Design and Leak-tightness Analysis of Volute Casing for Axially Split Multistage Centrifugal Pump

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Abstract - ***In axially split centrifugal pumps, leakage of internal fluid along the casing flanges due to high pressure of fluid is an important factor which critically affects the pump performance. This paper deals with the design and leak-tightness analysis of four stage BB3 pump casing. The design approach for pump components depends on its type. American Petroleum Institute (API) Standard 610 is the international standard used for classification, design and testing of pumps. The pumps are classified into many types depending on construction, geometrical features and working and centrifugal pump is one of the most common types. As per API standard 610, BB3 is a multistage axially split centrifugal pump which is normally used in chemical industries. The casing of the BB3 pump is designed on the basis of impeller outlet parameters to satisfy both strength and leak tightness criteria. Due to intricate profile of casing and sealing gasket, the casing needs to be analyzed using three-dimensional (3D) stress analysis. Geometric modeling of top and bottom casings is done by using 3D modeling software Creo 2.0 and analyzed for strength and leak tightness against the maximum allowable working pressure using finite element code. The design calculations are performed as per industry accepted handbook and API standard 610. The casing is analyzed in 3D using commercial software ANSYS®. The deflection and equivalent Von Mises stresses within casing are studied. The equivalent Von Mises stress results are used to check the strength of the pump casing. The deformation results on top and bottom flanges faces are evaluated for leak-tightness analysis.***

Key Words: *API 610, BB3, Centrifugal pump, FEA, MAWP, volute.*

1. INTRODUCTION

A pump is machinery used to transfer the fluids such as water, oil, chemicals, etc. One of the most common types of pump used in industry is the centrifugal pump. Centrifugal pumps can have single or multiple stages. For multistage pumps, several impellers are mounted on a single shaft and the stages operate sequentially, that is, the discharge from the first stage flows into the eye of the second stage, the discharge from the second stage flows into the eye of the third stage, and so on. The flow rate is same through all stages, but each stage develops an additional pressure rise to cumulatively develop the required total head. Thus, a very large discharge pressure or head can be developed by a multistage pump.

American Petroleum Institute Standards-610 [1] is the international standard used for design and testing of pumps. It also gives the pump classification and identification of types As per API, the centrifugal pumps are divided into three types on the basis of number of stages, orientation and the position of principal joint; the three types are: overhung, between bearing (BB) and vertically suspended. Between bearing pumps are further divided into BB1, BB2, BB3, BB4 and BB5. Type BB3 is a multistage axially split centrifugal pump (Fig -1) which is normally used in chemical industries, oil industries, descaling applications in steel industries, etc. The BB3 pump is usually employed for high flow rates up to 1500 m^3/hr and high pressure heads up to 1650 m. One of the major advantages of BB3 pump is its easy maintenance as compared to other types because the rotor can be removed by disassembling only upper half casing.

API standard recommends that the pumps are to be designed for minimum service life is 20 years and for uninterrupted operating service of at least for 3 years. The standard also includes basic design of pump casing. The casing is designed to operate without leakage while subjected to simultaneous action of maximum allowable working pressure (MAWP) and maximum operating temperature.

Fig -1: Typical BB3 Pump [2]

Proper selection of material is an important parameter while designing the casing of the centrifugal pump. For many pumping application, cast iron is preferred for pump casings primarily on the basis of cost. For corrosive or hazardous petroleum products, cast steel or cast stainless steel is preferred [3].

While there are many methods to perform the design of volute casing, the procedure recommended by Karassik [4] and Stepanoff [5] are widely used by industry. Towards the end of the basic design for pump, preliminary drawings for volute casing are generated. At this stage, detailed finite element analysis for the casing may be carried out prior to release of production drawings.

The centrifugal pump design is typically verified by FE techniques. Thin et al. carried out computational analysis and performance analysis of centrifugal pump. Shock losses, impeller friction losses, volute friction losses, disk friction losses and recirculation losses of centrifugal pump were considered in performance analysis of centrifugal pump. Prasad et al. [6] analyzed a centrifugal pump impeller which is made of three different alloy materials, namely, Inconel 740, Incoloy 803 and Warpaloy to estimate its performance. The investigation was done by using CATIA for modeling the impeller and ANSYS for determination of the variation of stresses, strains and deformation across profile of the impeller. HYPER MESH 9.0 was used to generate optimum FE discretization of the impeller to obtain accurate results. A structural analysis was carried out to investigate the stresses, strains and displacements of the impeller and modal analysis was performed to investigate the frequency and deflection of the impeller. By comparing the results obtained for three different alloys, the best alloy for impeller was suggested. Gowd et al. [7] carried out a finite element analysis for some mechanical components of the high-pressure hydraulic pumps by ANSYS software. For obtaining some proper results, the real model of the loading diagram had built. They found that the possibility of increasing the nominal operating pressure up to 420 bar, without changing the materials. Mona Golbabaei Asl et al. [8] carried out experimental and FEM failure analysis for optimization of a centrifugal-pump volute casing.

The axially split construction of pump is prone to leakage along the flange due to high working pressure and intricate gasket profile (Fig -2). Analytical procedure to check such leakage is complicated. The leak tightness along the gasket is an important aspect of casing design. Singer and Johne [9] have provided guidelines for rigorous pump casing design include the methods for leak tightness analysis.

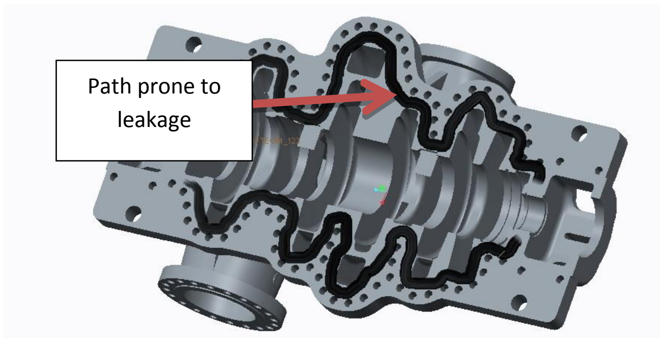

Fig -2: Path prone to leakage of BB3 pump

In this paper, initially the basic geometry of pump casing is calculated using procedure suggested by Karassik [4]. Later the CAD model for the pump volute casing is developed. This is used to create finite element model for the casing. The casing is checked for induced stresses against the maximum allowable working pressure (MAWP) with appropriate boundary conditions. Deformations at critical regions are checked to analyze the leak-tightness.

2. DESIGN OF CASING

The present study is performed on a new model of four stage BB3 pump being developed by KEPL, Kirloskarwadi. Material selected for BB3 pump casing is ASTM-A352-GR LC3 Cast/carbon steel (yield strength = 275 Mpa) based on API standard recommendation and customer requirement. Design parameters for the pump are follows.
Required head (H) = 520 m
Required flow rate (Q) = 1040 m^3/hr
Speed of Pump (N) = 1400 rpm

2.1 Basic Design of Volute Casing

First stage volute design:
For BB3 pump, first stage is having double suction impeller as per API standard.

Specific Speed (N_s):

$$N_s = \frac{N * \sqrt{Q}}{H_1^{0.75}} \tag{1}$$

where
N_s= specific speed
N= Speed of pump (r.p.m)
H_1= Head developed by first impeller

$N_s = 23.040$

Fig -3 correlates specific speed vs. ratio of throat velocity to impeller peripheral speed [10]. From this Fig. 3, average value for ratio,

$$\left(C_{thr}/u_2\right) = 0.505$$

$C_{thr.}$ = Throat velocity (m/s)
u_2 = Impeller peripheral speed

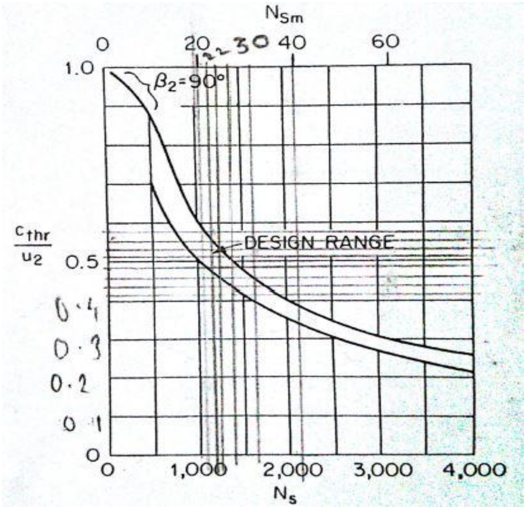

Fig -3: Specific speed vs. ratio of throat velocity to impeller peripheral speed [10]

Hence, $C_{thr.} = 23.804$ m/s
Tentative throat area ($A_{thr.}$):

$$A_{thr} = \frac{Q*10^6}{C_{thr}} \qquad (2)$$

$$A_{thr} = 12098.807\ mm^2$$

Assuming a circular throat section, its radius is

$$r_{thr} = \left(\sqrt{\frac{A_{thr}*4}{\pi}}\right)/2 \qquad (3)$$

$$r_{thr} = 62.058\ mm$$

Distance of the throat centre of the throat section from axis is calculated as:

$$r_4 = r_2 + t + r_{thr}$$

where
r_4= Distance of the throat centre from axis (mm)
r_2= Radius of impeller (mm)
t= Tongue distance (mm)

The intermediate volute areas are calculated on the basis of constant angular momentum. Stepanoff [10] recommends a constant mean velocity for all volute sections, which results in volute areas being proportional to the central angle θ (Fig. 4).

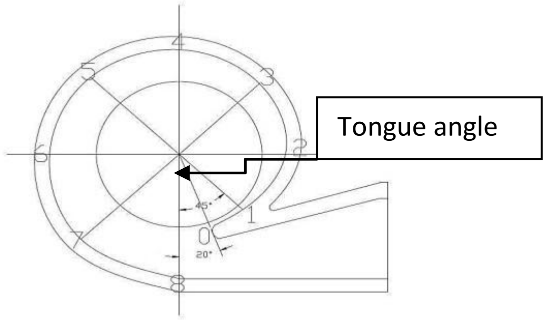

Fig -4: Volute Casing

Intermediate areas can be calculated as follows,

$$A_\theta = \frac{A_{thr}*\theta}{360} \qquad (4)$$

Radius of intermediate area can be calculated as,

$$r_\theta = \left(\sqrt{\frac{A_\theta*4}{\pi}}\right)/2 \qquad (5)$$

where
r_θ = Radius of intermediate section of volute at angle θ (mm)
A_θ = Area of intermediate section of volute at angle θ (mm^2)
θ = Central angle (degree)
The distances of section centre at angle θ from axis is calculated as follows,

$$r'_\theta = r_2 + t + r_\theta \qquad (6)$$

where
r_θ' = Distance of section centre at angle θ from axis (mm)

First, the velocities of volute areas at angle θ are calculated, for which previously calculated distance of centre of each section at angle θ from axis (r_θ') are used.

$$C_\theta = \frac{r_2*C*C_{u3}}{r'_\theta} \qquad (7)$$

where
C_θ = Velocity of volute areas at angle θ (m/s)
C = Constant factor
C_{u3} = Peripheral component of impeller discharge velocity (m/s)
Intermediate areas can be calculated as follows,

$$A_\theta = \frac{Q*10^6*\theta}{360*C_\theta} \qquad (8)$$

By Karassik method, the tentative choice of the throat velocity is checked by following equation,

$$C = \frac{r_4*Cthr.}{r_2*C'u_3} \qquad (9)$$

where
C'_{u3} =Peripheral component of impeller discharge velocity (m/s)

$$C = 1.2449$$

A modification of volute dimensions by considering the effect of constant factor C is suggested by Karassik [4]. The dimensions obtained for volute from above procedure may be acceptable, but for smoother volute and frictionless flow, factor C must be considered.

For frictionless flow, C=1, according to the law of constant angular momentum. Experience has shown that C=1 is indeed a good design value for volute of large pumps or very smooth volutes of medium size pumps. For commercially cast volutes of medium size and small pumps, the value C=0.9 gives a reasonable approximation.

The value for constant factor C is calculated from equation (9) is 1.2449. But according to the comment under volute casing this appears to be a very reasonable factor C. So the volute dimensions are modified from throat area which will be modified by the throat velocity by considering frictionless flow i.e. constant factor C=1.

Putting the value of constant factor C=1 in equation (9) and get modified throat velocity as follows

$$C = 1 = \frac{(C_{thr} * 385.733)}{(24.3818 * 302.5)}$$

$$C_{thr} = 19.120 \text{ m/s}$$

In this way, the throat velocity is modified. Here comparing this modified throat velocity with previous throat velocity, it is observed that the throat velocity are decreased (C_{thr}.(Modified) = 19.120 m/s and C_{thr}. = 23.804 m/s). As the velocities near the tongue are decreased, this implies that the throat area is increased, which is helpful to avoid the excessive losses due to higher flow rates. By using modified throat velocity and following the same design, the final volute dimensions such as A_θ and r_θ are finalized after doing three iterations.

The trapezoidal shape for intermediate area of volute is selected for better efficiency (Fig -5). The trapezoidal shape makes the entrance width of the volute to be appreciably greater than the impeller discharge width (b_2) which leads to smooth flow.

$$b_3 = b_2 + (2 * q) = 85 \text{ mm}$$

$$b_{max} = b_3 + (2 * x * \tan(\emptyset/2)) \qquad (10)$$

where
b_3 = Minimum entrance width of volute casing (mm)
b_2 = Width of impeller (mm)
b_{max} = Maximum entrance width of volute casing (mm)
q = Shroud thickness (mm)
x= Distance between any radius r_θ and impeller radius r_2 (mm)
$\emptyset$ = Angle with radial line of section (degree), generally considered as 60 degree

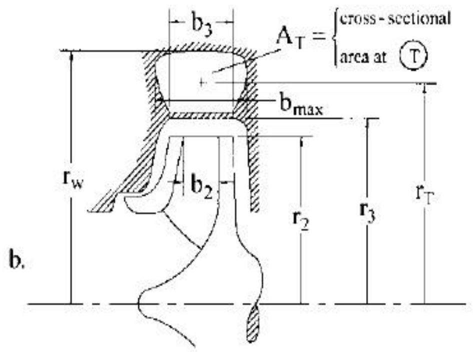

Fig -5: Trapezoidal shape for intermediate area of volute[4]

Hence, $b_{max} \approx 110$(mm)

The tongue angle ($\emptyset_t$) may found by assuming that the flow follows a logarithmic spiral.

$$r = r_2 * e^{\left(\frac{\pi * \emptyset}{180} * \tan\beta_2\right)} \qquad (11)$$

Hence, $\emptyset_t = 10.104°$

And to avoid shock losses, consider the tongue angle $\emptyset_t = \beta_2 = 20°$

Calculation of 2nd, 3rd and 4th stage volute casing:-
Next three stages produce equal heads and their impeller dimensions are same. Therefore the volute casing geometry is same for 2nd, 3rd and 4th stage. And design procedure is same as that for the first stage.

Design of pump casing volute thickness:
The maximum discharge pressure is the maximum suction pressure plus the maximum differential pressure that the pump is able to develop when operating with the rated speed and specified normal relative density. The maximum discharge pressure is obtained from following expression.

$$P_d = \rho * g * h \qquad (12)$$

where
P_d= Maximum discharge pressure (N/m^2)
ρ= Density of pumping fluid (kg/m^3)
= 1000 kg/m^3
h = 630 (m) = Shut off head

P_d= 1000*9.81*630 = 61.803 bar
Total head = Suction head + Max .Discharge Head +1.1% P_d
= 1000*9.81*11.7 + 61.803 + 1.1*61.803/100
= 63.629 bar

The pressure temperature rating chart given by American Society of Mechanical Engineers (A.S.M.E.) B 16.5 [11] for the given material is used to choose the pump rating class.

Table -1: Pressure-Temperature Ratings for Group 1.2 Materials [11]

Class Temp. (° C)	150	300	400	600	900
-29 to 38	19.8	51.7	68.9	103.4	155.1
50	19.5	51.7	68.9	103.4	155.1
100	17.7	51.5	68.7	103.0	154.6
150	15.8	50.2	66.8	100.3	150.5
200	13.8	48.6	64.8	97.2	145.8
250	12.1	46.3	61.7	92.7	139.0

For maximum differential pressure P_d=63.629 bar and operating temperature T= 65° C, 600 rating class is found suitable for the pump casing material (Table -1). The maximum working pressure for 600 rating class at ambient temperature is considered as design pressure for the casing thickness calculations.

Maximum working pressure at ambient temperature for 600 rating class = 103.4 bar. Pump casing volute thickness is calculated as per the rules of ASME section VIII, division-2 [12].

Casing volute thickness is given by following equation.

$$T = \frac{P*r}{\{(\sigma*\eta)-(0.6p)\}} \tag{13}$$

where,
T = Casing volute thickness (m)
P = Maximum working pressure at ambient temperature (N/m^2) = 103.4 bar
r = Maximum radius of impeller which can fit in casing from volute CAD model (m) = 730 mm
η = Casting factor = 0.8
σ = Design stress (N/m^2)

Design stress value is taken as minimum value obtained from 0.25 times the tensile strength and 0.67 times the yield strength which is 1240 bar for casing material.

Thus, $T = 51.78\ mm$.

The provided thickness for volute casing is 52 mm.

Flange thickness calculation:
Using the geometry of 3D-model of casing, total volute area is calculated as,
Total Volute area = 1093074 mm^2
Considering, Casing as a cylindrical body,

$$\left(\frac{\pi * D^2}{4}\right) = 1093074$$

D = 1179.722 mm = 46.445 inches.

From ASTM B16.47-2006, thickness of casing at 103.4 bar pressure and at ambient temperature 46” class 600 flanges is 179.4 mm.

The thickness of casing at maximum discharge pressure is calculated as 110 mm.

And by considering corrosion allowance and for more safety thickness of casing is finalized as 140 mm.

The total separating force acting on casing halves is calculated using total volute area and the pressure exerted on volute. Size and minimum number of bolts required to withstand this force is calculated as M48 x 52 numbers.

2.2 Structural Analysis of Casing Using Ansys

Geometric Modeling of casing is done using 3D modeling software PTC Creo 2.0 and the geometry is imported in ANSYS software. Structural analysis of casing is performed to determine stresses and deformations induced in the casing. The details of FE analysis is summarized below.

FE discretization of Top and Bottom models:
The details of discretization are given in Table -2.

Table -2: Meshing specifications

Sr. No.	Model	Elements	Nodes
1	Bottom Casing	2562765	3916206
2	Top Casing	83824	143100

Figs. 6, 7 show FE discretization of bottom and top casing respectively.

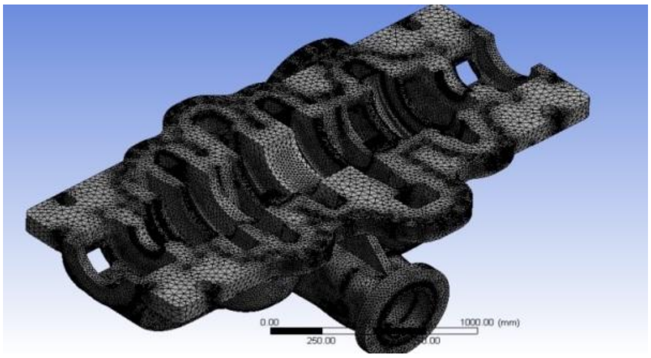

Fig -6: FE model of Bottom Casing

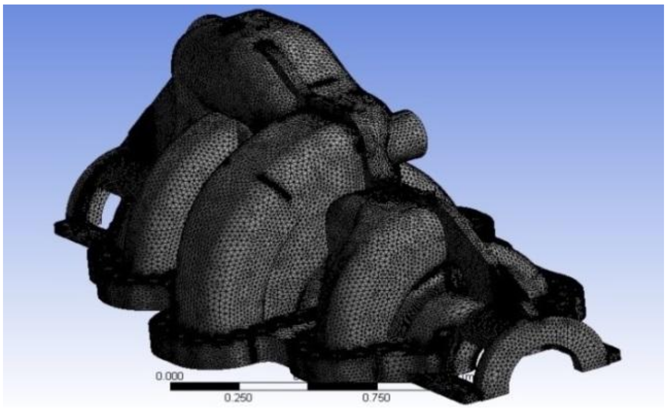

Fig -7: FE model of Top Casing

The inside surface of casing is subjected to the maximum allowable working pressure of 103.4 bar (Figs. 8, 9). Both top and bottom casings are analyzed independently. Both top and bottom casing models are fixed in all degrees of freedom at the bolt hole edges.

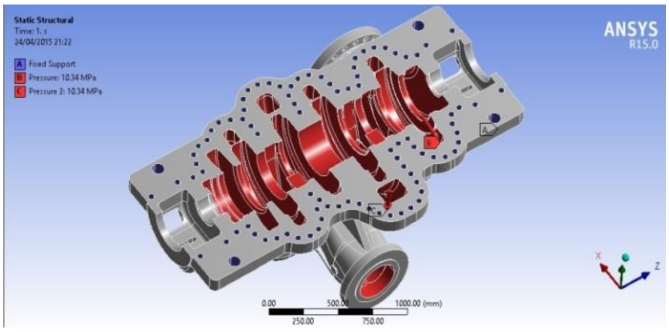

Fig -8: Applied pressure on bottom casing

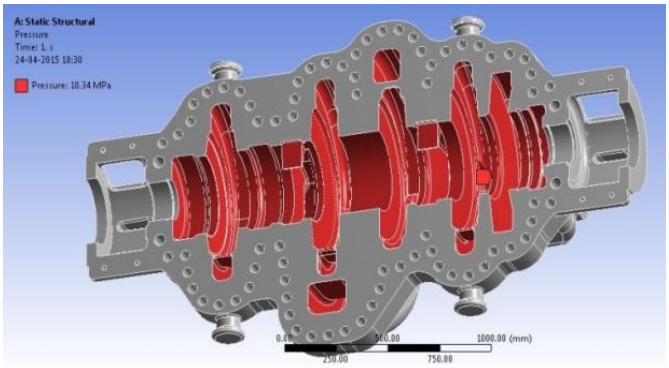

Fig -9: Applied pressure on top casing

3. RESULTS AND DISCUSSIONS

The total deformation and equivalent Von Mises stresses induced in the casing are shown in Figs. 10,11,12,13.

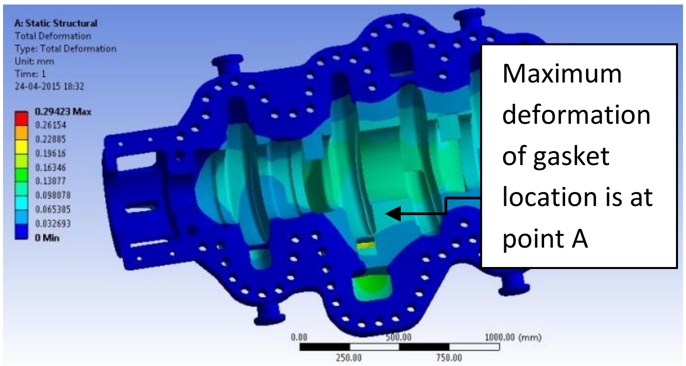

Fig -10: Total deformation of top casing at gasket area

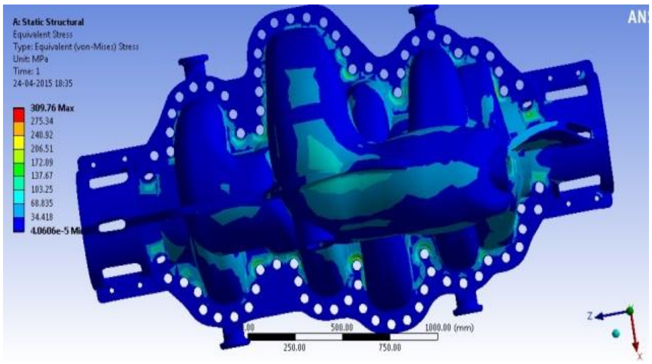

Fig -11: Von-Mises stresses in Top Casing

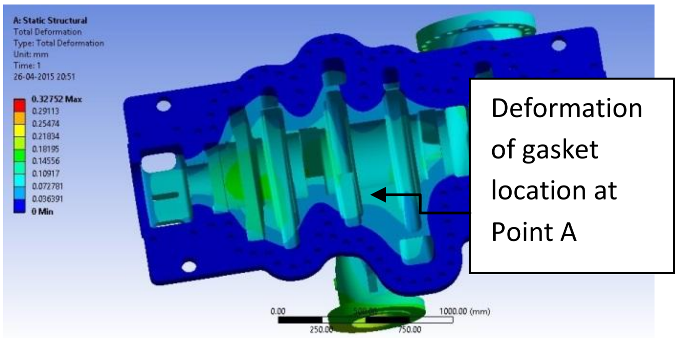

Fig -12: Total deformation of Bottom casing at gasket area

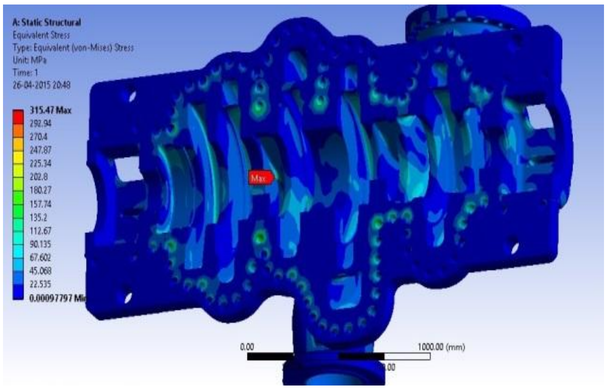

Fig -13: Von-mises stresses in Bottom Casing

Peak induced stress is observed at the bolt hole locations; however it can be neglected since it includes stress concentration effect due to fixity enforced on bolt hole edges. Equivalent Von Mises stresses caused are given in Table -4. And neglecting peak stresses and stresses at the boundary conditions, the maximum induced equivalent Von Mises stress is 157.74 MPa which is less than material allowable stress 183.34 MPa.

Table -3: Equivalent Von Mises stresses on casing

Casing	Equivalent stresses (Von Mises) (Mpa)		Material Allowable Stress = (yield strength/1.5) (Mpa)
	Minimum	Maximum	
Top casing	0.00004	103.25	183.34
Bottom casing	0.001	157.74	183.34

The total deformations in casing are given in Table -4. The maximum deformations of top and bottom casings are 0.29 mm and 0.33 mm respectively which are acceptable as per manufacturer's norms.

Table -4: Total deformation of casing

Casing	Pressure (Maximum)	Total deformation (mm)	
		Minimum	Maximum
Top casing	103.4 bar	00	0.29423
Bottom casing	103.4 bar	0.00004	0.32752

At gasket location, there are maximum deformations at point A in case of both top and bottom casings.

To avoid leakage along flange joint, it is necessary to limit the total deformation at gasket location, i.e., deformation of bottom casing plus deformation of top casing flange at point A.
Total Maximum deformation at flange (at point A)
=0.10917+0.098 (mm) = 0.20717 mm

The maximum deflection at the flange joints of casings is 0.21 mm which is less than gasket thickness of 0.50 mm. As per prevailing industry guideline, the computed deformation levels will restrict leakage at the flange joint.

4. CONCLUSIONS

The volute casing design for BB3 type multistage axially split centrifugal pump was successfully developed using procedure given by Karassik [4]. The stress values obtained from the finite element analysis results are within allowable limits thus indicating sufficiency of the strength of casing wall thickness.

The total deformation along flange joint is found to be less than the industry guideline, thus signifying leak-tightness of the critical gasket joint. Also, the overall deformation of casing body is within the norm followed by manufacturer.

This work successfully demonstrates a simplified method to design volute body and to analyze leak tightness across bolted flange joint of axially split centrifugal pump casing.

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BIOGRAPHIES

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