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Design and Analysis of round Flange for Pressure Vessel Application to Comply with ASME Code

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ABSTRACT

The Flanged joints are essential components in nearly all pressurized systems; however, they are also one of the most complex parts to design. Various factors enter into the determination of the successful design and operation of a flange joint in service. The bolted flange joint involves interaction between bolting, flange, and gasket. For applications such as vapour absorption chillers, unfired pressure vessels having flange size more than 600mm; there is a need of non-circular flanges having reduced height than traditional flanges. The proposed shape for flange (Slip-on type without hub) is obround shape whose investigation involves finding an appropriate analytical method to be used for design of obround flange which can comply with ASME code. This obround flange subjected to internal pressure is designed using equivalent circular flange method. The finite element analysis (FEA) is used to predict levels of stress and deformation of a particular flanged joint and stresses are linearized for stress categorization. These FEA results are compared with ASME allowable limit and are on safe side. The analytical design method is approximate method and results on positive error side. For selected application; there is 22% reduction in height observed with the use of obround flange.

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I. INTRODUCTION

Flanged joints with gaskets are very common in pressure vessel and piping systems, and are designed mainly for internal pressure consideration. Prevention of fluid leakage is the prime requirement of flanged joints. Many design variables affect joint performance and it is difficult to predict the behaviour of joints in service. The ASME boiler and pressure vessel code (BPVC) contains rules for non-circular pressure vessels of unreinforced and reinforced construction and their end covers. There is no standard design procedure available for bolted flange connections; so these flanges cannot be designed directly with the rules of ASME BPVCSection VIII div.-1 due to complexity of the shape. Hence the method is formulated for manual design of obround flange which can use ASME BPVC section VIII div.-1. Guidelines of the ASME BPVC Section VIII div.-2 are to be used with the allowable stress limits of ASME BPVC section VIII div.-1 and Finite Element Analysis (FEA) is done to meet requirements of ASME BPVC Section VIII Div-2 as specified in U-2(g).Structural integrity and leakage tightness of bolted flanged connections are one of principal factors to ensure a safe and extended service life of critical engineering structures such as reactors, steam generators, boilers, heat exchangers, piping systems, and others that operate under critical process conditions including internal pressure and a variety of operating temperatures. From structural integrity point of view safe design of the bolted flange connections (BFC) has been solved and satisfactorily standardized by American Codes which is based on Taylor forge method of flange design [5]. One of the most common methods used for flange design is found in ASME BPVC section VIII Division 1, Appendix 2, rules for bolted flange connections with ring type gaskets. Australian Standard AS1210 also follows this approach. These methods is adapted from of the Taylor-Forge method developed by Waters, Wesstrom, Rossheim and Williams of the Taylor-Forge company in Chicago in the 1930's and subsequently formed the basis of the ASME code for flanged joint design[5]. The assumptions made by this method are now generally regarded as simplistic. This method gave rise to the ‘m’ and ‘y’ gasket factors in ASME section VIII as well as other codes. The calculation is based on the axial forces balance between the bolt load, the resulting axial force due to the end thrust effect of the internal pressure and the reaction on the gasket.

Adolf E. Blach [1] in one of his work describes the two design methods for bolted flanged connection of non-circular cross-section of obround and rectangular type. One method is applicable to unreinforced “almost square" rectangular shapes, using an "equivalent circular flange", and standard flange design methods. The other is based on a decomposition of frame and flange bending stresses and may also be used for rib-reinforced pressure vessel flanges.Calculations, experimental values and finite element results were obtained for flange with ring gaskets (gasket fully inside the flange bolt line) and full face gaskets. Comparing numerical values with experimental data, he proved that the method of equivalent circular flanges is suitable for obround and rectangular pressure vessel flanges within certain limits. The results are on the safe side and become increasingly conservative as the length to width ratio increases.

NOMENCLATURE

a	:	Nominal diameter of bolts (mm)	n	:	Total number of bolts
A	:	Outside diameter of flange (mm)	N	:	Gasket width (mm)
A	:	Area of cross-section (mm^2)	P	:	Design pressure (MPa)
b	:	Effective gasket seating width (mm)	*P*	:	Wetted perimeter (mm)
B	:	Inside diameter of flange (mm)	R	:	Radial distance from bolt circle to point of intersection of hub
b_o	:	Basic gasket seating width (mm)			and back of flange (mm)
C	:	Bolt circle diameter (mm)	S_a	:	Allowable bolt stress at gasket seating temperature
c	:	Clearance between OD of shell and ID of flange (mm)			(atmospheric temperature) (MPa)
d	:	Bolt hole diameter (mm)	S_b	:	Allowable bolt stress at operating temperature (MPa)
D	:	Hydraulic diameter (mm)	S_{fa}	:	Allowable stress for flange material at gasket seating
D'_m	:	Equivalent mean gasket diameter assumed for design			temperature (atmospheric temperature) (MPa)
		purpose (mm)	S_{fb}	:	Allowable stress for flange material at operating temperature (MPa)
D_L	:	Maximum inside diameter of obround shell (mm)			
D_{m1}	:	maximum mean gasket diameter for non-circular flange (mm)	S_H	:	Calculated longitudinal stress in hub (MPa)
D_{m2}	:	minimum mean gasket diameter for non-circularflange (mm)	S_R	:	Calculated radial stress in flange (MPa)
D_S	:	Minimum inside diameter of obround shell (mm)	S_T	:	Calculated tangential stress in flange (MPa)
E	:	Radial distance from bolt circle to outside diameter of	S_{TS}	:	Minimum tensile strength (MPa)
		flange (mm)	S_Y	:	Minimum yield strength (MPa)
G	:	Diameter at location of gasket load reaction (mm)	T	:	Design temperature (°C)
h_D	:	Radial distance from bolt circle to circle on which H_D acts(mm)	t	:	Flange thickness (mm)
H_D	:	Hydrostatic end force on area inside flange (N)	t_g	:	Thickness of gasket (mm)
h_G	:	Radial distance from gasket load reaction to bolt circle (mm)	t_n	:	Nominal thickness of the shell, pipe or nozzle to which the
H_G	:	Gasket load = W_{m1}- H for operating condition (N)			flange is attached (mm)
h_T	:	Radial distance from bolt circle to circle on which H_T acts(mm)	W	:	Design bolt load for the gasket seating condition (N)
H_T	:	Difference between total hydrostatic end force and the	w	:	Width of the straight portion of obround flange (mm)
		hydrostatic end force on area inside flange = H-H_D (N)	y	:	Gasket / joint contact surface unit seating stress (MPa)
J	:	Flange rigidity	z	:	Factor for conversion of obround shape to circular shape
m	:	Gasket factor			

Muhsen Al-Sannaa and Abdulmalik Alghamdi [3] studied the results obtained using Finite Element Analysis (FEA) of large diameter welded neck steel flanges under different loading conditions. They give the stress analysis of flanged joint made up of the flange and the gasket for large diameter steel flanges. The parametric study done by them; include the effect of clamping pressure, internal pressure, axial end pressure, temperature, and gasket material on the contact pressure. They showed that clamping pressure is a determinate factor for the sealing condition and that 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.

M. Murali Krishna, M.S. Shunmugam and N. Siva Prasad [4] worked on the finite element analysis of bolted flange joint considering non-linearity of the gasket material under various loading and operating conditions.Gaskets behaviour is complex due to nonlinear material properties combined with permanent deformation. They found that variation of contact stresses due to the rotation of the flange and the material properties of the gasket play important roles in achieving a leak proof joint. Analysis shows that the distribution of contact stress has a more dominant effect on sealing performance than the limit on flange rotation specified by ASME.Flange rotation causes variable compression across the gasket from the inner radius to the outer radius. Due to the variation in compression, the contact stresses also vary along the radius.

This paper aims to find the appropriate analytical method to be used for the obround flange which can comply with ASME code. The obround flange is designed using equivalent circular flange method. The finite element analysis (FEA) is used to predict levels of stress and deflection of a particular flanged joint and stresses are linearized. These FEA results are compared with ASME allowable limit and are on safe side. The analytical design method is approximate method which results on positive error side.

I. Flange Design

The obround flange designed here is for the particular application in generator shell of vapour absorption chiller unit. The chiller unit is mainly used in industries and hotels for food storage or air-conditioning purpose. Mostly the fitment of this unit is in parking area or basement where the height is restricted. So to reduce the machine height; one of the option is to reduce the height of the generator shell and its flange with the use of non-circular shape.

A. *Forces Acting on Flange*

The forces acting on flange joint subjected to internal pressure with ring type gasket i.e. gasket is wholly within the circle enclosed by a bolt hole and no point of contact beyond this circle [7],[8]are as shown in figure 1.

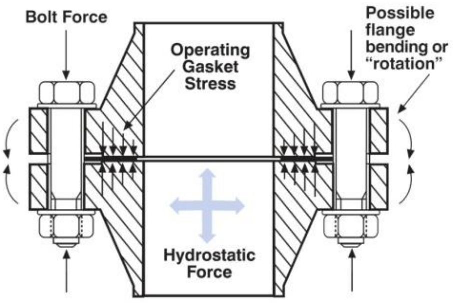

Fig. 1 forces acting in a bolted flange joint assembly

The initial bolt load generated upon tightening is transferred to the gasket via the flanges. This initial seating stress compresses the gasket and tightens it within itself. The hydrostatic force generated by the system pressure, tends to 'unload' and reduce the stress on the gasket. The stress remaining on the gasket is considered to be the 'operating' or 'residual' stress. It should be seen that on a raised face assembly as shown in figure 1, there will be some deflection of the flanges themselves ('flange rotation'). This is a function of the load applied, the flange material and the geometry of the flanges. Thus, the operational stress towards the outside edge of the gasket tends to be greater than on the inside edge.

The various forces acting on flange can be represented on cross-sectional view in the required directional sense [7], [11]as shown in figure 2

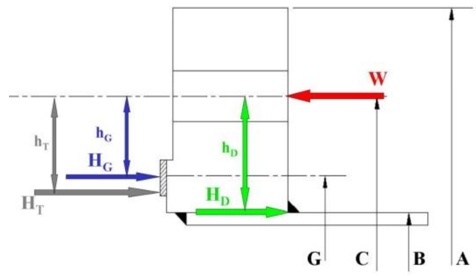

Fig. 2 Forces represented on flange (Slip-on flange without hub)

The calculations use four loads bolt loads, gasket load, face pressure load and hydrostatic end force represented in figure 2 and two conditions seating and operating [11]. *Load* $H_D$is created by the pressure on the pipe attached to the flange. During operation, Pressure is applied to the exposed edge of the gasket and gasket tries to expand but is held in place by the flange faces. The flange faces push back and gives rise to uniformly varying pressure along gasket width whose average value is represented by *load* H_T. *Load* H_G is the force required to seat the gasket into the flange gasket face which is based on gasket physical properties.

B. *Methods for Non-Standard Flange Design*

As per the literature survey, the different methods which can be used for non-standard flange design either obround or rectangular are:

1) As per Swedish standard for piping
2) Equivalent circular flange method
3) Frame bending method
4) Hydraulic diameter method

These methods either convert the non-circular shape into circular shape and then we can design that circular flange or considers the change in shape into various formulae in design process for designing flange to be safe.

1) As per Swedish standard for piping[15]:In the Swedish piping code for flange joint, the design procedure of rectangular, oval and obround flange is given by converting these shapes into equivalent circular shapes. This method uses the factor k_4 multiplied with the maximum mean gasket diameter D_{m1} to get equivalent mean gasket diameter D'_m

.... (a)

2) Equivalent circular flange method [1], [10], [11]:The method for design of non-circular pressure vessel flange is developed by Adolf E. Blach [1]. It covers the design of obround and rectangular shaped flanges which comply with the ASME BPVC codes. It has given two methods of designing flange as per the shell construction. The first method is for flange mounted on unreinforced pressure vessels.

This method uses the part of the procedure used in the ASME code section VIII div 1 article UG-34 for the design of non-circular flat covers. In the code, a factor z is defined which relates a flat cover of obround/rectangular shape to a circular one. The factor z is given by

.... (b)

But factor z should not be larger than 2.5 and length to width ratio should be less than 2 is the requirement. The square root of this factor is used as a multiplier of the small side of the obround/rectangular flange to obtain an equivalent circular shape.

......................................(c)

Any obround or rectangular flange which satisfies the above criteria can then simply be designed or analysed as an equivalent circular flange, and all flange design code rules as per appendix 2 of the ASME BPVC section VIII division 1 are applicable without modification.

3) Frame bending method [1]:The other method given by Adolf Blach is frame bending approach for non-circular pressure vessel flanges. This method is applicable to the flanges with obround or rectangular shapes mounted on reinforced vessels. The method uses a combination of frame analysis for the ability of the flange to retain its shape, and bending of an infinitely long flanged section in a perpendicular plane with respect to the frame.

In this case, the flange must also act as a stiffener for the vessel side plates, in addition to providing a tight seal between components. Thus, such flanges have to resist frame bending stresses, stresses which occur when a frame is subjected to internal pressure. These stresses cause deflections in a plane perpendicular to the vessel axis. In addition, flanges also have to resist flange bending stresses in planes parallel to the axis of the vessel, stresses which occur when a flange is bolted-up about the gasket, or when internal pressure effects tend to open up the bolted connection.

*4) Hydraulic diameter method:*This method uses the conversion of non-standard shape into circular shape using the logic of hydraulic diameter. Hydraulic diameter is the commonly used term when handling flow in non-circular tubes and channels. Using this term one can calculate many things in the same way as for a round tube. This is given by

.... (d)

This gives the equivalent inside diameter of flange which can be used in design calculations as per ASME BPVC section VIII appendix 2.

C. Input Parameters for Design

The generator on which this obround flange is going to be mounted contains hot water on tube side and weak solution of Li-Br on shell side. The working parameters needed for design of flange are as given in table I.

Table I. INPUT PARAMETERS FOR FLANGE DESIGN

Parameter	Label	Value	Unit
Design Pressure	P	1.054	MPa
Design temperature	T	200	°C
Corrosion allowance		1.5	mm
Vessel or nozzle wall thickness	t_n	16	mm
Minimum inside diameter of shell	D_S	500	mm
Maximum inside diameter of shell	D_L	856	mm
Clearance between OD of shell and ID of flange	c	1	mm
Flange thickness (assumed)	t	70	mm

The existing circular flange outer diameter is 873.76mm which is needed to be replaced for height reduction. The bolting is selected as in [7] and its dimensions are taken from [17] which is given in table II below. Similarly for the given operating parameters, gasket is selected and which has following parameters required in design as specified in table III.

Table II. BOLTING SPECIFICATIONS [17]

Selected Bolt Parameters			
Parameter	**Label**	**Value**	**Unit**
Selected Bolting		**M30**	
Bolt diameter	a	30	mm
Bolt hole diameter	d	33	mm
Radial distance from bolt circle to point of intersection of hub and back of flange	R	33.34	mm
Radial distance from bolt circle to outside of flange	E	33.34	mm
Root area of selected bolt		502.96	mm^2
Total bolts being used	n	28	Qty

Table III. GASKET SPECIFICATIONS [11]

Selected Gasket Parameters			
Selected Gasket	Spitman AF 154		
Parameter	Label	Value	Unit
Gasket factor	m	1	
Gasket contact surface unit seating stress	y	1.4	MPa
Gasket width	N	20	mm
Gasket thickness	t_g	5	mm

D. Material of construction

The material required for standard component is selected as per the operating temperature to which it is subjected and the form of the material. The working temperature range for generator is from 150°C to 200°C i.e. 302°F to 392°F.

1) *Flange and bolt material:* The material for flange is selected as per guidelines given in ref. [6].The material used for the flange is SA-516 Gr70 which is in the plate form and for bolts; it is SA-193-B7. Material properties are as shown in table IV.

2) Table IV. MECHANICAL PROPERTIES OF FLANGE AND BOLT MATERIAL [13]

Parameter	Value		Unit
Selected Material	**SA-516 Gr70**	**SA-193-B7**	
Nominal Composition	Carbon steel	Carbon Steel 1Cr-1/2Mo	-
Product Form	Plate	Bolting	-
Size / Thickness	-	≤64	mm
UNS No.	K02700	G41400	-
Tensile Strength	485	860	MPa
Yield Strength	260	725	MPa
Max. Temperature limit	538	538	°C
Density	7750	7750	kg/m^3
Modulus of Elasticity	185	185	GPa

The maximum allowable stress value is the maximum unit stress permitted in a given material used in a vessel constructed. The criteria for maximum allowable tensile stress values permitted for different materials are given in mandatory appendix 1 & 2 of ASME BPVC Section II, Part D. As per it, the allowable stress value for selected flange material is as given in table V.

Table V. ALLOWABLE STRESS VALUES OF FLANGE MATERIAL [13]

Quantity	Allowable value	Value	Unit
Allowable tensile strength	$S_{TS}/3.5$	138	MPa
Allowable yield strength	$2S_Y/3$	173	MPa

Allowable stress values for selected bolting material are as given in table VI.

Table VI. ALLOWABLE STRESS VALUES OF BOLT MATERIAL [13]

Quantity	Allowable value	Value	Unit
Allowable tensile strength	$S_{TS}/5$	172	MPa
Allowable yield strength	$S_Y/4$	181.25	MPa

1) *Gasket Material:* Material used for gasket compressed non-asbestos fiber reinforced type (CNAF). We took Spitman AF 154 (Champion seals Pvt. Ltd.) having superior performance compressed jointing sheet incorporating a blend of special heat resisting aramid fibres with a high quality nitrile elastomer binder.Colour: Yellow

Table VII. PROPERTIES OF SPITMAN AF 154 GASKET

Parameter	Value	Unit
Max operating pressure	150	Bar
Max. Short term service temperature	450	°C
Max. Continuous service temperature	250	°C
Max. Operation temp. for steam	290	°C
Density	1800	Kg/m^3
Tensile strength	15	MPa
Compressibility	7-11%	
Recovery	≥50%	
Water absorption	<5%	

E. Selection of Appropriate Design Method

The selection of the appropriate method to design the obround flange will affect the flange thickness considerably. One of the key requirements of the work is that the design should comply with the ASME code.

We are having 4 methods for the approximate design calculation of obround flange and each method gives different stress values. This graph is plotted with 65mm flange thickness, M30 (28) bolting and 20mm gasket width. We plot the stresses in gasket seating and operating condition of flange as shown in figure 3.

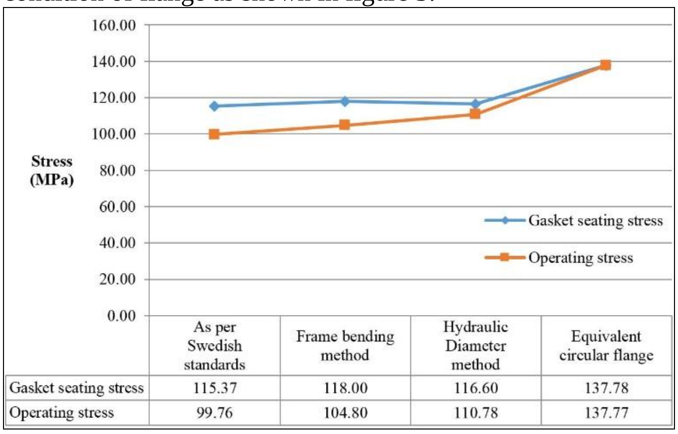

Fig. 3 Trend of the stress vs. various methods

It is clear from the above figure that stresses generating in equivalent circular flange method are maximum of all. So if we design the flange as per equivalent circular flange method, we will design it for maximum stress condition and it will be safe for other methods. The other advantage of using this method is; this method uses the ASME code for flange design with slight modification so it will comply with ASME codes as well. So the most appropriate method for designing of obround flange is equivalent circular flange method.

F. *Manual Design*

The manual design of the obround flange is done with the equivalent circular flange method as its give the safest design. The manual design of slip-on type flange considers only tangential stress S_T in flange which is the combination of membrane + bending stresses. The hub stress $S_H$is considered zero as slip-on flange has nearly hub-less design and radial stress S_R is consider to be entirely carried away by shell so they are ignored.

With the use of the given basic dimensions we found the factor z to be 1.9567 which is below the limit of 2.5. Then with the help of equation (b), the inner diameter of equivalent circular shape of flange is calculated as 751.181mm. It is just the virtual dimension of circular flange required for finding the approximate stresses in the actual obround flange.This dimension along with the required specifications of bolting, gasket and their materials is used to find the stress occurring in flange with the help of procedure given in appendix 2 of ASME BPVC section VIII division 1. The flange thickness appropriate for the given loading condition and dimension is found to be 70mm. The max stress generating in this flange is 127.93 MPa in operating condition and 118.8 MPa in gasket seating condition. The height of the designed flange is 680.68mm. The flange dimensions also satisfies the rigidity criteria with J = 0.987 which is below allowable limit of 1. Cover flange is also design based on the UG-34 given in [11] and the minimum thickness required for flange is 45mm.

II. Finite Element Analysis

The modelling of generator flange for chiller is carried out in SolidWork, Release 2015 (Student Edition) and analysis on its Cosmos solver. Flanges, nozzle opening on shells are important for illustration, process and inspection. They will not only weaken the strength of shell but also generates boundary stress on the joint of vessel flange, leading to severe stress concentration. So the joint is most vulnerable part to failure. It's of great importance to study the influence of various parameters on stress distribution of the Flange. Due to different loadings applied to flange, a local stress state of the flange connection characterized by high stress concentrations occurs in the intersection region. The significant stress concentration almost always occurs in the vicinity of the flange-to-shell junction due to the inherent structural discontinuity that is formed by the intersection.

A. *Design by Analysis (DBA) Approach[12]*

Traditionally, to determine the acceptability of the design, formulas and charts based on analytical solutions and empirical data have been used. This is today called as Design by Formula (DBF) which is described by ASME VIII Division 1. In the 1960's an alternative to DBF was established known as Design by Analysis (DBA). It served as a complement for the design cases which were not covered by DBF, and was based on a method where stresses are classified into different categories. Formulas are not always applicable and therefore design by analysis serves as a complement within both the ASME code as allowed by U-2(g).

ASME BPVC section VIII division 2 Part 5 [12] states the design by analysis requirements and includes several methods for evaluating the design against four different failure modes. Methods are described for evaluating against plastic collapse, local failure, buckling, and cyclical load. The "Stress classification" method, which this work is limited to, is included in the analysis for plastic collapse and thus, this is the only failure mode that will be considered.

For the elastic stress analysis method a linear elastic model is used. This means that the stress in the material is assumed to be linearly proportional to the strain. The "Stress categorization" or "elastic stress analysis-method" is used since it is a relatively straightforward way to obtain a result.

B. *Analysis:*

1) *Basic Assumptions:* In order to simplify the analysis of the flanged joint, a number of assumptions were made. These basic assumptions are:

- Gasket material was assumed to have linear properties with the non-linear behaviour of the gasket section ignored.
- All materials for the model of slip-on flange, shell, bolts, gasket and blind flange, are assumed isotropic.
- Analysis will be linear static analysis.
- Temperature effects will not be considered (already considered while selecting material).
- Bolt loads will be averaged over the area where the bolt head are located in the circular ring.

2) *Boundary conditions and Loads:* The fix support is applied to the tubesheet which is going to be attached to the evaporator shell and back side of the generator shell to restrain it in all direction as shown in figure 4. The pressure of 1.054 MPa is applied at the inside of the obround shell as shown in figure 4. Then the gasket load reaction (H_G) of 1449.81 N & difference between total hydrostatic end force and hydrostatic end force acting inside the shell (H_T) of 1091.67 N is applied at distance of 26.4mm and 32.45mm respectively from bolt circle position in inward direction on flange face. The load imparted on the flanged joint by the twenty eight bolts is calculated using the bolt load calculation during flange design. That bolt load calculated is applied on the effective bolt contact area near the every bolt holes. The total value of bolt force 538272.530N is divided by number of bolts and then applied.

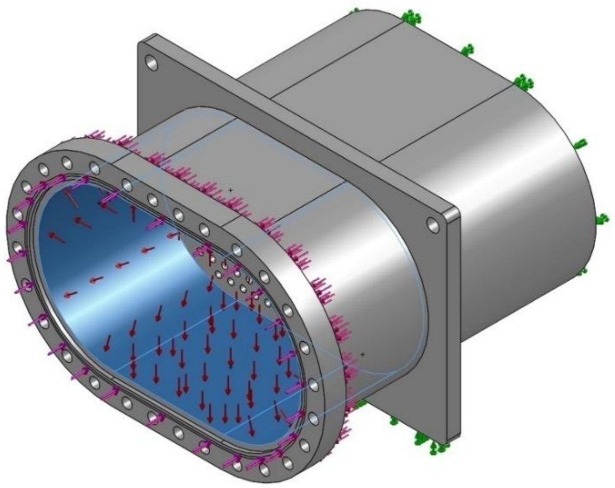

Fig. 4Boundary conditions and loading

The model is fine meshed with the solid elements. Mesh used is the standard mesh using parabolic tetrahedral solid elements defined by four corner nodes, six mid-side nodes, and six edges. For the optimum selection of the mesh size and plotting of the results, the stress convergence is established and the percentage variation must be kept under 5%.

3) *Results:*The von Mises stress plots of flange analysis are given in figure 5 and figure 6.

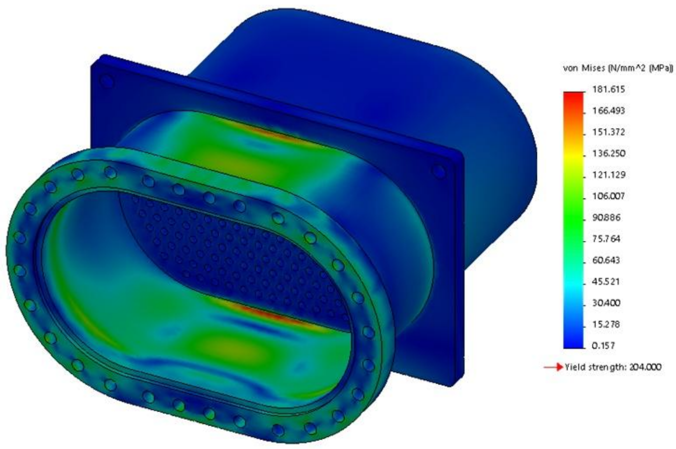

Fig. 5von Mises stress plot of obround flange (view-1)

The maximum stress generating is 181.615 MPa. This stress value is higher than the allowable stress but below the yield strength of material. The stresses are localized at the shell and tubesheet junction and also near the bolt holes at straight to circular transition of flange; so these need to be linearized to compare with allowable limit. The stress in the other regions is within allowable limit.

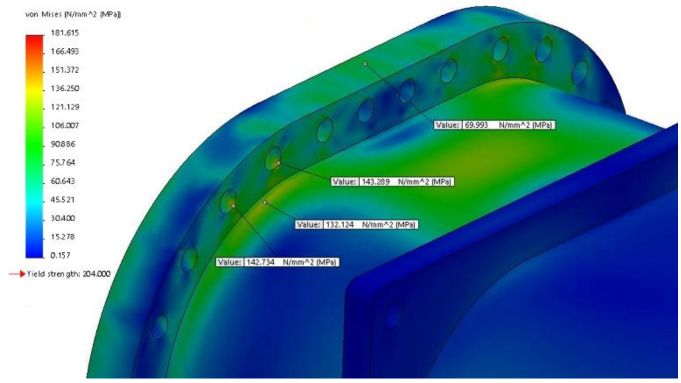

Fig. 6 von Mises stress plot of obround flange (view-1)

The deformation plot of the obround flange subjected to given loading conditions is as shown in figure 7.

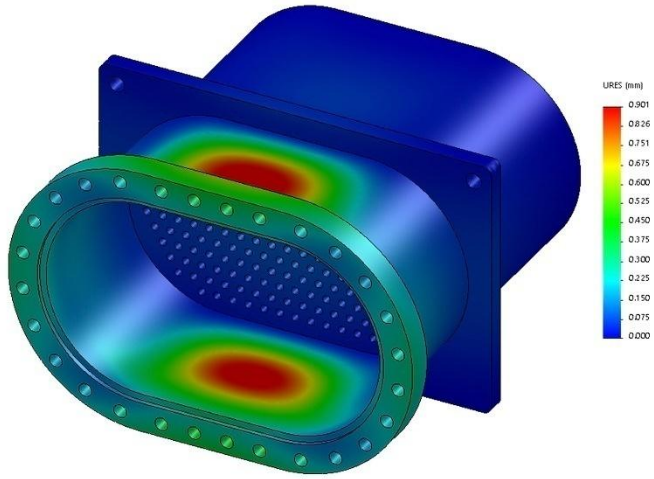

Fig. 7 Deformation plot of obround flange

The maximum deformation of 0.901mm is generating in the straight portion of shell. Maximum deformation occurring in flange is within the range of 0.600mm to 0.715mm. The deformation value allowed is 5 mm so the design is safe for deformation. It is occurring because of the non-symmetric shape of the shell. The shell straight portion tries to adopt more stable shape at these locations subjected to internal pressure and hence the deformation is occurring maximum there. Also this deformation is negligible as compared to overall size of the flange.

III. LINEARIZATION

In the finite element method, when continuum elements are used in an analysis, the total stress distribution is obtained i.e. von Mises stress. The ASME code does not use principal stress or von Mises stress as comparable. These values cannot be directly compared with the analytical values of stresses as calculations give the membrane and bending stress. Therefore, to find membrane and bending stresses, the total stress distribution shall be linearized on a stress component basis and used to calculate the equivalent stresses.

Linearization is a decomposition of the stress distribution we see in FEA of pressure vessels. It decomposes a basically parabolic distribution into a uniform value (membrane stress), a linearly changing value (bending stress), and possibly an extra component (peak stress) [14].By doing this, we can use finite element distribution and pick one or more stress classification lines to decompose the stresses such that we can apply the code. A Stress Classification Line or SCL is a straight line running through thickness of a vessel. It is perpendicular to both the inside and outside surfaces. The guidelines of linearization of stress results as per annexure 5-A given in [12] are used for stress classification.

The maximum stress concentration zones can be identified from the stress plot of the obround flange as given in figures 5 & 6. The stress concentration is at transition of flange from straight portion to curve portion, flange thickness near bolt hole and also at the intersection of shell and tubesheet. Allowable stress values as per category are as given below in table VIII.

Table VIII. ALLOWABLE STRESS VALUES FOR LOCALIZED STRESS [14]

Quantity (MPa)	Allowable stress	Value
Membrane Stress	S_a	138
Membrane Stress + Bending	$1.5*S_a$	207

C. Stress Linearization / Stress Classification

1) *At flange thickness:* Firstly we perform the linearization at the flange thickness near the bolt hole in flange

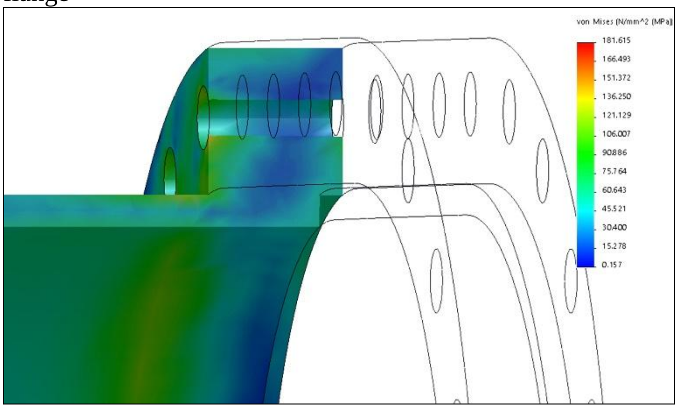

Fig. 8 Stress concentration zone in flange

Stress concentration line is marked on the section of stress concentrated zone with following points :

Start point Point 1(218, 290.59, -57.764)mm

End point Point 2(218, 290.59, 11.915)mm

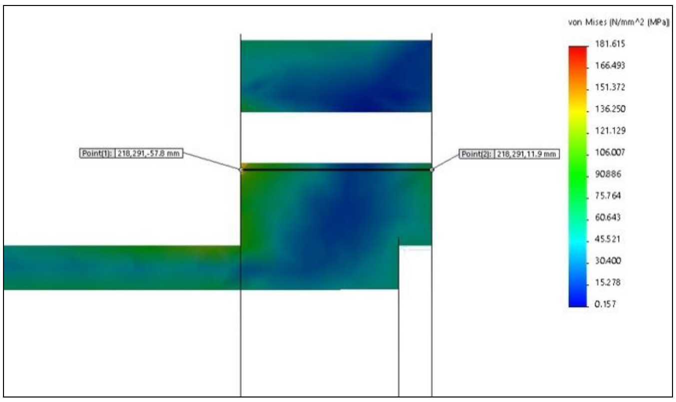

Fig. 9 Stress classification line through flange

The component wise membrane and bending stresses are as given in table XI-

Table IX. COMPONENT WISE MEMBRANE AND BENDING STRESSES AT FLANGE

Quantity(N/mm^2)	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ
Membrane Stress	6.5149	9.8379	0.4168	-8.3892	-7.2917	2.7082
Bending (Point 1)	49.919	35.95	-1.0882	-38.305	-1.1323	1.5911
Membrane Stress + Bending (Point 1)	56.434	45.787	-0.6714	-46.695	-8.424	4.2993
Bending (Point 2)	-49.919	-35.95	1.0882	38.305	1.1323	-1.5911
Membrane Stress + Bending (Point 2)	-43.404	-26.112	1.5051	29.916	-6.1593	1.1171

The maximum values of membrane and membrane+bending stresses as per von Mises is as shown in table X.

Table X. EQUIVALENT MEMBRANE AND BENDING STRESSES AT FLANGE

Quantity (MPa)	Max. Prin.	Mid. Prin.	Min. Prin.	von Mises
Membrane Stress	19.233	2.524	-4.988	21.474
Membrane Stress + Bending (Point 1)	98.948	4.992	-2.391	97.854
Membrane Stress + Bending (Point 2)	3.128	-4.778	-66.892	69.488

1) *At flange to shell transition:*The stress concentration is also seen near the shell and flange intersection and equivalent stress in that location is more than allowable stress value as shown in figure 10. So it needs to be linearized so that it can be compared to allowable stresses as per code.

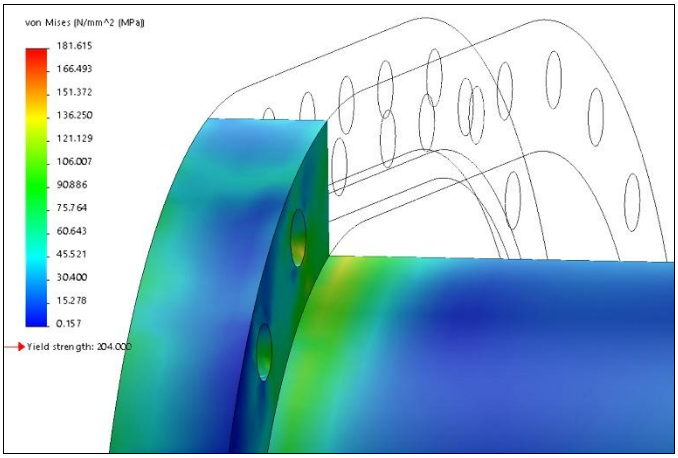

Figure 10. Stress concentration zone near flange to shell transition

Stress concentration line is marked on the section of stress concentrate

Start point Point 1(-255, -238.04, -70.366)mm

End point Point 2(255, -254.47, -70.362)mm

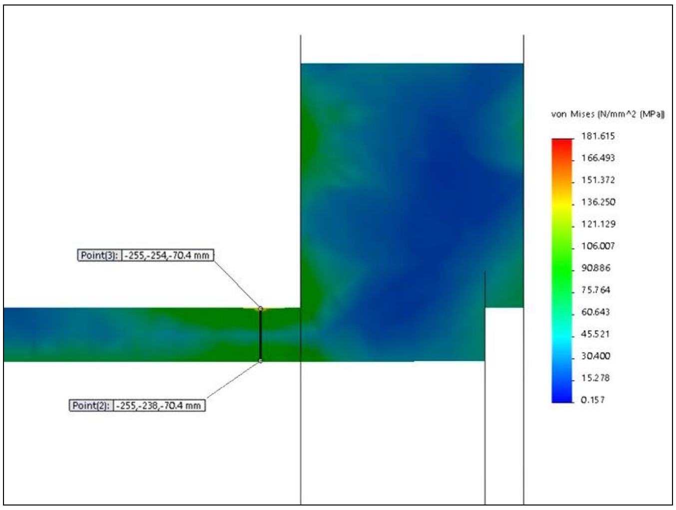

Figure 11. Stress classification line near flange to shell transition

The component wise membrane and bending stresses are as given in table XI-

Table XI. COMPONENT WISE MEMBRANE AND BENDING STRESSES NEAR FLANGE TO SHELL TRANSITION

Quantity(MPa)	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ
Membrane Stress	43.517	11.268	14.426	-14.492	4.4883	-9.5668
Bending (Point 1)	-15.685	-13.037	-85.387	8.7169	-14.398	8.9527
Membrane Stress + Bending (Point 1)	27.832	-1.7692	-70.962	-5.775	-9.9093	-0.6141
Bending (Point 2)	15.685	13.037	85.387	-8.7169	14.398	-8.9527
Membrane Stress + Bending (Point 2)	59.202	24.305	99.813	-23.209	18.886	-18.52

The maximum values of membrane and membrane+bending stresses as per von Mises is as shown in table XII.

TableXII. EQUIVALENT MEMBRANE AND BENDING STRESSES NEAR FLANGE TO SHELL TRANSITION

Quantity (MPa)	Max. Prin.	Mid. Prin.	Min. Prin.	von Mises
Membrane Stress	50.724	17.254	1.232	43.740
Membrane Stress + Bending (Point 1)	29.840	-2.773	-71.966	90.043
Membrane Stress + Bending (Point 2)	114.841	56.618	11.860	89.438

1) *Near shell to tubesheet joint:* The stress concentration is maximum at the shell and tubesheet intersection and equivalent stress in that location is 31% more than allowable stress value as shown in figure 12. So it needs to be linearized so that it can be compared to allowable stresses as per code.

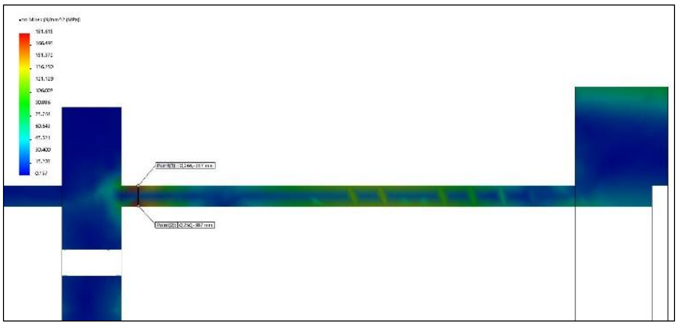

Figure 12. Stress classification line on shell

Stress concentration line is marked on the section of stress concentrated zone with following points :

Start point	Point 1(0, 265.76, -387.48)mm
End point	Point 2(0, 250.39, -387.05)mm

The component wise membrane and bending stresses are as given in table XIII-

Table XIII. COMPONENT WISE MEMBRANE AND BENDING STRESSES AT SHELL

Quantity(N/mm²)	Normal X	Normal Y	Normal Z	Shear XY	Shear XZ	Shear YZ
Membrane Stress	3.4499	11.167	-11.296	-0.0635	0.4351	9.379
Bending (Point 1)	-41.597	-11.773	-133.97	0.296	0.2869	1.2273
Membrane Stress + Bending (Point 1)	-38.147	-0.6063	-145.26	0.2324	0.7220	10.606
Bending (Point 2)	41.597	11.773	133.97	-0.296	-0.2869	-1.2273
Membrane Stress + Bending (Point 2)	45.046	22.939	122.67	-0.3596	0.1482	8.1517

The maximum values of membrane and membrane+bending stresses as per von Mises is as shown in table XIV.

Table XIV. EQUIVALENT MEMBRANE AND BENDING STRESSES AT SHELL

Quantity (MPa)	Max. Prin.	Mid. Prin.	Min. Prin.	von Mises
Membrane Stress	14.568	3.459	-14.708	25.598
Membrane Stress + Bending (Point 1)	0.169	-38.144	-146.042	131.315
Membrane Stress + Bending (Point 2)	123.333	45.052	22.272	91.816

D. Results:

The stresses at the most stress concentrated zones are compared with the allowable limits as per ASME given in table XV.

TableXV. RESULTS OF THE STRESS LINEARIZATION

Quantity (MPa)	Allowable stress	At Flange thickness	At flange to shell transition	Near shell to tubesheet joint
		von Mises	von Mises	von Mises
Membrane Stress	138	21.474	43.740	25.598
Membrane Stress + Bending (Point 1)	207	97.854	90.043	131.315
Membrane Stress + Bending (Point 2)	207	69.488	89.438	91.816

As the linearized stress values are far below the allowable stress values, the design is safe. The analytical value of stress is 127.930 MPa and the maximum value found after linearization in flange is 97.854 MPa; this gives the clear idea that the analytical method formulated gives the safer design and its errors on the safe side.

IV. PROPOSED BOLT TIGHTENING SEQUENCE

One of the activities essential to leak-free performances is the joint assembly process. The required bolt tightening sequence during the assembly of bolted flange joint is proposed as given in figure 13. The bolt tightening sequence is decided based on the guidelines given in [18].

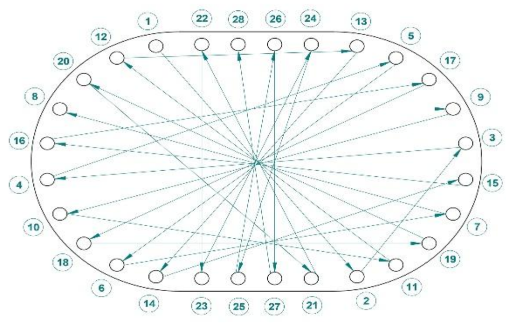

Figure13. Bolt tightening sequence

V. Conclusions

The design of obround flange has be done analytically with the equivalent circular flange method and its finite element analysis has been conducted which shows the equivalent stresses generating in flange. These stresses are linearized to obtain the membrane and bending stresses and are compared to allowable limit. The key conclusions are as listed below-

- The equivalent circular flange method is the most appropriate method for obround flange design as it gives the maximum stress for which we have to design the flange and which also complies the ASME codes.
- Equivalent circular flange method is approximate analytical method for design of obround flange with some limitations that factor z should not be larger than 2.5 and length to width ratio should be less than 2.
- The stress values calculated using with this method falls on the high side as required for a safe design.
- The stress concentration in flange is maximum near bolt hole at the transition area of flange from straight portion to curve portion and in shell it is maximum at the shell to tubesheet attachment.
- The results from stress linearization of stress concentration zones show that the stresses occurring in the flange and shell are within allowable limits as specified by ASME.
- Though it is an approximate method, it can be used successfully to reduce the height of the pressure vessel equipment considerably. The height reduction achieved in our case is 22%.

Acknowledgment

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References

[1] Adolf E. Blach; “Non circular pressure vessel flanges: New design methods”, Fluid sealing, Springer-science & business Media; pp.247-265.

[2] Hildegard Zerres, YannGuerout, “Present calculation methods dedicated to bolted flanged connections”, International Journal of Pressure Vessels and Piping 81 (2004), pp. 211–216.

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

[4] Muhsen Al-Sannaa and Abdulmalik Alghamdi, “Two dimensional finite element analysis for large diameter steel flanges”, 6th Saudi Engineering conference (2002), Vol.5, pp. 397-408.

[5] Waters EO, Rossheim DB, Wesstrom DB, Williams FSG. Development of General Formulas for Bolted Flanges. Chicago, IL: Taylor-Forge and Pipe Works; 1937.

[6] Dennis R. Moss, “Pressure Vessel Design Manual”, Fourth Edition, 2013, Gulf Professional Publishing.

[7] Kanti K Mahajan, “Design of Process Equipment”, Second Edition, Pressure Vessel Handbook Publishing INC.

[8] Lloyd E. Brownell and Edwin H. Young, “Process Equipment Design”, Wiley-Interscience.

[9] John H. Bickford, “Gaskets and Gasketed Joints”, CRC press.

[10] B. S. Nau, “Fluid Sealing”, Springer-Science &Business Media.

[11] ASME BPVC section VIII Division-1, “Rules for Construction of Pressure Vessels”, 2013, ASME International.

[12] ASME BPVC section VIII Division-2, “Alternative Rules for Construction of Pressure Vessels”, 2013, ASME International.

[13] ASME BPVC section II Part D, “ASME Boiler and Pressure Vessel Code – Materials (Metric)”, ASME International.

[14] ASME PTB-3, “ASME BPVC section VIII- Division 2 Example Problem Manual”, 2013, ASME International.

[15] Swedish piping code for flange joint

[16] British standards PD5500:2012 enquiry case 5500/133; “Flat unstayed ends of non-circular shape and associated flanges”.

[17] Standards of the tubular exchanger manufacturers association, Eighth edition.

[18] ASME PCC-1-2013, “Guidelines for Pressure Boundary Bolted Flange JointAssembly”, ASME international.