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Design and Analysis of High Efficiency Cross-Flow Turbine for Hydro-Power Plant

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The cross-flow turbine has gained much attention as it is low head turbine and can be used at remote places where a small waterfall of 10-15 meters is located. The objective of this study is to design a cross-flow turbine with maximum efficiency and doing static and model analysis of it. Complete design calculations of turbine have been performed along with static and model analysis of the turbine. The design parameters include runner diameter, runner speed, runner length, turbine power, number of blades blade spacing, radius of blade curvature, attack angle and the blade and exit angles.

Keywords: Cross-flow turbine, design parameters, maximum efficiency, analysis.

I. INTRODUCTION

Hydro-power is considered as one of the most desirable source of electrical energy as it is environmental friendly. In recent years there is increasing research in the field of small hydro-electric power plants. Several types of small hydro-power turbines include radial, axial, and propeller type turbines. There are many sites which are suitable for low-head hydro turbines. Low head hydro turbines include Kaplan Turbine which is axial inlet and axial outlet, Cross-flow Turbine which is radial inlet and outlet of water is across the runner, etc. have been the main field of interest. Nowadays, the Cross-flow hydraulic turbine is gaining popularity in low head and small water flow rate establishments, because of its simple structure and ease of manufacturing in the site of the power plant. Cross-flow turbine is also called Banki-Mitchell turbine as they were the inventors and early developers of this turbine. Cross-flow turbine is low head turbine which works with tremendous flow rate. Thus, it is low specific speed turbo machine. The major parts of cross-flow turbine are: the runner and the nozzle. The runner is a hollow circular section of turbine on which the curved blades are supported at the ends. The nozzle is the component of the turbine through which emerges a high speed jet of water. This water jet strikes the runner blades imparting the potential energy which is converted into the kinetic energy. The components of the cross-flow turbine are shown in the figure (1). The water from the jet strikes the curved blades due to which the runner starts rotating. The water after passing through the blade passes across the runner to the other side to again strike the blades which forces the runner once again. Here again, there will be the energy transfer between the water and the runner which maximizes the efficiency of this type of turbine.

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The efficiency of the cross-flow turbine is a function of several design parameters. These parameters include runner diameter, runner length, runner speed, turbine power, radius of blade curvature, blade spacing, number of blades, the blade angles, exit angles and attack angle. The research is still going on the above mentioned parameters to maximize the efficiency further.

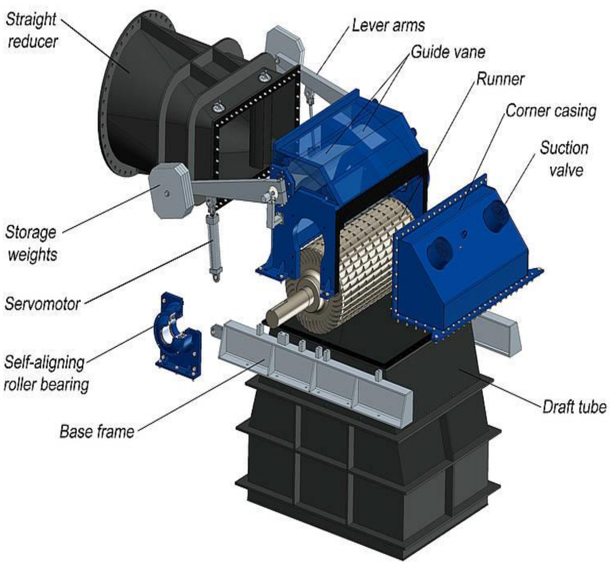

Figure (1) Components of cross-flow turbine

II. DESIGNING PROCEDURE

The designing procedure consists of finding the different parameters associated with the turbine. They are as follows:

a. *Net Head(H_n)*

The net head is the differences between the gross head and the head equivalent of losses. The losses accounts for the losses in pipes, channels, trash rake, penstock, etc.

$H_n = H_g - H_l$ (mm).

H_l is the head equivalent for different losses which takes place in the pipes, channels, etc. This loss is approximately 6% of gross head.[3]

H_g is the gross head at the intake of the turbine. The head for the turbine was selected as 10m.

b. *Water Flow Rate (Q)*

Water flow rate can be calculated by the measurement of velocity of water and the cross-section area of the inlet of turbine.

Q = A*V (m^3/s).

The flow rate for the turbine was selected as Q = 0.315 m^3/s

c. *Turbine Efficiency (η)*

The maximum turbine efficiency can be calculated as [2]:

$\eta = 0.5 * C^2 * (1+\psi) * (\cos\alpha)^2$.

From the given equation we can see that low value for angle of attack (α) is beneficial. It is seen that 16° for attack angle can be manufactured without inconvenience [2].

Ψ = Blade roughness coefficient (0.98).

C = Nozzle roughness coefficient (0.98).

η = 88%

d. *Turbine Power (P)*

It is the output power that is produced by the turbine.

$P = \rho * g * Q * H_n * \eta$ (Watts).

P = 26.945 kW

e. *Turbine Speed (N)*

The turbine speed is the rotational speed at which the runner of the turbine rotates. The turbine speed can be calculated from below mentioned formula [4]

$N = 513.25 * ({H_n}^{0.745} / \sqrt{P})$

N = 550 rpm

f. *Runner Outer Diameter (D_o)*

The outer diameter of the runner is the diameter of the outermost point of the runner from the centre of the shaft.Itsselection depend upon the flow conditions. Larger diameter turbine is selected for more flow and smaller diameter is selected for less flow.

$D_o = 40 * \sqrt{H} / N$ (mm).

$D_o = 230\ mm$

g. *Blade Spacing (t_b)*

The tangential blade spacing can be given as [2]:

$t_b = 0.174 * D_o$ (mm).

$t_b = 40mm$

h. *Radial rim width (w)*

It can be defined as the difference between the outer and the inner radius of the runner of the turbine. It can be given as:

$w = 0.174 * D_o$ (mm).

w = 40 mm

i. *Number of runner blades (n)*

The number of blades in turbine should be optimum because if number of blades is excess than it would cause pulsating power and reduce the turbine efficiency and if it is less, it would result in incomplete utilization of water. The number of runner blades can be determined as [2]:

$n = \pi * D_o / t_b$ n = 18

j. *Runner Length (L)*

It is the length of the runner along the axis of turbine. It can be calculated as [2]:

$L = (210 * Q) / (D_o * \sqrt{H})$ (mm).

L = 346 mm

k. *Runner inner diameter (D_i)*

It is the inner diameter of the runner. It can be calculated as [2]:

$D_i = D_o - 2a$ (mm).

$D_i = 150mm$

l. *Blade radius curvature (r_b)*

It is defined as the radius of the curvature of the curved blades of the turbine. The radiusis very important for the efficient working of the turbine. It is varies directly with the size of turbine. It can be calculated as [2]:

$r_b = 0.163 * D_o$ (mm).

$r_b = 37mm$

m. *Diameter of shaft (Ds)*

The diameter of shaft should be capable of carrying the load of the turbine. The size should be optimum for better flow of water across turbine. It can be calculated as:

$D_S = 0.22 * D_o$

D_S = 50mm

n. *Drawings of the proposed turbine:*

Based on the above calculations, the different parts of the turbine are designed. The detailed drawings of different components of the turbine are:

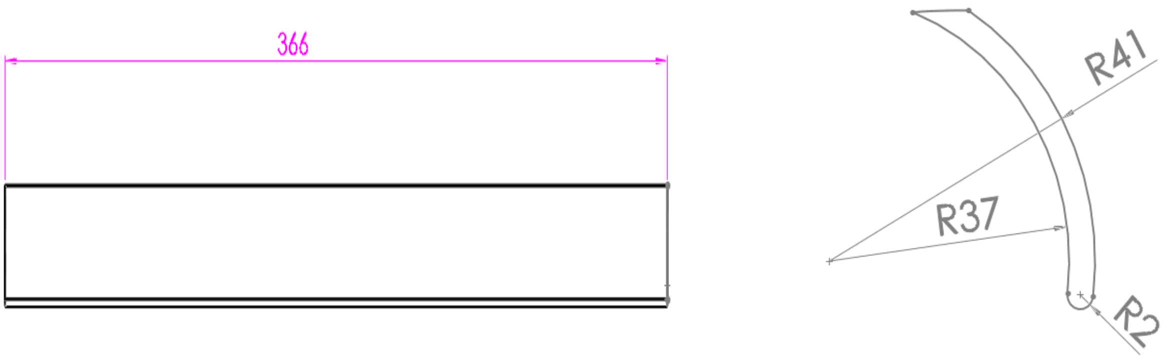

Fig. 2: Turbine Blade

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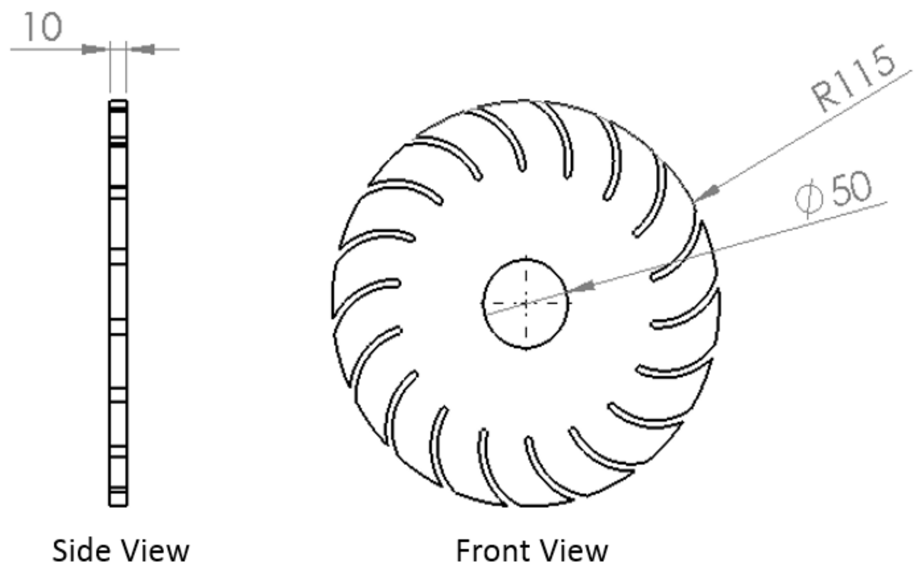

Fig. 3: Side disks

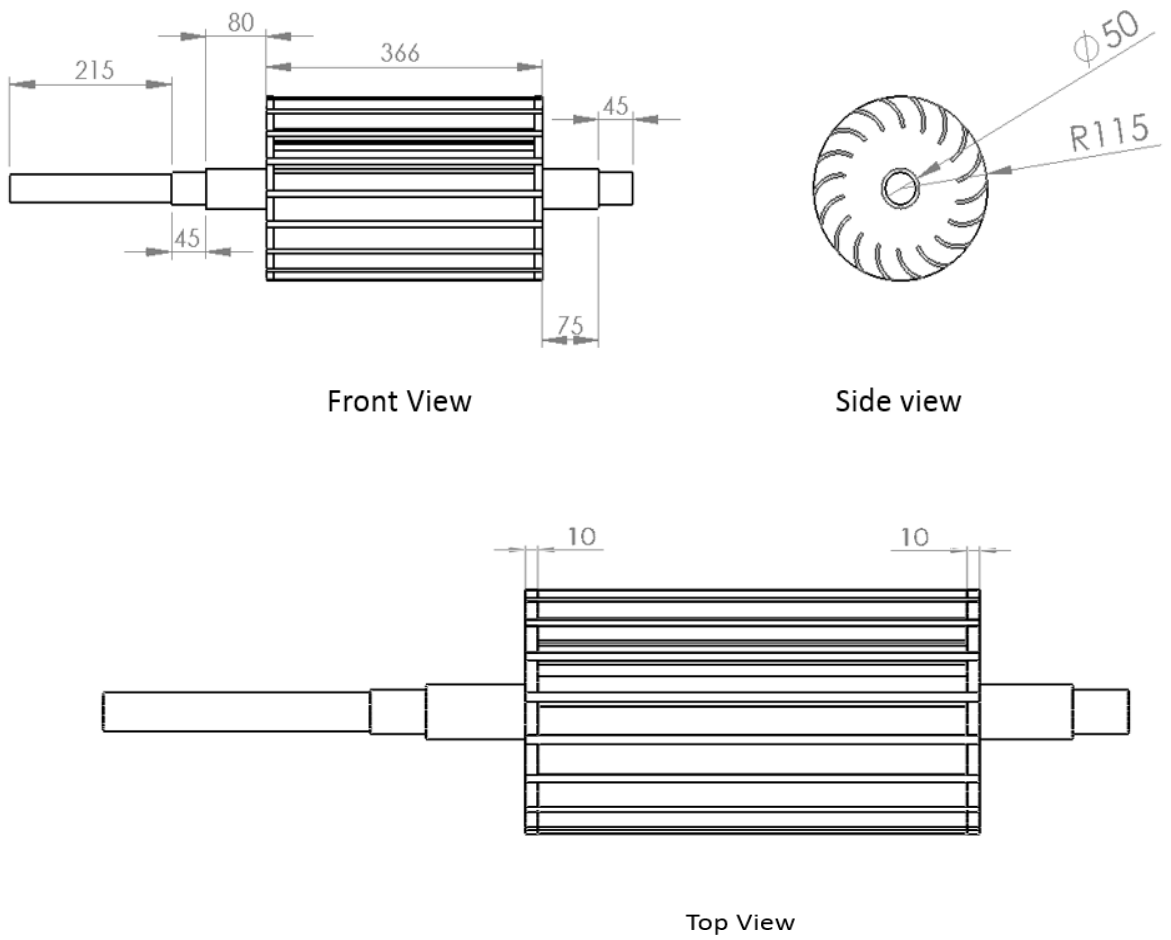

Fig. 4: Turbine rotor assembly

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III. RESULT ANALYSIS

Material properties of steel are selected with young's Modulus, E = 210 Gpa and Poisson's ratio, v = 0.3. The following loads were applied on the turbine.

Gravity: The turbine is analysed under its self-weight by applying gravitational acceleration of 9.81 m/sec^2 in ANSYS16.

Hydraulic load: The turbine blades are analysed under a head of 10 m and discharge of 315 L/sec. Equivalent pressure of hydraulic load on blades is calculated and applied on 6 blades in ANSYS16. Static analysis was performed at 300 rpm. Constraints are applied to both ends of the shaft and initially simple static Analysis is performed. The ANSYS results are discussed below.

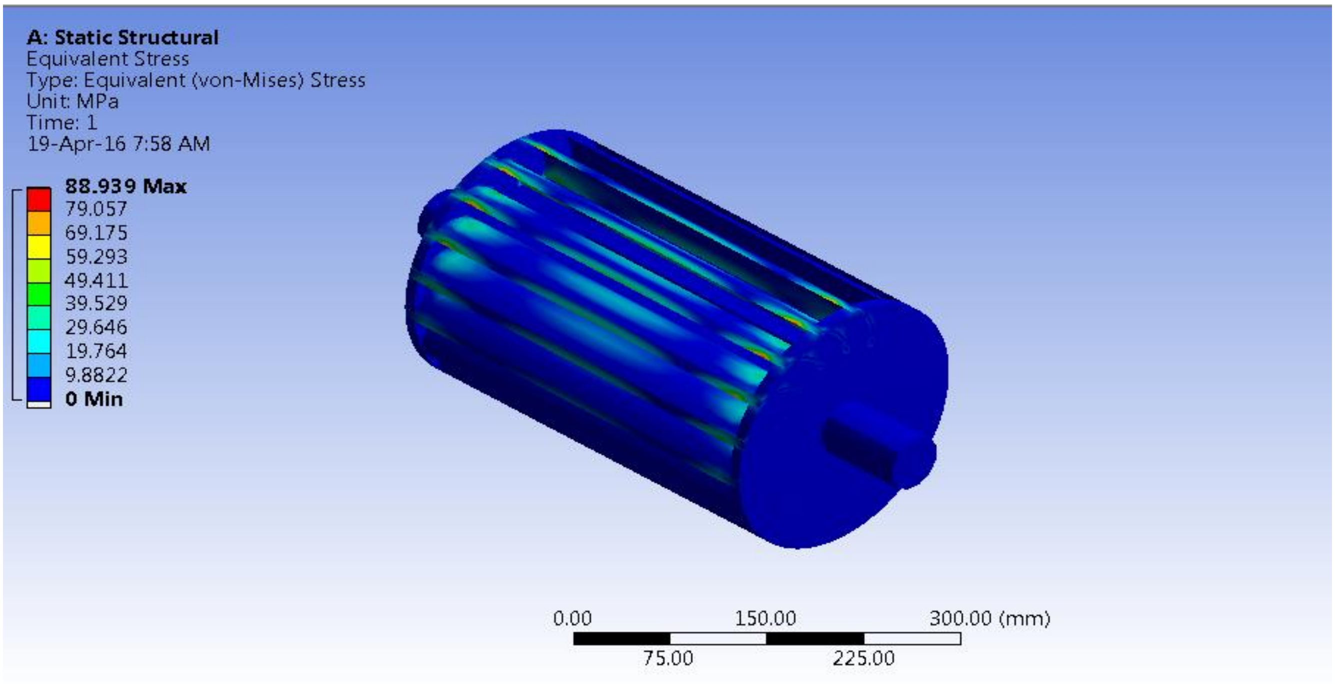

Fig.5 3-D Von-misses stresses on cross-flow turbine.

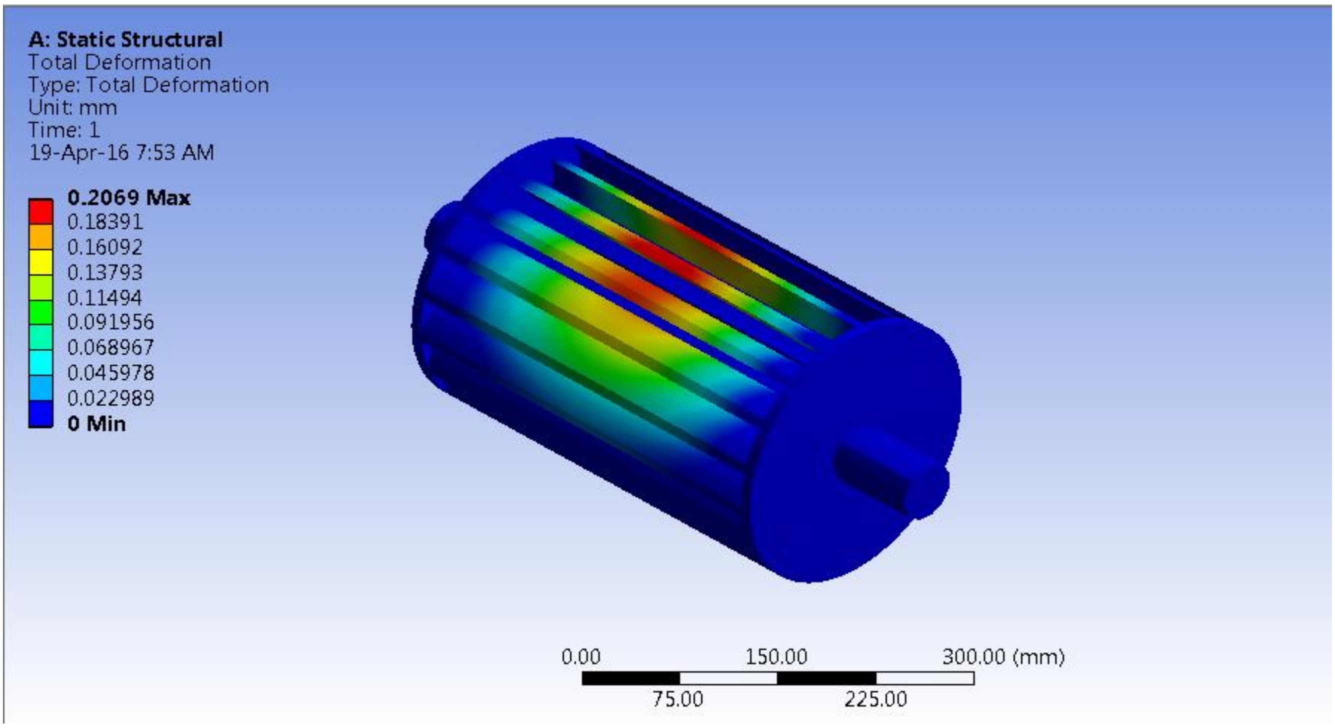

Fig. 6 Maximum displacement on cross-flow turbine.

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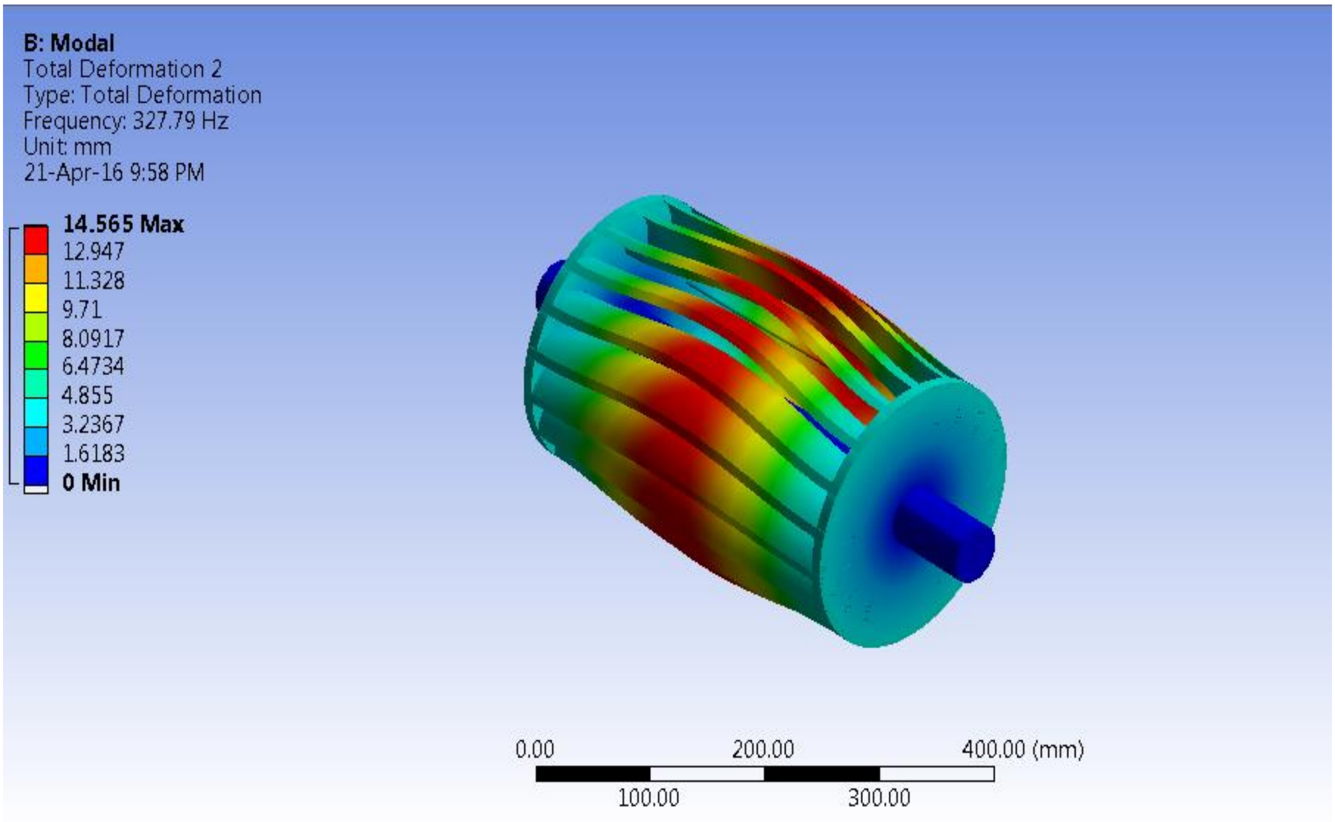

Fig. 7 Maximum stress and deformation on global mode shape 1

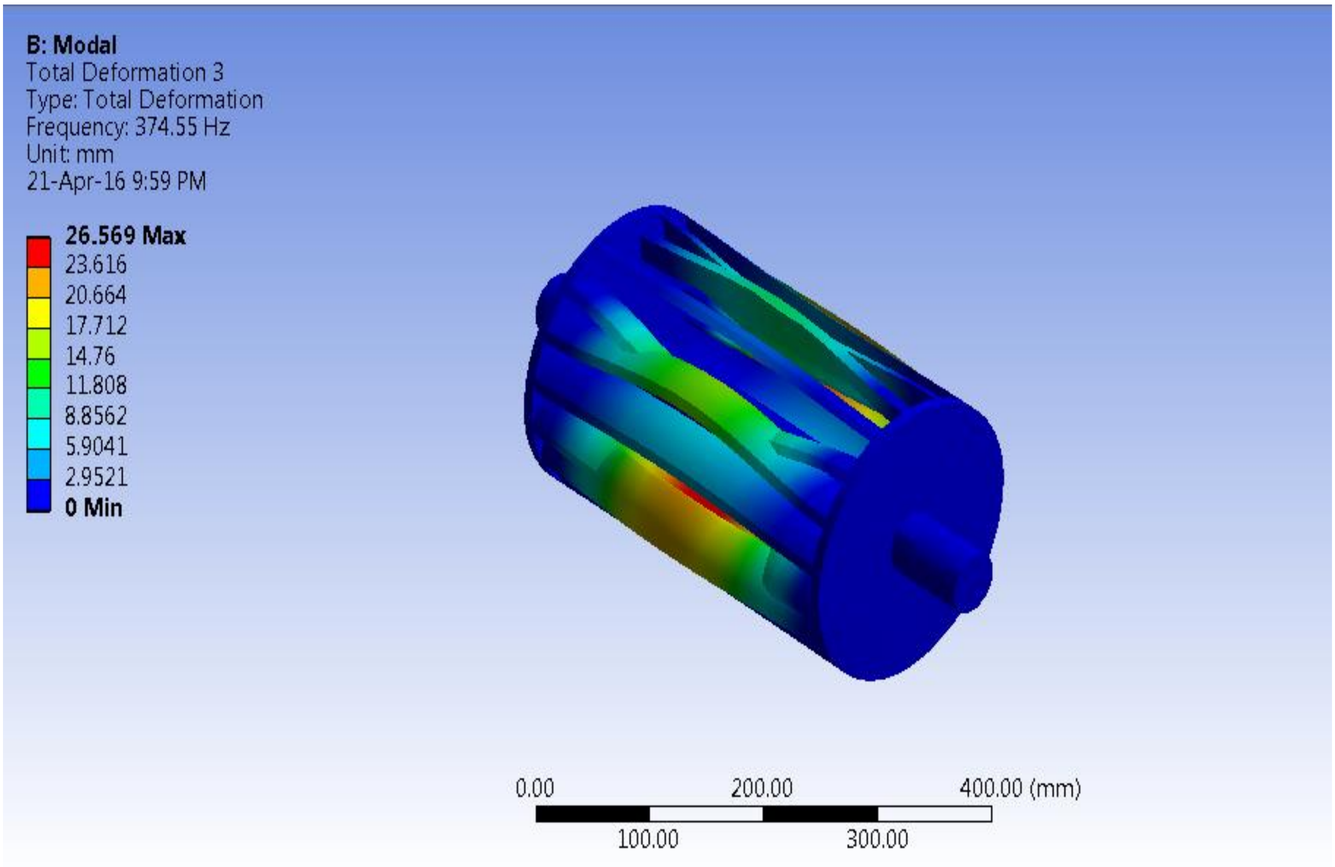

Fig.8 Maximum stress and deformation on global mode shape 2

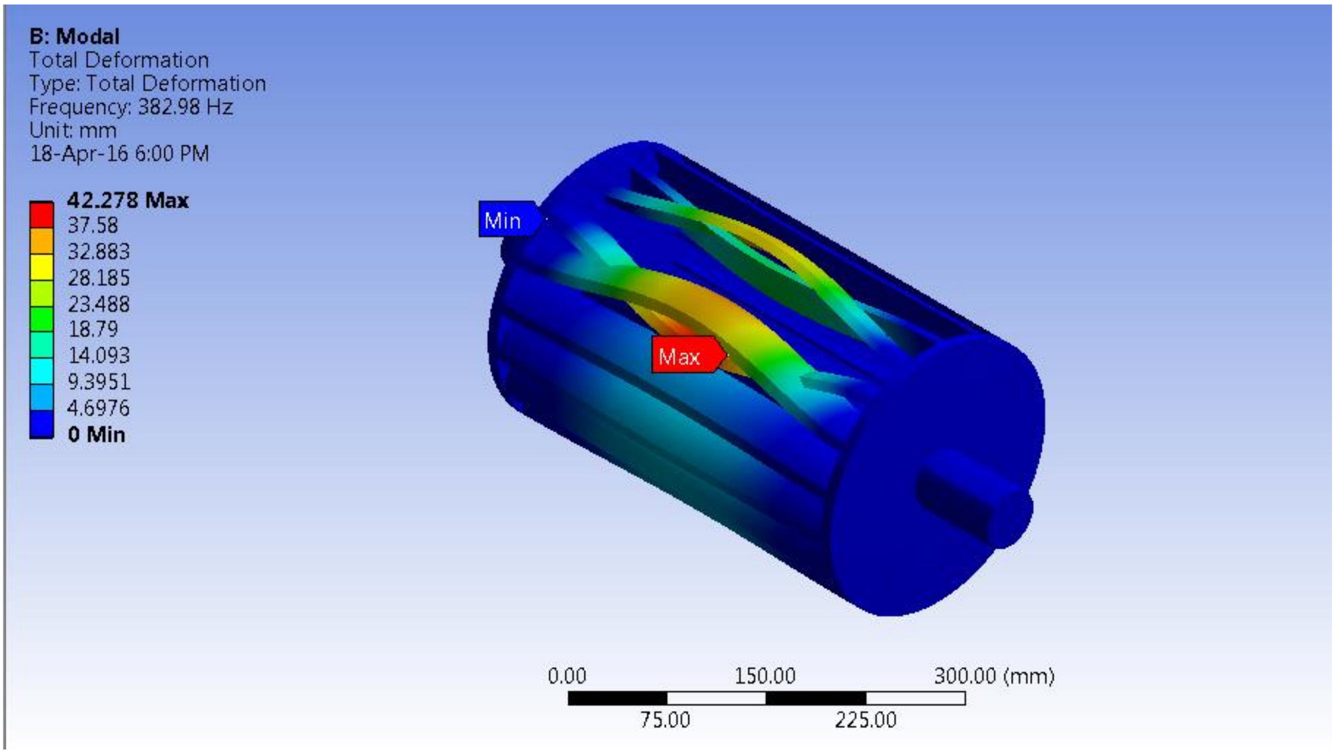

Fig.9 Maximum stress and deformation on global mode shape 3.

***3-D von-misses stresses on cross-flow turbine*:**
Considering the whole turbine, the stresses developed on the structure are 88 Mpa. Yield point of steel is 250 Mpa, if we consider the factor of safety as 2, then also the stress safe limit is 125 Mpa. The maximum possible stress on the turbine is 88 Mpa which is much less than 125 Mpa, so the overall structure of turbine is safe. Thereby proving that the design of the turbine is safe for the Hydraulic load. The maximum stress are shown in Fig. 5

***Maximum displacement on turbine*:** The maximum deformation noted is 0.206mm at the centre of the blade which is very less to cause any distortion or to cause any problem during operation of the turbine. The maximum displacement is shown inFig. 6

***Mode shape 1*:** For global mode shape 1, the natural frequency measured by ANSYS is 327.79 Hz.The working frequency of our turbine is 5 Hz, the difference between both the frequencies is very huge. As a result of which, the tendency of the turbine to reach at a frequency where resonance occurs is very less. The distortion of the blades as shown in Fig. 7 will occur at 1960 RPM which is very high as compared to our operating RPM. The Maximum stress and deformation produced at 327.79 Hz are showed in Fig. 7.

***Mode shape 2*:** For global mode shape 2, the natural frequency measured by ANSYS is 374.55 Hz.The working frequency of our turbine is 5 Hz, the difference between both the frequencies is very huge. As a result of which, the tendency of the turbine to reach at a frequency where resonance occurs is very less. The distortion of the blades as shown in Fig. 8 will occur at 2200 RPM which is very high as compared to our operating RPM. The Maximum stress and deformation produced at 374.55 Hz are showed in Fig. 8.

***Mode shape 3*:** For global mode shape 3, the natural frequency measured by ANSYS is 382.98 Hz.The working frequency of our turbine is 5 Hz, the difference between both the frequencies is very huge. As a result of which, the tendency of the turbine to reach at a frequency where resonance occurs is very less. The distortion of the blades as shown in Fig. 9 will occur at 2300 RPM which is very high as compared to our operating RPM. The Maximum stress and deformation produced at 382.98 Hz are showed in Fig. 9.

IV. DISCUSSION

The cross flow turbine design is very unique from conventional turbine as it is designed in such a way to give maximum efficiency as stated in theory. It is dependent on several factors such as runner diameter, runner speed, runner length, turbine power, number of blades blade spacing,, radius of blade curvature, attack angle and the blade and exit angles. This parameters play very important role in attaining the desired efficiency.

The Angle of attack is kept 16°. The dimension for outer and inner diameter plays an important role. If the ratio of outer diameter to inner diameter is increased then, there will be more curvature of the blade and energy will be transferred in shorter distance. The optimum diameter ratio is 0.68 for design of high efficiency turbine. The inner discharge angle is recommended to be kept as 90°. As it will also help the water to move in radial direction. After analysis, the result reflects the number of blades to be kept as 18.

The design was done considering the head as 10m and discharge as 315L/sec. The design for cross-flow turbine was analysed in ANSYS software after being modelled in SOLIDWORKS.

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Material properties of steel are selected with Poisson's ratio = 0.3, young's modulus = 210 Gpa.
The stresses and displacement revealed after analysis were nominal as compared to material property of steel confirming the design as safe. The modal analysis which resulted the natural frequency as 327.79 Hz in mode shape 1, 374.55 Hz in mode shape 2, 382.98 Hz in mode shape 3 shows that the operating RPM is under safe range and shows no sign of resonance and thus the design is safe. Static analysis reveals that the design of turbine has a factor of safety 2.84. This turbine design with such high efficiency is very useful for the location where electricity is not supplied in adequate amount and it can be used as source of energy as it will generate nearly 25kW power.

V. CONCLUSION

Thus, we can see from the observations that Cross-flow turbine is suitable for very low head and a large flow. The maximum efficiency was found to be 88% for different values of head and water flow rate. The different design parameters such as water jet thickness, runner diameter, runner length, blade spacing, and radius of blade curvature, turbine speed, turbine power and number of blades were determined at maximum turbine efficiency.

The stress analysis was done for the deformation of blades. And also that maximum 18 runner blades are required for smooth running of the turbine. The maximum efficiency was calculated to be around 88% which is very good for a low head and small scale plant.

In future,the scope of small scale plant is high as there is very much need of energy at mobile places like hilly areas or some rural villages where there is power cut or maybe no electricity in todays developing age.

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