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Design and Optimization of Weld Neck Flange for Pressure Vessel

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Abstract: *This paper presents the design analysis of a weld neck pressure vessel flange. Flanges are often used for applications where pressure is a factor. Flange data given in ASME B16.5 for flanges upto 24 inch and ASME B16.47 for flanges greater than 24 inch. Rating for flanges is decided based upon design pressure & temperature limits given in both codes. In our research, a model of the WNRF is created in ANSYS and pressure is applied beyond the given limit of pressures given in ASME B16.5. Stresses are calculated using mathematical approach and ANSYS software. This analysis reveals the zone of high localized stresses on the flange. The results obtained by both the methods are compared with allowable stress value for safe & optimized designing.*

Keywords: *ANSYS 15, Finite element analysis, Pressure vessel, Flange*

1. INTRODUCTION

Flanged joints on large diameter flanges can prove problematic to seal successfully with many factors contributing to ensuring a successful operation. One such factor is stud bolt loading contributing to stress and deflection of the flanged joint. Flanged joints with gaskets are very common in pressure vessel and piping systems, and are designed mainly considering internal pressure. Prevention of fluid leakage is the prime requirement of flanged joints. Many design variables affect joint performance and it is difficult to predict the behavior of joints in service. Nozzle and flange provide a way to haulage of pressurised fluid. So it is very essential to provide a leak proof and safe joint.

The flange structure itself is stressed due to this the design of a flange and loads generated due to bolting, with or without flow of fluid. Usually flange is weld over the nozzle and makes a permanent joint and flanges are joined together by the nut and bolt. A flange joint is considered in two conditions first is initial and after tightening and in operating condition. So that ASME gives the pressure criteria for different flange material. ASME VIII-1 Appendix 2 provides a method of sizing flanges. This investigation involves the use of finite element analysis (FEA) to predict levels of stress and deflection of a particular flanged joint when the stud bolts are tightened and flange pressurised. The level of stud bolt force selected must ensure the joint is sufficiently tight to avoid leakage. However, the force must not be excessive causing damage. Rachkow et. al. [1] studied the strength of the flange by using basic design criteria and found the strength of the flange on given pressure. Modestova et.al. [2] studied the different materials of the flange which provided more pressure sustainability. Voloslin [3] study the added stress on the flange due to the internal pressure and the bolt load. Abid et. al. [4] studied the performance of a flange joint using different gasket under combined internal pressure and thermal loading. The FEA by David Heckman on pressure vessel flange also suggest that the analyzing of the high stress area and different end connections have better result obtain if it is done by computer programme rather than hand calculation.

2. FLANGE DESIGN

This study work has been carried out on the following aspect

1. To calculate the stresses by manual calculation.
2. To develop the FEM based model using ANSYS.
3. To develop the geometry of flange using ANSYS.
4. To compare the results of ANSYS and manual calculation.

High-pressure flanges require a large bolt area to counteract the large hydrostatic end force. Large bolts in turn increase the bolt circle with a corresponding increase in the moment arm.

Thicker flanges and large hubs are necessary to distribute the bolt loads. A balance between the quantity and size of bolts, bolt spacing, and bolt circle diameter is necessary. In this study flange is designed by two approaches namely design by rules & design by analysis.

Some of the principal assumptions and simplifications involved in this method are summarized below [9]

1. Materials of all of the elements are assumed to be homogenous and remain elastic under the loading conditions assumed in the design.
2. The effect of the bolt holes in the flanges is neglected.
3. Axial symmetry is used to reduce the problem to consideration of the conditions on a single flange, hub and shell cross section, and neglecting variations due to location of bolts.
4. All loading applied to the flange is reduced to a 'couple' involving a pair of equivalent loads located at the extremities of the flange.
5. Stretching of the middle surface of the flange ring due to the applied couple is negligible.
6. Displacements of the joint are small such that the theorems of superposition are valid.
7. When a ring moment is applied to the flange, the point of connection between the flange and the hub is assumed to have zero radial displacement.
8. Hub and shell are assumed to act as thin shells.
9. The inside bore of the hub and shell is used in the shell theory analysis instead of the mean thickness diameter.
10. Effects due to interaction of elements are neglected.

TABLE 1: Vessel design data

Parameter	Symbol	Value	Unit
Design Pressure	P	1.36E+05	N/m2
Design Temperature	T	200	°C
Corrosion Allowance	C	3	mm
Vessel or Nozzle Wall thickness	t_n	16	mm
Flange Thickness (Assumed)	t	70	mm

TABLE 2: Flange properties [7]

Parameter	Value	Unit
Material	SA-105	
Tensile Strength	485	MPa
Yield Strength	260	MPa
Max. Temperature Limit	583	°C
Density	7750	KG/M3
Modulus of Elasticity	185	GPa
Poisson Ratio	0.3	

3. MATERIAL AND METHOD

3.1 MODELING PROCEDURE

1. Mathematical Procedure on the flange
2. Simulation of developed model to get the results

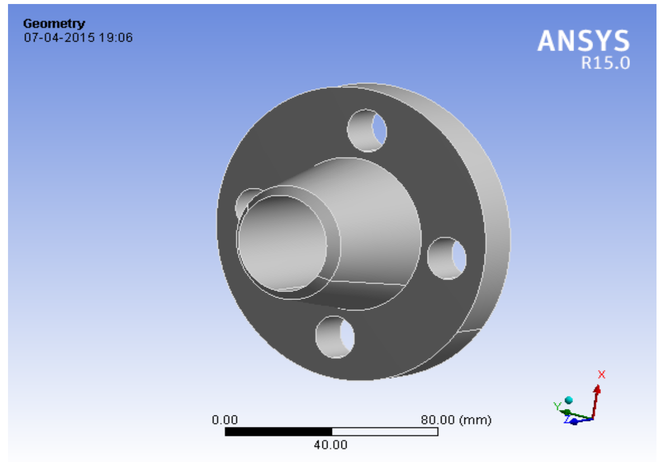

Fig. 1. Pressure vessel weld neck flange.

TABLE 3: Flange dimensional parameter [8]

Parameter	Symbol	Value
Outside diameter of flange	O	125
Diameter of the hub	X	65
Thickness of the flange	T_f	15
Length of the weld neck	Y	60
Chamfer diameter of weld neck	A	48.5
Weld neck diameter	B	40.9

3.2 METHODOLOGY

In this study the following methodology is adopted

The Maximum stress on flange is calculated by the mathematical approach.

The analysis of strain in flange by varying pressure is the area of research attention.

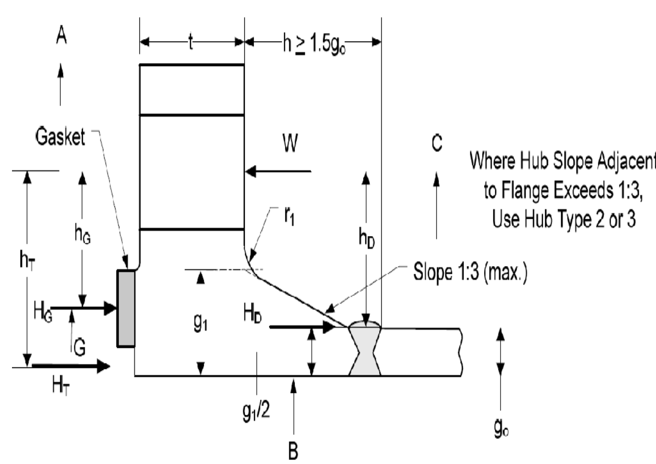

Fig. 2. Force represented on flange.

4. MATHEMATICAL APPROACH (DESIGN BY RULES)

Gasket contact width (N) = $(G_o-G_i)/22$ ……..(1)

= (65-48.5)/2 =8.25mm

Basic gasket seating width (b_0) =N/2 ……...(2)

= 8.25/2

=4.125

Outside diameter of the gasket Contact area ………(3)

G_C = if ($G_O \leq F_{od}$)

$G_C = G_O$= 65mm

Diameter at the location of the gasket load reaction

$G= (G_C-2*b)$ ………. (4)

= 65-284.125

= 56.75

Total force work on the flange (P_e)

$P_e = (16M/\pi G^3+4P_T/\pi G^2+P)$ ………… (5)

Here M=2000 N-mm

P_T=10000 N

P_e=17.6

P_e=21.6 bar

Total hydrostatic end force

$H= 78.5*G^2*P_e$ ………(6)

= $78.58*56.75^2*1.86$

= 5609 N

Contact load on flange surface

$H_P=2*b*\pi*G*m*P_e$ ……… (7)

Value of constant m=1

= 2*4.125* π*56.75*1*1.86

=3846.1756 N

Total hydrostatic end force on the area inside the flange

$H_D=0.785*B^2*Pe$ ……… (8)

=0.785*40.92*2.16

=2803.07N

Difference between the total hydrostatic force and end force on the area inside the flange

$H_T=H-H_D$ ………. (9)

=5609- 2803.07

=2805.93

Moment arm for load H_G =

h_G= C-G/2 …10)

=80-56.75/2

=11.625

Moment arm for load H_T=

h_T=.5(C-B/2+hG) ……… (11)

=.5(80-40.9/2+11.625)

= 15.8875

Moment arm for load H_D=

h_D=C-B/2 ……… .(12)

=80-40.9/2

=19.55

TABLE 4: Force analysis of flange with force

Loading	Force	Distance	Moment
	N	mm	N-mm
End pressure (MD)	2803.07	19.55	54800.019
Face pressure(MT)	2805.93	15.8875	44579.212

Flange design moment for the operating

$M_O=(M_D+M_T)$absolute ……. (13)

=49689.61525 N-mm

Flange maximum operating stress

S_{max}=(1.33*t*e+1)*MO/(26.76*t2*B) (14)

S_{max}= (1.33*15.9*.011*+1)*49689.61525/(26.76*15.92*40.9)

= 221.352 N/mm^2

5. FINITE ELEMENT APPORACH (DESIGN BY ANALYSIS)

A model of flange is created in ANSYS and pressure and load are applied beyond the given limit of pressures and thus study the finite element analysis of the flange. This analysis reveals the zone of high localized stresses on the flange. Different stress values were obtained corresponding to the different pressures. All stresses were compared with allowable stress value and it has been optimized that how much pressure can be applied with same initial bolt load beyond the ASME pressure limit.

5.1 BOUNDARY CONDITIONS AND MESH SENSITIVITY ANALYSIS

The boundary conditions for the flange of the pressure vessel are pressure applying on the inside diameter of the flange and the chamfer weld neck is a constraint. The movements in Z-axis directions of flange are given. The other movements $U_y = U_x = 0$ of flange are zero, while the boundary conditions of the flange are fixed. The movements in all the three axes are constrained, $U_x = U_y = U_z = 0$. Force is applied in the X direction on the flange. A mesh analysis is done on the FEA mode of flange, to ensure the optimum mesh size of FEA model for proper convergence and exact numerical results. The maximum Von Mises stress value occurring in the FEA model is used as a convergence criterion. The Fig. 4 Indicates that with the element number 491037 the mesh becomes appropriate and the number of element reaches 340442, the meshing in the model of pressure vessel flange has enough sensitivity.

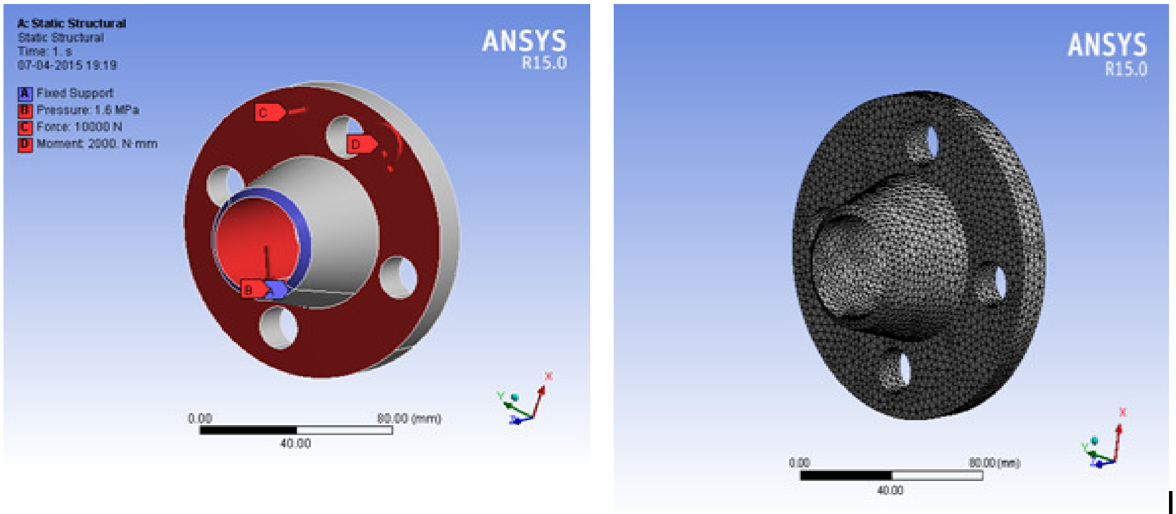

Fig. 3. Boundary Conditions of weld neck flange and Meshed finite element model of weld neck flange

5.2 FEA RESULTS

In this part, the results obtained gave a detailed distribution of stresses in the local part of the flange area. It is found that the maximum intensity of stress is obtained near fixed chamfered zone of the flange. The consequences of various design parameters on the calculated stress values are investigated. The flange is suitable to withstand more pressure value as per given in the ASME for same material. When pressure increase then the stress also increases and force which is applied to hold the flange also increases the flange stress. The stress distribution over various geometric parameters of wear plate, web plate, saddle, base plate and flange is analyzed for the selection of optimal size of flange for given pressure criteria. This signifies that the analytical method of designing is used to decide the safety pressure range in ASME.

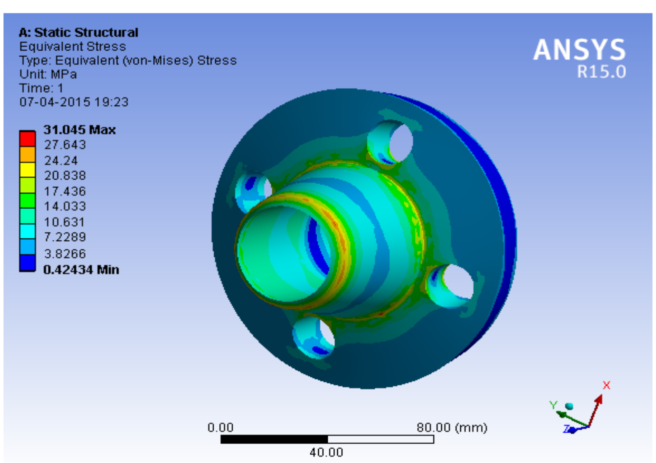

Fig. 4. Stress distribution over the flange.

6. CONCLUSIONS

The Mathematical approach is used to design by rule and for designing of pressure vessel the analytical method used is finite element analysis and calculations is done as per Megyesy, (Pressure Vessel Handbook). The stress distribution over various geometric parameters of wear plate, web plate, saddle, base plate and flange is analyzed for the selection of optimal size of flange for given pressure criteria. This signifies that the analytical method of designing is used to decide the safety pressure range in ASME. The comparative study of these results helps to decide the most optimized design to fulfil the desired requirements and facilitates to find the economical solution. After analysis of the result of the stress on the flange to greater pressure than as given in ASME for safe design, it can be concluded that the weld neck flange is safe by increasing the pressure value greater than the 13.6 bar given in ASME. This flange can bear 18.4 bar pressure after comparing the maximum stress to the allowable stress.

NOMENCLATURE:

h_D: Radial distance from bolt circle to circle on which HD acts(mm)
H_D: Hydrostatic end force on area inside flange (N)
h_G: Radial distance from gasket load reaction to bolt circle (mm)
H_G: Gasket load = Wm1- H for operating condition (N)
h_T: Radial distance from bolt circle to circle on which HT acts(mm)
H_T: Difference between total hydrostatic end force and the hydrostatic end force on area inside
flange = H-H_D (N)
m: Gasket factor

y: Gasket / joint contact surface unit seating stress (MPa)
W: Design bolt load for the gasket seating condition (N)
w: Width of the straight portion of obround flange (mm)
b_o: Basic gasket seating width (mm)
N: Gasket width (mm)
G: Diameter at location of gasket load reaction (mm)

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