Design and Analysis of Pelton Turbine for Organic Rankine Cycle application

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***Abstract*—Depleting fossil fuel and growing energy demand have led to renewed emphasis on extracting more and more energy from exhaust gases let out from internal combustion engines. Organic Rankine cycle uses organic fluids and hence helps to extract low grade energy from exhaust gases. It is one of the most efficient cycles for waste heat recovery. Organic fluid absorbs heat from exhaust gases and vaporizes. This gaseous fluid is then taken to turbine to extract mechanical energy. This work intends to design and analyze a Pelton turbine that will generate mechanical work using gaseous Toluene as working fluid. Pelton turbine, until now, has only been developed using water as the working fluid. In this sense, this is a unique work that intends to develop Pelton turbine using gaseous fluids. Detailed analytical design procedure with determination of critical parameters and dimensions is presented in this work. With the inputs from analytical design, CAD models are developed and detailed Finite Elements Analysis is performed for bucket which is the most critical component. Further, efficiency calculations are also performed in this work.**

***Index Terms*— ORC, FEA, Pelton Turbine, Bucket, Design, Analysis**

I. Introduction

WITH depleting fossil fuels all over the world and growing concerns over global warming, considerable research work is being done for improving the efficiency of Internal Combustion (I.C.) engines. Waste heat recovery using Organic Rankine cycle (ORC) appears to be a promising technique in this regard. It serves the dual purpose of enhancing engine efficiency and reducing the temperature of exhaust gases emitted to atmosphere. It works the same as conventional Rankine cycle except that it uses organic fluids instead of water or steam. This change allows it to convert low temperature heat energy into useful work. Turbo expander is the heart of ORC system since it is the power generating component of the cycle.

Turbo expander is essentially a turbine. According to the way of energy transfer, there can be two types of turbines namely Impulse turbine and Reaction turbine. In Impulse turbine, the entire pressure drop occurs in the nozzle whereas in the reaction turbine, pressure drop occurs in nozzle as well as the vanes and runner. Further, Reaction turbine is much costlier to build and it has much lesser efficiency as compared to Impulse turbine. Also, for same power output, Impulse turbine is of lesser size. With these considerations, we have selected tangent flow impulse turbine as the turbine type for turbo expander. In more specific terms, we intend to develop Pelton Wheel turbine.

Pelton Wheel has been designed using only water as working fluid until now. Uniqueness of this work lies in the fact that, Pelton Wheel will be designed for organic fluids in this work. In more precise terms, organic fluids would be in gaseous form after gaining heat from exhaust gases. This makes the design work that much more challenging. Selection of organic fluid becomes an important task since the selected fluid has to be stable with the material of turbine components.

Design of Pelton wheel has two different but connected aspects namely thermodynamic and structural. Thermodynamic inputs required for structural design are taken from extensive literature review. Working fluid selection is also done using literature. Using these inputs, this work intends to provide a systematic approach for design of Pelton Wheel. Force and strength calculation of important components is included in the scope of work. Further, detailed finite element analysis of bucket is also carried out.

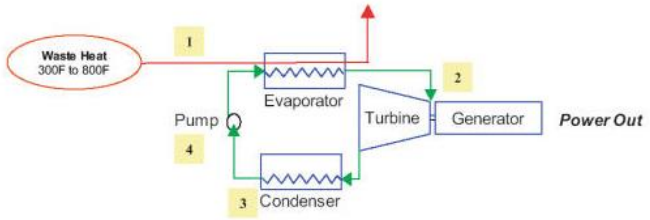

Fig. 1. Organic Rankine cycle

II. Literature Review

The Pelton Turbine has been given increasing interest by the research community within multiple fields. This is due to increasing demand for energy on a global basis in addition to the growing focus on meeting the energy demand by utilizing renewable energy sources. An increase in efficiency in the order of 0.1 % would lead to large increase in electrical energy production [1].

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Reference [2] shows the basic formulae in the design of Hydraulic Pelton turbine. A numerical methodology to investigate complex unsteady free surface flow in a Pelton turbine using Lagrangian approach is presented by J.S. Anagnostopoulos et al [3]. The influence of jet velocity and jet quality on turbine efficiency was investigated by Vesley and Staubli in [4], [5]. Unsteady analysis of a Pelton runner with mechanical simulation was presented by Parkinson in [6].

The effect to Jet to runner speed ratio on Pelton turbine efficiency is tested experimentally by Bryan in [7]. A modification in bucket design to increase efficiency is suggested by S. Yadav [8]. Stress distribution inside a Pelton bucket has been investigated by V. Sharma et al. for hydraulic application [9]. The Effect of bucket parameters on flow in hydraulic Pelton Turbine and its efficiency was investigated using a parametric design by Solemslie et al. [10]. Chen et al. discussed the selection criteria for working fluid in ORC and presented a screening of 35 fluids for their suitability in ORC turbines [11].

It is clear from the review of available literature that, while considerable effort has gone into design of Pelton Wheel for hydraulic application; Pelton Wheel using gaseous fluids is an unexplored area. Also, turbine design in ORC applications has been mostly Reaction type. Thus, design of Pelton Turbine for ORC application becomes a novel research topic.

III. DESIGN METHODOLOGY

Design methodology of Pelton Wheel starts with input gathering using literature and ends with complete structural design of Pelton Wheel for ORC application. Important steps in the methodology are as shown in Fig. 2.

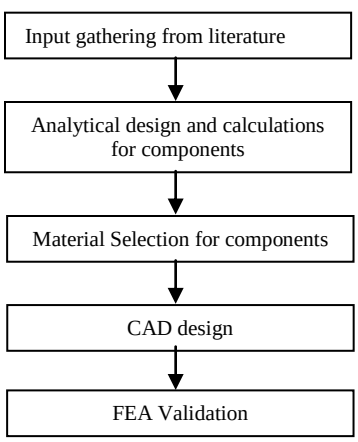

Fig. 2. Design Methodology

Abbreviations used in Fig. 2 are as follows
CAD: Computer aided drafting
FEA: Finite Elements Analysis

IV. DESIGN INPUTS

A. Selection of working fluid

Based on review of literature, Toluene was found to be most suitable working fluid for present operating conditions. Toluene is considered as isentropic fluid and has a relatively high critical temperature of 318.75 °C. Toluene has good vapour density and hence, it reduces the size of power plant and amount of material required. Further, Toluene does not form condensate on turbine blades and corrosion is also absent. All these properties make Toluene suitable candidate for working fluid

B. Design Considerations

Following assumptions or considerations are made in the design process using design guides and literature:

1. As per general exhaust gas temperature, it is assumed that working fluid (toluene) is heated to 300 °C and 10 bar pressure by exhaust gases
2. Flow across the nozzle is considered to be adiabatic
3. As per standard nozzle and tubing dimensions, inlet diameter of nozzle (d_1) is taken as 12 mm
4. Inlet angle of nozzle, α is taken as 0°
5. Maximum flow rate through the turbine, Q is taken as 0.5 Kg/s due to material considerations

V. ANALYTICAL DESIGN AND CALCULATIONS

A. Nozzle

Elemental equations from Fluid machinery design are used to determine critical dimensions of nozzle. We shall use the following notations to designate the parameters of nozzle:

d_1 = Inlet diameter of nozzle = 12 mm (standard nozzle)
d_2 = outlet diameter of nozzle
D = Nozzle outer body diameter
t = Thickness of nozzle body
P_1, P_2 = Pressures at nozzle inlet, outlet (P1=10 bar)
ν_1, ν_2 = Velocity of fluid at nozzle inlet, outlet
ρ_1, ρ_2 = Density of fluid at nozzle inlet, outlet
L = Length of nozzle
A_1, A_2 = Area of nozzle at inlet, outlet
$\dot{m}$ = Mass flow rate of fluid (assumed as 0.5 kg/s)

Areas of nozzle at inlet and outlet are calculated as follows:

$$A_1 = \frac{\pi}{4} d_1^2 = 1.13 \times 10^{-4}\ m^2$$

At a temperature of 300 °C, density of Toluene is ρ_1= 477.28 Kg/m^3

Mass flow rate through a nozzle where minimum pressure is equal to critical pressure can be expressed as:

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

In the above relation, polytrophic index of Toluene (*n*) is taken as 1.2 and then area A_2 at throat of nozzle is found as 35.29 mm^2. Diameter of nozzle at throat is

$$d_2 = \sqrt{\frac{4A_2}{\pi}} = 7\ mm$$

Velocity at nozzle inlet can be found as follows:

$$\dot{m} = \rho_1 A_1 \nu_1$$

$$\therefore\ \nu_1 = \frac{\dot{m}}{\rho_1 A_1} = 9.27\ m/s$$

Pressure of fluid at nozzle outlet is found by applying critical

pressure ratio principle.

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 5.64 \text{ bar}$$

Temperature at outlet of nozzle can be found from elementary adiabatic process relation as follows

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

Let us denote coefficient of volumetric expansion for Toluene by β. As for most of the liquids, let us assume β as 0.001 /K. Using β, value of density of fluid at nozzle outlet can be found as follows:

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

Now, velocity at the exit of nozzle can be found as follows:

$$v_2 = \frac{\dot{m}}{\rho_2 A_2} = 28.13 \text{ m/s}$$

Dimensions of the nozzle are found from empirical relations as follows:

Outer Diameter, D = 2.5 x d = 30 mm
Length of nozzle, L = 8 x d ≈ 100 mm
Thickness of nozzle, t = (D – d_1)/2 = 9 mm

B. Wheel

Design of wheel starts with jet ratio which is defined as ratio of mean or pitch diameter of wheel to diameter of jet. We know that diameter of jet is 7 mm from our previous section. For maximum efficiency, jet ratio should be between 10 and 14. We select jet ratio of 12 for maximum efficiency.

Mean diameter of wheel, D_m, can then be found as follows:
D_m = Jet Ratio x d_j = 84 mm (d_j is jet diameter)
Shaft diameter, d_s = 0.3 x D_m = 25 mm
Diameter of shaft collar, d_c = 1.25 x d_s ≈ 32 mm
Number of buckets, Z = 15 + (D_m/2d_j) = 21

Further design of wheel depends on analysis of velocity diagram. Following are the notations used in velocity diagram
V_1, V_2 = Absolute velocities at bucket inlet and outlet
V_{r1}, V_{r2} = Relative velocities of fluid at bucket inlet and outlet
V_{w1}, V_{w2}= Tangential component of velocities at bucket inlet and outlet
u_1=u_2= Bucket speed

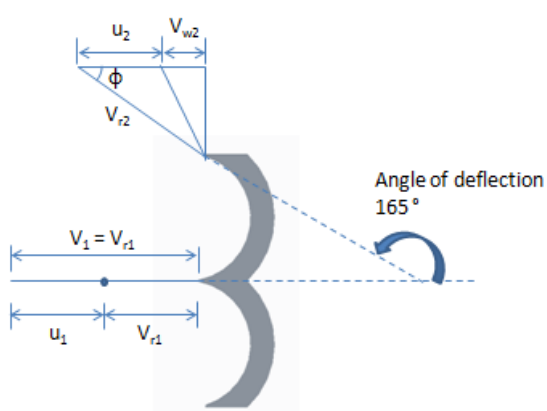

Fig. 3. Velocity diagram

Here, V_1 = 28.13 m/s (velocity from nozzle)

Entire component of velocity V_1 is along tangential direction. Hence V_{w1}=V_1 =28.13 m/s

For maximum efficiency, bucket speed (u) should be as close as possible to half of velocity of jet. However, considering u=0.5V_1 is not realistic. In realistic design of Pelton turbine, bucket speed is taken as 0.44-0.46 times velocity of jet.

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

Angle of deflection of Pelton bucket should ideally be 180° for maximum momentum change of fluid and hence maximum driving force on the wheel. However, if the angle of deflection is kept at 180°, all fluid leaving a bucket will hit following bucket. Hence, angle of deflection is limited to 165 °.

Relative velocity, Vr1 is determined from velocity diagram as

$$V_{r1} = V_1 - u = 15.47 \text{ m/s}$$

Ideally, relative velocity at outlet should be equal to relative velocity at inlet in absence of friction from blade surfaces. In practice, however, blade surfaces offer considerable friction to fluid passage and hence relative velocity at outlet reduces by 15% as compared to relative velocity at inlet.

$$V_{r2} = 0.85\, V_{r1} = 13.15 \text{ m/s}$$

Angle, φ, in outlet triangle is calculated as,
Φ = 180 -165 =15°

Tangential velocity of fluid at outlet is calculated from velocity triangle as follows,

V_{w2}=V_{r2} (cosφ) – u = 0.043 m/s.

Power transferred to wheel by fluid is calculated as follows,
Power transferred = ρAV_1 (V_{w1}+V_{w2}) × u = 178.43 W

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

Hydraulic Efficiency of turbine is defined as ratio of power transferred to wheel by jet to kinetic energy of jet per second.

η_{Hyd} = Power Transferred / KE of jet per second = 90.14 %

Speed or turbine shaft assuming no mechanical losses in the system is calculated as follows,

$$N = \frac{60 \times u}{\pi \times D_m} = 2878 \text{ RPM}$$

C. Bucket

Bucket design is based on standard empirical relations to determine the dimensions of bucket. Following notations are used in the course of design:
B = bucket axial width
L = bucket radial length
T = depth of bucket
t = thickness of bucket
Calculation for bucket dimensions is as follows:
B= 3.4 × d_j ≈ 24 mm
L = 3 × d_j = 21 mm
T = 1.2 × d_j ≈ 9 mm
t = 0.5 × d_j = 3.5 mm

Force acting on the nozzle, F_X, is calculated as follows:

$$F_X = \rho A V_1 \times (V_{w1} - V_{w2}) = 14.04 \text{ N}$$

VI. MATERIAL SELECTION AND CAD DESIGN

From cost and material strength standpoint, material AISI 1018 mild/low carbon steel is chosen for bucket, wheel, shaft and nozzle. Though friction in bucket will be high for steel as compared to Aluminum and copper, cost of steel is significantly less than these materials. Hence we select this steel grade. Incorporation of appropriate coating on bucket steel faces to reduce friction can be a topic for further research. Material properties of AISI 1018 mild/low carbon steel are as follows:

Density = 7870 Kg/m^3
Young's modulus, E = 205 GPa
Tensile Strength, Yield = 370 MPa
Tensile Strength, Ultimate = 440 MPa
Poison's ratio = 0.29

Taking critical dimensions from the calculations discussed in previous section, we proceed for CAD modeling of Bucket, Wheel, Nozzle and Shaft. CAD modeling is executed using ProE software. Some of the pictures of CAD design are included in the below sections.

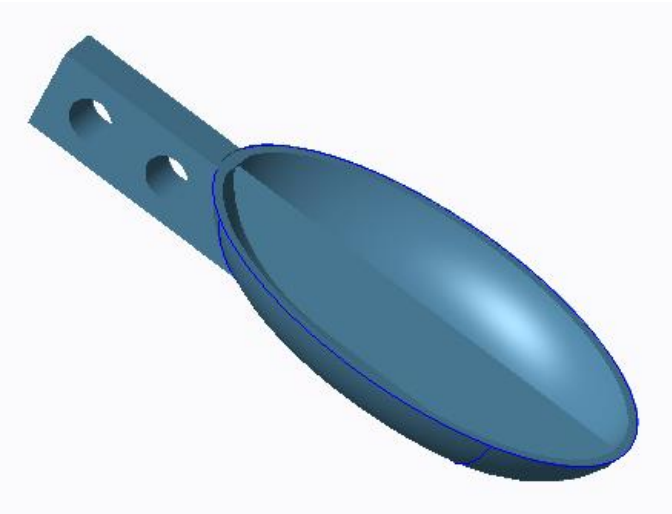

Fig. 4. Bucket model

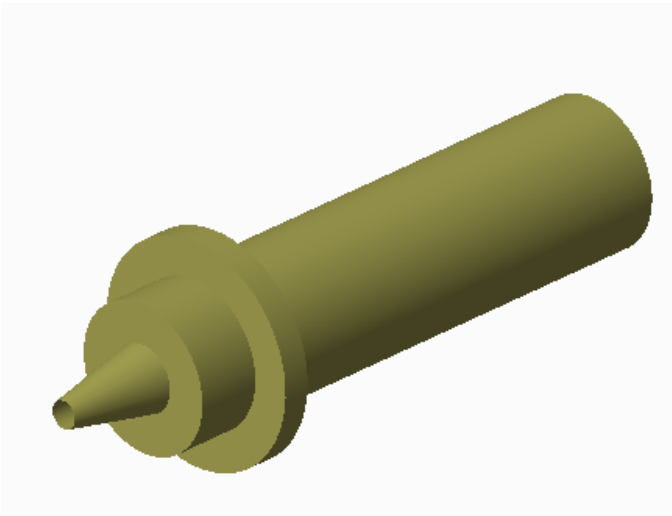

Fig. 5. Nozzle model

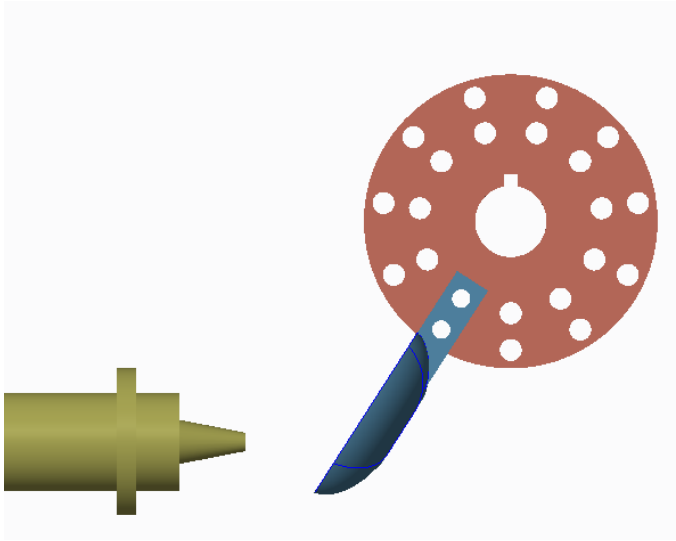

Fig. 6. Assembly model with one bucket

VII. FEA VALIDATION

FEA is finite elements analysis. It deals with discretizing the domain into nodes and elements. Elemental displacement equations are then solved and assembled to get the displacement for entire components. We have used Ansys R12.0 workbench for executing the analysis.

Bucket is the main mechanical component in the Pelton turbine assembly. Hence, our analysis effort is dedicated to bucket for the project. Bucket is subjected to impact force in the tangential direction due to velocity of incoming fluid. Further, the wheel rotates at 2878 RPM after jet impacts the bucket. Hence, exciting frequency of 48 Hz (2878/60 RPS) can be considered for system. If bucket natural frequency matches the exciting frequency, resonance would occur in the bucket leading to large scale vibration, deformation and noise in the system. This will lead to premature failure of bucket. Hence, it is very important that bucket natural frequency does not match with exciting frequency. Thus, two types of finite element analysis are selected for this work and presented below. These are as follows:

1. Bucket Structural analysis – Determination of stresses and deflection due to impact force
2. Bucket Modal analysis – Determination of natural frequency of bucket

FEA starts with meshing of the component. Meshing includes selection of right type of elements that are suitable for structural and modal analysis. Further, elements should be able to follow the intricate shape of the bucket. This will guide us to select the element size and order. Considering all these factors, we have selected second order tetrahedral elements for analysis. Element size is kept at 1 mm. Workbench mesher is used for this work.

Second step in analysis is application of suitable boundary conditions. Since the bucket is going to be screwed to wheel at two locations, these locations are applied the boundary condition of Fixed Support. Fixed Support constrains all degrees of freedom of the component. Third step in analysis is

application of loads. We have already calculated the impact force as 14 N in preceding section. Since our calculation makes simplifying assumptions in order to use standard formulae, we have to apply factor of safety to this force to take care of realistic scenario. We apply a factor of safety of about 2 and the force applied in FEA is 28 N. Pictures of analysis and results are given below.

Fig. 7. Meshed Bucket

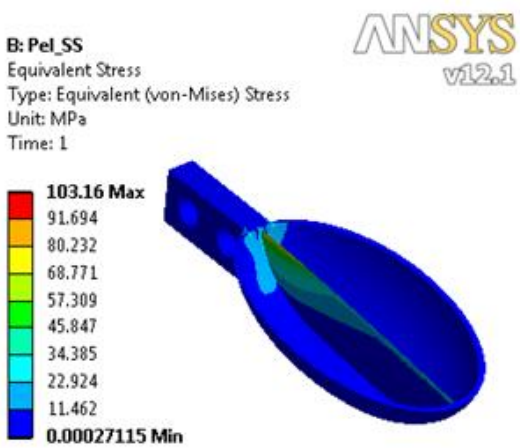

Fig. 8. Stresses in Bucket

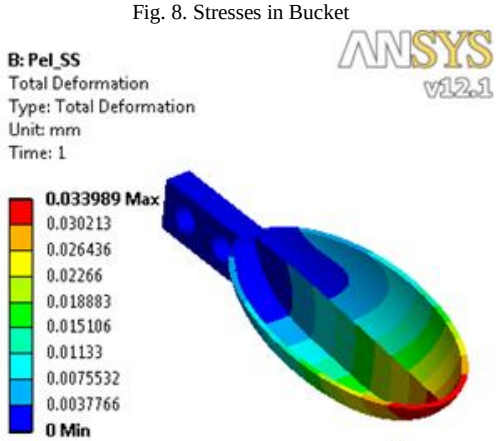

Fig. 9. Deflection of Bucket

Maximum Von mises Stress generated in bucket is 103.1 MPa which is lesser than material Yield limit of 370 MPa. Hence, bucket is safe for given loading. Further, deflection of the bucket is 0.033 mm which is negligible. Hence the bucket passes the structural analysis.

TABLE I
RESULTS FOR MODAL ANALYSIS

Mode number	Mode Frequency (Hz)
1	428
2	680
3	1158
4	2170
5	2441
6	3682

First or lowest modal frequency calculated from analysis is 428 Hz, which is considerably higher than exciting frequency of 48 Hz. Hence, there will be no resonance in the bucket. Thus, Bucket design passes the modal analysis. Next steps in the project would be experimental validation of stresses and strains. Further, system level modal analysis of assembly would also be conducted.

VIII. CONCLUSION

Pelton turbine for Organic Rankine cycle application was successfully designed using Toluene as working fluid. General exhaust gas conditions were used as a starting point for design. Design procedure for Pelton turbine for gaseous fluids was developed and implemented in the project which is the novelty of this work. Basic relations from Thermodynamics were used as a input for structural design of Pelton Turbine.

Critical dimensions of the turbine and forces acting on turbine were calculated using the developed design procedure. Hydraulic efficiency of turbine ignoring friction losses in the bucket was found to be 90.14 %. Using dimensions calculated, a CAD model of turbine was developed. Further, a detailed Finite Element analysis for Bucket is executed using the force calculated as input. Results from FEA show that maximum stress generated in Bucket is 103.16 MPa and deflection is 0.033 mm. Thus, stress generated is less than material Yield limit of 370 MPa. Modal analysis shows that lowest mode frequency is 480 Hz which is greater than exciting frequency of 48 Hz. Thus, it is concluded that Bucket is safe for given loading and resonance would not occur in Bucket. Further steps in the project include experimental validation and system level fatigue analysis.

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