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Modelling and Analysis of a very Low Head Kaplan Turbine Runner Blades for Rural Area of Punjab

Muhammad Abubakar, Saeed Badshah, Talal Ahmad, Noor Rahman

Abstract— The world have a huge potential of small hydro kaplan turbine power plants. The world energy demands are increasing. In this senario the micro hydro Kaplan power plants gains special attention. The development of micro hydro kaplan power plants on large scale will generate enough energy for the world inhibitants.This paper presents the modeling and Analysis of the runner of a low head Kaplan for a specific site at RD 216+852 Jhang branch Canal in Punjab, Pakistan where the net head and rated flow of water is 1.16m and 7.07m^3/s respectively. The turbine is Single regulated Kaplan turbine with runner diameter 1.25m, hub diameter is 0.36m, runaway speed is 13s^{-1} and net power capacity is 60 KW. The Kaplan turbine runner was modelled in 3-D model of runner of Kaplan turbine in Pro-E engineer software and after calculating the blade operating conditions from the hydrodynamics properties of the water flow at the jhang branch canal in Punjab we performed Analysis on runner blade in ANSYS 14 software.

Index Terms— Kaplan Turbine; Low head; Jhang branch canal; runner; stress; blade; net head

1. Introduction

Kaplan turbine's high efficiency under low pressure and head accounts for its growing popularity today because many installations have high flows but very low head. [1]. Micro hydro power is one of the major sources of renewable energy and also it is major contributor of electricity for all over the world. However, during last two decades Government of different under developing countries are renewed their interest in the development of micro hydro power projects due to its long lasting benefits specially in environment handling and ability to produce power in rural and hilly areas where access of government is impossible [2]. Small hydro power plants are very suitable in less populated areas where they can serve to many scattered communities as well. Pakistan is very rich in these sites where we can easily install thousands of small hydro power plants, the Himalayans and Hindukash range of mountains, major canals of Punjab, Sindh are the considerable potential sites which could be utilized by micro hydro power.[3]

Turbines are the most important component of any hydro power plant. They cover about 15-35 percent of the total project cost[4].

The low head Kaplan Turbine at Jhang branch Canal in Punjab, Pakistan consists of four blades operating at low head. The flow rate at this specific site is 7.07 m3./sec and the velocity of water is 1.86 m/sec. The turbine runner diameter is 1.25 m with a hub diameter of 0.36 m. The Maximal Net Head for the turbine is 1.12 m. The nominal speed of the turbine is 144 RPM and run away speed of the turbine is 300 RPM. The turbine electricity production at the above

Classification of turbines the low head Kaplan turbine at the Jhang branch Canal in Punjab, Pakistan is a micro hydro power turbine.

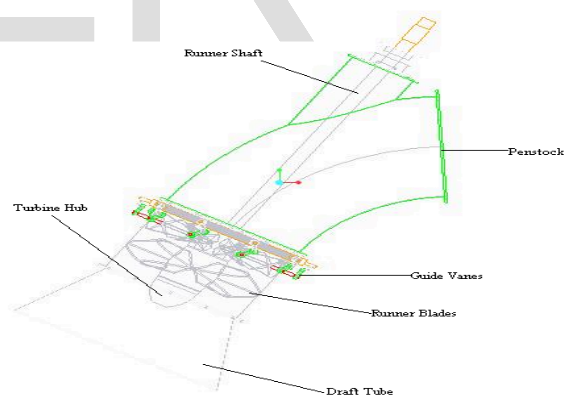

Fig.1. Layout of Kaplan Turbine at Jhang branch Canal in Punjab, Pakistan

Some features of the low head Kaplan turbine are the following

- Turbine runner Diameter D = 1.25m
- Runner Hub Diameter is d = 0.36m
- Net Head water H=1.16m
- Rated water flow Q= 7.07m^3/s
- Velocity of water v = 1.86 m/sec.
- Nominal speed n = 144 RPM
- Run away speed n_r = 300 RPM
- Maximum Power Capacity P = 60KW

Turbines are the most important component of any hydro power plant. They cover about 15-35 percent of the total pro-

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ject cost. [5] discussed the hydropower on small scale, or micro hydro, is one of the most cost-effective energy techniques to be considered for rural electrification in less developed countries. Micro hydro is one of the most environmentally friendly energy technologies available.

[4] discussed the key factors for success of village hydroelectric programs. He addresses the key technical, organizational and economic factors for successful village hydroelectric programs in order to promote a wider and more sustainable use of small-scale hydro power for rural development.

[6]. Cost of the electro- mechanical equipment is very essential part of micro hydro power plant and this cost can b kept down by the carefully selection, standardization and local manufacture. In Indonesia, Nepal, Pakistan locally manufactured turbines are available at less than one tenth of the cost of imported models.

[7]discuss in this paper about the conducted a review of very low head hydro-electric equipment including several conventional propeller designs. It was analsys that propeller turbines are very suitable for the micro hydro applications, commonly due to high rotational speed that allow the turbine to direct couple with the generators. This results in dense design with enhanced efficiency, lifespan and reduced costs and maintenance.

[8] this paper compares large, small and Pico hydropower generation and focused on relative adverse environmental and social effect of each and their economic performance. Cost and population densities were based on data for Pakistan and Peru. The results show that investment is recouped 25% more quickly where a number of very small hydro schemes are used instead of one large scheme, this paper concluded that micro and Pico hydro can be cost effective for supply of rural electricity.

[9] In addition, as with any new technology, capacity training and proper management of the system is required. This is deficient in many occurrences and has led to the miscarriage of the system and wasting of

[10] has develop a micro hydro turbine to be used for specific site of lower head as run of river, which head is less than 1.2 meter ,flow rate is 293.15 l/s and the power generated is 20kw with 293.11 N-m torque , 90% efficiency. A simple civil construction is need for the development of this micro hydro site that resulting the economical viable.

[11] design of two different specific speed micro hydro turbines operating at head 6m to 12m at small scale and up to head of 50m at large scales. These turbines are mixed and radial flow members of a family of turbines developed to cover the micro hydro range from 2 to 50m of head with the efficiency of 70% for all type of models.

[12] present a model of micro hydro turbine with working head 6.7m and flow rate of 86 m^3/sec and the hydraulic efficiency is 80% located at Nandi Pur near Gujranwala. He has done the optimizing of the blades by using static and dynamic analysis by using the software Ansys. And the maximum power output is 4.6MW.

[13] the department of mechanical engineering of the University of Canterbury developed a propeller turbine with a 2.8m operating head, 0.4 m^3/s flow rate and a speed of 612 rpm. The rated power is 3.7 kW with an overall efficiency 20%.

[14] created and applied a qualitative and quantitative selection criteria approach and resolute that a single jet turgo turbine would be the most appropriate technology for 1.3 kW power generation with a 3.5 m operating head, 304 rpm running speed, and 435 mm diameter wheel . A test rig is currently under construction and will be used to optimize the design before field testing and grid connection.

[15] has done the performance analysis of very low head Kaplan turbine. The rated head and discharge rate for the turbine design is 1.5 m and 7.03 m^3/s to develop a required Kaplan type turbine model at Indian institute of technology, Roorkee, India And rated output power is 100kw.

2. TURBINE SELECTION

Selection of the hydraulic turbines type is not always easy for the system design as we can use various turbine types in different applications. The selection of the turbine usually depends upon the following factors.

- Net head
- Flow rate of water
- Rotational speed
- Cavitation problem
- Cost

The specified table completely describes the detail available for each type of the turbine on the basis of net head.

Table 1
Range of heads for specific type of turbines[16]

TURBINE TYPE	HEAD RANGE IN METERS
KAPLAN AND PROPELLER	1<HN< 40
FRANCIS	25 <HN< 350
PELTON	50 <HN< 1'300
CROSSFLOW	50 <HN< 200
TURGO	50 <HN< 250

The figure below describes the various regions for which turbines are typically used for head and flow ranges.

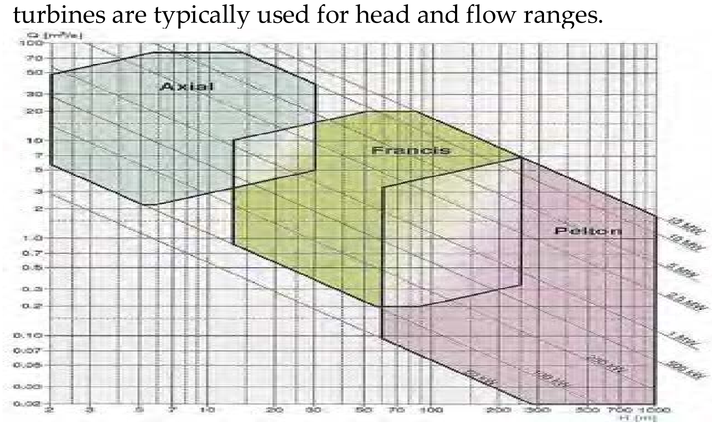

Fig 2 typical flow rate for various type of turbine with respect to net head[16]

Calculated earlier the net head and rated flow of water is

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1.16m and 7.07m³/s respectively.
From this calculation it is concluded that best turbine for this site(RD 216+852) is propeller type Kaplan turbine.

3. TURBINE DESIGN

Specific speed is very important in selecting the desire water turbine; this could be one of the methods to classify the water turbine. The specific speed is define as

$$n_{QE} = \frac{n\sqrt{Q}}{E^{3/4}} \qquad (1)$$

Where:

E specific hydraulic energy of the machine [J/kg]

n rational speed of the turbine [s^{-1}]

The specific hydraulic energy of machine can be determined with the following equation:

$E = H_n * g$

Where:

H_n is net head [m]

$$n_{QE} = \frac{2.294}{H_n^{0.486}} \qquad (2)$$

Since the rational speed is unknown, the specific speed has to be calculated with the up mentioned equation. Hence, the calculated specific speed is **2.09**.
The rational speed can be calculated by the specific speed equation. The calculated value for the rotational speed is **5s⁻¹**.
The runaway speed is the maximum speed which a turbine can attain theoretically, this speed can be achieved when turbine run on no load. Following table gives us the guidelines to determine the runaway speed.

TABLE 2:
TURBINE TYPE WITH RESPECT TO RUNAWAY CONSTANT

TURBINE TYPE	RUNAWAY SPEED
SINGLE REGULATED KAPLAN TURBINE	2.0 – 2.6
DOUBLE REGULATED KAPLAN TURBINE	2.8 – 3.2

The selected turbine for this project is single regulated Kaplan turbine, so from the above table we can calculate the runaway speed.[16]

$n_{max} = n * 2.6$ (3)

$n_{max} = 5 * 2.6$

$n_{max} =$ **13s⁻¹**

Runner diameter can be calculated from the following equation:

$$D_e = 84.5*(0.79+1.602*n_{QE})*\frac{\sqrt{H_n}}{60*n} \qquad (4)$$

By putting all the value, which are previously calculated we can get the value of runner diameter 1.25m, which is very important for the design of the turbine.
The hub diameter can be calculated by the following equation

$$D_i = \left(0.25+\frac{0.0951}{n_{QE}}\right)*D_e \qquad (5)$$

A hub diameter of .36m is calculated from the above equation.[16]
Cavitation is the natural phenomenon affecting the operational restrictions of the fluid flow in pips, particularly liquids. In fact, the vapor pressure of a liquid overdoes the hydrodynamic pressure of water in the turbine, a small part of water changes into vapor state cause the development of steam bubbles. As the pressure of the fluid increase the steam bubbles are not able to withstand the higher pressure and they condense in a collapsing manner. This collapse releases very quick micro streams and pressure peaks up to some hundred MPa. Cavitation occurs usually when air added by some mean to the water at lower pressure end. In the case of Kaplan turbine, the inlet of the runner is relatively subject to it. Due to high water flow velocity cavitation might also occur in Kaplan turbines. 4.2)
There are many disadvantages of cavitation on the turbine. First it decreases the efficiency and creates crunching noise. However, the main problem is the distortion or the damage of the turbine part specifically turbine blades. Cavitation does not just leads towards the distortion but also changes the chemical properties of the material.
The suction head H_s is the head where turbine actually installed; if the suction head is positive, the turbine is positioned above the path of water; if it is negative, the turbine is located under the path of water. To avoid cavitation, the range of suction head is restricted. So, we can calculate the maximum suction head by using the following equation.

$$H_s = \frac{p_{atm}-p_v}{\rho*g} + \frac{c_4^2}{2*g} - \sigma*H_n \qquad (6)$$

Where:

P_{atm} (atmospheric pressure) = p_v (water vapor pressure) =101,300[Pa]

ρ (water density) = 998 kg/m3 [kg/m³]

g (acceleration of gravity) = 9.81m/s2 [m/s²]

c_4 (outlet average velocity) = 2m/s [m/s]

H_n (net head) = 1.2m [m]

σ (cavitation coefficient) [-]

The cavitation coefficient calculated by the model testing, is usually given by the turbine manufacture. However, the cavitation coefficient for the Kaplan turbine can also be calculated by the following equation.

$$\sigma = 1.5241*n_{QE}^{1.46} + \frac{c_4^2}{2*g*H_n} \qquad (7)$$

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Where n_{QE} is specific speed [-]

The specific speed is known from the equation (1) and has $2.09s^{-1}$. Thus a cavitation coefficient of 4.6 is calculated.

$$H_s = \frac{101300 - 9112.67}{1000*9.81} + \frac{2^2}{2*9.8} - 4.6*1.2$$

Hence, the maximum suction head of 4.1m result from the above equation (6). Suction head of 2.9m is chosen.

Centrifugal force occur on blade of the turbine which is define as,

$F_c = mr\omega^2$ (8)

Where:

m= mass of the blade

r = Radius of the runner

ω= angular velocity

Total mass of the blade is 525kg and radius of the runner is .625m.

Angular velocity, $\omega = 2*\pi*n = (2*3.14*5) = 31.4s^{-1}$

Where:

n= rotational speed

$F_c = 525*0.625(31.4)^2 = 323518.125N$

Water force on the blade is the Force due to water is define as,

$F_w = \rho QV = \rho Q(r\square)$ (9)

Where:

ρ	Density of water
Q	Discharge flow of water
r	Radius of runner
ω	Angular velocity

$\omega = (2*3.14*5) = 31.4s^{-1}$

$F_w = 1000*7.07*0.625*31.4 = 138748.75N$

Suppose the blade is at 20□□then the axil and tangential components of water force is given by[15]

Tangential component of water force

$F_{wt} = 138748.75(\cos 19.1^o) = 130381.18N$ (10)

Axial component of water force

$F_{wa} = 138748.75(\sin 19.1^o) = 47454.87N$ (11)

F_{wa} is the total force that is induced due to the action of water. So the total force acts on blades are 47454.87N and for single blade we can take the force by dividing on number of blades of our turbine.

Table 3:
Total Forces

S.NO.	FORCE TYPE	VALUE(NEWTON)
1	WEIGHT	17640 N
2	CENTRIFUGAL FORCE	323518.125N
3	WATER FORCE	138748.75 N
4	WATER FORCE(AXIAL)	130381.18 N
5	WATER FORCE(TANGENTIAL)	47454.87 N

Design of the blade just does not depend on the stress analysis. Many other factors are also involved for the design of the blades. The properties of blades should be like the leading edge is thicker than the trailing edge for a streamlined flow. Although, the blades should be as thin as possible to minimize the effect of cavitations characteristics; normally Kaplan turbine blade is thicker towards the flange and becoming thinner and thinner near the tip. In addition, the blade has to be distorted on the basis of the tangential velocity. The "airfoil theory" also plays an important factor in defining the shape of the profile and distortion of the blade.

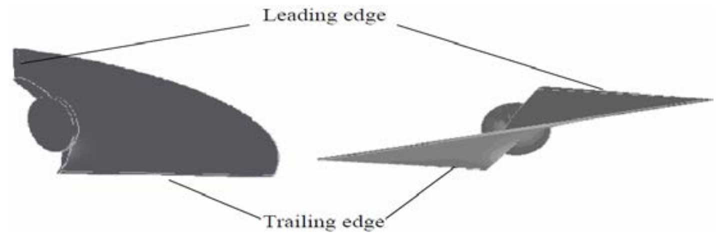

Fig.3: Blades leading and trailing edge

4. Drawings of the proposed Turbine

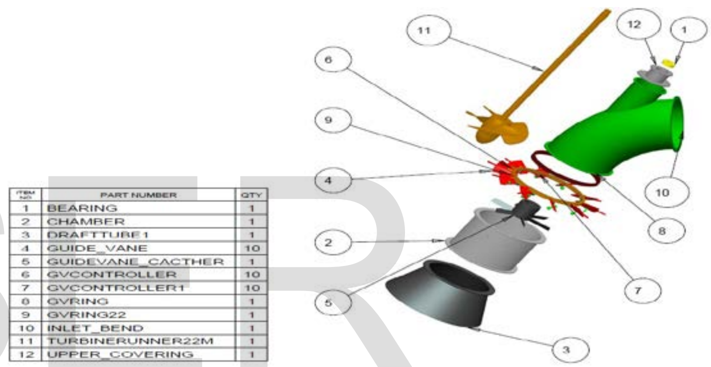

Fig. 4 Turbine geometry detail

4.1 Runner of Turbine

Runner of the turbine consists of shaft, hub, and blades. Detail of runner design mentioned in the following drawing.

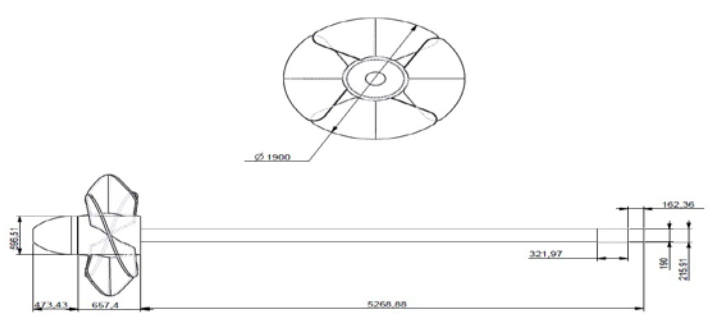

Fig.5: Detail of Kaplan turbine Runner

4.2 Guided vane catcher

The complete model of the low head Kaplan turbine is prepared in the Pro-E software. Every Part of the turbine is sketched separately

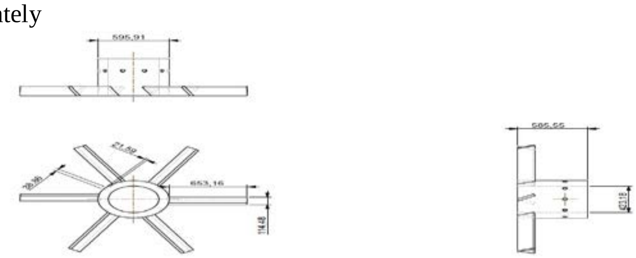

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Fig.6: Guided vane catcher

4.3 Inlet Bend

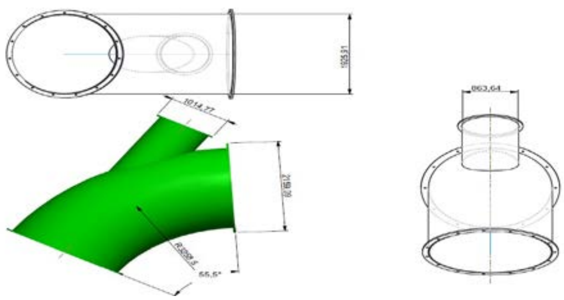

Fig.7: Inlet Bend

4.4 Chamber

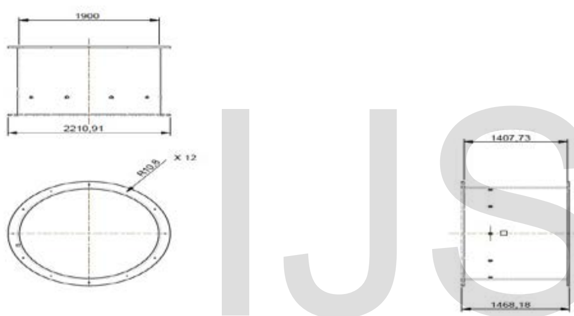

Fig.8: Chamber

4.5 Upper cover

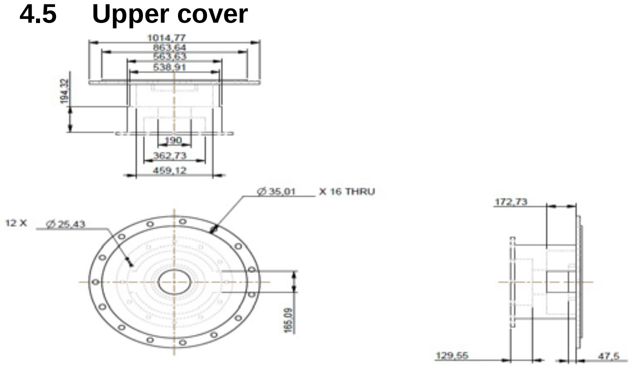

Fig.9: Upper cover

4.6 Draft tube

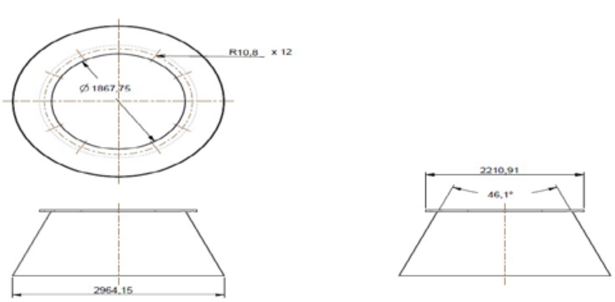

Fig.10 Draft Tube

5. 3D SOLID MODELLING OF THE TURBINE RUNNER

The complete model of the low head Kaplan turbine is prepared in the Pro-E software. Every Part of the turbine is sketched separately in the Pro-E and then all parts are assembled using the assembly command in the Pro-E. We save the IGS file of the model for further processing.

Fig.11 Kaplan Turbine Runner (Blades) in Pro-**E**

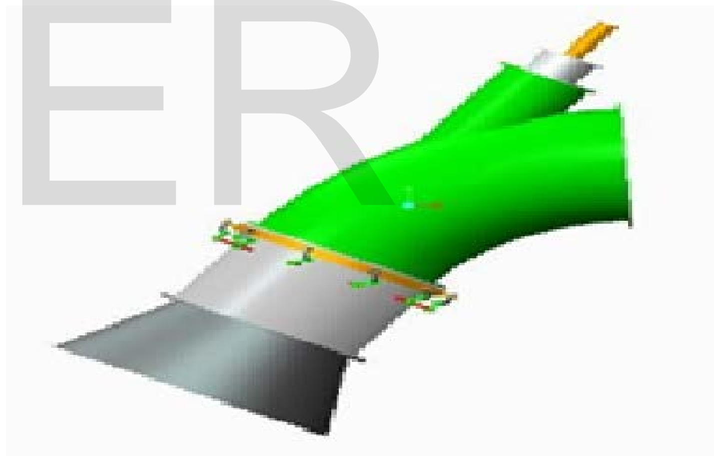

Fig.12 Kaplan Turbine Assembly in Pro-E

6. STATIC AND MODAL ANALYSIS OF KAPLAN TURBINE RUNNER

The analysis on the Kaplan Turbine Runner was performed in the ANSYS 14. The Pro-E file is not directly incorporated in the ANSYS 14. We opened the IGS file in the ANSYS 14 and performed all the required analysis on it.

6.1 MESH GENERATION

For meshing of the Kaplan Turbine Runner we used the sweep method. The numbers of nodes are 83564 and the numbers of elements are 18937.

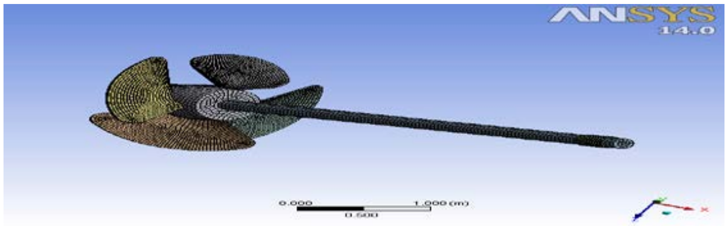

Fig.13 Meshing of Kaplan Turbine Runner in ANSYS

Table 4
Summary of Mesh data

Component	Total No. of nodes	Total No. of element
Shaft	14214	2764
Runner blades	69350	16173
both	83564	18937

6.2 BOUNDARY CONDTIONS

Turbine is analyzed under the head of 1.2m and discharge rate of 7.07m3/s. equivalent force of hydraulic load on blade is calculated (eq. 4.) and applied on 4 blades in ANSYS14.0. Beam rod is fixed at point A and forces is applied on the blades at point B,C,D,E respectively

7 STATIC STRUCTUAL ANALYSIS OF RUNNER BLADE

The maximum stress on the blade is 244 Mpa while the ultimate tensile strength of the blade material (16Cr-5Ni) is 890 Mpa. He the factor of safety is 3.64. The blade will work safely at this stress.

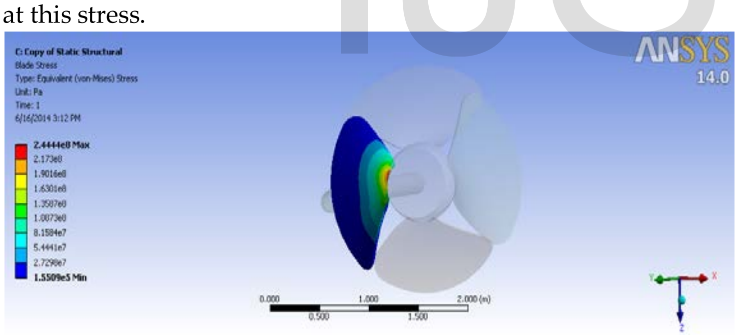

Fig.14 Blade Stress on one blade

The Maximum principle stress on the blade shown in the below diagram is 302 Mpa and the ultimate tensile strength of the blade material (16Cr-5Ni) is 890 Mpa. He the factor of safety is 2.94. The blade will work safely at this stress.

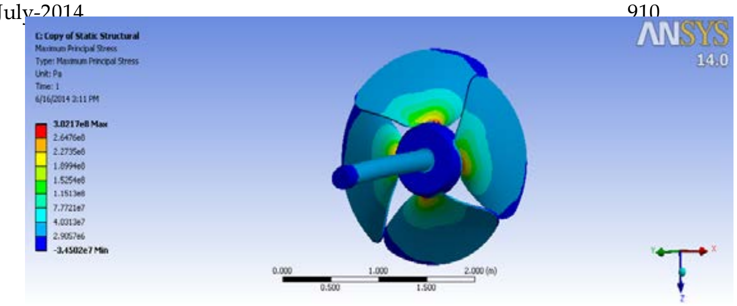

Fig.15 Maximum Principle Stress on runner

The maximum principle stress on the hub is 51.959 Mpa while the ultimate tensile properties of the hub (AISI 4140 alloy steel) material is 1242 Mpa. Hence the factor of safety for the hub is 23.90.

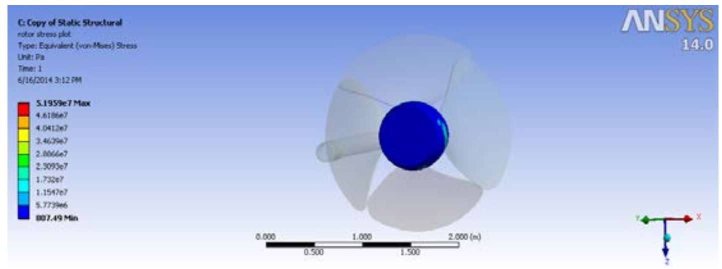

Fig.16 Equivalent (Von-Mises) Stress on runner hub

The equivalent (Von-Mises) stress on the runner shaft is 10.34 Mpa while the runner shaft material (S355) tensile strength is 355Mpa. Hence the factor of safety for the runner shaft is 34.33.

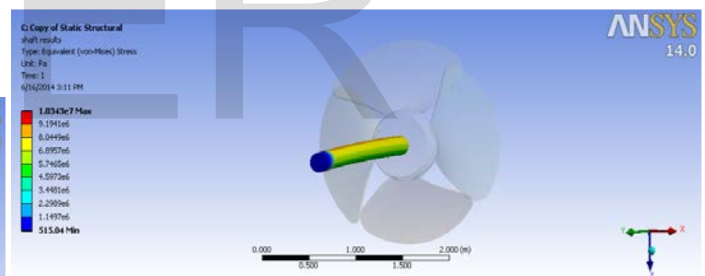

Fig.17 Equivalent (Von-Mises) Stress On runner shaft

The maximum displacement in the blades of the runner is 0.0077 m at the edges of the blades. The blades behave like a fixed beam and show no failure or major events during the analysis.

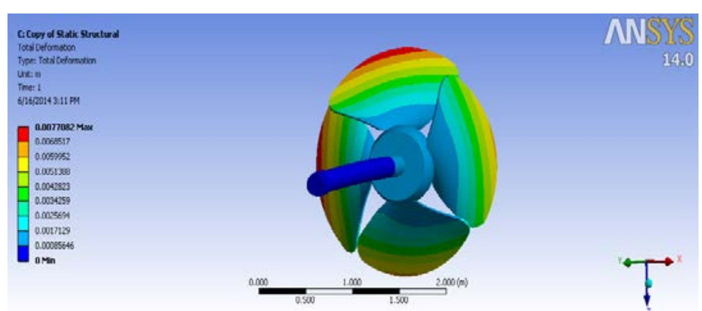

Fig.18 Total Deformation in the shaft

8 Modal Analysis

The designed turbine operates at 144RPM so the corresponding working frequency is 13.53 Hz. By performing modal analysis, the following mode Global mode shapes are obtained.

8.1 Mode Shapes

In first mode shape the designed turbine operates at 144RPM so the corresponding working frequency is 13.53 Hz and the natural frequency of the runner blade is 40.055 Hz. As turbine runner blades working frequency and the turbine runner blade natural frequency do not match so the turbine structure has no tendency of resonance. The Maximum displacement in the structure if the structure exhibits this model shape is shown in first figure.
In second Mode shape the designed turbine operates at 144 RPM so the corresponding working frequency is 13.53 Hz and the natural frequency of the runner blade is 41.66 Hz. As turbine runner blades working frequency and the turbine runner blade natural frequency do not match so the turbine structure has no tendency of resonance. The Maximum displacement in the structure if the structure exhibits this model shape is shown in second figure.

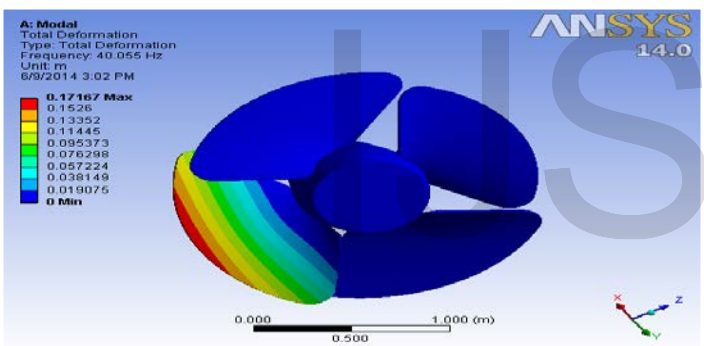

Fig.19. Max Total Deformation in the blade in First Mode Shapes

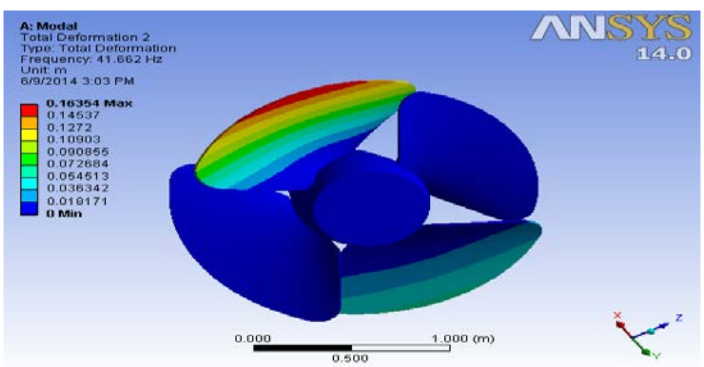

Fig.20 Max Total Deformation in the blade in second Mode Shape

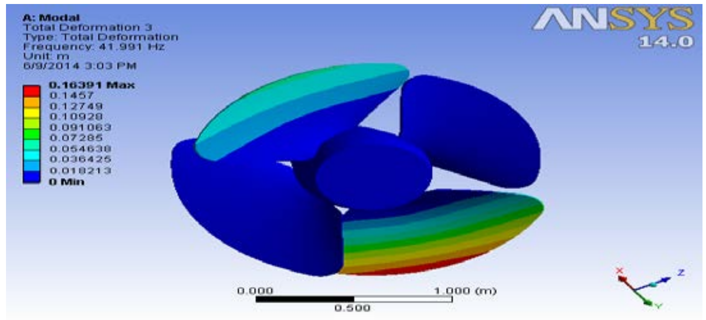

Fig.21 Max Total Deformation in the blade in Third Mode Shape

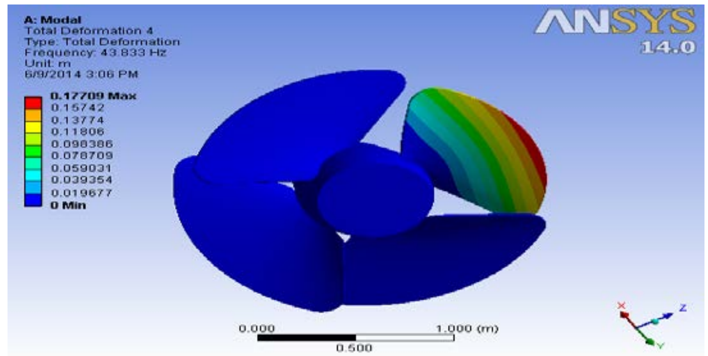

Fig.22 Max Total Deformation in the blade in FourthMode Shape

In third mode shape in the above diagram the designed turbine operates at 144 RPM so the corresponding working frequency is 13.53 Hz and the natural frequency of the runner blade is 41.99 Hz. As turbine runner blades working frequency and the turbine runner blade natural frequency do not match so the turbine structure has no tendency of resonance. The Maximum displacement in the structure if the structure exhibits this model shape is shown in third figure above.
In fourth mode shape the designed turbine operates at 144 RPM so the corresponding working frequency is 13.53 Hz and the natural frequency of the runner blade is 43.833 Hz. As turbine runner blades working frequency and the turbine runner blade natural frequency do not match so the turbine structure has no tendency of resonance. The Maximum displacement in the structure if the structure exhibits this model shape is shown in fourth figure above.

9 Conclusion

The analysis in ANSYS is very important prior to the fabrication of the turbine. The static structural and modal analysis is carried out in the ANSYS 14 software for low head Kaplan turbine runner. The stress (Von-Mises) and maximum stress developed at the runner blades are maximum at joints between the hub and runner blade whoever their valves are less the ultimate tensile strength of the runner blade material. The factor of safety for stress (Von-Mises) and Maximum stress are 3.64 for stress and 2.94 respectively. Maximum principle stress is also in the safe limits. Hence all the stresses developed at the turbine runner blades are safe and no major failure is recorded during the static structural analysis. The modal analysis shows no resonance in any of the four mode shapes. The natural frequency of all mode shape does not match with the natural frequency of the runner blade. Hence no resonance produced during the modal analysis. The blade acts as a fixed cantilever beam during the modal analysis where the displacement is high but in safe limits at the edges of the runner blade for all mode shapes.

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