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Research Article

Design and Analysis of Cross Flow Turbine for Micro Hydro Power Application using Sewerage Water

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Abstract: The objective of this study endeavor is to describe the design of an environment friendly captive micro hydel power plant at the sewerage treatment plant outlet fall at Lai Nallah located in I-9 Islamabad. It will use sewerage water of CDA as discharge water to run the turbine of the micro hydel power plant to generate electricity. The important phases of the project are to carry out survey for collection of data about hydrology, quantity of flow of water, fall head, geology and design of an efficient turbine. Complete design calculations of turbine have been performed along with static and model analysis of the turbine. Key parameters to increase efficiency of cross-flow turbine are discussed.

Keywords: Analysis, CDA, cross-flow turbine, efficiency

INTRODUCTION

One of the most efficient and well established renewable electrical power generations is Hydro power generation. Hydro power is being produced for almost ten decades at competitive prices. In nearly 30 countries, it serves as prime source of electricity. Almost 25% of total electricity of the world comes from hydel power (Bhutta, 2008). Depletion of fossil fuels, short term policies and political inefficiencies lead to current worst energy crisis of Pakistan. Short fall is soaring to the height of 6000 MW. Industries and general public are effected the most. Paying heed to the resources, Pakistan has great potential to generate electric power from hydel power. Currently, only 20% of this potential is being used. Moreover, hydro power is 100% environment friendly and cheap (Saket and Anand Kumar, 2006). It is very rationale to build such hydro power projects to meet ever increasing demand of electricity. Significant amount of power can be generated on run of river in different locations with different capacity. Proportion of these micro hydro power projects is very remarkable and cannot be ignored. Cross flow hydraulic turbine are best choice of such micro hydro power projects in Pakistan due to their efficiency, cheap manufacturing and low maintenance cost (Jiandong *et al*., 1995; Bhatti *et al*., 2004).

MATERIALS AND METHODS

Site selection and basic lay out: The powerhouse layout depends upon the head and quantity of water and type of water supply forebay, foundation conditions, location of spillway, turbine size to locate adjacent to each powerhouse, width of channel condition below and above the fall, location of control panels, synchronization system and accessibility and overall project development (Bhutta, 2008). The layout of proposed hydropower plants envisages positioning of the various components of the plant to ensure optimum use of the available head, water and space for efficient and convenient erection, operation and maintenance as being designed. Keeping in view all the above mentioned parameters it was decided to install the micro hydel power plant at CDA sewerage treatment plant near Nalla Lai named as Phase IV by CDA (Fig. 1).

Fig. 1: Proposed site for installation of MHP

Evaluation of water resources and generating potentials: Available quantity of water = 4 MG/D or 0.175 m^3/sec.
Using formula:

$$P = Q * g * h * \eta * \rho \quad (1)$$

where,
P = Power in Kw

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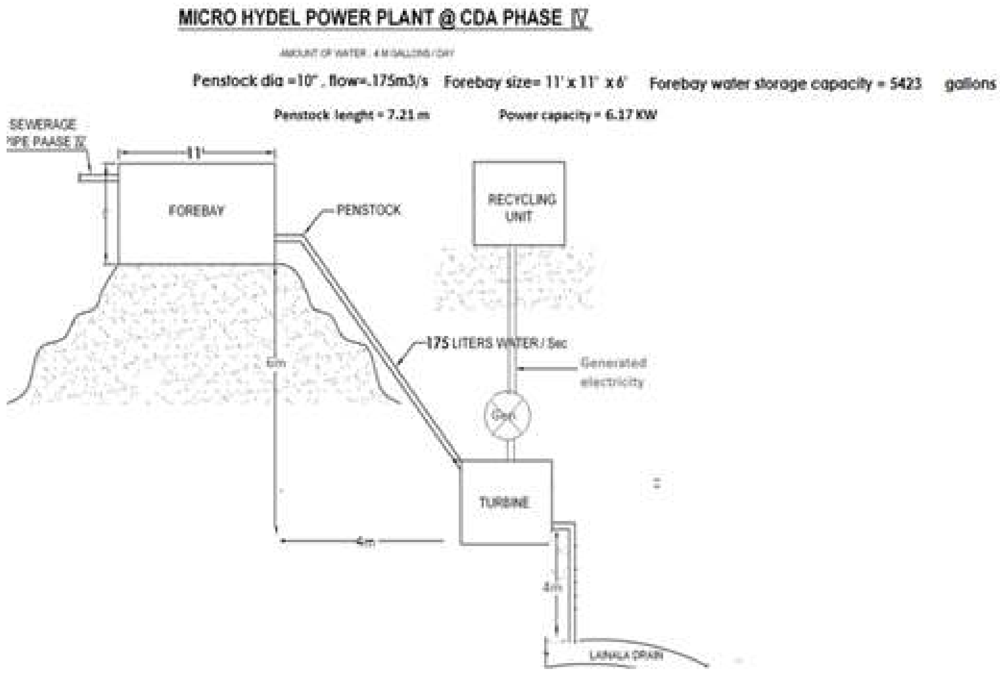

Fig. 2: Schematic diagram of the MHP

Q = Discharge in m^3/sec
H = Head in m
g = Acceleration due to gravity in m/sec^2
η = Efficiency coefficient

Putting H = 6 m, Q = 0.175 m^3/sec, g = 9.8 m/sec^2, η = 0.6.

The total generation potential at phase IV of site is calculated as 6.17 KW. Keeping in view the existing condition of topography infrastructure it is recommended to design and install the turbines on forebays of fall sewerage treatment water outlet. It is assumed that water will regularly flowing about 0.175 m^3/sec (4 MG/D).

Design of forebay: In order to make the operation of turbine smooth a forebay is designed to store water for 2 min. The size of forebay was calculated using formula:

$$L \times W \times D = \text{Cubic Feet} \tag{2}$$

$$\text{Cubic ft} \times 7.47 = \text{Gallons (US)}$$

Length and width of the designed forebay will be 11 ft and depth will be 6 ft (Fig. 2).

Designed turbine calculations: A Cross-flow Turbine was designed for given available conditions of 6 m head and discharge 175 L/sec using basic turbine design formulae (Chattha *et al*., 2010).

Runner diameter: Diameter of runner is selected depending upon the flow conditions. If there is larger flow through the turbine select a large diameter of turbine and for the low water flow conditions select a smaller diameter of turbine. Runner diameter was selected as 350 mm for safe mode of operation.

Blade angle: Its value is taken as 73° for the Cross flow turbine.

Radius of blade: The curvature of blade accounts a lot for the efficient working of the turbine. It is varies directly with the size of turbine. The following relation is used for the blade radius:

$$r = 0.326\,R \tag{3}$$

where,
r = The curvature of blade
R = The radius or turbine runner Putting the value of R in above relation:

$$r = 57 \text{ mm}$$

Thickness of blade: The blade thickness should be kept at an optimum value, to bear the stresses induced by the water. Blade thickness should not be too low to break down under the given load conditions. Alternatively too great thinness of blade may cause interruption to the water admission. The following relation holds true for the blade thickness:

$$t = K_1 D \tag{4}$$

where, t is the blade thickness, K_1 is the constant of proportionality and D is the outer Diameter of Turbine Runner. The Value of K1 lies between "0.0177" to "0.0185". Take the value of $k_1 = 0.0177$:

$$t = 6.1 \text{ mm}$$

Number of blades: The selection of optimum number of blades is very important in the design of Turbine Runner, fewer no of blades may cause incomplete utilization of water available to the turbine and excessive number of blades may cause the pulsating power and reducing the turbine efficiency. The following relation exists for the number of Blades in a Turbine runner:

$$n = k_2(\pi * D) \tag{5}$$
$$n = 36$$

Pitch circle diameter: It is the circle from which the profile of the blade radius is drawn the side plates of the runner. It has the following relation:

$$D_p = 0.7532D \tag{6}$$
$$D_P = 263 \text{ mm}$$

Blade spacing: Proper blade spacing allows the water to strike, on the blades for maximum thrust production, the blade spacing depends upon the number of blades used in the turbine runner. Blade spacing can be calculated as:

$$S = [\pi * D - n(t)]/n \tag{7}$$
$$S = 30.53 \text{ mm}$$

Diameter of shaft: The diameter of shaft should have a value to bear the load on the turbine. It should not be too larger that water strike the shaft after passing through the first set of blades at the inlet:

$$d = 0.22D \tag{8}$$
$$d = 77 \text{ mm}$$

Runner width: Runner width is calculated using Eq. (9):

$$B_t = \frac{Q_t}{D_t \cdot \sqrt{H_t} \cdot q_{11}} \tag{9}$$
$$B_t = 25 \text{ mm}$$

Turbine speed: Equation (10) has been used for calculation of turbine speed:

$$n_t = \frac{\sqrt{H_t}}{D_t} . n_{11} \tag{10}$$
$$n_t = 266 \text{ rpm}$$

Runway speed: Using Eq. (11):

$$n_{tr} = 1.8 * n_t \tag{11}$$
$$n_{tr} = 478 \text{ rpm}$$

Drawings of the proposed turbine: Based on the above calculations various parts of the turbine have been designed the detail of which are given in the following drawings.

Runner of the turbine: The Runner of the turbine consists of Side Disks, Blades and Shaft (Fig. 3 to 6).

The overall turbine Rotor assembly is shown in the Fig. 7.

Fig. 3: Turbine blade

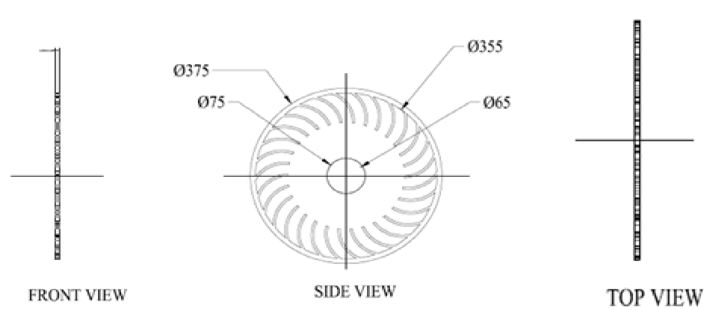

Fig. 4: Side disks

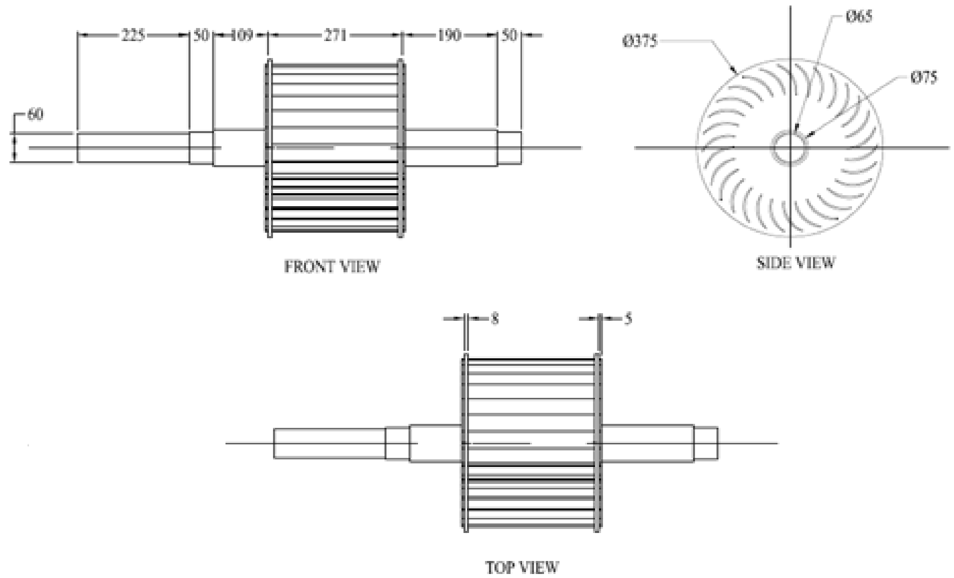

Fig. 5: Turbine rotor assembly

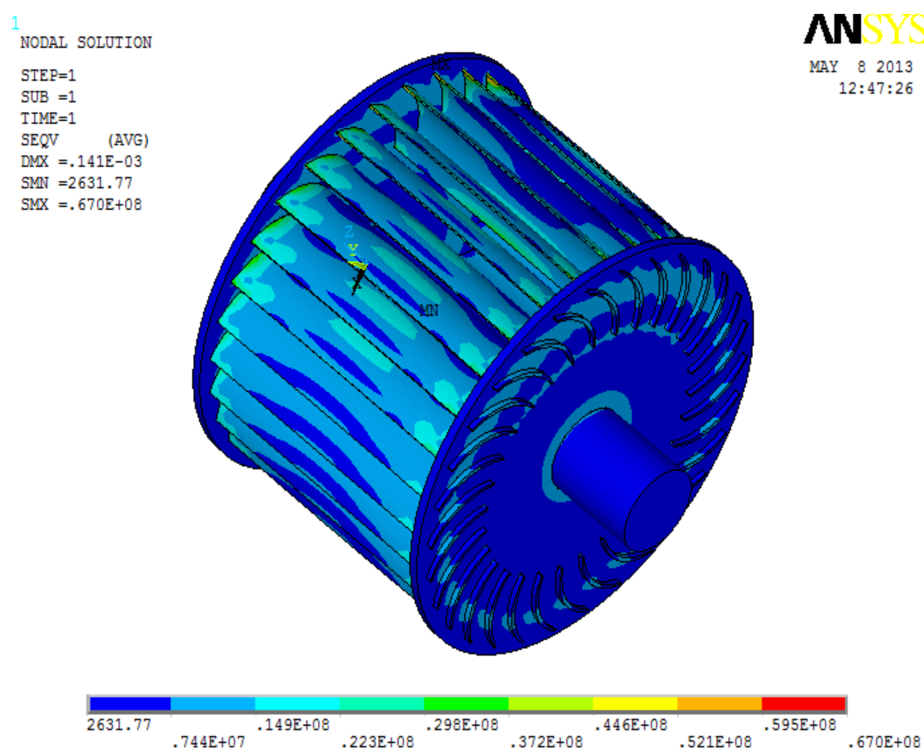

Fig. 6: 3-D von-misses stresses on overall hydraulic turbine

RESULT ANALYSIS

The designed hydraulic turbine was analyzed using ANSYS13. The following two types of analysis were performed.

Static analysis: Pro-E model of the turbine was imported to ANASYS13 in. IGES format. Element type SOLID186 is selected and model is meshed. 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 analyzed under its self weight by applying gravitational acceleration of 9.81 m/sec^2 in ANSYS13.

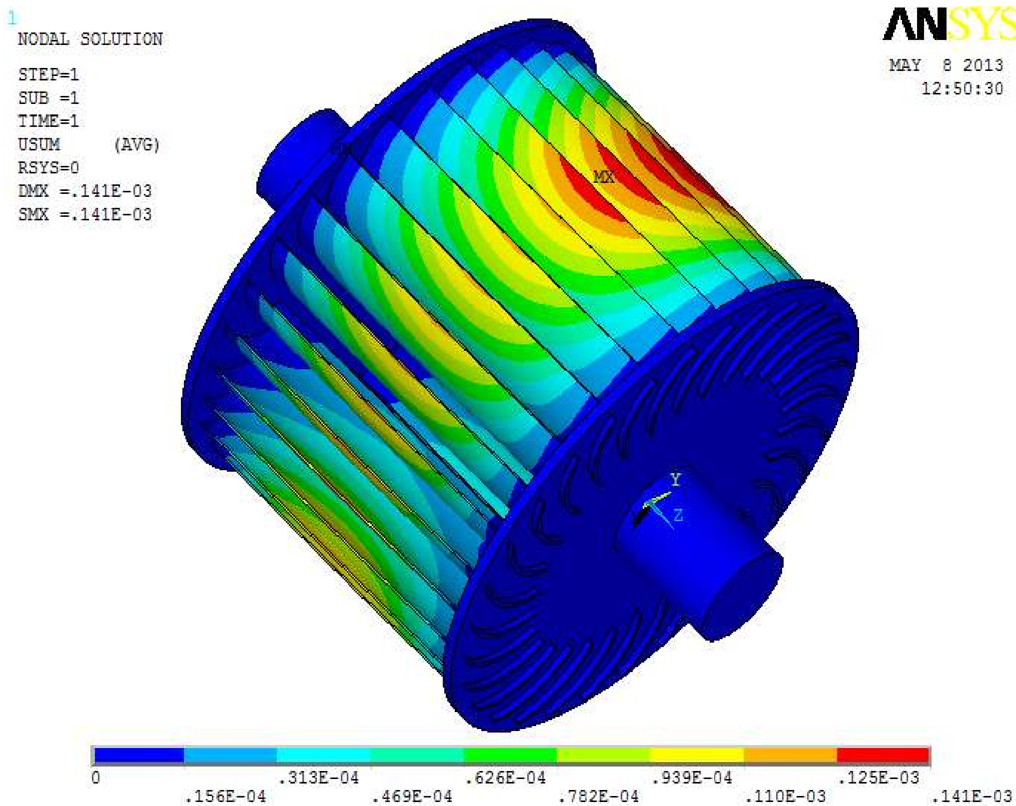

Fig. 7: Displacements on overall turbine

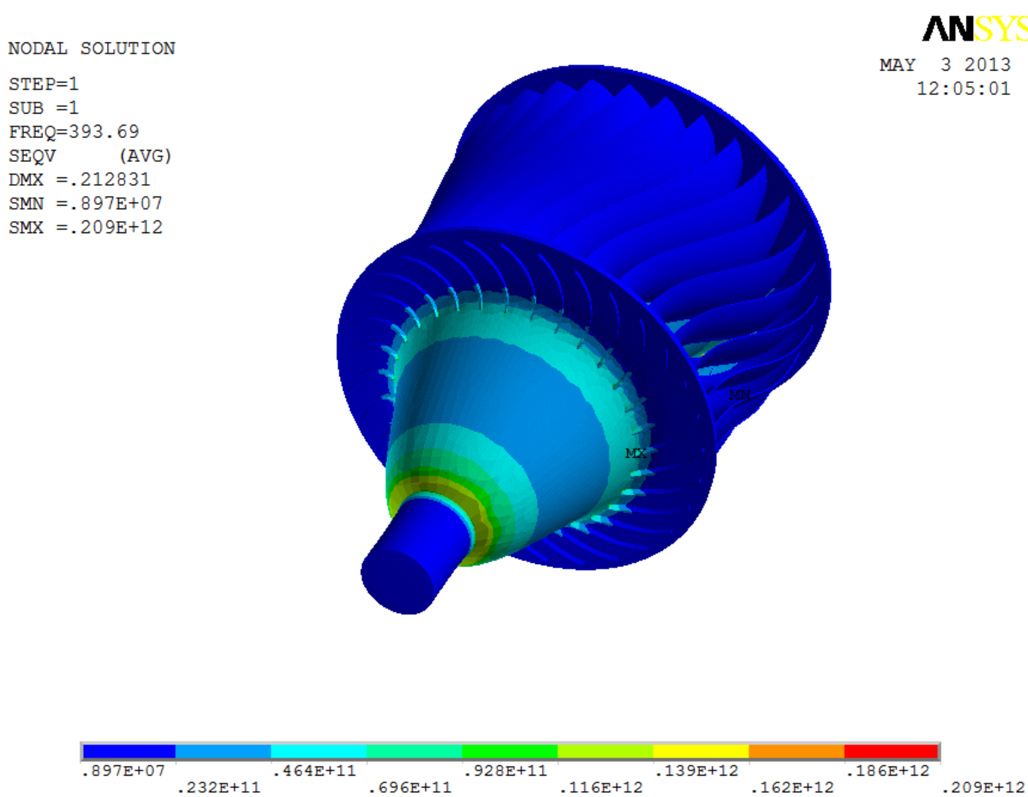

Fig. 8: Max stresses on global mode shape 1

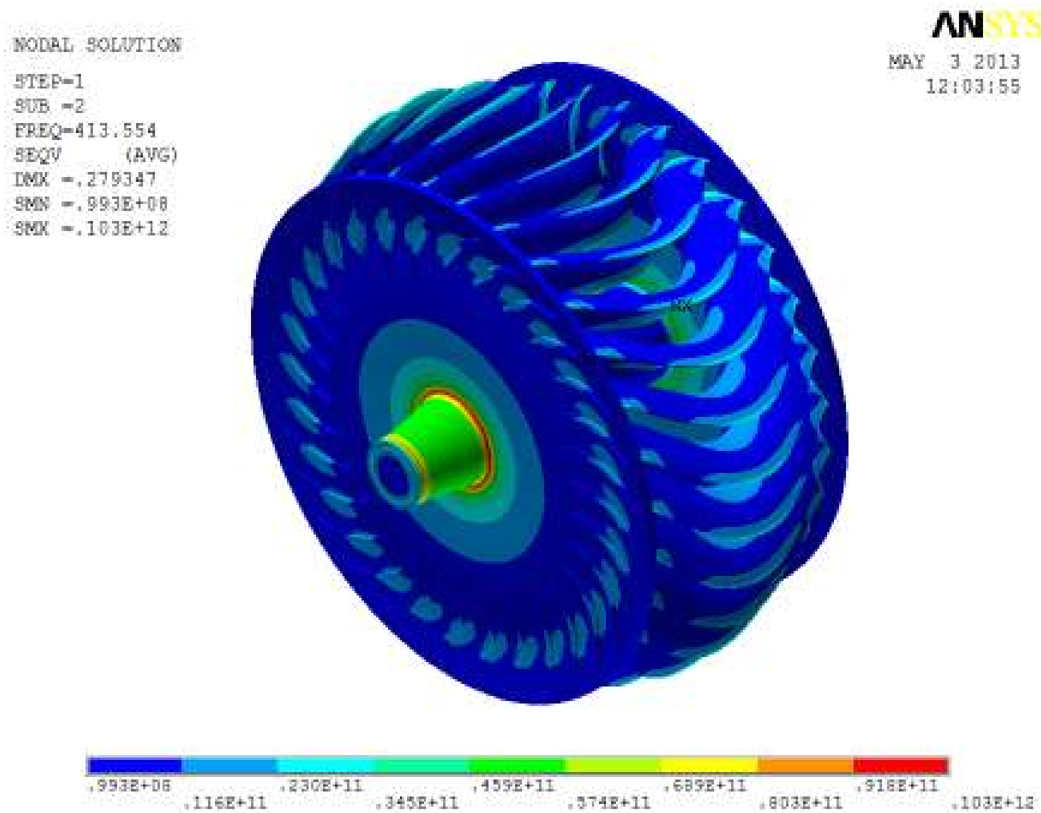

Fig. 9: Maximum stresses at global mode shape 2

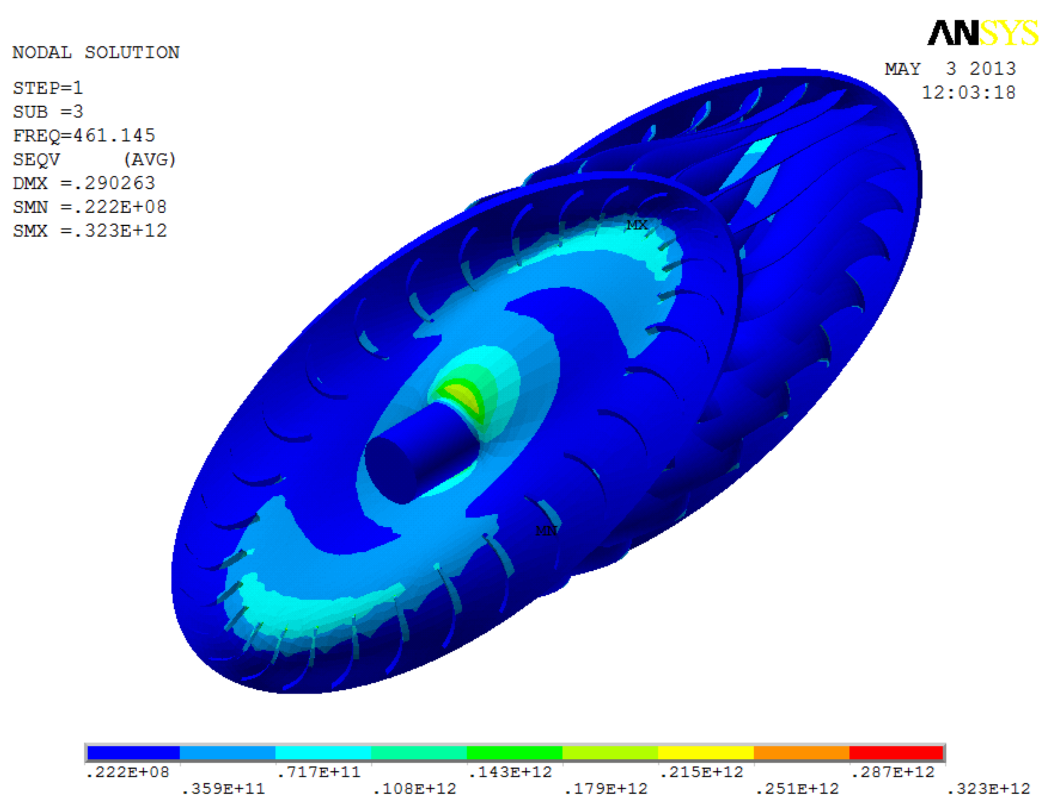

Fig. 10: Maximum stresses at global mode shape 3

Hydraulic load: The turbine blades are analyzed under a head of 6 m and discharge of 175 L/sec. Equivalent pressure of hydraulic load on blades is calculated and applied on 10 blades in ANSYS13. Static analysis was performed at 1500 rpm as the designed turbine can be used for various speed up to 1500 rpm.

Constraints are applied to both ends of the shaft and initially simple static Analysis is performed.
The ANSYS results are discussed below.

3-D von-misses stresses on overall hydraulic turbine: Considering the whole structure, the stresses on the overall structure are 67 Mpa. Yield point of steel is 245 Mpa and using a material factor of safety of 1.5 we have a stress safe limit of 163 Mpa. As maximum stress on overall structure is 67 Mpa which is much lower than 163 Mpa so overall structure is safe with the factor of safety 2.43.

Maximum displacement on overall turbine: Maximum displacement on overall turbine is 0.141 mm on the centre of the blade which are much lower to cause any distortion in the given structure.

Modal analysis: The designed turbine operates at 266 RPM. So the corresponding working frequency is 4.43 Hz. By performing modal analysis, following mode Global mode shapes are obtained.

Mode shape 1: For global mode shape 1, the natural frequency is 393.69 Hz while our working frequency is 4.43 Hz (or 25 Hz in case of 1500 rpm). As both the frequencies do not match so the structure has no tendency of resonance. The Maximum stresses in the structure if the structure exhibit this model shape are shown in Fig. 8.

Mode shape 2: For global mode shape 2, the natural frequency is 413.55 Hz while our working frequency is 4.43 Hz. As both the frequencies do not match so the structure has no tendency of resonance. The maximum stresses in the structure if the structure exhibit this model shape are shown in Fig. 9.

Mode shape 3: For global mode shape 3, the natural frequency is 461.145 Hz while our working frequency is 4.43 Hz. As both the frequencies do not match so the structure has no tendency of resonance. The maximum stresses in the structure if the structure exhibit this model shape are shown in Fig. 10.

As in all Global Mode shape, the structure is safe so our designed turbine is safe.

DISCUSSION

Increase in efficiency of the designed turbine was obtained by selection of optimum values for certain key parameters which makes the design different form conventional turbines. These include reduction of Angle of attack to 16°. As increase in the diameter ratio gives stronger curvature of the runner blades with shorter distance for energy to be transfer. Mathematical analysis revealed that keeping inner to outer diameter ratio 0.68 gives high efficiency. The inlet discharge angle was kept 90° instead of 120° as experimental studies shows that for a given flow rate smaller inlet discharge attack gives higher inlet velocities (Fiuzat and Akerkar, 1989). Increasing the number of blades provides more efficiency as it provides regular velocity profile inside the space between each couple of blades. However number of blades in the proposed design are kept to 36 as too much blades gives rise to higher energy dissipation. This results in higher efficiency with low energy dissipation. Also the value of β_2 was kept 90° to provide radial direction to the outlet velocity inside the runner. The value of W/B was selected to 1 which results in higher efficiency (Sammartano *et al*., 2013).

The designed hydraulic turbine was analyzed using ANSYS13 software. Material properties of steel are selected with young's Modulus, E = 210 Gpa and Poisson's ratio, ν = 0.3. The turbine blades are analyzed under a head of 6m and discharge of 175 L/sec and its self weight by applying gravitational acceleration of 9.81 m/sec^2. Static analysis reveals that max stress on blade, runner and overall assembly are 60, 6.5 and 67 Mpa, respectively and max displacement on blades runner and overall assembly are 0.141, 0.0027 and 0.141 mm, respectively. Static analysis reveals that designed turbine have a safety factor of 2.43. Model analysis of the designed turbine was performed having natural frequency respectively for model shape 1 model shape 2 and model shape 3, respectively showing no tendency of resonance and the designed model is safe.

CONCLUSION

To meet the demands of electricity in hilly rural areas, number of off grid micro hydel power plants is installed but it requires improved layout for power house and optimum design of turbines to extort considerable efficiency form them, especially for urban areas. A layout for power house at Nalla Lai and efficient design of a cross flow turbine is presented in this study and simulation results show that proposed design is safe and effectively efficient. Further work is needed to find exact value of W/B ratio and equation of determining optimal no of blades for the design of efficient cross flow turbine.

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