Design Methodology and Structural analysis of Cross flow Turbine

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ABSTRACT

Cross flow turbine was developed and patented by A. G. M. Michell during 1903 and further it was developed by Donat Banki. He has worked in this area, for blade profile design, development of expression for maximum efficiency, ratio of the inner to outer diameter of the runner, and energy transfer at each stage. Now it is known as Banki or Michell-Banki Turbine or cross flow turbine. The turbine developed by Banki is having theoretical efficiency of 87% and actual efficiency obtained is far below the theoretical efficiency.

In order to improve its performance, during the past decades an extensive amount of research works has been conducted and documented on flow analysis of Cross-Flow turbine. One of the area, less concentrated and much needed to serve for its design life period is system integrity. For checking the structural integrity, structural design analysis needs to be done. Very few analysis publications are available in this regard. In this present study a turbine was analyzed and optimized for its structural integrity and for its modal frequencies for various flow conditions. Based on the analysis results, further it was strengthened and analyzed. Finite element software package ANSYS Workbench 15.0 is used for this analysis. Obtained modal frequencies, stress and deformation contours for design inlet pressure and design load are discussed and presented.

Keywords: Cross-Flow Turbine, Design, Development, structural design, Finite element analysis

1. INTRODUCTION

Hydropower is one of the efficient sources for clean and renewable energy used for power generation. It has been utilized for many centuries across the globe. It is the leading renewable source for electricity generation globally, and supplying 71% of all renewable electricity at the end of 2015. Undeveloped potential is approximately 10000 TWh/y worldwide. The global hydropower capacity increased by more than 30% between 2007 and 2015 accounting to a total of 1 209 GW in 2015[1]. It accounts for 6.8% of global power generation. In a World Bank report it was shown that pico-hydropower represented the cheapest opportunity for off-grid generation under 5 kW in 2005 and was projected to be at least 25% cheaper than the nearest alternative still after ten years [3]. Apart from economic viability, hydro plants are having long life span than any other power generation unit. Generally, hydropower is produced by turbines installed behind large dams; however, environmental challenges associated with dams, makes local power generation an attractive and efficient option, especially in rural, remote and mountainous areas. Pico turbines can be installed on small dams, water streams, run off rivers, to provide enough electricity in remote areas. However, the turbine used for this purpose should have unique characteristics, including an affordable price and easy to maintain and low maintenance cost. Also, a high efficiency under varying load conditions, that is present in many local water sources is expected. These features are all present in a Cross-Flow turbine.

In a Cross-Flow turbine, water enters the rotor two times, thereby transmitting the majority of its kinetic energy to the rotor. During both the time, energy transfer will happen [9,10]. The specific speed of Cross-Flow turbines is greater than impulse Pelton turbines but lower than mixed-flow Francis turbines. Banki, has demonstrated that the theoretical efficiency of Cross-Flow turbine is 87%, and which is slightly less than other types of hydraulic turbines [2]. It is suitable for applications where water head is between 2.5 and 200 meters and volume flow rate is between 0.025 and 13 m^3/s. Cross-Flow turbine are ideal for pico and micro power plants. It is particularly of interest because of its flat efficiency curve over wide flow/ head range. When needed, the turbine can be built as a multi-cell turbine with a 1:2 division. This enables the turbine to handle low-flow conditions and operate at optimum efficiency for any water flow from 1/6 to full design flow rate [4]. Goodarz Mehr [5] has discussed about the design methodology of cross flow turbine and documented the work conducted in design development and CFD analysis in this area. Many researchers have worked in the area of design optimization of cross flow turbine in efficiency improvement. As many as the design optimization using CFD as a tool, testing activities are also carried out and documented [6, 7]. But very limited studies were conducted in structural analysis of cross flow turbine [8]. In this paper, design procedure is briefly discussed and structure analysis of a turbine was done using the software package ANSYS.

2. CROSS FLOW TURBINE DESIGN

Banki/ Cross flow turbine is an impulse turbine, optimized to work with wide variation of flow and head. It consists of major parts such as nozzle, impeller, casing and guide vane. The impeller has the shape of an empty wheel, consisting of two or more circular plates equally spaced and linked by a series of blades, shaped so that the jet is directed towards the blade of impeller and pass through center of the wheel and then again crossing other set of blades before exiting. Since the jet crosses through the impeller and hit again the blades, it is called as cross flow turbine. Basic design of cross flow turbine is reported by Mockmore [2]. An extensive literature review on the development of hydraulic cross-flow turbines, is reported in [9,10]. It have been shown that the water flow within the impeller still possesses an appreciable energy content when it reaches the blade output, where there is a second energy transfer from the water flow to the turbine. Figure 1 [2] illustrates the path of water through turbine. The water starts enter from point A and strikes a blade AB. Then through the interior of the runner, water strikes again to a blade CD and pass through the exit.

2.1 Efficiency

First and foremost, cross flow turbines are designed based on the available head, flow rate and efficiency. For a turbine design, power available at the impeller outlet, Mockmore [2] defined the hydraulic efficiency of the machine 'η' as the ratio between the power available for production by the machine P_T and the input power P_{in}.

$$\text{Turbine efficiency } (\eta) = \frac{P_T}{P_{in}} \qquad (1)$$

The turbine efficiency defined in Equation (1) is strongly affected by the geometry of the nozzle and the impeller.

$$\text{In this,} \qquad \text{Inlet power } P_{in} = \rho g Q H \qquad (2)$$

Where

P_{in} – Inlet power in watts

ρ – Density in kg/m^3

Q = Discharge in m^3/sec

H = Head in m

g- Acceleration due to gravity in m/sec^2

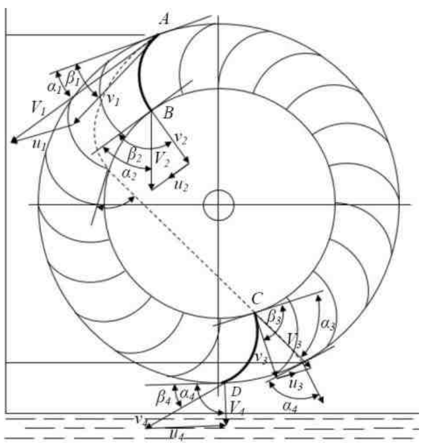

Figure 1: path of water through turbine

Power available at turbine shaft output is the power generated by flowing water at two stages. First one is due to the flow directly striking on the runner blades and second one is due to the cross flow travel through the centre of impeller and hitting on another set of blades. Mockmore [2] derived equations for power generated at the shaft of turbine is as in equation 3.

Power developed $P_T = \rho Q u_1 \left(V_1 \cos \propto_1 - u_1\right)\left(\left(1 + \frac{cos\beta_2}{cos\beta_1}\right)\right)$ (3)

Where

u_1 – peripheral velocity of impeller

d_1 – outer diameter of impeller

d_2 – inner diameter of impeller

$\beta_1 -$ angle between relative velocity and peripheral velocity

$\beta_2 -$ blade exit angle

α_1 – angle of attack

For simplification purpose he has optimized the attach angle as α = 16° and considered $\beta_1=\beta_2$. With that the maximum theoretical efficiency derived is 87.8 %.

Further the angle of attack was optimized by various researchers and documented by Vincenzo Sammartano [11], that optimum angle as 22°. With this, efficiency of 90% was obtained by various researchers. It was also documented, that the optimum value of β_2 is 90°.

2.2 Impeller diameter

Impeller is having two diameters. First one is outer diameter d_1 and second one is inner diameter d_2. Outer diameter can be determined from the equation 4.

Outer diameter $d_1 = \frac{30\sqrt{2gH}\cos\alpha_1}{\pi N}$ (4)

Where 'N' – rotational speed of turbine

Outer diameter is depend on angle of attack of jet and is also based on the available head and rotational speed.

Increase in head will increase the diameter and speed is inversely proportional to it.

In literatures it is suggested to have a optimum diameter ratio of 0.68, ie $\frac{d_2}{d_1} = 0.68$. Aziz and Totapally [13] also suggested to have a diameter ratio of 0.68 for achieving maximum efficiency.

2.3 Blade design

Selection of number of blades is very important in the design of turbine runner. Less number of blades may cause incomplete utilization of water available to the turbine and cause pulsating power and excessive stress on blades. More number of blades may increase the friction lass.

Number of blades in the propeller is proportional to the jet thickness and outer diameter. Jet thickness of the flow from nozzle is as in equation (5) and number of blades is as in equation (6)

Jet thickness $'t' = \frac{kd_1}{sin\beta_1}$ (5)

Where 'k' = 0.075 to 0.1

Number of blades $'n' = \frac{\pi d_1}{t}$ (6)

Blade thickness is optimised by various researchers. Blade thickness can be determined by equation (7).

Blade thickness $t_b = k_b\ d_1$ (7)

Mockmore. C.A., Merryfield Fred [2] has used the factor 'k_b' as 0.0088 and Muhammad Adil Khan, Saeed Badshah [12] has suggested to have 'k_b' as 0.0177 to 0.0185.

Another important factor in blade design is radius of curvature of blade. Or this relationship was developed with outer diameter of impeller and is as in equation (8)

Radius of curvature $= 0.163\ d_1$ (8)

Width of the impeller suggested by Mockmore is

Width $B_w = 0.074\ \frac{NQ}{H\ cos\alpha_1}\ to\ \ 0.055\ \frac{NQ}{H\ cos\alpha_1}$

Aziz & Desai [14] has optimized that the Width $B_w = 1.44\ d_1\ \ to\ \ 1.92\ d_1$

2.4 Shaft Diameter:

Shaft of the turbine has to have the capability to take the torque produced by the impeller and it should not be too large that water striking the shaft after passing through the first set of blades at the inlet. Since the torque is produced by the flowing fluid through impeller, it is having direct relation with the diameter of impeller. The diameter of shaft for turbine runner can be calculated as per ASME code is as in equation (9).

$d_s^3 = \frac{16}{\pi\ \tau_{allowable}}\sqrt{(C_{bm}M)^2 + (C_tT)^2}$ (9)

Where

$\tau_{allowable} -$ Allowable stress

M – Bending moment

T-Torque

C_{bm} and C_t are the bending and torsion factors as in ASME code.

Muhammad Adil Khan, Saeed Badshah [12] has optimized the shaft diameter in terms of impeller outer diameter, and is as in equation (10)

$$\text{Shaft diameter } d_s = 0.22\, d_1 \qquad (10)$$

2.5 Nozzle & Housing design:

Nozzle is a critical part of the system design and together with the turbine to determine the efficiency of system. Losses must be as small as possible and the flow must be uniform through the nozzle to achieve the highest possible transformation of potential energy to kinetic energy. One of the largest variations in the nozzle design is the nozzle entry arc which is the angle over which the water enters the turbine. Banki’s calculations translated in Mockmore and Merryfield’s paper are based on a narrow jet of water that only hits one turbine blade at a time. However, the design of the Cross flow turbine seems to have evolved to have a larger nozzle where the water jet hits multiple blades at a time. Apart from optimization for maximum efficiency, it has to be designed to withstand for structural integrity. The nozzle will be subjected to the pressure head available in the system.

Housing is mainly to hold the impeller assembly and to support the nozzle. It has to be stiff enough to withstand the load transmitted by nozzle and impeller assembly. Due to the complexity of structural geometry, optimization using Finite element method is more appropriate.

3. TURBINE SPECIFICATION

Turbine available at FCRI is used to study for structural analysis and design verification. For analysis, available commercial software ANSYS Workbench is used. Design specification of turbine is as given in table 1.

Turbine design output	10 kW	Nozzle sheet thickness	5mm
Maximum discharge	400 m^3/hr	Casing thickness	5 mm
Available head	50 m	Shaft diameter	36 mm
Inlet diameter	150 mm	Bearing size	30 mm
Transition length	1 m		
Nozzle size	120x130 mm		
		Number of blades	28
Runner outer Diameter	150 mm	Blade thickness	2 mm
Runner inner diameter	110 mm	Impeller side plate thickness	3 mm

Table 1: Turbine specification

4. ANALYSIS

Hydro turbine need to be designed and optimised for achieving maximum efficiency and for structural integrity. Efficiency optimization can be done using Computational Fluid Dynamic software packages. It includes, nozzle flow profile optimization, blade profile optimization, number of blades etc.,. Since it is not scope of this paper, it is not included. In this, it is discussed about the structural integrity analysis of two major components, ie., impeller and housing including nozzle. Two types of analysis required are stress analysis and model analysis. Stress analysis is used to get its stress distribution, stress concentration and deformation of the object due to various loading acting on it. In the case of cross flow turbine, load acting in the casing is the static head available. In this present study it has the head of 50 m.

Impeller design it mainly due to the torque generated by the flowing fluid, acting on the blade. Due to flowing fluid, torque will be generated in two stages. Shepherd [15] has reported that, cross-flow turbines 75% of the available energy is transferred with greater efficiency in the 1st stage, when the water flows towards the interior of the runner blade, and the remaining 25% is transferred with lesser efficiency in the 2nd stage. De Andrade *et al.* [16] have numerically analyzed and shown that the energy transfers at first stage are 68.5 % and 31.5% in the second stage. So for analysis purpose, it is conservative to consider 75% torque transfer in first stage and 25% in second stage. As per Banki's calculations translated in Mockmore and Merryfield's paper [2] narrow jet of water that only hits on one turbine blade at a time. But further studies shows that only 50% of flow is through one blade and remaining shared by other two adjust ant blades. The right proportion of water sharing by blades can be determined by performing CFD analysis.

Modal analysis is another structural analysis tool to determine the mode shapes and natural frequencies. As it is essential that natural frequency of the cross flow turbine does not match the frequency of excitation generated by the rotational speed. If the natural frequency of turbine is same as the rotational speed or its harmonic, it may continue to resonate and experience structural damage. Modal analysis of casing and impeller assembly are simulated separately.

For both the analysis, commercially available Finite Element Analysis software ANSYS work bench has been used. It has various modules to perform analysis. From this required modules of static structural and modal are selected and used. For both the analysis, common geometry was created and meshed. For meshing, three dimensional solid elements are selected.

5. RESULTS & DISCUSSION

Structural analysis of casing and impeller are performed separately for its static structural and modal analysis. Stress analysis results are in the form of stress/strain distribution and deflection. Modal analysis results are in the form of natural frequency and mode shapes.

5.1 Impeller

Stress analysis

Stress analysis of impeller assembly was carried out due to the torque produced by the flowing fluid at two stages by considering 75% torque is produced by inlet side blades and 25% by the outlet side blades. Photograph of impeller considered for analysis is as in figure 2 [7]. Analyzed stress and deformation results of impeller is given in table 2. Torque considered for this analysis is, converting the entire inlet power in to mechanical power. Stress and deflection contours of impeller is as in figure 3 & 4. Since the stress levels are above the allowable limit of 1e+8N/m^2, required structural modification is carried out. As part of stiffening, 2 mm thick plate having outer diameter of 150 mm and inner diameter of 110 mm is considered. Two such stiffening plates are inserted in the blade assembly at equal distance. Geometry of modified impeller is as in figure 5. Stress and deflection contours of modified impeller are as in figure 6 & 7 and the maximum values are as in table 2.

	Without stiffening	With additional stiffener
Maximum deflection in mm	0.5	0.043
Maximum deflection in N/m^2	4.9e+8	0.76e+8

Table 2: Maximum stress and deformation

Modal analysis

Modified impeller is further analyzed to determine its mode shape and natural frequency and mode shape. First six natural frequency of the impeller is as given in table 3 and its mode shapes are as in figure 8 to 13.

Mode number	Natural frequency in Hz
1	818.0
2	1095.3
3	1096.4
4	1366.3
5	1775.8
6	1777.0

Table 3: Impeller natural frequency

In this, first mode corresponds to the side plate of impeller. Second and third modes corresponds to the mode of entire assembly in two perpendicular directions. Fourth mode corresponds to the blade and fifth and sixth corresponds to the blade frequency in two perpendicular directions. For qualifying the impeller by modal analysis, its first natural frequency should be minimum of 1.3 times of the rotational speed of impeller. In this design, maximum speed of the impeller is 3000rpm and is corresponds to 50Hz. For this, natural frequency of the impeller should be above 65 Hz. Observed first natural frequency of impeller is 818 Hz, which is well above the allowable limit.

Figure 2: Impeller

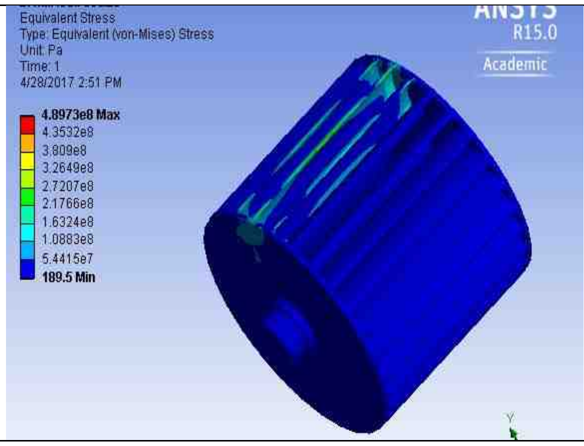

Figure 3. Stress distribution of impeller

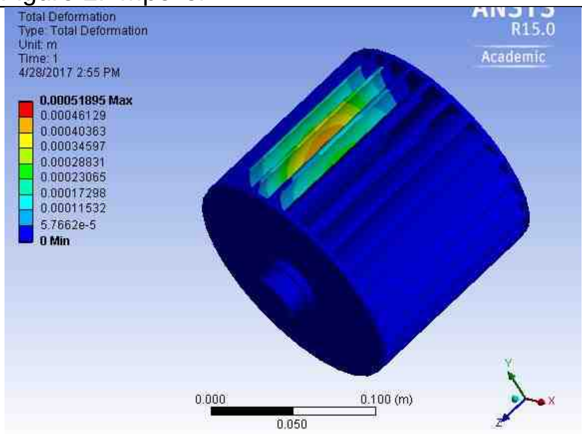

Figure 4. Deflection distribution of impeller

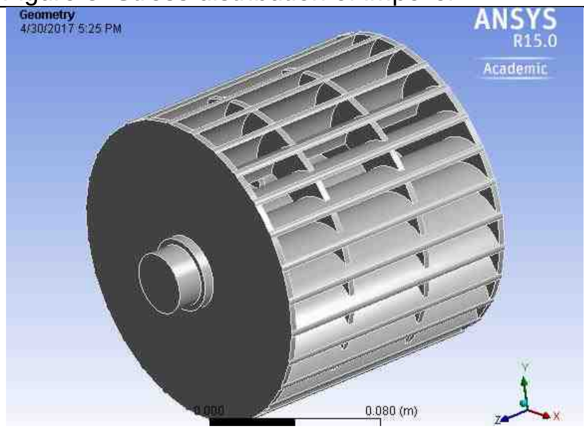

Figure 5: Modified impeller

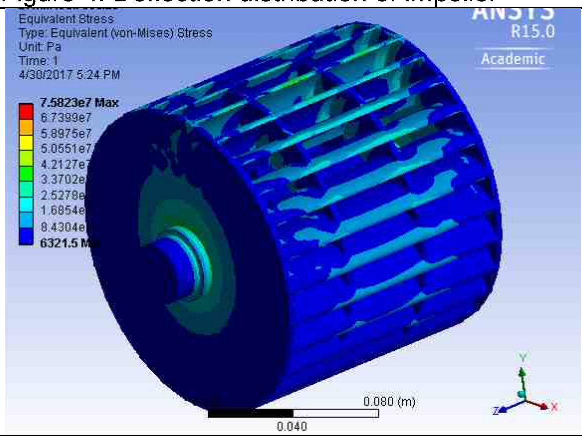

Figure 6. Stress distribution of modified impeller

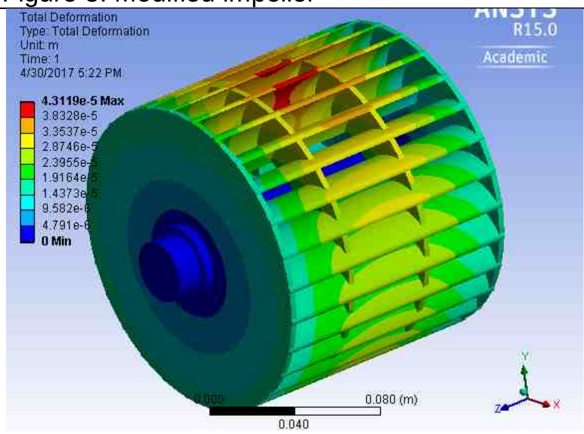

Figure 7. Deflection distribution of modified impeller

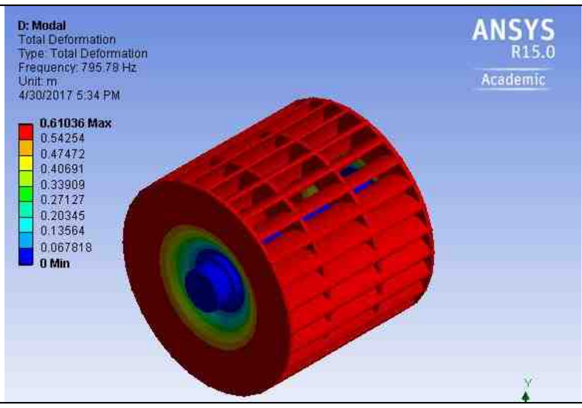

Figure 8: First mode of impeller

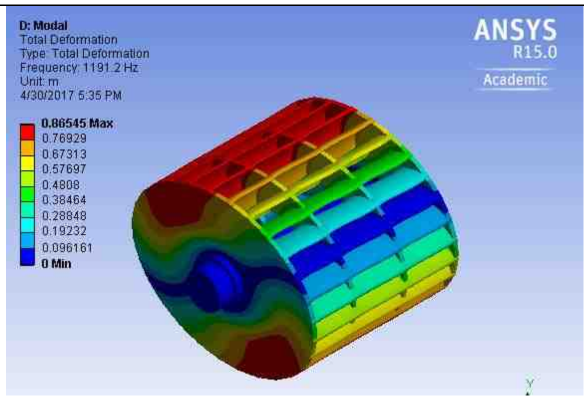

Figure 9: Second mode of impeller

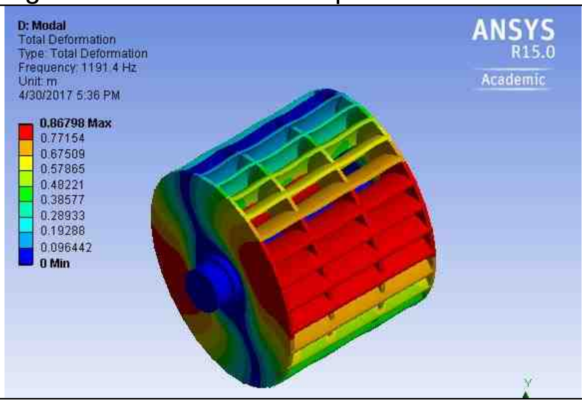

Figure 10: Third mode of impeller

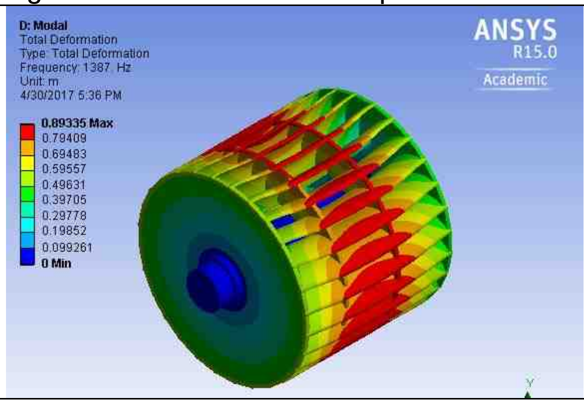

Figure 11: Fourth mode of impeller

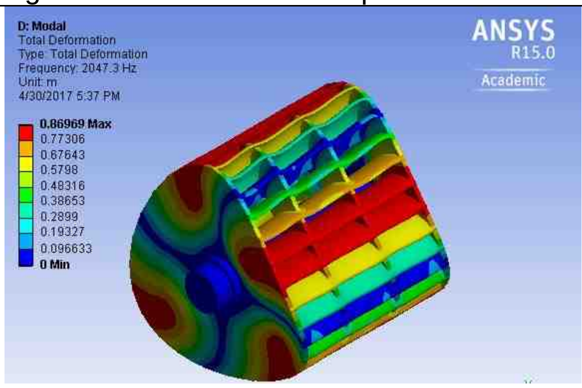

Figure 12: Fifth mode of impeller

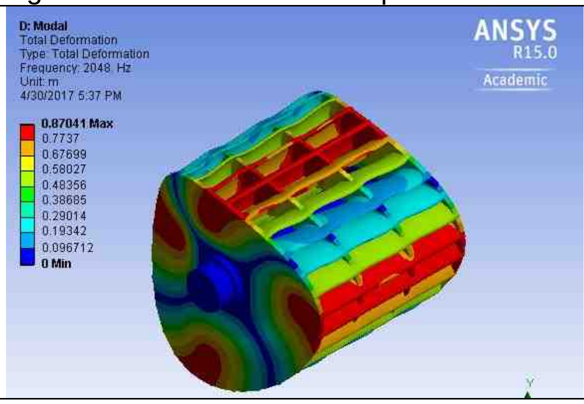

Figure 13: Sixth mode of impeller

5.2 Casing

Stress analysis of casing with inlet nozzle was carried out for the inlet pressure of 5 bar. Photograph of casing used for analysis is as in figure 14 [7]. Analyzed stress and deformation results of casing is as in table 3. Stress and deflection contours of casing is as in figure 15 & 16. Since the stress levels are above the allowable limit of 1e+8 N/m^2, required structural modification is carried out. As part of stiffening, 5x30 mm thick ribs are provided on top and bottom of the nozzle section, where higher stress levels are indicated. Geometry of modified casing is as in figure 17. Stress and deflection contours of modified casing are as in figure 18 & 19 and the maximum values are as in table 4.

Modified casing is further analyzed by modal analysis for its natural frequency and mode shape. First three natural frequencies of the casing is as given in table 5, and its mode shapes are as in figure 20 to 22. In this first mode corresponds to the side plate of casing where impeller is situated. Second mode corresponds to the back side plate of casing and third modes corresponds to inlet side.

	Without stiffening	With additional
Maximum deflection in mm	0.45	0.3
Maximum deflection in	1.36e+8	0.96e+8

Table 4: Maximum stress and deformation

Mode number	Natural frequency in Hz
1	675.9
2	807.3
3	1201.7

Table 5: Casing natural frequency

Figure 14: Casing

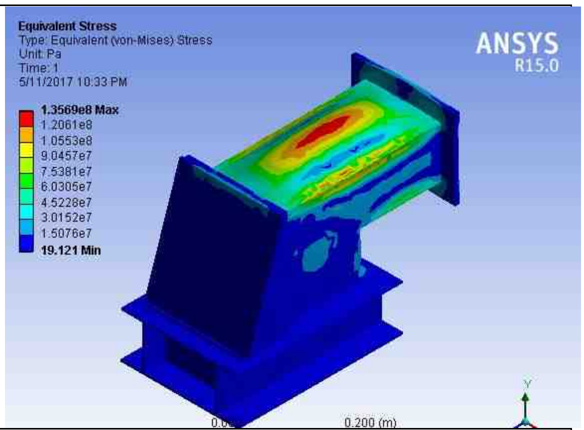

Figure 15. Stress distribution of casing

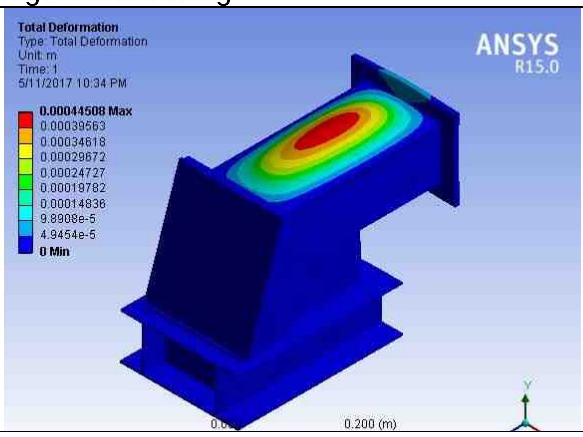

Figure 16. Deflection distribution of casing

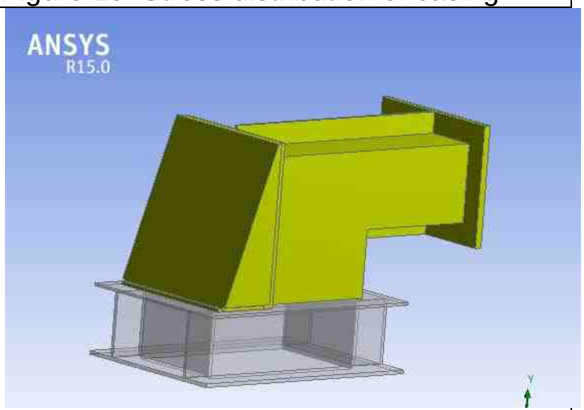

Figure 17: Modified casing

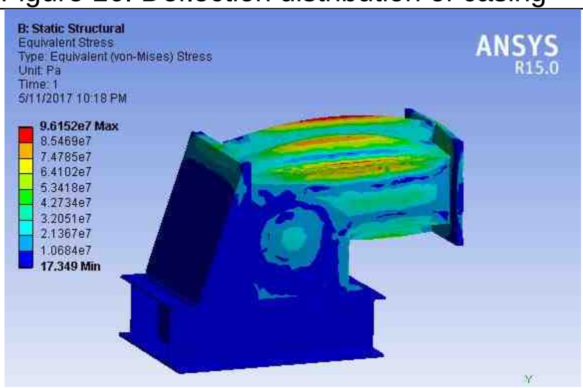

Figure 18. Stress distribution of modified casing

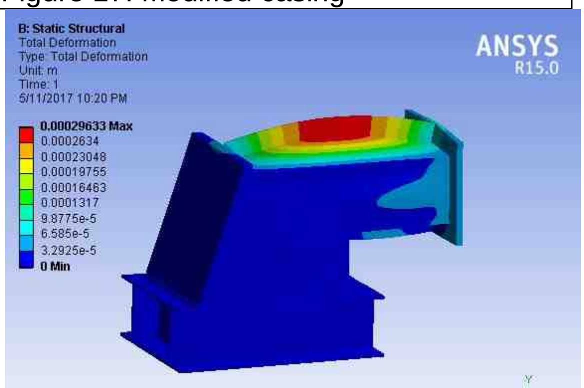

Figure 19. Deflection distribution of modified casing

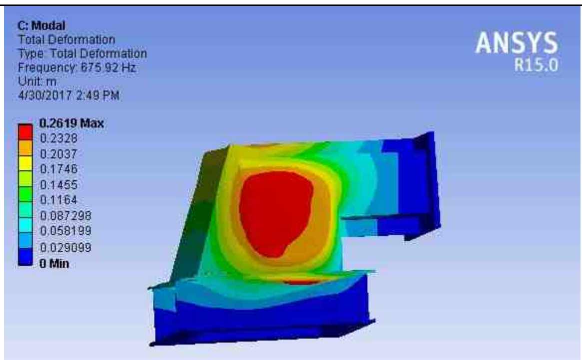

Figure 20: First mode of modified casing

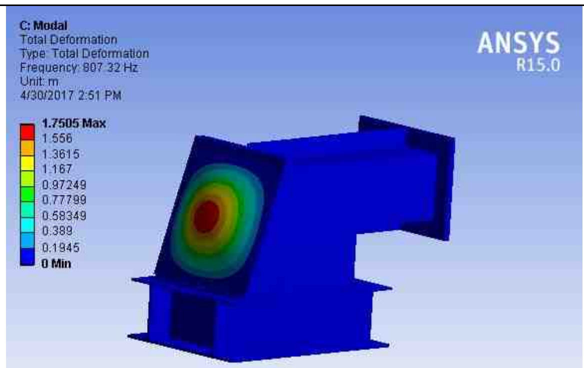

Figure 21: Second mode of modified casing

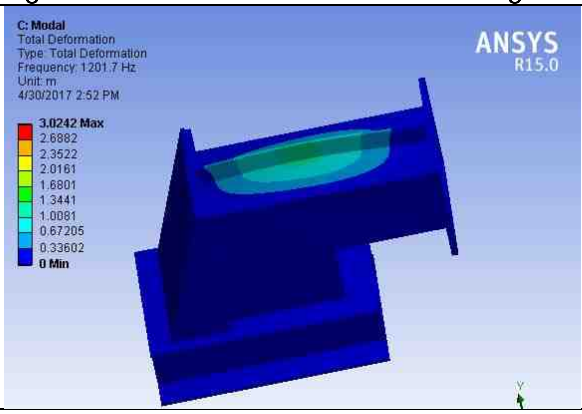

Figure 22: Third mode of modified casing

6 CONCLUTION

The cross-flow turbine is suitable for pico-hydropower production in case of low to high head with variable flow rate. The designed cross-flow turbine impeller and casing was analyzed with static and modal analysis. From the obtained stress analysis results, it is clear that both turbine runner and casing fail to meet the allowable limits. For strengthening propose, suitable strengthening location for both casing and impeller are identified. Identified locations are modified and further analyzed. With proposed modification, stress levels are within allowable limit. It was further analyzed for its modal frequencies and mode shapes. It was identified that the natural frequencies of both impeller and casing are well above the working frequency range of turbine. Hence, further modification is not required. The modified design of cross flow turbine is structurally integral and safe for continuous use.

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