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"Design and analysis of High-Pressure Casing of a Steam Turbine"

Laxminarayan.K[a],Jayanth.H.S [b],M.Venkatarama Reddy[c]

[a]Research Scholar, Mechanical Department, Bangalore Institute of Technology, Bangalore,560004,India
[b]Research Scholar, Mechanical Department, East West Institute of Technology, Bangalore560091,India
[c]Prof & Head, Dept. of Mechanical Engineering, Bangalore Institute of Technology, Bangalore560004,India

Abstract

Contact pressure and pretension in bolts-analysis has been made easier in recent years due to the availability of high computational capabilities and flexibility in the computational methods using finite element analysis. In the present work, one such analysis is carried out blending the hand calculations and steady-state finite element analysis to evaluate the contact pressure in a high pressure steam turbine casing. The work involves design considerations, design checks, validation and sensitivity analysis to achieve the design criteria to fulfill the structural requirements for mechanical integrity. During the last several years the primary changes to the design of steam turbines have focused on improving their efficiency, reliability and reducing operating costs. Siemens Power Generation, for example, has improved the overall efficiency and availability of its steam turbines by decreasing the steam flow energy losses in each of the steam turbines components. The steam turbine unit largely influences the efficiency and reliability of power stations. Any improvement in the design of steam turbine enables more efficient use of fuel and results in reduced cost. The high pressure steam at 565 °C and 156 bar pressure passes through the high pressure turbine. The exhaust steam from this section is returned to the boiler for reheating before being used. On leaving the boiler reheater, steam enters the intermediate pressure turbine at 565 °C and 40.2 bar pressure. From the intermediate pressure turbine, the steam continues its expansion in the three Low pressure turbines. The steam entering the turbine is at 306°C and 6.32 bar. To get the most work out of the steam, the exhaust pressure is kept very low. The casing thus witnesses, energy of the steam turned into work in HP and IP stages. So, the design of the casing is a very important aspect.

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*Keywords:*Steam turbine, casing, flange, contact pressure, finite element modelling.

* Corresponding author. Tel.: +91-9980193211 ;.
*E-mail address:*mtechlax@gmail.com

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1. Introduction

The ASME Boiler and Pressure Vessel Code (2012) contain rules for circular and non-circular pressure vessels of unreinforced and reinforced construction. These rules cover the sides, reinforcing ribs, and end plates of such vessels. For bolted flanged connections of such non-circular pressure vessels, which are employed extensively in industry, however, no design rules are presently included in the code. As early as 1890's there were some pioneer literature publications on flange design by Bach (Bach, C, 2013)and Westphal. Bach's work discussed deflections and stresses of various shapes; this formed the basis of 'Bach-method' in flange design which was used for many years. Westphal (2012) further showed that flange thickness could be reduced by a few modifications in Bach's approach. The theory of elasticity has been extensively employed in analysis and design of bolted flanged connections. Waters and Taylor, Timoshenko and others have proposed analytical methods based on theory of elasticity for the analysis of bolted flanges. McKenzie et al (2012) used a two-dimensional photo elastic test method to measure and analyze the stress and strain distribution in steam turbine flanges. The photo elastic test method proved to be very useful in the design of flanges. Later he employed a finite element plane stress program to analyze flange structure. The results showed a good agreement between experimental and calculated values only under certain conditions.P.Shlyakhin, A.Kostyuk and V.Frolov have proposed methods to design flanges and bolts of a steam turbine casing. The method proposed by Shlyakhin stands out since it incorporates the bolt design along with flange design. So, in this work the design calculations have been carried out based on the method proposed by P. Shlyakhin. The method proposed by him has also been validated in the present work.

Nomenclature

P_i inside pressure in MPa.
R inner radius in mm.
S allowable stress in MPa.
E joint efficiency. [0.9-0.8 for welded joints and 1 for bolted joints]
t_c casing thickness in mm.
h flange thickness in mm.
t distance between bolt hole centres in mm.
b distance between the edges of the bolt holes in mm.
P_o outside pressure in MPa.

1.1. Design

The design procedure adopted in the present work is as given below:

Because of the very complicated shape of the turbine cylinder the exact calculation of the wall thickness becomes very difficult. Neglecting the effect of side walls, stiffening ribs, flanges, the pressure and temperature variation along the length, etc., we may consider the cylinder to be drum shaped.

In this case the tensile forces acting on the stator walls may be expressed by the equation 1.

$$t_c = \frac{P_i R}{SE - 0.6P_i} \tag{1}$$

The flanges of a turbine cylinder operate under conditions of compression and bending. Their design is however, based only on the bending forces present.Figure1 shows the section of a flange with the basic dimensions and the forces acting on the flange and the bolts. The flange thickness is h, distance between the bolt hole centres is t. The distance b between the edges of the bolt holes is chosen according to the strength of the material used for the flange and the bolts. The remaining dimensions shown in Figure1 are arbitrarily chosen to suit the design under consideration.

The flange design is based on the gauge pressure $\Delta p = P_i - P_o$. The main forces acting on the flanges due to the pressure difference are shown in figure 1. Nomenclatures of the equations from 2 to 22 are shown in table 2.

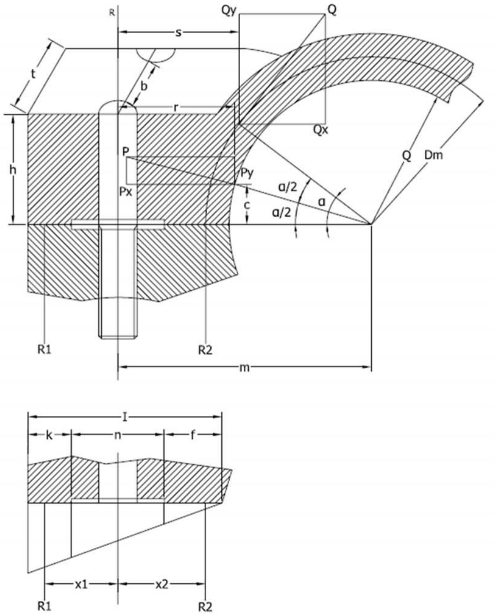


Fig.1. Basic dimensions of a Steam Turbine Flange

$$Q = \frac{Dt}{2}\Delta p \tag{2}$$

$$Q_x = Q\sin\alpha = \frac{Dt}{2}\Delta p\sin\alpha \tag{3}$$

$$Qy = Q\cos\alpha = \frac{Dt}{2}\Delta p\cos\alpha \tag{4}$$

$$P = Dt\Delta p\sin\frac{\alpha}{2} \tag{5}$$

$$P_x = P\cos\frac{\alpha}{2} = Dt\Delta p\sin\frac{\alpha}{2}\cos\frac{\alpha}{2} = Dt\Delta p\sin\alpha \tag{6}$$

$$P_y = P\sin\frac{\alpha}{2} = Dt\Delta p\sin^2\frac{\alpha}{2} \tag{7}$$

The moment acting on the flange due to the pressure difference is given by:

$$M_p = Q_y s - Q_x h + P_y r + P_x c \tag{8}$$

$$s = m - \frac{D_{av}}{2}\cos\alpha \tag{9}$$

$$r = m - \frac{D}{2}\cos\frac{\alpha}{2} \tag{10}$$

$$c = \frac{D}{2}\sin\frac{\alpha}{2} \tag{11}$$

The value of α is obtained from the constructional details and dimensions of the turbine stator and flanges.

To ensure a tight joint between the flanges we shall assume that for some tensile force R exerted on the bolts, and at the pressure under consideration (reaction force) the pressure exerted on the flange face along its length l varies in a linear fashion. The moment of the two opposing reaction forces are R_1 and R_2 acting on the flange face.

$$M_t = R_2 x_2 - R_1 x_1 \tag{12}$$

From elementary mechanics we know that for any system in equilibrium the sum of the moments of all forces acting in it is equal to zero. Since the flange is to be designed for sufficient strength at its minimum section 'bh' we shall take the sum of moments of all the forces acting with respect to this section

$$M_p + M_t = Q_y s - Q_x h + P_y r + P_x c + R_2 x_2 - R_1 x \tag{13}$$

The forces acting on the section 'bh' of the flange can be obtained from the equation 14.

$$R_1 x_t = M_p - R_2 x_2 \tag{14}$$

The unknowns x_1 and x_2 are determined from the following relations,

$$x_1 = \frac{n}{2} + \frac{k}{3} \times \frac{2l + n + f}{l + n + f} \tag{15}$$

$$x_2 = \frac{n}{2} + \frac{f}{3} \tag{16}$$

The values of R_1 and R_2 can be determined from the following equations,

$$R_1 = \frac{M_p}{x_1 + x_2 N} \tag{17}$$

$$R_2 = NR_1 \tag{18}$$

$$N = \frac{f^2}{k(l + n + f)} \tag{19}$$

The bending force on the section 'bh' of the flange is determined from the relation,

$$\sigma_f = \frac{6R_1 x_1}{bh^2} \tag{20}$$

The force exerted on the bolts will be,

$$R = R_1 + R_2 + P_y + Q_y \tag{21}$$

The tensile stress at the minimum bolt section, i.e., at the root section of the threads will be,

$$\sigma_b = \frac{4R}{\pi d_{\min}^2} \tag{22}$$

An excel sheet is prepared using the equations (1 to 22), and tabulated in Table 1 and Table 2 which gives the results according to the analytical method in less time. The results are comparable with analysis (Ansys output) and analytical method (Shlyakhin approach). Since the derived forces have good correlation with analysis output, it can be concluded that further calculations carried out in the analytical method are valid. Hence this procedure shall be adopted to design the steam turbine casing. The finite element analysis gives a complete picture of mechanical behaviour of the flange structures, and design guidelines without costly experiments.

Table 1. Inputs considered

Description			Units
Inside pressure	P_I	5	MPa
Outside pressure	P_O	0	MPa
Inner diameter	D	610	mm
Allowable stress	σ	72.5	MPa
Assumed ratio of flange height to casing thickness	h/t_c	2.9	
Bolt diameter	d	42	mm
Bolt thread pitch	p	4.5	mm
Bolt pitch	t	84	mm
Cap nut diameter	n	58	mm
Distance from casing outer edge to bolt	k	20	mm
Distance from bolt to casing inner edge	f	18.47	mm

Table 2. Output obtained from equation 1-22

Description			Units
Casing thickness	t_c	36.94	mm
Flange height	h	107.13	mm
Force	Q	128100	N
Distance from bolt centre to casing centre	m	352.47	mm
Included angle between flange tip to casing centre	α	19.34°	
Force	Q_x	42426.49	N
Force	Q_y	120870.19	N
Force	P	43038.09	N
Force	P_x	42426.49	N
Force	P_y	7229.81	N
Distance between bolt centre and Q_y	s	47.26	mm
Distance between bolt centre and P_y	r	51.81	mm
Distance between bolt centre and P_x	c	51.24	mm
Moment acting on the flange	M_p	3714900.00	Nmm
Dimension	x_1	39.39	mm
Dimension	x_2	35.16	mm
Factor N	N	0.099	
Reaction force R1	R_1	86688.51	N
Reaction force R2	R_2	8551.09	N
Flange bending stress	σ_f	42.50	MPa
Force exerted on the bolts	R	223339.59	N
Bolt bending stress	σ_b	202.21	MPa
Required initial bolt tension (Preload)	$F_{P\text{-}Bolt}$	312675.43	N

Table 2 are the outputs obtained which have been calculated using the equations 1 to 22.Obtained flange bending stress (42.58 MPa) < allowable stress (72.5 MPa), hence the design is safe

1.2. Validation

The forces are considered to be the basis of the whole method proposed by Shlyakhin. These forces are resolved and solved further to obtain the reaction forces $R_1$and R_2.

The validation is done by taking into consideration a single stage (1st stage) and applying the similar boundary conditions using ANSYS 12.0 .The problem is assumed to be axi-symmetric about Y-axis; hence only the resolving area is analyzed to reduce the considerable time of computations and tedious computer efforts Hexamesh is applied to obtain accurate results. The required model is modelled using Unigraphics software.

The following are the loads & boundary conditions applied shown in Figure 2:

- An internal pressure of 5MPa on the inside surface.
- The casing is fixed in Y-direction.
- The bolt is applied with a pretension of 3.3943e5 N shown in Figure 3.

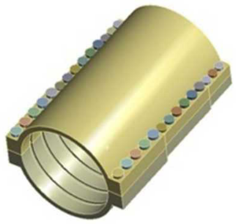


Fig.2. Boundary Conditions

Fig.3. Bolt Pretension applied on bolts

Similarly the force *P*, reaction force (*R1+R2*) and the reaction force on the bolt *R* are determined.

Table 3. Analysis and Calculated Results

Force	Calculated Value (in N)	Obtained value from analysis (in N)	%error
Q	128100	127200	0.70
P	43038.09	42999	0.09
R1+R2	95239.59	96587	1.41
R	223339.59	246200	10.24

The results obtained by analysis are almost equal to the calculated values which are shown in Table 3. The error except in the case of force R can be neglected. Since the whole face of the bolt which is in contact with the top casing cannot be selected to determine the resultant, the error can be justified. As, the forces calculated have good co-relation, it can be concluded that further calculations carried out in the analytical method are valid. Hence this procedure can be adopted to design the steam turbine casing.

2. MODELLING

The analytical procedure is adopted to determine the casing and flange thicknesses at various stages. The flange width and the required bolt sizes are also determined. This dimensional data is then used to construct the 3-D model of the casing using Unigraphics as shown in Figure 4.Table 4 and Table 5 gives the calculated and adopted dimensions of the casing:

Table 4. Calculated Values

Sl No.	Parameter	At nozzle chest	At 1st Stage	At 9th Stage	At end of I.P Stage
1	Casing thickness	26.62	36.94	32.16	30.75
2	Flange thickness	106.47	107.13	64.32	61.50
3	Flange width	78.00	78.00	66.00	66.00
4	Bolt Diameters	M 42	M 42	M 30	M 30

(All dimensions are in mm)

Table 5. Adopted Values

Sl No.	Parameter	At nozzle chest	At 1st Stage	At 9th Stage	At end of I.P Stage
1	Casing thickness	27.00	37.00	32.00	32.00
2	Flange thickness	108.00	108.00	60.00	60.00
3	Flange width	78.00	78.00	66.00	66.00
4	Bolt Diameters	M 42	M 42	M 30	M 30

(All dimensions are in mm)

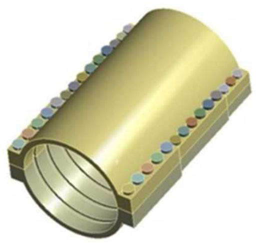


Fig.4. 3-D model of Steam Turbine Casing with Bolts

3D casing model is taken up for contact pressure and structural analysis. By carrying out the contact pressure analysis it will be taken care that required contact pressure is maintained at the parting plane and thus no steam leaks out of the casing. By carrying out the structural analysis the stresses and deflections in the casing can be determined.

3. Problem Description

The main objective of this work is to develop a procedure to validate and evaluate the parameters required to design a steam turbine casing. The finite element analysis gives a complete picture of mechanical behaviour of the flange structures, and design guidelines without costly experiments. 3D model of the top and bottom casing is generated and subjected for analysis. Contact pressure analysis was performed to validate structural integrity of casing. Figure 5 indicates base line geometry of parting plane of steam turbine casing without relief. In this base line design, the required contact pressure was not achieved. Hence, a few design modifications were done to the bottom casing and a relief was provided as indicated in Figure 6. The modifications in the design at the parting plane with relief have resulted in the desired contact pressure. The required contact pressure (3 times the pressure at respective stage) is achieved in the high pressure as well as in the intermediate pressure stages.

4. Methodology

Steam turbine casing is an ideal example of Axisymmetric structure. The Axisymmetric model of steam turbine casing is analysed to reduce the computational time in FEA package. Hence, one sector of the system is considered for the analysis. The analysis is carried out by applying design pressure of 5MPa and bolt pretension load of 3.39e5 N in ANSYS and results are validated with analytical method. Linear static analysis is carried out to develop a custom made methodology, to verify the contact pressure at parting plane. Contact pressure achieved at the parting plane should be three times the internal pressure. This ensures that the casing is safe and there will not be any steam leakage. Chromium steel is used for casing and bolts material withyield stress of 585 MPa, Young's modulus 210000 MPa, density 7900 kg/m^3 and Poison's ratio 0.3.

Fig.5.Geometry without relief

Fig.6.Geometry with relief

4.1. Finite Element Model

Fig.7. Finite Element Model

Figure 7 shows the FEA model used for analysis. The element type used for analysis is Solid 187 (10 noded Tetrahedron), No of elements= 259133, No of nodes= 457339. Table 6 and Table 7show the loads considered on the Finite Element Model. Figure 8 shows temperature and pressure applied inside casing.

Table 6. Applied Pressure Load

Description	Region	Pressure (MPa)
WCP	A	5.5
2nd Stage	B	3.8
3rd Stage	C	3.0
4th Stage	D	2.4
5th Stage	E	1.9
6th Stage	F	1.5
7th Stage	G	1.2
8th Stage	H	0.8
9th Stage	I	0.7
10th Stage	J	0.5
After 10th Stage	K	0.5
At Gland seal location	L	2.50

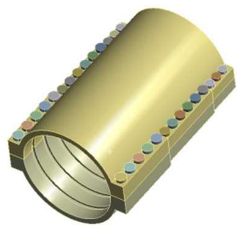


Fig.8. Applied Temperature and Pressure load inside casing

Table 7. Pretension in Bolts

Region	Temperature (ºC)	Bolt Size (mm)	No. of Bolts	Pretension (N)
A	450	42	6*2	295958.39
B	450	48	2*2	378966.11
C	450	48	1*2	378966.11
D	425	42	1*2	311835.98
E	395	42	1*2	329380.73
F	365	42	1*2	342733.78
G	335	42	1*2	353721.08
H	335	30	1*2	165592.42
I	310	30	1*2	169688.00
J	280	30	2*2	173783.59
K	250	30	2*2	178243.39
L	220	30	2*2	182554.53
M	220	30	2*2	182554.53
N	450	20	2*2	59232.67

5. Results and Discussions

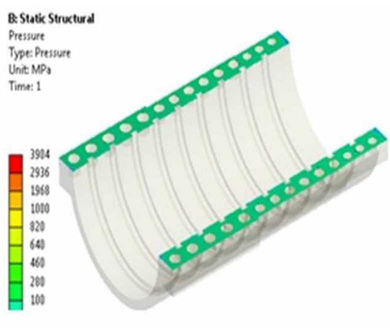


Fig.9. Contact Pressure not achieved

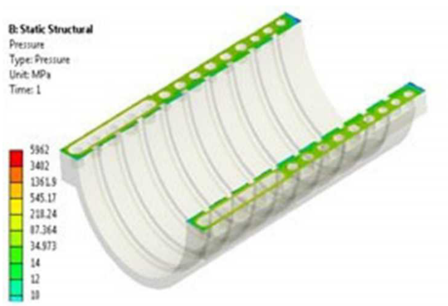
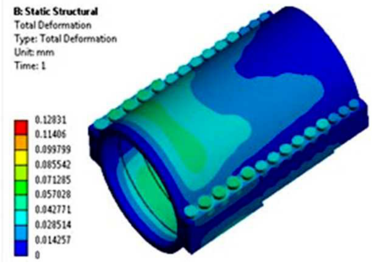


Fig. 10. Contact Pressure achieved

Figure 9 explains the required contact pressure is less. Due to the lack of contact pressure the steam may leak at the wheel case. This requires some design modification to be done at the parting plane to achieve the required contact pressure. So, a relief is provided to the bottom casing up to end of high pressure stage so as to increase the contact pressure to the required extent. Figure10 shows the contact pressure is increased after providing relief.



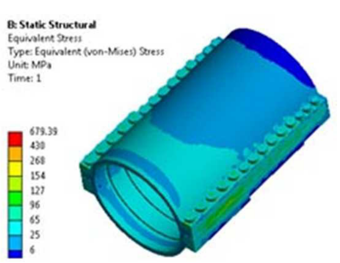


Fig.11. Total Deformation

Fig. 12. Von-Mises stress distribution for Casing

The total maximum deformation is found to be 0.12 mm as shown in figure 11.The Von-Mises stress distribution for the casing is shown in figure 12. The peak stresses are found to be 679MPa.

6. Conclusions

The classical method in combination with knowledge based engineering is utilized to identify the trivial areas in the design of turbine casing. Custom made methodology is developed to achieve the structural integrity of the casing, with the accomplishment of simulation engineering. Sensitivity analysis is made possible with the aid of modern computers which allows the user to explore the criticality in the core design. The study has made it possible to meet the design considerations successfully and achieve the required safety factors for existing manufacturing and design uncertainties.The present study on strength of steam turbine casing for a given operating conditions reveals that the optimized casing geometry shall be used as design modification for future. Comparing the results from analytical method and the finite element analysis, it shall be summarized that a good correlation is observed and methodology adopted has promised the confidence in design.

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