INTERNATIONAL JOURNAL OF ADVANCED RESEARCH IN ENGINEERING AND TECHNOLOGY (IJARET)

ISSN 0976 - 6480 (Print)
ISSN 0976 - 6499 (Online)
Volume 5, Issue 4, April (2014), pp. 103-110
© IAEME: www.iaeme.com/ijaret.asp
Journal Impact Factor (2014): 7.8273 (Calculated by GISI)
www.jifactor.com

IJARET

©IAEME

# FLOW RATE CHARACTERISTICS OF KAPLAN WATER TURBINE

**Valentin Obretenov**

Department of Hydroaerodynamics and Hydraulic Machinery,
Technical University of Sofia, Bulgaria

## ABSTRACT

Results of experimental tests of Kaplan water turbine in Ivaylovgrad HPP, Bulgaria, have been presented in the paper. The flow rate values for the turbine have been determined over the whole output range. Two different flow rate measuring methods have been applied: by measuring the spiral case inlet section mean velocity and by the Winter-Kennedy method.Measuring method has been developed and on its base the required facilities and measuring system have been designed and manufactured. The possibility for reliable determination of the value of the base factor k in the flow equation in the Winter-Kennedy method by calculations has been proven experimentally.

**Keywords:** Kaplan Turbine, Flow Rate, Winter-Kennedy Method, Flow Characteristic.

## I. INTRODUCTION

Results of experimental tests of Kaplan water turbine in Ivaylovgrad HPP, Bulgaria, have been presented. Three hydro aggregates have been installed in the power plant, each of them consisting of Kaplan vertical turbine and synchronous generator. Each turbine has a spiral case with circular and oval cross sections, cylindrical guide vanes and elbow-shaped draft tube. The subject of investigations is to determine experimentally the flow rates for several operating modes in the operating range of Turbine No. 1 prior to its modernization. Two independent methods have been used to measure the flow rates.

## II. INVESTIGATION LAYOUT

Figure 1 shows the measuring system layout.

### 2.1. Instrumentation

Pressure transducer of P31AP/100B1 type (accuracy class of 0.05) manufactured by Hottinger Baldwin Messtechnik (HBM) has been used for pressure measurements at the turbine inlet (measuring point 1 – Fig. 1).

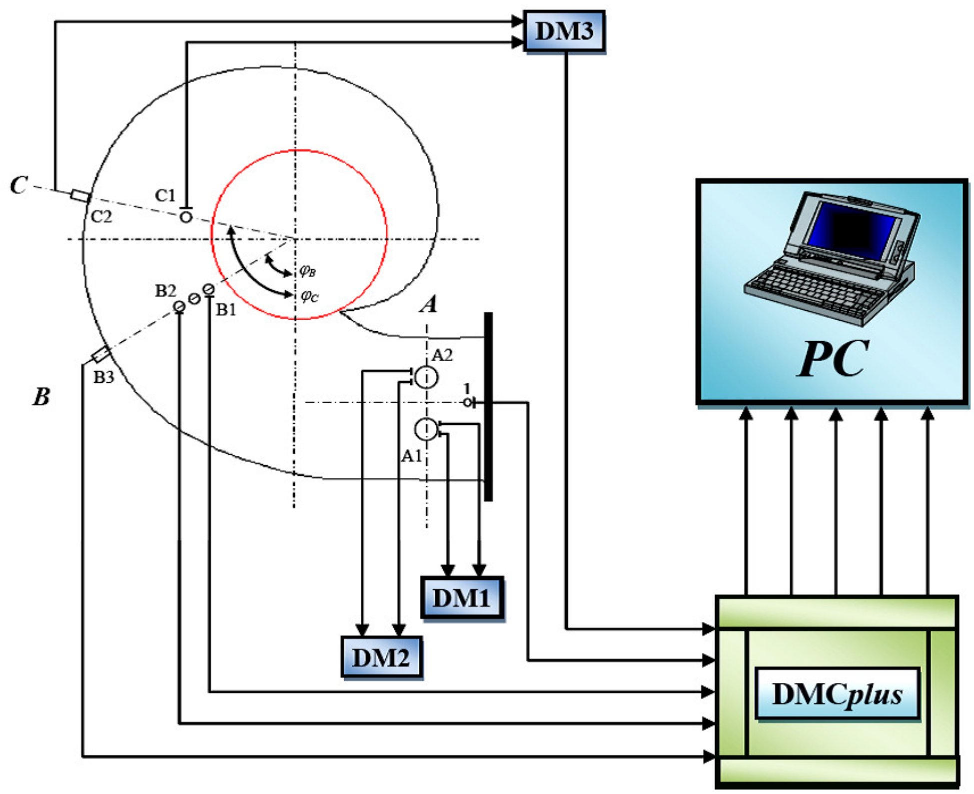


**Fig. 1:** Measuring system layout

In the measuring section B, the pressure is measured in 3 points (B1, B2 and B3) where pressure transducers have been connected. These pressure transducers are of P31AP/100B1 (point B3) and P8AP/100B (points B1 and B2) types and of accuracy class 0.3. In the measuring section C, the pressure difference between measuring points C1 and C2 is measured using differential pressure gauge of PDE 200/20 type.

The signals from the pressure transducers are transferred to the DMCplus digital measuring device. The values transformed into digital codes are monitored and are saved online by personal computer (PC – Fig. 1). The computer processes the experimental data using CATMAN® specialized software developed by HBM company. The data are recorded for 40 seconds with 1 second interval in files and then are processed by specialized software together with the data from other instruments. The measurements have been performed for several turbine operation modes according to the actual regulator settings (according to the gate-vane relationship [5]).

### 2.2. Calculation of the flow rate using the mean flow velocity

In the A section (Fig. 1) two identical flow velocity measuring tubes (tube A1 and tube A2) have been installed. They have been designed and manufactured specially for this investigation. The dimensions of the active part of these probes (known as Pitot differential tube or Pitot-Prandtl tube) comply with the Central Institute of Aerohydrodynamics (CIA) recommendations [2]. As we know, for the CIA tube, the maximum measuring deviation of the dynamic pressure with a change of orientation by ± 20° does not exceed ± 3 %. Because of the limited lengths of the flow velocity measuring tubes as well as the limited (by the operators) measuring time for the investigations, determination of full velocity profile for each operation mode was not possible. Therefore, the following method has been adopted: with maximum turbine load, the velocity profile has been determined within the probe range. On the base of the obtained results has been found the mean value of the flow velocity and its position in the section, respectively. Obviously, this is a case of developed turbulent velocity profile in cylindrical tube (the Reynolds number over the whole range of investigated ranges is within $Re = 9.244.10^6 \div 1.307.10^7$) for which the mean velocity can be measured for $r_s = \kappa_r R$ radius and the factor $k_r = f(Re)$ [4] and $R$ is the radius of the circular section($R$=*2.2m*). With the specified values of the Reynolds number($Re$) for the $k_r$ factor it is recommended to apply the value $k_r = 0.862 \div 0.868$. In this specific case the value of the radius $r_s$ is determined experimentally by the flow velocity profile in the measuring section. The determination of the mean value of the flow velocity is performed by averaging the flow rate. In this case the flow rate through the discussed section the following can be presented, as follows:

$$Q = 2\pi \int_0^R c(r)dr \tag{1}$$

According to the continuity principle the mean velocity will be:

$$c_s = \frac{Q}{S} = \frac{2\pi \int_0^R c(r)dr}{\pi R^2} = \frac{2\int_0^R c(r)dr}{R^2} \tag{2}$$

The velocity in any measuring point is calculated using the dynamic pressure which is measured by the flow velocity measuring tube and the differential pressure gauge (Fig. 1). Once the dynamic pressure $\Delta p_d$ has been measured, the flow velocity can be calculated by the following expression:

$$c = \sqrt{\frac{2\Delta p_d}{\rho}} \tag{3}$$

where $\rho$ is the water density (determined according to the IECStandard 60041 [7] recommendations according to the pressure and temperature). The integral of the mean velocity equation is solved numerically and the radius value related to the mean value is calculated by cubic interpolation. The measurements showed practically equal values of the dynamic pressure for both probes.

An assessment of the measuring errors has been made (the measuring errors of the flow rate equation values have been taken into account). In such conditions, the following is obtained for the mean relative quadratic error of the flow rate measurements: $\sigma_{Q1} = \pm 0.019$.This value is within the recommendations of Standard 60041 [7] for this method.

### 2.3. Flow rate measurements according to Winter-Kennedy method

This flow rate measuring method for water turbines is based on the principle of the so called centrifugal flow rate transducers [4]. There, the pressure difference along the radius for flows in curved tubes is measured. This difference is caused by the continuous action of the centrifugal acceleration. In this case, the pressure increase can be determined by the following equation [4]:

$$dp = \rho \frac{c^2}{r} dr \tag{4}$$

If some mean velocity (determined for example according to the continuity equation) is assumed for the velocity between two points of $r_1$ and $r_2$, respectively, the pressure difference between these points is:

$$\Delta p = \frac{\rho Q^2}{S^2} ln \frac{r_2}{r_1}. \tag{4a}$$

From this equation, the flow rate can be determined:

$$Q = S\sqrt{\frac{\Delta p}{\rho \, ln \frac{r_2}{r_1}}}, \tag{5}$$

where $S$ is the cross-section area.

In the reaction turbine spiral cases, conditions for the use of this method are available; therefore it is applicable especially for low-pressure water turbines where in general the options are limited. This method is called the Winter-Kennedy method after the names of both researchers who suggested it for the first time as a measuring method for the flow rate in reactive water turbines [10]. The kinematic configuration of the flow in the spiral case is considerably more complex than the configuration described above. The difficulties are related mostly with the determination of the radial velocity distribution. In general, the flow rate determination formula according to this method is of the following type:

$Q = kh^m$, (6)where $h = \Delta p / \rho g$ is the pressure difference between the pressures at the two points (in $mWS$), and $k$ is a dimension factor which takes into account the measuring section geometry, the pressure measuring points, the flow parameters and the modes. The exponent value $m$according to the recommendation in [10] is $m=0.48\div0.52$. Also, the same recommendation underlines the ISO Standard 41 [7] (Art. 15.5). There are several methods suggested by various authors to determine the $k$ factor [1, 3, 5, 7]. The most reliable method is calibration by means of other method or model investigation. In the latter case $k = \lambda k_m$, where $k_m$ is the value of the factor obtained during the model investigations and $\lambda$ is a similarity factor. The value of $\lambda$ remains the same for similar turbines (e. g. model and original) as the hydraulic losses have an impact over the pressure in both points and the pressure has to be measured. For cases where it is possible to perform model investigations, Sheldon L. [8] offers the use of the following formula in order to determine the exponent: $m = a + blog(h)$.

There are also theoretical (computational) methods for determination of the $k$ factor on the base of analysis having impact on the flow kinematics in the spiral case [3, 6]. For example, in [3] it has been proved that if the spiral canal has been sized with constant velocity momentum ($c_u r = const$, [5]) the $k$ factor value can be obtained according to the following equation:

$$k = \frac{2R_1R_2\sqrt{2g}}{\sqrt{R_2^2 - R_1^2}} \int_{x_N}^{x_K} \frac{y(x)}{x} dx \tag{7}$$

Where$R_1$ and $R_2$ are the radiuses of the external and internal measuring points, respectively (Fig. 1, Fig. 2);

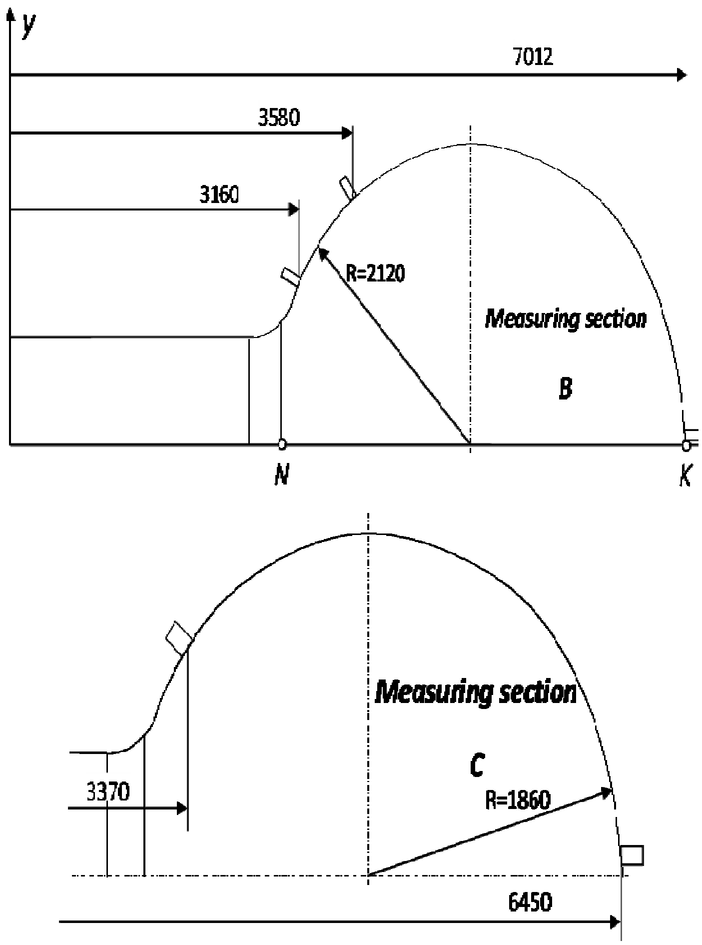


**Fig.2:** Layout of the measuring sections B and C

- $y, x$ are the coordinates for the measuring section of the spiral case (Fig. 2);
- $x_N, x_K$ - coordinates of end points of the cross section along the $x$ axis.

Actually, the equation given in [6] is a special case of equation (4) for spiral case of circular sections. The integral of equation (4) for the $k$ factor is solved by numerical integration as the integrand has been determined (the spiral case geometry is known).

The measurements are made in both measuring sections – B and C (Fig. 1, Fig. 2). The locations of these sections is in conformity with the standard requirements [7] and in order to determine their coordinates and dimensions more accurately they are selected according to the locations of the guide vanes (along the radius of blades No. 5 and No. 8, respectively, downstream of the spiral inlet section), and the standard requirements [7] have been met. Figure 2 shows schemes and basic dimensions of the measuring sections.

The determination of the flow rate value has been performed by equation (6) and the equation (7) has been used for the $k$ factor; and a value of $m=0.5$ has been used as an exponent. It is relevant to note that the flow rate value calculated by equation (6) is valid for the measuring section (actually, this is the flow rate entering the turbine vanes along the spiral range downstream of the measuring section). Therefore, the total turbine flow rate should be calculated by the following equation:

$$Q_t = Q\frac{360}{360-\varphi} \quad (8)$$

where $\varphi$ e is the angle between the measuring section and the spiral inlet section. The angle values for both sections (Fig. 1) are $\varphi_B = 67.5^0; \varphi_C = 112.5^0$, respectively (within the limits recommended by the Standard 60041[7]). The equation (8) is valid in case that the spiral distributes evenly the flow rate over the turbine guide vane perimeter (this is one of the main requirements to each turbine case [5]). The check of the spiral case of the HG1 turbine of Ivaylovgrad HPP with the theoretical layout showed that this requirement has been met.

Error estimation for the measurements has been carried out and the measuring errors for the flow rate equation have been taken into consideration. The mean relative square error of the measurements in both sections is calculated to be: $\sigma_{Q2B} = \pm 0.0154;\ \sigma_{Q2C} = \pm 0.0157$.These values meet the recommendations of Standard 60041.

## III. RESULTS

Main results of the field investigations performed on turbine No. 1 of Ivaylovgrad HPP have been presented graphically on Fig. 3 and Fig. 4. Fig. 3 shows the change of the flow rate based on various values of the generator active power ($P_g$) in the hydraulic unit operation range. This curve is valid for the measurements for mean velocity in section A (Fig. 1).

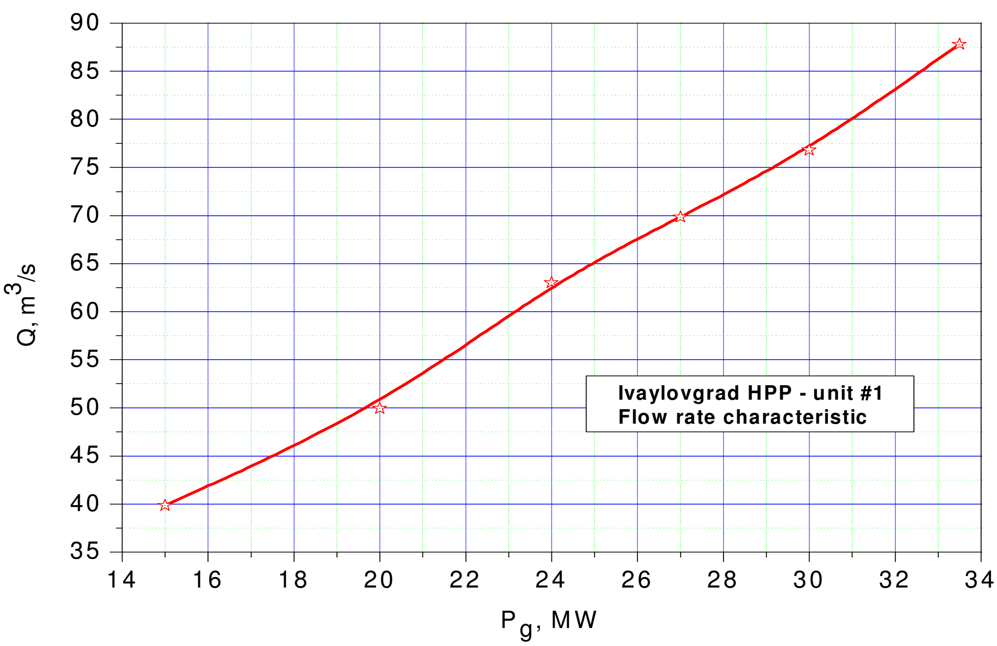


**Fig.3.** Flow rate characteristic (through measurements of the mean velocity)

Similar curve of Fig. 4 is valid for the flow rate measurements using the Winter-Kennedy method in both measuring sections (B and C – Fig. 1). For Section B, the measurements have been carried out in the B1 – B2 and B1 – B3 measuring points.

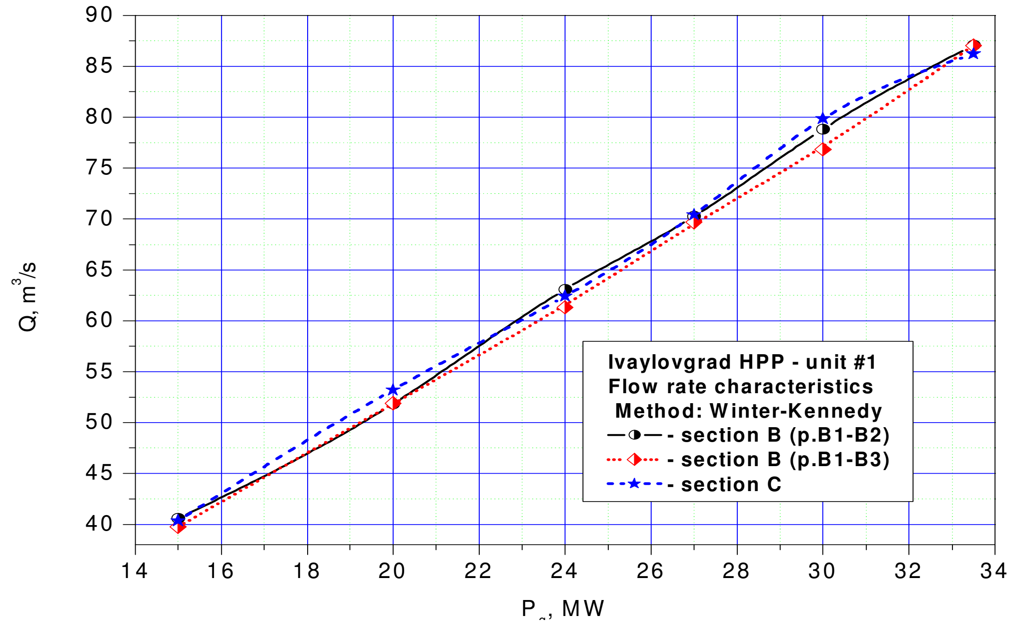


**Fig. 4.** Flow rate characteristic (acc. to Winter-Kennedy method)

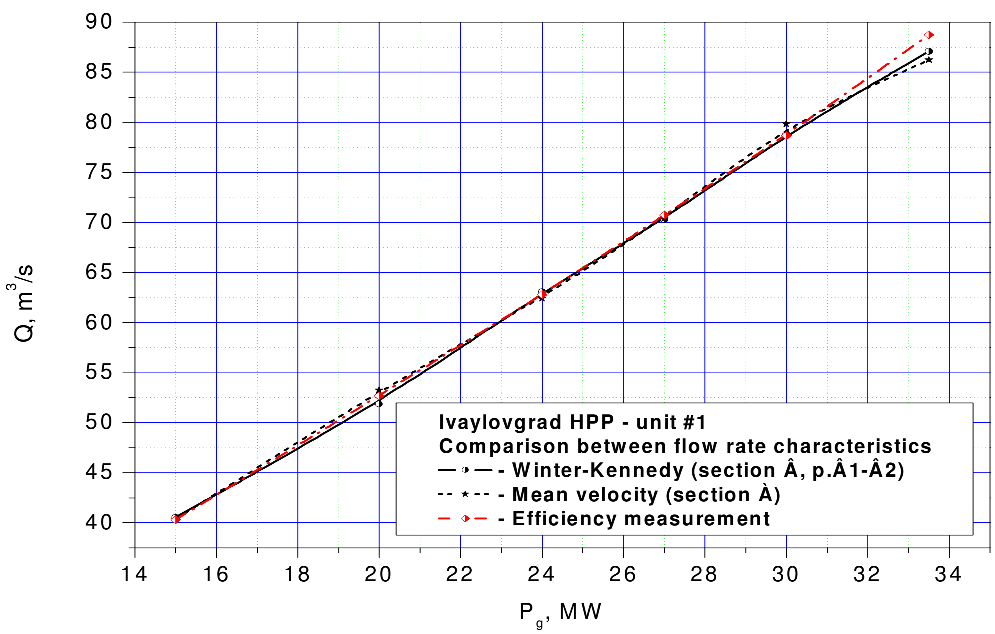


**Fig. 5.** Comparison between flow rate characteristics

Fig. 5 shows the comparison between the measuring results following both methods with the flow rate values calculated indirectly by measuring the turbine output, pressure and efficiency. It is obvious that there is good coincidence between the flow rate values in the whole investigated range. The maximum difference between flow rate values measured by the Winter-Kennedy method (section B, pointB1-B2) and through the mean velocity is $\Delta Q=0.843\ m^3/s$ (1 % of the flow rate average value); and compared to the indirect flow rate measurements the difference is $\Delta Q=1.65\ m^3/s$(1.88%). These differences are valid for maximum output operation.

## CONCLUSIONS

The main results of the field test performed on Kaplan water turbine No1 of Ivaylovgrad HPP are the following:

1. The flow rate values for the turbine have been determined over the whole output range. Two different flow rate measuring methods have been applied: by measuring the spiral case inlet section mean velocity and by the Winter-Kennedy method.
2. Measuring method has been developed and on its base the required facilities and measuring system have been designed and manufactured. Also, specialized software has been developed for measuring data processing.
3. The possibility for reliable determination of the value of the base factor $k$ in the flow equation in the Winter-Kennedy method by calculations has been proven experimentally.
4. The investigation results show that the methods and systems developed for determination of the flow rates and the experience gained can be implemented successfully to solve similar problems with various types of low-pressure water turbines equipped with spiral turbine cases.

## REFERENCES

[1] Volkov D. Spiral case of excess pressure as a flow transducer and flow rate indicator, Dissertation Wrok, Bauman MBTU, Moscow, 1953 (in Russian).
[2] Kersten I. Aerodynamic tests on mine blower installations, Nedra, Moscow, 1986 (in Russian).
[3] Kisselev, P. Determination of turbine flow rates by pressure difference in spiral case sections, GidrotechnicheskoeStroitelstvo, No. 11, Moscow, 1949 (in Russian).
[4] Kremlevskiy, P. Flow meters and quantity counters, Mashinostroenie, Sankt-Petersburg, 1975 (in Russian).
[5] Obretenov V. Water turbines, Ekoprogres, Sofia, 2008 (In Bulgarian).
[6] Schapov N. Turbine equipment for hydraulic power plants, Gosenergoizdat, Sankt-Petersburg, 1961 (in Russian).
[7] Field acceptance tests to determine the hydraulic performance of hydraulic turbines, storage pumps and pump-turbines. Standard 41 of IEC, Genève, 1991.
[8] Sheldon, Lee H. A New Form of the Calibration Equation for the Winter-Kennedy Piezometer System. Waterpower XV Technical Papers CD-Rom, HCI Publications, Kansas City, Mo., 2007.
[9] Urquiza G., Basurto M., Castro L., Adamkowski A., Janicki W. Flow Measurement Methods Applied to Hydropower Plants. Flow Measurement, chapter 7, pp. 151-167, Intech, 2012.
[10] Winter I.A. and Kennedy A.M. Improved Type of Flow Meter for Hydraulic Turbines. ASCE Proceedings, Vol.59, No 4, Part 1, April 1933, pp. 565-584.
[11] Bilal Abdullah Nasir, "Design of High Efficiency Pelton Turbine for Micro-Hydropower Plant", International Journal of Electrical Engineering & Technology (IJEET), Volume 4, Issue 1, 2013, pp. 171 - 183, ISSN Print : 0976-6545, ISSN Online: 0976-6553.
[12] Naveen Rathi, "Estimation of Misalignment in Bearing Shaft by Signal Processing of Acoustic Signal", International Journal of Mechanical Engineering & Technology (IJMET), Volume 2, Issue 1, 2011, pp. 60 - 69, ISSN Print: 0976 – 6340, ISSN Online: 0976 – 6359.
[13] Valentin Slavov Obretenov, "Analysis of the Load of Pelton Water Turbine Shaft and Guide Bearing", International Journal of Mechanical Engineering & Technology (IJMET), Volume 4, Issue 5, 2013, pp. 257 - 265, ISSN Print: 0976 – 6340, ISSN Online: 0976 – 6359.