Design and Analysis of Pelton Turbine for Organic Rankine Cycle Application

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Abstract – *Waste Heat Recovery is a promising technique for extracting maximum work from outgoing exhaust gases of IC engines. It serves the dual purpose of reducing exhaust gas temperature and increasing IC engine efficiency. However, conventional power generation cycles fail to extract the energy from exhaust gases because they form a low grade heat source. Organic Rankine Cycle (ORC) helps to extract energy from low grade sources due to the fact that it uses organic fluids instead of water. Hence, ORC can be effectively utilized for Waste Heat Recovery. Turbine is the heart of any power generating cycle. Challenge in this work is that turbine needs to be developed and analyzed for ORC application. A Pelton turbine is developed for ORC application in this work with complete evaluation using theoretical knowledge and Finite Elements analysis. Analysis is restricted only to structural aspects of design. Pelton turbine has generally been designed for hydraulic applications. Novelty of this work lies in the fact that Pelton turbine is being designed for organic fluid which would be in gaseous form.*

Key Words: ORC, Pelton turbine, Bucket, FEA, Structural analysis etc

1. INTRODUCTION

With depleting fossil fuels and growing concerns over global warming, Research fraternity is putting a renewed emphasis on use of renewable energy. Also, there is a growing recognition for the need of improving efficiency of conventional engines. This is especially important because of the fact that fossil fuels are limited. Hence, it is important to make efficient use of available fossil fuels. About 40 % of the energy in fossil fuels is wasted in the form of exhaust gases. General Exhaust gas temperature is 300-500°C for standalone IC engines. Waste Heat Recovery is a promising technique in this regard. Waste Heat Recovery means to extract maximum energy from exhaust gases and convert it into useful mechanical work. It serves the dual purpose of enhancing engine efficiency and reducing the temperature of exhaust gases let out to the atmosphere. Energy present in the exhaust gases is moderate temperature heat energy. It is also called low grade heat energy. This low grade heat energy cannot be efficiently converted into electrical power by conventional power generation cycles. Hence, it generally gets wasted. In this regard, research on how to convert this low grade temperature heat sources into electrical power is of great significance.

Organic Rankine Cycle appears to be a promising technique of power generation when heat sources are of low grade. It works the same as conventional Rankine Cycle except the fact that it uses organic fluids in place of water or steam. Organic fluids generally have lower boiling points than water. Hence, low grade heat energy can be effectively used to vaporize organic fluids. Hence, because of organic fluids, Organic Rankine Cycle can be used to tap low grade heat energy. Further, Organic fluids also have higher molecular mass and vapor density. Good vapor density means that size of turbine required for expansion of gaseous fluids will be less. This will lead to considerable savings in materials and cost. Thus, Organic Rankine Cycle will also be cost effective. Henceforth, we shall refer the Organic Rankine Cycle as ORC. Turbo expander is the heart of Organic Rankine Cycle since it is the power generating unit of ORC. Turbo expander is essentially a turbine

B.A. Nasir [1] has elaborated the procedure for design of Pelton Wheel for hydraulic application. J. S. Anagnostopoulus [2] has investigated the best hydraulic design of inner surface of a Pelton turbine bucket to achieve maximum efficiency. E. Parkinson et.al [3] has performed various numerical simulations on Pelton turbine using CFD and mechanical structural analysis. V. Sharma et.al [4] has performed the structural analysis on Pelton turbine and experimental correlation of strains. H. Chen [5] has done a screening of 35 working fluids as possible candidates for use in Organic Rankine Cycle. He has studied and compiled important thermodynamic and mechanical properties of these fluids. Chukwuneke J.L. et.al [6] has studied the effect of bucket splitter angle on the power output of a Pelton turbine.

2. ANALYTICAL DESIGN

Design procedure for Pelton turbine for organic fluids starts with the selection of working fluid. Selection of working fluid would determine the dimensions of turbine and help us to select compatible materials for turbine components. In order to achieve maximum efficiency for Organic Rankine cycle, fluid needs to be selected based on following general criteria:

- ✓ Isentropic or dry fluids are considered appropriate for Organic Rankine Cycle to avoid liquid droplet impingement on the turbine blades during expansion process.
- ✓ Fluids with high latent heat are desirable for Organic Rankine cycle as such fluids will absorb higher energy and transfer this energy to turbine.
- ✓ Fluids with high density are desirable as such fluids have higher vapor density. Higher vapor density helps to reduce the size of turbine.
- ✓ Fluids with low specific heats are preferred in Organic Rankine Cycle. Lower specific heat reduces the heat required to raise temperature of organic fluid. Hence, efficiency of the cycle would increase.
- ✓ Critical temperature of the fluid selected should be higher than highest operating temperature of the cycle
- ✓ Freezing temperature of the fluid selected should be lower than the lowest operating temperature of cycle
- ✓ Fluid selected should be stable at the operating conditions of cycle and should be compatible with turbine material

As per general exhaust conditions for Standalone IC engines like those of Diesel Generator sets, inlet conditions of working fluid are considered. Generally, exhaust gases from these engines are at 300 – 600 °C. Considering a general heat exchanger, working fluid will be heated to following conditions:

Temperature, $T_1 = 300$ °C

Pressure, $P_1 = 10$ bar

Since, highest operating temperature for working fluid is 300°C; critical temperature of working fluid selected should be above 300°C. Toluene is one such organic fluid whose critical temperature is 318°C. Molecular structure of Toluene consists of polymeric chains of Benzene *(CH3)*. Hence, chemical formula for Toluene can be written as $(CH_3)_n$. Toluene is an aromatic hydrocarbon. Benzene and toluene are considered as isentropic fluids with relatively high critical temperatures, which are desirable characteristics for Organic Rankine Cycle. Toluene is also chemically stable at the potential operating condition. Toluene also does not form wet condensate on turbine blades. Hence, corrosion of turbine blades can be avoided if Toluene is used as a working fluid. All these properties make Toluene suitable candidate for Organic Rankine Cycle for given operating conditions. Density of Toluene changes with temperature. *Hence, Toluene is selected for current application*. As per data from National Institute of standards and Technology, density of Toluene at 300°C is 477.28 kg/m³.

2.1 Nozzle

Main function of Nozzle in the Pelton turbine is to accelerate the fluid. In this regard there can be two types of nozzles Convergent Nozzle and Convergent-Divergent Nozzle. Convergent Nozzle can provide acceleration up to sonic velocity in local medium, while Convergent-Divergent Nozzle can accelerate fluid to supersonic velocities. Supersonic flow can lead to shock waves in the casing. Hence, we select Convergent Nozzle for design. Design of Nozzle is done using critical pressure ratio principle. Ratio of Pressure at Nozzle inlet to that at Nozzle throat is equated to critical pressure ratio to find dimensions of Nozzle. Inlet diameter of Nozzle is assumed to be 12 mm as per standard piping dimensions while the maximum mass flow rate is limited to 0.5 kg/s due to material considerations. With these inputs, dimensions of Nozzle are found as follows.

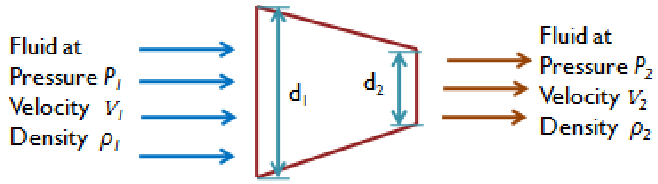


Fig -1: Nozzle Schematic

$$\frac{P_C}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad \text{Hence, } P_C = P_2 = 5.64 \text{ bars} \tag{1}$$

$$A_2 = \frac{\dot{m}}{\sqrt{nP_1\rho_1 \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}} \text{, Hence } A_2 = 35.29 \text{ mm}^2$$

$$d_2 = \sqrt{\frac{4A_2}{\pi}} \text{, Hence } d_2 = 7 \text{ mm} \tag{2}$$

$$v_1 = \frac{\dot{m}}{\rho_1 A_1} = 9.27 \text{ m/s} \tag{3}$$

$$T_2 = T_1\left(\frac{P_1}{P_2}\right)^{\frac{1-n}{n}} = 520.8 \text{ K} \tag{4}$$

$$\rho_2 = \frac{\rho_1}{1+\beta(T_2 - T_1)} = 503.5 \text{kg/m}^3 \tag{5}$$

$$v_2 = \frac{\dot{m}}{\rho_2 A_2} = 28.13 \text{ m/s (Velocity at nozzle exit)} \tag{6}$$

Other dimensions of Nozzle are found from following empirical relations.
Outer Diameter, $D = 2.5d_1 = 30$ mm
Length of Nozzle, $L = 8d_1 \approx 100$ mm
Thickness of Nozzle, $t = (D-d_1)/2 = 9$ mm

2.2 Wheel

Input from Nozzle design is the jet diameter. Jet diameter is same as diameter of Nozzle throat. Hence, we can write the following Jet diameter value.
$d_{jet} = d_2$ = 7 mm
Jet ratio is defined as the ratio of mean or pitch diameter of Wheel to jet diameter. As per literature review, Optimum efficiency is obtained at the *Jet ratio* of 10 to 14 [1]. Hence, we select a *Jet Ratio* of 12. Thus, mean diameter of Wheel (D_m) is calculated as follows.
D_m = *Jet Ratio* x d_{jet} = 84 mm
Some of the other dimensions are calculated based on following empirical relations.

Shaft Diameter, $d_s = 0.3D_m$ = 25 mm ≈ 30 mm
Diameter of Shaft Collar, $d_c = 1.25ds$ ≈ 32 mm
Number of Buckets, $Z = 15 + (D_m/2d_{jet})$ = 21

Further, design of Wheel is done using velocity diagram given below.

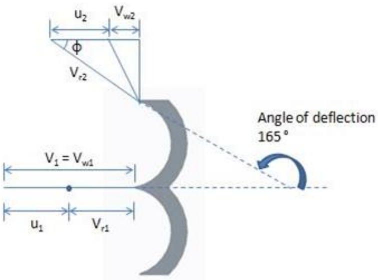


Fig -2: Velocity diagram

Here, V_1 = 28.13 m/s (velocity at bucket inlet is same as velocity at Nozzle outlet)

As seen from the above velocity diagram, Entire component of inlet velocity is along tangential direction to Bucket.

Hence, $V_{w1} = V_1$ = 28.13 m/s

For hydraulic efficiency to be maximum, Bucket speed should be half the Jet Speed [1]. This can be proved mathematically by equating the derivative of hydraulic efficiency to zero. Note that, Hydraulic efficiency is the ratio of power transferred by the jet to the wheel to kinetic energy of jet. We will state the expression for Hydraulic Efficiency in the later course of design.

$$\frac{d(\eta_{Hyd})}{d(U/V_1)} = 0$$

The above equation yields the following expression.

$$\frac{u}{V_1} = 0.5 \text{ (Ratio practically limited to 0.44-0.46)} \quad (7)$$

Hence, $u = 0.45V_1$ = 12.66 m/s

As per the above velocity diagram, relative velocity at bucket inlet (V_{r1}) is the difference between velocity of jet and velocity of Bucket.

$$V_{r1} = V_1 - u = 28.13 - 12.66 = 15.47 \text{ m/s} \quad (8)$$

Ideally, relative velocity at outlet is equal to relative velocity at inlet if the friction in the bucket is absent. But, practically, this is not possible. Bucket surfaces offer considerable friction to fluid passage. Hence, relative velocity at outlet is less than relative velocity at inlet. In general, friction coefficient value is taken as 0.85. Hence, relative velocity at outlet can be calculated as follows.

$$V_{r2} = 0.85V_{r1} = 13.15 \text{ m/s} \quad (9)$$

Angle ø in the Velocity triangle is calculated as follows.

Ø = 180° - 165° = 15°

Tangential Velocity at outlet is calculated from Velocity triangle using Pythagoras theorem as follows.

$$V_{w2} = V_{r2}\,(Cos\phi) - u = 0.043 \text{ m/s} \quad (10)$$

Power transferred by the jet to the wheel and kinetic energy of the jet are calculated by the following standard formulas.

Power Transferred = $\rho A V_1 (V_{w1} + V_{w2}) \times u$ = 178.43 W

KE of the jet per second = $\frac{1}{2} \times (\rho A V_1) \times V_1^2$ =197.9 W

Hydraulic Efficiency of the Pelton Turbine is defined as ratio of power transferred to the wheel to kinetic energy of the jet per second.

$$\eta_{Hyd} = \frac{\text{Power Transferred to the Wheel}}{\text{KE of the Jet Per Second}} = 90.14\,\% \quad (11)$$

Speed of the turbine shaft assuming no mechanical losses in transmission is calculated as follows.

$$N = \frac{60 \times u}{\pi \times D_m} = 2878 \text{ RPM} \quad (12)$$

2.3 Bucket

Bucket design is based on standard empirical relations derived from detailed CFD and Experimental Studies available in literature. These empirical relationships are derived from maximum efficiency standpoint. Following notations will be used in the course of bucket design.
h = Height of bucket
b = Bucket Width
t = Depth of bucket
T = Material Thickness

$$h = 3 \times d_{jet} = 21mm$$
$$b = 3.4 \times d_{jet} = 24\ mm$$
$$t = 1.2 \times d_{jet} = 8\ mm$$
$$T = 0.5 \times d_{jet} = 3.5\ mm$$

Impact force exerted by the jet on Bucket is denoted by FX and is calculated as follows.

$$F_X = \rho A V_1 \times (V_{W1} - V_{W2}) = 14.04 \text{ N} \quad (13)$$

3. CAD DESIGN

Considering the high temperature that turbine has to operate at, plastic parts are ruled out as they will deteriorate in this temperature. Metals like steel, aluminum and copper are candidates for materials. However, considering cost advantage of steel, AISI 1018 mild carbon steel is selected as the material for all turbine components. Its Young's modulus is 205 GPa while its Yield strength is 370 MPa.

Software package Pro-E is used for 3D modeling of parts of Pelton turbine. Important parts designed in CAD software are Casing, Nozzle, Wheel, Bucket and Shaft. After designing the parts, they are assembled in CAD software. Some of the pictures of parts designed are as follows.

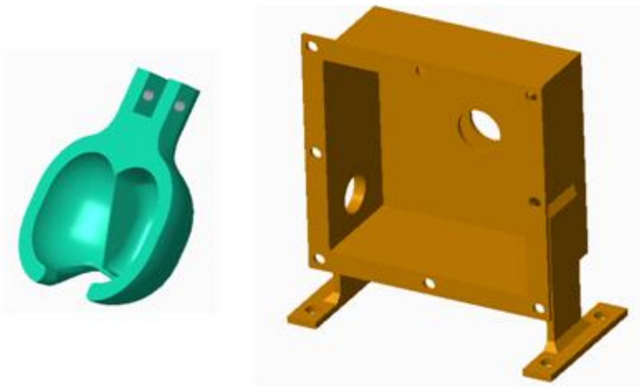


Fig -3: Bucket and Casing

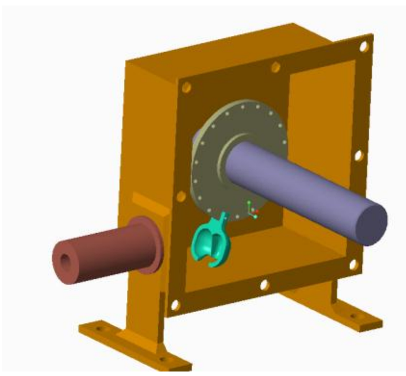


Fig -4: Turbine assembly

4. THEORETICAL ANALYSIS

4.1 Estimation of Bucket deflection

Deflection of Bucket under the action of jet impact force is calculated below. Approximate calculation for deflection can be done using cantilever beam deflection formula.

$$\delta_{\max} = \frac{P\ell^3}{3EI}$$

In the above formula, *I* is the moment inertia of bucket. *I* is calculated by assuming a rectangular cross-section of bucket. This is an approximation we make to get the deflection value in the early phase of design.

$$I = \frac{bd^3}{12}$$

In the above formula, *b* is the width of bucket which is 34 mm as per CAD design. *d* is thickness of bucket which is 3.5 mm.

Hence, I = 121.47 mm^3

In above equation, ℓ is the length of the bucket from mounting location. Mounting location means the screw location here. This length is 34.5 mm as per CAD design. This length is measured from CAD model as shown in below picture. E is Young's modulus of material which is 205000 MPA. *P* is the jet impact force acting on the bucket. As per analytical calculations, this force is about 14.1 N. We consider a factor of safety of above 2 to account for assumptions made in this calculation. Hence, 'P' value is taken as 28.7 N.

From all these values, Deflection at the bucket tip can be calculated.

$$\therefore \delta_{\max} = 0.0157\text{mm} \quad (14)$$

4.2 Estimation of Stresses in Nozzle

Nozzle is similar to Cylindrical Pressure Vessel. Cylindrical Pressure Vessels are subjected to internal and external pressures. Stresses generated in the pressure vessels are of three types

- ✓ Axial Direction stress (σ_A)
- ✓ Circumferential direction Stress or Hoops Stress (σ_H)
- ✓ Radial Direction Stress (σ_R)

These three stresses are also the Principal stresses in case of Pressure Vessels.

Cylindrical Shells can be thick or thin based on which formulation changes. A Pressure vessel in which thickness is not more than one tenth of its radius is said to be a thin Shell. Pressure vessel in which thickness is more than one tenth of radius is said to be a thick shell. In case of Nozzle, thickness is 9 mm while the radius is 6 mm. Hence, Nozzle falls in to the thick shell category. Stress calculation in case of thick shells is done using *Lame's* Equations. Some of the symbols used in *Lame's* Equation are as follows:

P_i = Inside Pressure = 10 bar = 1 MPa (at Nozzle Entrance)

P_o = Outside Pressure = 1 bar = 0.1 MPa (Atmospheric Pressure)

R_i = Inside Radius = 6 mm

R_o = Outside Radius = 15 mm (Nozzle body) = 20 mm (in Collar area)

Lame's Equation Calculations for Nozzle body are as follows:

$$\sigma_A = \frac{[P_i R_i^2 - P_o R_o^2]}{[R_o^2 - R_i^2]} = 0.0714\text{MPa}$$

$$\sigma_H = \left[\frac{P_i R_i^2 - P_o R_o^2}{R_o^2 - R_i^2}\right] - \left[\frac{R_i^2 R_o^2 (P_o - P_i)}{R^2 (R_o^2 - R_i^2)}\right] = 1.14\text{MPa}$$

$$\sigma_R = \left[\frac{P_i R_i^2 - P_o R_o^2}{R_o^2 - R_i^2}\right] + \left[\frac{R_i^2 R_o^2 (P_o - P_i)}{R^2 (R_o^2 - R_i^2)}\right] = -1\text{MPa}$$

Above three stresses are also the Principal Stress values. Numerically highest value is σ_1, Second highest is σ_2 and the lowest is σ_3. Hence three Principal stresses are as follows.

$$\sigma_1 = \sigma_H = 1.14\text{MPa}$$
$$\sigma_2 = \sigma_A = 0.0714\text{MPa}$$
$$\sigma_3 = \sigma_R = -1\text{MPa}$$

Using the above values of Principal Stresses, Von-mises stress (σ_{VM}) induced in Nozzle in Cylindrical portion is calculated as follows.

$$\sigma_{VM} = \frac{\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}}{\sqrt{2}} = 1.853\text{MPa}$$

In Nozzle collar area, outside radius (*Ro*) has to be changed to 20 mm in the above equations to get three Principal Stresses. Three Principal Stresses and Resultant Von Mises Stress in collar area of Nozzle are as follows.

$$\sigma_1 = \sigma_H = 0.98\text{MPa}$$
$$\sigma_2 = \sigma_A = \text{-}0.01099\text{MPa}$$
$$\sigma_3 = \sigma_R = -1\text{MPa}$$

$$\sigma_{VM} = \frac{\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}}{\sqrt{2}} = 1.714\text{MPa}$$

Since the Von-Mises stress in Nozzle cylindrical portion and collar area are below material yield limit of 370 MPa, We can say that Nozzle is safe for pressure loading.

5. FINITE ELEMENTS ANALYSIS

FEA analysis is an effective tool for finding the solutions of differential equations when applied to complex part geometries. It deals with discretization of domain into nodes and elements. Elemental displacement equations are solved separately and assembled using matrices to give displacement of entire domain or assembly or part. We have used *Ansys R 16.1 Workbench* for Finite Elements Analysis.

FEA requirements are first found out by identifying failure modes and loading on each component. Bucket has to sustain jet impact force. Further, it has to also resist resonance failure which can occur if its natural frequency matches with frequency of rotation. Nozzle has to sustain internal pressure due to high pressure gas inside. Casing is a structural member which carries the load of entire assembly. Hence following FEA requirements are important for current design:

- ✓ Bucket Structural Analysis
- ✓ Bucket Modal Analysis
- ✓ Nozzle Body Pressure Vessel Analysis
- ✓ Casing Structural Analysis

5.1 Bucket structural analysis

Following are the details of Bucket Structural Analysis

Load: 28.7 N Force Perpendicular to Bucket Splitter profile

Boundary Condition: Fixed Boundary Condition at screwing location with Wheel

Mesh Details: Higher Order Patch Conforming Tetrahedral Elements, Element Size: 1 mm

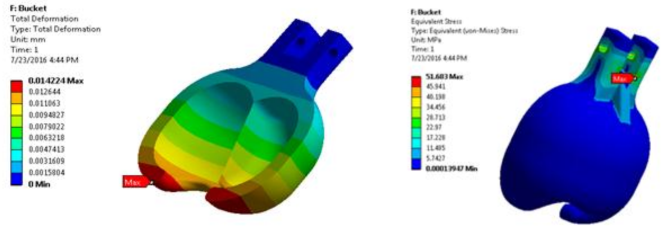


Fig -5: Deflection and stress in Bucket

Thus results obtained from FEA for Bucket are as follows:

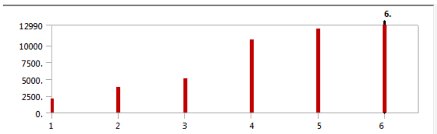
Deflection of Bucket = 0.014 mm

Von-Mises Stress = 51.6 MPa

Thus, deflection of bucket is negligible. Von-Mises Stress generated in the Bucket is well below Material Yield Limit of 370 MPa. Hence, based on FEA, *Bucket is safe for Jet Impact loading.*

5.2 Bucket Modal analysis

From modal analysis, first six natural frequencies of bucket are as shown in below figure.



	Mode	Frequency [Hz]
1	1.	2026.4
2	2.	3832.5
3	3.	5066.3
4	4.	10767
5	5.	12368
6	6.	12990

Fig -6: First six natural frequencies of Bucket

Thus, First Modal frequency of Bucket (*2016 Hz*) is well above the Turbine operating frequency of *48 Hz (2878 RPM).* Thus, there is no chance of Resonance. Hence, based on FEA, *Bucket is safe from Resonance Failure.*

5.3 Nozzle Pressure Vessel Analysis

Following are the details of Nozzle Analysis

Loads: 1 MPa (10 bar) internal pressure, 0.1 MPa (1 bar) external atmospheric pressure for Nozzle body outside Casing, 0.564 MPa (5.64 bar) external pressure for Nozzle body inside Casing

Boundary Condition: Fixed Boundary Condition on cylindrical area with Casing interface, 'Compression Only Support' for mating face with Casing

Mesh Details: Higher Order Patch Conforming Tetrahedral Elements. Element Size: 3 mm

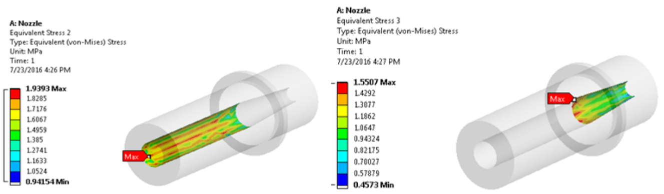


Fig -7: Von-mises stress induced in Nozzle

Thus, Stresses induced in the Nozzle are well below the material yield limit of 370 MPa for Steel. Hence, based on FEA*, Nozzle Body is safe under Pressure Loads.*

5.4 Casing Structural analysis

Following are the details of Casing Structural Analysis

Loads: Nozzle mass (0.48 Kg), Wheel Assembly with Bucket composite mass (0.75 Kg), Shaft mass (0.25 Kg), Key mass (0.009 Kg), internal pressure of 0.564 MPa (5.64 bar)

Boundary Condition: Fixed Boundary Condition at Casing Legs and Screwing Location

Mesh Details: Higher Order Patch Conforming Tetrahedral Elements

Mesh Size: Casing (5 mm), Wheel (4 mm), Shaft (3 mm), Key (2 mm)

Nozzle is modeled as a point mass at its centre of gravity and attached to its assembly location in casing. Gravitational acceleration of *9.806 m/s²* is applied to simulate the forces due to above mentioned component masses. Interface between Shaft and Casing is simulated using Frictional Contact with Frictional Coefficient of 0.15. All other interfaces like Shaft-Key, Shaft-Wheel and Key-Wheel are simulated using Bonded Contact.

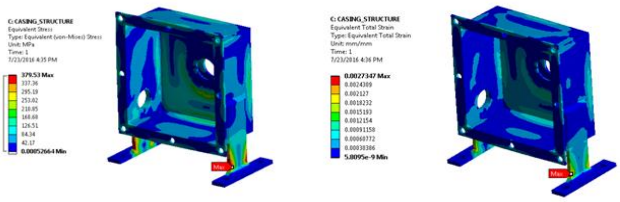


Fig -8: Stresses and Strain in Casing

Thus results obtained from FEA for Casing Structural analysis are as follows:
Von-Mises Stress- Casing = 379.5 MPa
Strain – Casing =0.2 %
Thus, Stresses induced in the Casing marginally exceeds the material yield limit of 370 MPa. However, the strain induced in Casing is just 0.2 % (which is within acceptable limits) and is concentrated near the legs. Strain limit for Steel as per Industry standards is 0.8 %. Beyond this point, Visible Strain marks appear in part. Hence, based on FEA, *Casing Assembly is safe for Component and Pressure Loads.*

6. CORRELATION- THEORY Vs FEA

Theoretical calculations have been done for two analyses which are Bucket deflection analysis and Nozzle Pressure Vessel Analysis. FEA has also been conducted for these analyses. Hence, we try to correlate the results from theory and FEA

6.1 Correlation of Bucket deflection

Deflection of Bucket as per Theory = 0.015 mm

Deflection of Bucket as per FEA = 0.014 mm

$$\%\,\text{Deviation} = \frac{\delta_{th} - \delta_{FEA}}{\delta_{th}} \times 100 = 6.3\%$$

$$\%\,\text{Correlation} = 100 - \%\,\text{Deviation} = 93.7\%$$

6.2 Correlation of Nozzle Stresses

Stress in Nozzle Cylindrical area as per Theory = 1.85 MPa

Stress in Nozzle Cylindrical area as per FEA = 1.93 MPa

Stress in Nozzle Collar area as per Theory= 1.71 MPa

Stress in Nozzle Collar area as per FEA = 1.55 MPa

$$\%\,\text{Deviation}_{\text{Nozzle Cyl}} = \frac{\sigma_{th} - \sigma_{FEA}}{\sigma_{th}} \times 100 = 4.3\%$$

$$\%\,\text{Correlation}_{\text{Nozzle Cyl}} = 100 - \%\,\text{Deviation} = 95.7\%$$

$$\%\ \text{Deviation}_{\text{Nozzle Collar}} = \frac{\sigma_{th} - \sigma_{FEA}}{\sigma_{th}} \times 100 = 9.4\%$$

$$\%\ \text{Correlation}_{\text{Nozzle Collar}} = 100 - \%\ \text{Deviation} = 90.6\%$$

7. CONCLUSIONS

1. Pelton turbine for Organic Rankine cycle application was successfully designed using Toluene as working fluid. Design procedure for Pelton turbine with gaseous fluids was developed and implemented in the project which is novelty of this work.
2. Hydraulic Efficiency of developed turbine is *90.14 %* assuming friction losses of *15 %*
3. This Turbine develops a power of *178.43 W* for considered input conditions
4. A detailed Theoretical analysis is conducted for deflection of Bucket and Nozzle stresses
5. FEA analysis is conducted for determination of deflection of Bucket, Nozzle Stresses, Casing Structure stresses and Bucket modal frequencies. *Maximum Stresses induced is all components in above FEA simulations are below Material Yield Limits*. Even though stress in casing exceeds yield limit marginally, strain of *0.2%* is within the limit.
6. FEA results correlate well with Theoretical calculations with maximum Correlation for Bucket deflection as 93.7 %
7. Thus as per detailed FEA and Theoretical analysis, *developed Pelton Turbine is safe for considered loading scenarios*
8. Thus, *a comprehensive working turbine design has been developed and proved using FEA and theoretical analysis*
9. Future scope of this work includes *experimental investigation, CFD analysis, pressure distribution study and estimation of losses due to friction*.

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