

# Design of Obround Flange for Pressure Vessel Application by Analytical Method and FEA to Comply with ASME code

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## ABSTRACT

Flanged joints are essential components in nearly all pressurized systems and piping; however they constitute one of the most complex parts of design. A wrong design can lead to major leakage, affect the system performance and can be hazardous for operators. Various factors influence the successful design and operation of a flange joint in service. The bolted flange joint involves interaction between bolting, flange, and gasket. For various industrial applications such as vapour absorption chillers, unfired pressure vessels having flange size more than 600 mm; there is a need of non-circular flanges having reduced heights than traditional flanges. The shape considered for flange (Slip-on type without hub) is obround shape; whose investigation involves finding an appropriate analytical method to be used for design of this obround flange and which can also 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 flange 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.

**Keyword:** - Flange, Obround, Non-circular, Pressure vessel, Linearization, FEA

## 1. INTRODUCTION

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 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 procedure available for bolted flange connections; so these flanges cannot be designed directly with the rules of ASME BPVC Section 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).

## 2. LITERATURE REVIEW

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 Div.1,

| 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 <sup>2</sup> )  | 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 and back of flange (mm)                          |
| b <sub>0</sub> : Basic gasket seating width (mm)   | S <sub>a</sub> : Allowable bolt stress at gasket seating temperature (atmospheric temperature) (MPa)                 |
| C: Bolt circle diameter (mm)   | S <sub>b</sub> : Allowable bolt stress at operating temperature (MPa)  |
| c: Clearance between OD of shell and ID of flange (mm)   | S <sub>fa</sub> : Allowable stress for flange material at gasket seating temperature (atmospheric temperature) (MPa) |
| d: Bolt hole diameter (mm)   | S <sub>fb</sub> : Allowable stress for flange material at operating temperature (MPa)                                |
| D: Hydraulic diameter (mm)   | S <sub>H</sub> : Calculated longitudinal stress in hub (MPa)   |
| D' <sub>m</sub> : Equivalent mean gasket diameter assumed for design purpose (mm)  | S <sub>R</sub> : Calculated radial stress in flange (MPa)  |
| D <sub>L</sub> : Maximum inside diameter of obround shell (mm)   | S <sub>T</sub> : Calculated tangential stress in flange (MPa)  |
| D <sub>m1</sub> : maximum mean gasket diameter for non-circular flange (mm)  | S <sub>TS</sub> : Minimum tensile strength (MPa)   |
| D <sub>m2</sub> : minimum mean gasket diameter for non-circular flange (mm)  | S <sub>Y</sub> : Minimum yield strength (MPa)  |
| D <sub>S</sub> : Minimum inside diameter of obround shell (mm)   | T: Design temperature (°C)   |
| E: Radial distance from bolt circle to outside diameter of flange (mm)   | t: Flange thickness (mm)   |
| G: Diameter at location of gasket load reaction (mm)   | t <sub>g</sub> : Thickness of gasket (mm)  |
| h <sub>D</sub> : Radial distance from bolt circle to circle on which H <sub>D</sub> acts (mm)  | t <sub>n</sub> : Nominal thickness of the shell, pipe or nozzle to which the flange is attached (mm)                 |
| H <sub>D</sub> : Hydrostatic end force on area inside flange (N)   | W: Design bolt load for the gasket seating condition (N)   |
| h <sub>G</sub> : Radial distance from gasket load reaction to bolt circle (mm)   | w: Width of the straight portion of obround flange (mm)  |
| H <sub>G</sub> : Gasket load = W <sub>m1</sub> - H for operating condition (N)   | y: Gasket / joint contact surface unit seating stress (MPa)  |
| h <sub>T</sub> : Radial distance from bolt circle to circle on which H <sub>T</sub> acts (mm)  | z: Factor for conversion of obround shape to circular shape  |
| H <sub>T</sub> : Difference between total hydrostatic end force and the hydrostatic end force on area inside flange = H - H <sub>D</sub> (N) |  |
| J: Flange rigidity   |  |
| m: Gasket factor   |  |

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.

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. 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 results 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. 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.

### 3. 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.

#### 3.1 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. 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

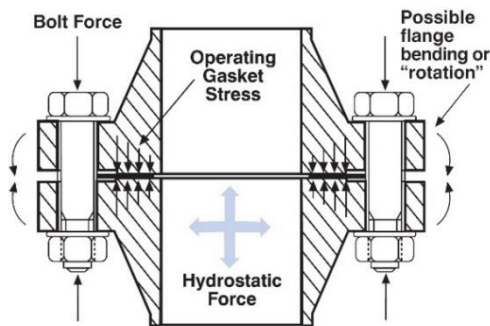


Fig -1: forces acting in a bolted flange joint assembly

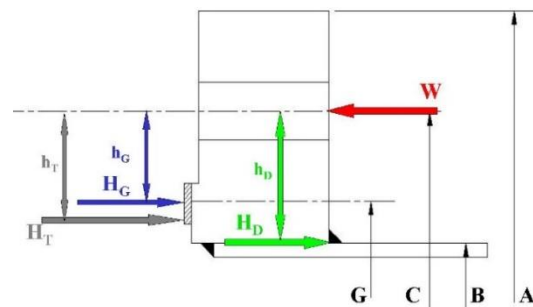


Fig -2: Forces represented on flange (Slip-on flange without hub)

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 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  $H_r$ . Load  $H_G$  is the force required to seat the gasket into the flange gasket face which is based on gasket physical properties.

#### 3.2 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  $D'_m$

$$D'_m = k_4 D_{m1} \dots \dots \dots (a)$$

**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

$$z = 3.4 - \frac{2.4 D_s}{D_L} \dots \dots \dots (b)$$

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.

$$B = D_s \sqrt{z} \dots \dots \dots (c)$$

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

$$D = B = \frac{4A}{P} \dots \dots \dots (d)$$

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

**3.3 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 -1:** 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 below. Similarly for the given operating parameters, gasket is selected and which has following parameters required in design as specified in table III.

**Table -2:** 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 <sup>2</sup> |
| Total bolts being used  | n     | 28     | Qty             |

**Table -3:** 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   |

### 3.4 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 4. 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 138 MPa and for selected bolting material is 172 MPa.

**Table -4:** 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 <sup>3</sup> |
| Modulus of Elasticity  | 185          | 185                    | GPa               |

**Table -5:** 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 <sup>3</sup> |
| Tensile strength                    | 15    | MPa               |
| Compressibility                     | 7-11% |                   |
| Recovery                            | ≥50%  |                   |
| Water absorption                    | <5%   |                   |

2) **Gasket Material:**

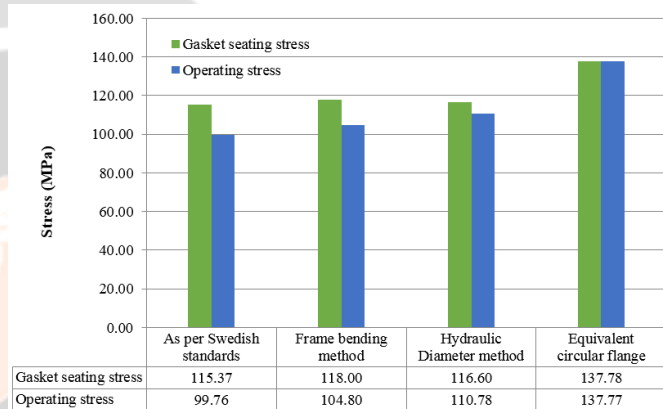
Material used for gasket compressed non-asbestos fiber reinforced type (CNAF). We have considered Spitman AF 154 (Champion seals Pvt. Ltd.) as it has superior performance compressed jointing sheet incorporating a blend of special heat resisting aramid fibres with a high quality nitrile elastomer binder. Properties of gasket are as given in table 5.

**3.5 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 65 mm flange thickness, M30 (28) bolting and 20 mm gasket width. We plot the stresses in gasket seating and operating condition of flange as shown in chart 1.

It is clear from the chart 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.



**Chart -1:** Trend of the stress vs. various methods

**3.6 Design by Formula (Analytical Method)**

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.

**4. 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.

#### 4.1 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.

#### 4.2 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 evaporator shell and back side of the generator shell to restrain it in all direction as shown in figure 3. The pressure of 1.054 MPa is applied at the inside of the obround shell. 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 26.4 mm and 32.45 mm 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. 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%.

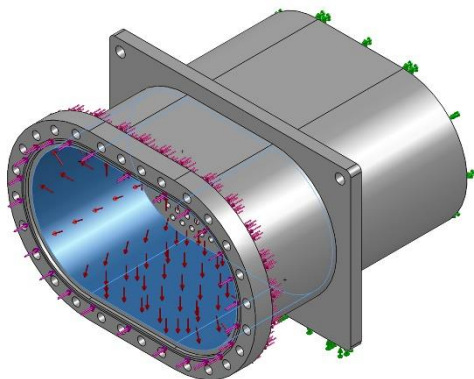


Fig -3: Boundary conditions and loading

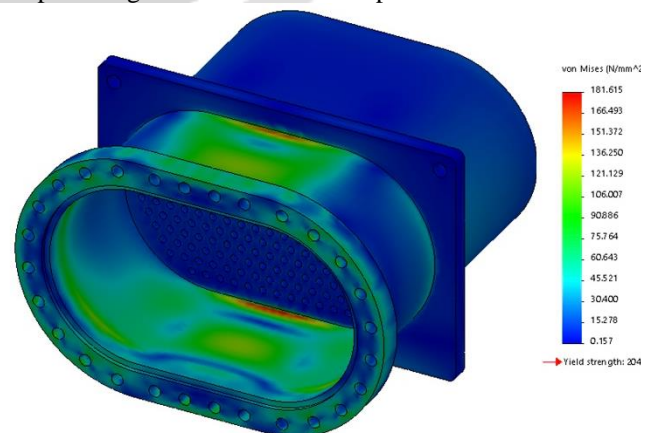


Fig -4: von Mises stress plot of obround flange (a)

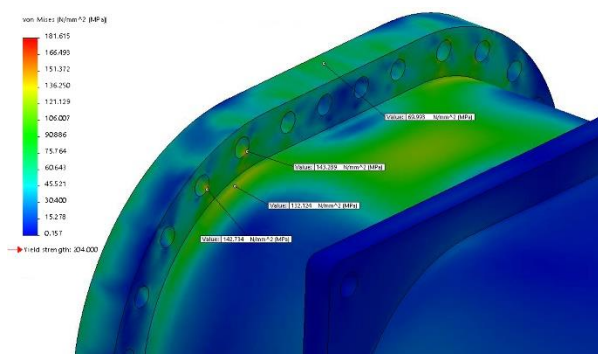


Fig -5: von Mises stress plot of obround flange (b)

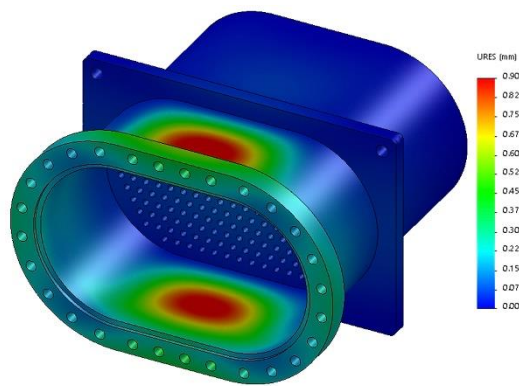


Fig -6: Deformation plot of obround flange

3) **Results:**

The von Mises stress plots of flange analysis are given in figure 4 and figure 5. The maximum stress generated 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.

The deformation plot of obround flange subjected to given loading conditions is as shown in figure 6. The maximum deformation obtained is 0.901 mm which is in the straight portion of shell. Maximum deformation occurring in flange is within the range of 0.600 mm to 0.715 mm. The deformation allowed is 5 mm; so the design is safe for deformation & also deformation is negligible as compared to overall size of the flange. 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.

**5. 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 4 & 5. 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 [14] are as given below in table VIII.

**Table -6:** Allowable Stress Values for Localized Stress

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

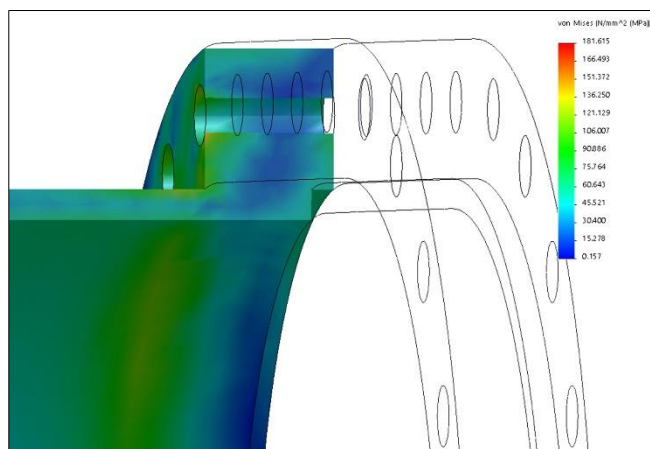
**5.1 Stress Linearization / Stress Classification**

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

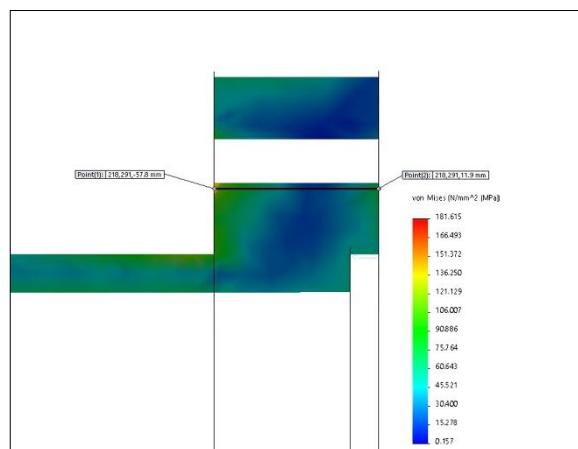
Stress concentration line is marked on the section of stress concentrated zone with following points :

- Start point Point 1(218, 290.59, -57.764)mm
- End point Point 2(218, 290.59, 11.915)mm





**Fig -7:** Stress concentration zone in flange



**Fig -8:** Stress classification line through flange

The maximum values of membrane and membrane + bending stresses as per von Mises is as shown in table 7.

**Table -7:** Equivalent Membrane and Bending Stresses at Flange

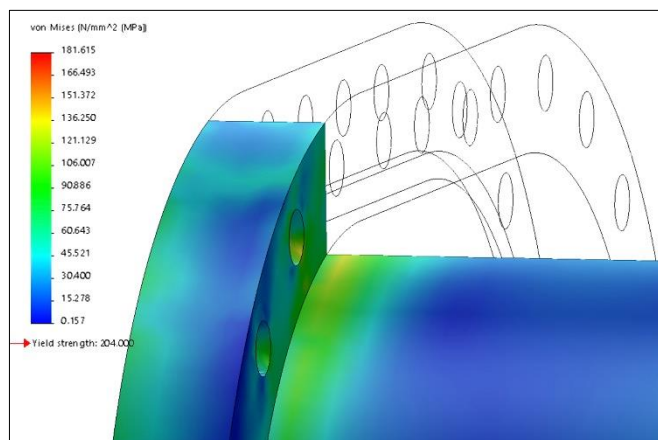
| Quantity (MPa)                      | Max. Prin. | Mid. Prin. | Min. Prin. | von Mises |
|-------------------------------------|------------|------------|------------|-----------|
| Membrane Stress                     | 19.233     | 2.524      | -4.988     | 21.474    |
| Membrane Stress + Bending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

2) **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 9. So it needs to be linearized so that it can be compared to allowable stresses as per code.

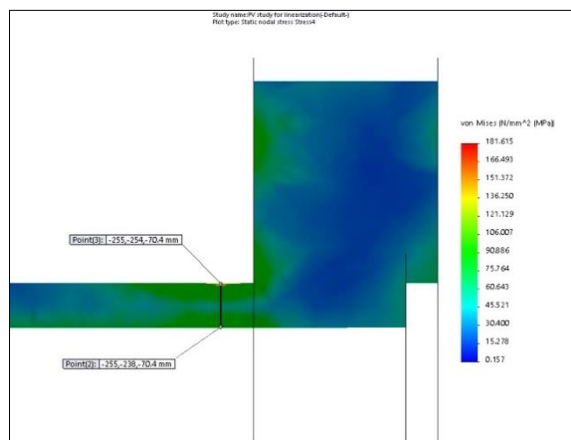
Stress concentration line is marked on the section of stress concentrated zone with following points :

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

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



**Fig -9:** Stress concentration zone near flange to shell transition



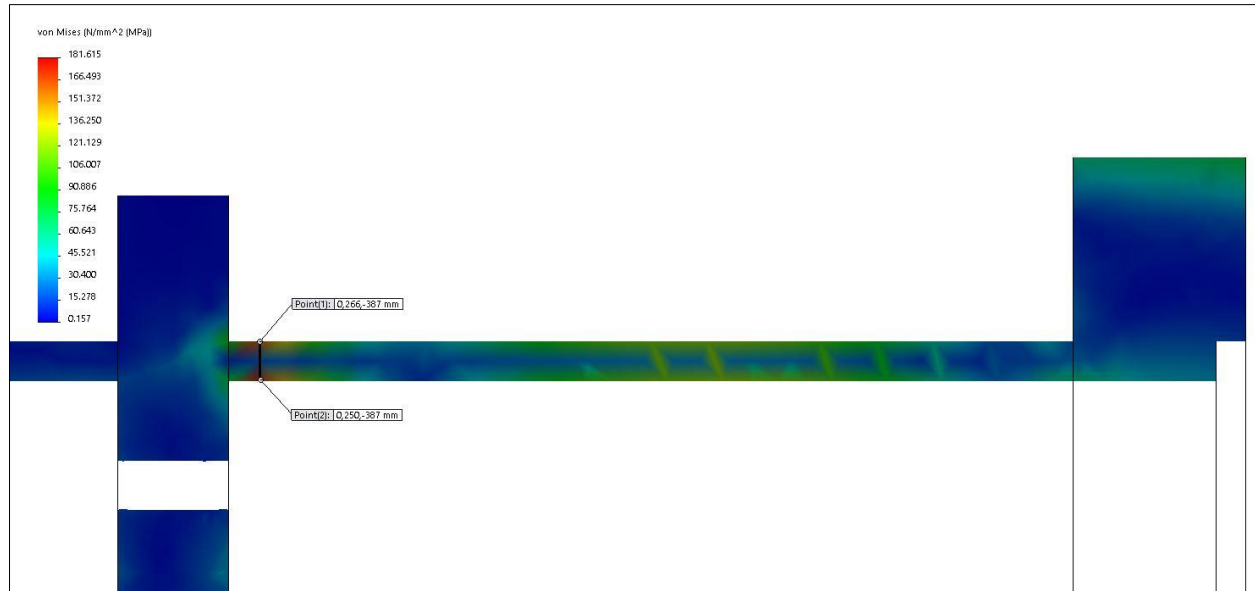
**Fig -10:** Stress classification line near flange to shell transition

The maximum values of membrane and membrane + bending stresses as per von Mises is as shown in table 8.

**Table -8:** 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    |

- 4) **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 11. So it needs to be linearized so that it can be compared to allowable stresses as per code.



**Fig -11:** 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 maximum values of membrane and membrane + bending stresses as per von Mises is as shown in table 9.

**Table -9:** 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    |

**5.2 Results**

The stresses at the most stress concentrated zones are compared with the allowable limits as per ASME given in table 10.

**Table -10:** Results of the Stress Linearization

| Quantity (MPa)                      | Allowable stress | At flange thickness | At flange to shell transition | Near shell to tubesheet joint |
|-------------------------------------|------------------|---------------------|-------------------------------|-------------------------------|
|                                     |                  | von Mises stress    | von Mises stress              | von Mises stress              |
| 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.

**6. 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 code.
- 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%.

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