# Vibration Analysis of Pelton Wheel Turbine by Theoretical, Numerical & Experimental Approach

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***Abstract*—Pelton wheel turbine is used in hydroelectric power plant for mechanical power generation high head applications. The dynamic characteristics of a hydro turbine power depend on set input parameter and load condition. In this paper the study of Dynamic analysis of the Pelton wheel turbine is presented. When the Natural frequency of the turbine coincides with actual frequency of the turbine causes the formation of the resonance. This resonance forms the increase in chances of failure of the turbine by buckling of deformation of the shaft. Hence to reduce such failure mode due to resonance it is needed to observe and find out the possible conditions of increase in natural frequency. In this paper the methods of investigation of the natural frequency is given using theoretical calculations and FEA simulation and the proposed methods are validated using experimental values. The theoretical calculations are based on the mass of the system and its material properties. FEA simulation is carried out using Solid Works. The experimental validation of the results from these methods is finding the natural frequency using FFT analyzer. In this paper the Comparision of the natural frequency of the pelton wheel turbine between the theoretical calculations, FEA values and the Experimental values and the results are shown and it is seen that they are good in result as showing less error with safe conditions.**

***Index Terms*—forced vibration, dynamic analysis, mathematical modeling, FFT analyzer**

## I. INTRODUCTION

THE study of dynamic behavior of pelton wheel turbine has been of great importance in order to understand the operations and the failures related with the machines. Machines are set to vibration under number of excitation modes. So, accurate prediction of vibration characteristics is important in the design stage of hydraulic machinery considering the requirements of quality, performance and safety. Pelton wheel Turbine operates at high rotational speed for the purpose of electricity production. So, there is a need to develop methodologies that allow for more realistic dynamic analysis of Pelton turbines because prototyping and testing cost are exceptionally high and failure is generally causing great damage in the practical applications and testing of these systems. [1]

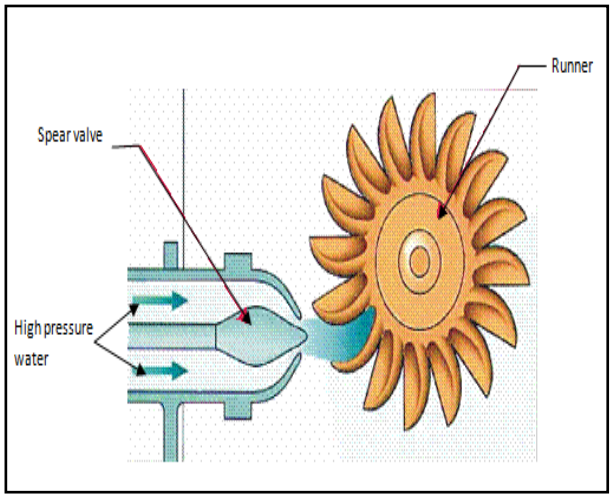


Fig. 1 pelton wheel turbine

Rotating machines nowadays are designed such that they can be safely operated beyond or passing through many critical speeds. Not only each machine structure reveals its own local dynamic characteristics, but the whole machine as an assemblage of part structures also reveals global dynamic characteristics. The dynamic properties of most common interest in rotating machinery typically include the critical speeds, stability of modes and forced response. The critical speeds of a rotor are defined as the rotational speeds at which the speed-dependent modal (natural) frequencies. [1]

One of the most important and correct graphical presentations for predicting critical speeds is known as the Campbell diagram, it is very useful in the design and manufacture of a turbine. It is the graph of rotational speed v/s natural frequencies of its disk wheels and buckets. [4] It is also referred to as the whirl speed frequency-speed diagram, natural frequency-speed map, and frequency interference diagram, where the equivalently the modal frequencies, whirl speeds and the order lines of possible excitation forces are plotted against the rotational speed. The Campbell diagram has been mostly considered in the design of rotors with bladed disks such as turbines and fans, to find out natural frequencies.

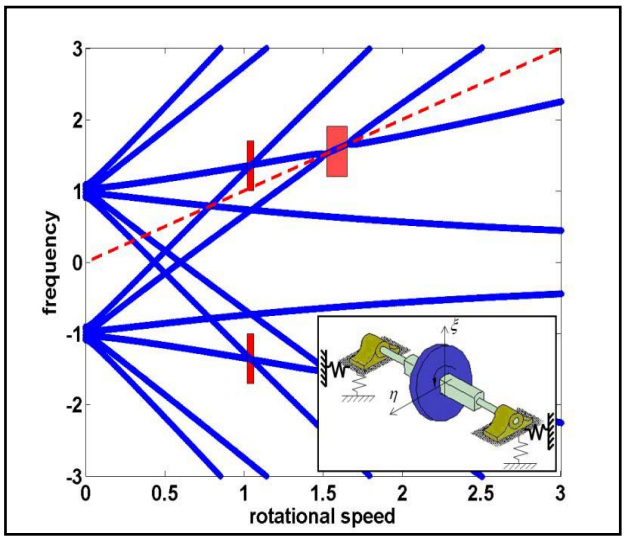


Fig. 2 Conventional Campbell diagram for the simple general rotor [4]

## II. LITERATURE SURVEY

The dynamic behavior of flexible rotor systems subjected to base excitation (support movements) is investigated theoretically and experimentally by Aman Rajak et al. [1]. They have developed a mathematical model for total energy of the system and the equation of motion has been derived using energy method. The study focuses on the critical speeds of rotation.

Lixiang Zhang et al. [2] have studied ANSYS finite Element software to model the main shaft system in the hydro-turbine generating unit. In that paper they have done Modal analysis and calculate the critical speed of rotation. These results can provide a reference for dynamic analysis and a foundation for the design safety or improvement.

Uzma Nawaz et al. [3] have studied the differential equations for hydro system and based on the differential equations, a transfer function is obtained. The analytical method for calculating the flow of water through the penstock uses equation of motion and continuity. These equations used arithmetical and graphical methods for solving transient stability problems.

Chong-Won Lee [4] has studied Frequency-speed diagram, also known as Campbell diagram that has an important tool in the design and operation of rotating machinery. The construction of the conventional frequency-speed diagrams are explained in relation to the desired rotor dynamic properties of turbine and fans rotor systems. Thus, from Campbell diagram drawn for rotating machines, engineers can understand which modes are likely to be excited by the excitation sources of interest and which speed regions are safe for operation.

Junyi Li and Qijuan Chen [5] have proposed a nonlinear mathematical model for hydro turbine governing system (HTGS). All the components of HTGS which is conduit system, hydraulic servo system, turbine, generator, and are considered in the model. There are three main contributions of this paper compared with prior works. First, a new fully coupled nonlinear mathematical model of HTGS was presented and the parameters were from a practical power station, which made the work more consistent.

Prakash K. Dhakan and Abdul Basheer Pombra Chalil [6] have presented the design of casing for four jet vertical pelton wheel turbines is carried out considering the size and shape of the casing. Casing should have enough strength to meet the mechanical/structural requirements such as to withstand the dead weight of the generator, forces developed in the manifold/branch pipes, load due to the four jets in different combinations and load due to various actuation mechanisms. After satisfying above aspects, the casing should be checked for vibration behavior by modal analysis. Through the structural analysis using ANSYS Mechanical software, casing design is optimized and a weight reduction of around 12% is achieved. Vibration behavior of the casing is analyzed through the model analysis and ensured the natural vibration of casing is well above the operating frequency of turbine unit.

Sridharan. P [7] has presented the unbalanced mass in the generator rotor causes an abnormal vibration when the turbine shaft attains it critical speed and resonance condition is created. Therefore a modification in the stiffness of the supporting structure for the bulb turbine is made to avoid the resonance condition in the operating range of the turbine.

Jiaqi Liang [8] has explained the dynamics of an elastic water model for the case of a common water tunnel connected to multiple penstocks. The result shows that water elasticity effects are negligible for transient studies with temporary disturbances, but may be necessary for long-term dynamic studies and hydraulic system studies.

From the past literature review it is observed that, very less work has been done in the field of the dynamic behavior of pelton wheel turbine and their effects in design and operation. So the experiment investigation of dynamic characteristics (natural frequency) of pelton wheel turbine is not done.

## III. PROBLEM DEFINITION

Dynamic characteristics of the Pelton wheel turbine are to be investigated experimentally in the present work. Natural frequency of forced vibration caused by rotation of Pelton wheel is to be investigated. Finally the experimental results are to be compared with that obtained from suitable analysis software using solid works simulation.

## IV. METHODOLOGY

In this paper investigation of dynamic analysis for Pelton wheel turbine test rig is carried out using following two methods and the validation of these two methods is done by the comparison with the experimental results.

### *A. Theoretical model:*

For the theoretical model the shaft, the disks and the unbalance mass are considers for the system of pelton when assembly. The natural frequencies are calculated using total system energy derivation and equation of motion. As the model is developed in reference paper 1 the natural frequency is calculated using the same formula of natural frequency.

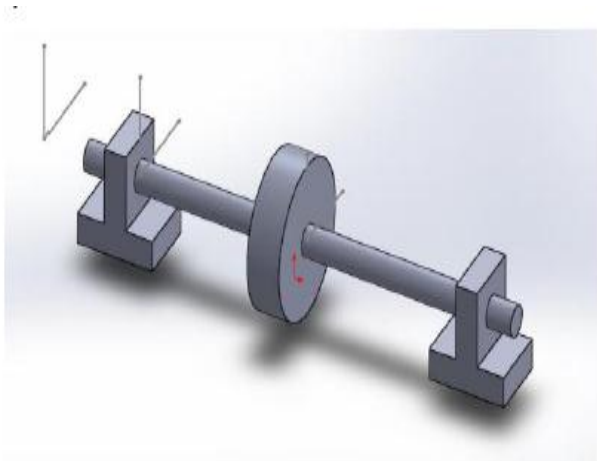

Fig. 3 Simple Rotor Disk Systems [1]

The natural frequency is calculated using following formula,

$$s^2 = \frac{-(2mK+a^2\omega^2)\pm\sqrt{4mka^2\omega^2+a^4\omega^4}}{2m^2} \quad 1$$

The solution of the above expression leads to two pairs of complex conjugate roots. The real part of the complex conjugate roots represents the rate of decay of the vibration. Due to our assumption of undamped system the rate of decay of vibration is '0' in the solution.

The extracted parameters on which natural frequency depends are then used for further calculation of natural frequency are given below in table 1.

Table 1: Parameters used in calculation on which natural frequency depend [1]

| Sr. no. | Parameter | Values |
|---|---|---|
| 1 | Mass of runner, $M_D$ (kg) | 3.25 |
| 2 | cross sectional area of shaft, S ($m^2$) | $1.25\times10^{-3}$ |
| 3 | Density of shaft material, ρ ($kg/m^3$) | 7850 |
| 4 | Length of shaft, L (mm) | 560 |
| 5 | Mass moment of inertia Ixx ($m^4$) | $1.256\times10^{-7}$ |
| 6 | Mass moment of inertia Izz ($m^4$) | $2.513\times10^{-7}$ |
| 7 | Modulus of elasticity E ($N/m^2$) | $2\times10^{11}$ |
| 8 | Runner speed, ω (rad/s) | 78.84 |

The solution for natural frequency of the system is obtained from the above equation. Now substituting the value of the parameter as provided from model in a, k and m.

$$m = M_D + \frac{\rho SL}{2} + \frac{\rho I\pi^2}{2L} \quad (2)$$

$$a = \frac{\rho I_{Sxx}\pi^2}{L} \quad (3)$$

$$K = \frac{\pi^4 EI}{2L^3} \quad (4)$$

Then we have,

$S^2$=13292907.2($i^2$)

Hence, the natural frequencies are

$$S_1 = 3645.94 \text{ Hz}$$

$$S_2 = 3639.05\text{Hz}$$

$$\omega_1 = \frac{S1}{2\pi} = 580.27\ Hz$$

$$\omega_2 = \frac{S2}{2\pi} = 579.17\ Hz$$

The natural frequency of the system along the respective degree of freedom U and V were found to be 580.27 Hz and 579.17 Hz respectively.

*B. FEA simulation*

The Pelton turbine installed at R. H. Sapat College of Engineering, Management Studies and Research, Nashik by using modeling software A Pelton turbine model is developed in Solid works 2013 for a jet diameter of 15mm as shown in the Figure below

Fig 4 Pelton Wheel 3D models in Solid Works

Fig 5 Bucket of Pelton Wheel 3D models in Solid Works

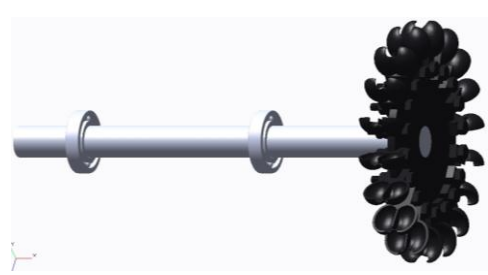

Fig 6 Assembly of Pelton Wheel 3D models in Solid Works

The simulation for the above geometry is carried out in Solid Work Simulation software. The natural frequency is calculated from the simulation.

### *C. Experimental Methodology*

Experimental validation is done by using FFT (Fast Fourier Transform) analyzer. The FFT spectrum analyzers samples the input signal, computes the magnitude of this in sine and cosine components, and display the spectrum of these measured frequency. The advantage of this technique is its speed.

Figure 7(A) shows Experimental Set up of pelton wheel test rig model for dynamic Analysis, in which proper connections of accelerometer, modal, laptop and FFT Analyzer were made. Then by analyzer test various frequencies are obtained with the help of FFT Analyzer. Figure 7(B) shows the connection of two accelerometer 1$^{st}$horizontal and 2$^{nd}$ vertical attached to system.

Fig. 7 (A) Pelton Wheel Turbine Test Rig. (B) Connections of Accelerometer

The connections i.e. accelerometer, laptop and other power connections were made. The surface of the pelton wheel turbine model was cleaned for proper contact with the accelerometer. The accelerometer was then attached with the surface of the pelton wheel turbine model. We are using SKF series two channel FFT analyzer.

## V.RESULT & DISCUSSION

Investigation of dynamic analysis for Pelton wheel turbine test rig is carried out. Here the results from theoretical calculation results, FEA results and Experimental results are shown in following table 2

Table 2 Result table

| Sr. No. | Direction | Results | | |
|---|---|---|---|---|
| | | Theoretical | FEA | Exp. |
| 1. | Vertical | 580.27 Hz | 549Hz | 605 Hz |
| 2. | Horizontal | 579.17 Hz | 550.91Hz | 610 Hz |

### *A. FEA simulation results*

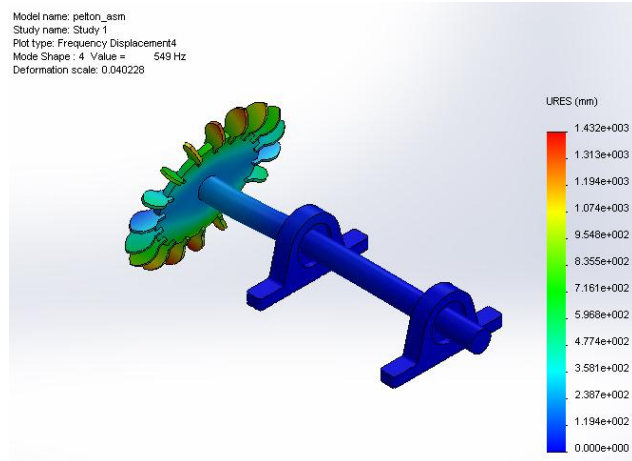


Fig. 8 Pelton Wheel simulation Vertical dimension in Solid Works

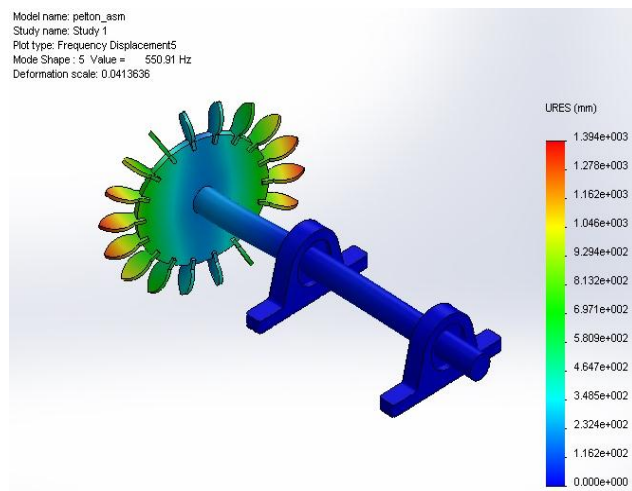


Fig. 9 Pelton Wheel simulation horizontal dimension in Solid Works

### *B. Experimental FFT results*

a) Horizontal direction (U):

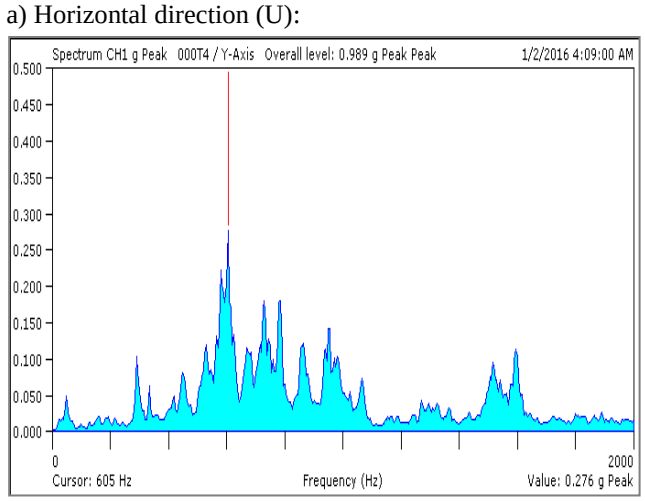


Fig. 10 Result In Horizontal Direction

b) Vertical direction (v)

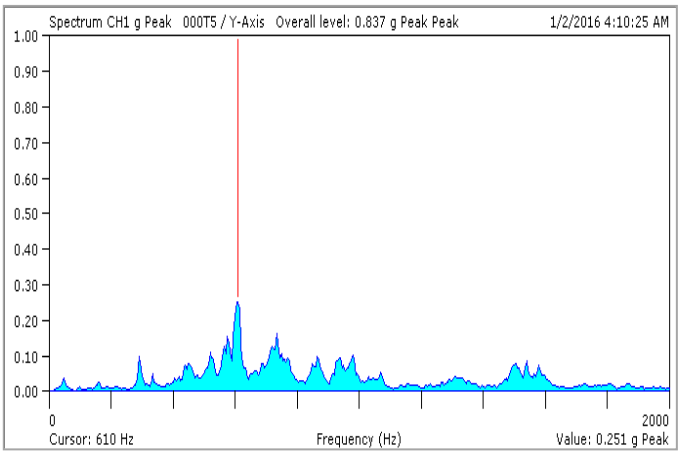


Fig.11 Result In Vertical Direction

The analytical results for a Pelton turbine laboratory setup shows that the natural frequency of the system lies in a good safe range. The natural frequency of the system along the respective degree of freedom U and V were found to be 580.27Hz & 579.17Hz respectively.

### *C. Sensitivity of various parameters affecting dynamic response of system*

Based on the model developed in this paper shows the various parameter and relation between sensitivity of natural frequency of system following are the basic parameter affecting sensitiveness in natural frequency of system:

1. change in length of shaft
2. change in diameter of shaft
3. change in stiffness of bearing

1. Change in length of shaft: from equation no. 4 it is seen that the stiffness of shaft is inversely proportional to cubical value of length of shaft (L3). Due to this correlation, as length of shaft increases the natural frequency of system decreases and vice versa.

2. Change in diameter of shaft: It is seen that shaft diameter is directly proportional to stiffness of shaft. Hence it also directly proportional to natural frequency of system.

3. Change in stiffness of bearing: it is observed that decrease in bearing stiffness the natural frequency of the system dramatically decreases.

## VI. CONCLUSION

In this presented paper, the dynamic behavior of Pelton Turbine has been studied based on its natural frequency. The analytical model for dynamic behavior of the Pelton turbine assembly was thus formulated and the analytical solution of natural frequency was performed. The theoretical, FEA & experimental results for a Pelton turbine laboratory setup shows that the natural frequency of the system lies in a good safe range. The error between theoretical and experimental is about 4% and the error between experimental and FEA is 9% this is due to the unbalance mass, maintenance, bearing lubrication, attachment of sensor at the time of experimentation.

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