulated with 70% bolt load with 46260.52124 N pretension applied on beam joints as shown in Fig 3.6 for all 8 bolts and the steel gasket of 1 mm is placed in between pump side flange and pipe side flange fig 4.11 (a) represents 0.11783 mm deformation achieved with 1mm steel gasket fig 4.11 (b) represents 0.11783 mm deformation with 2mm steel gasket where red color indicates maximum deformation near flange OD.



(a)



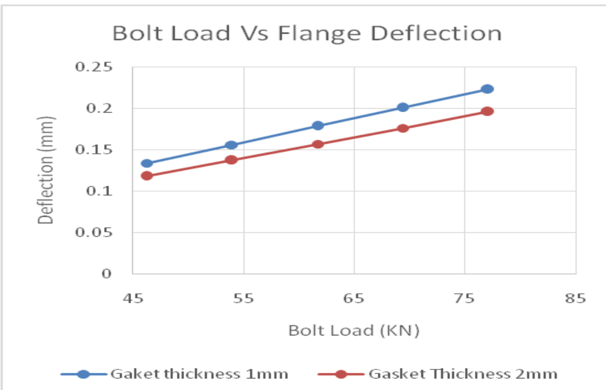
(b)

Fig 4.12 Stress generated for 60 % bolt load.
(e) Stress for 60% bolt load with 1mm gasket thickness. (b) Stress for 60% bolt load with 2mm gasket thickness.

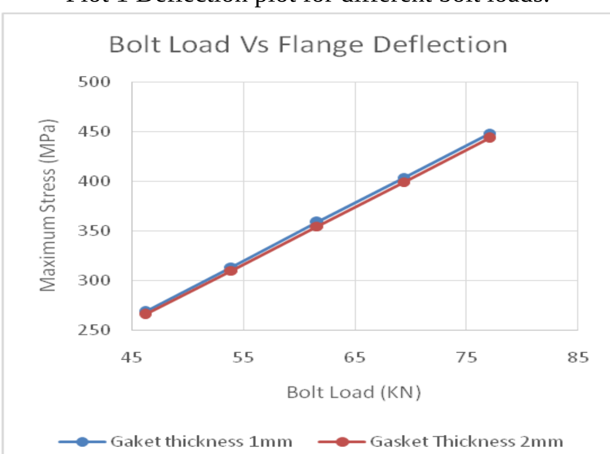
The above figure represents the stress generated on the flange due to applied bolt load and pressure the flange with 1mm thickness have maximum stress generated on the edge at pipe connection as shown in the fig (a) which is 268.52 MPa with 2mm flange thickness the stress generated is 265.65 MPa.

V. CONCLUSION

The Model done as per ASME B16.5 for 2" NPS the designed model is test with two different gaskets with two different thickness for different tightening percentages where the simulation results show the deflection is very minimum when compared to the Large deflections the stress induced on the joints very high which is as high as 448.31 MPa with 100 % Tightening of the bolt when we use 1mm Steel Gasket although stress slightly reduces where we increase the gasket thickness it is concluded from the results obtained that we cant tight the bolts more than 65 % to ensure a factor of safety of 2 For this joints.



Plot 1 Deflection plot for different bolt loads.



Plot 2 Stress Plot for different bolt loads

REFERENCES

[1] G. Mathan and N. Siva Prasad , "Studies on gasketed flange joints under bending with anisotropic Hill plasticity model for gasket," International Journal of Pressure Vessels and Piping , 88, 495-500, 2011.

[2] M. Murali Krishna, M.S. Shunmugam,and N. Siva Prasad, A study on the sealing performance of bolted flange jointswith gaskets using finite element analysis, International Journal of Pressure Vessels and Piping, 84, (2007) 349–357

[3] M. Abid, and D.H. Nash, A parametric study of metal-to-metal contact flanges with optimized geometry for safe stress and no-leak conditions, International Journal of Pressure Vessels and Piping, 81 (2004) 67–74

[4] V.H.Vishwanath, S. J. Sanjay and V. B. Math, The Study Of The Behaviour Of Bolted Flanges With Gaskets, International Journal of Engineering Research & Technology, 2(7), 2013.

[5] T.Sawa, N. Ogata and T. Nishida,Stress Analysis and Determination of Bolt Preload in Pipe Flange Connections With Gaskets Under Internal Pressure, http://pressurevesseltech.asmedigitalcollection.asme.org

[6] P.P. Bhale and A.K. Lavnis, Static& Dynamic Analysis of Motor Flange,International Research Journal of Engineering and Technology, 03 (04), 1209-1211, 2015..

[7] P. M. Desai,B. C. Pathak, V. A. Patel and P. B. Rana, Design, Analysis and Optimization of Body Flange and Cover Flange Using Finite Element Analysis, International Research Journal of Mechanical Engineering and Technology (IRMET), 4(5), 81-87, 2013.

[8] SauravRajgadia et. al, "Design and Stress-Analysis of a Rigid Flange Coupling using FEM", International journal of professional engineering studies. Vol. 4, Issue 10. 2015

[9] KondruNagendraBabu et. al, "Design and Failure Analysis of Flange Coupling with Uniform Strengthen Bolts", International journal of professional engineering studies. Vol. 6, Issue 1. 2017

[10] Somvir Singh et. al, "Finite Element Analysis and Weight Reduction of Flange Coupling Using CAE Tools", international journal of professional engineering studies. Vol. 03, Issue 4. 2017