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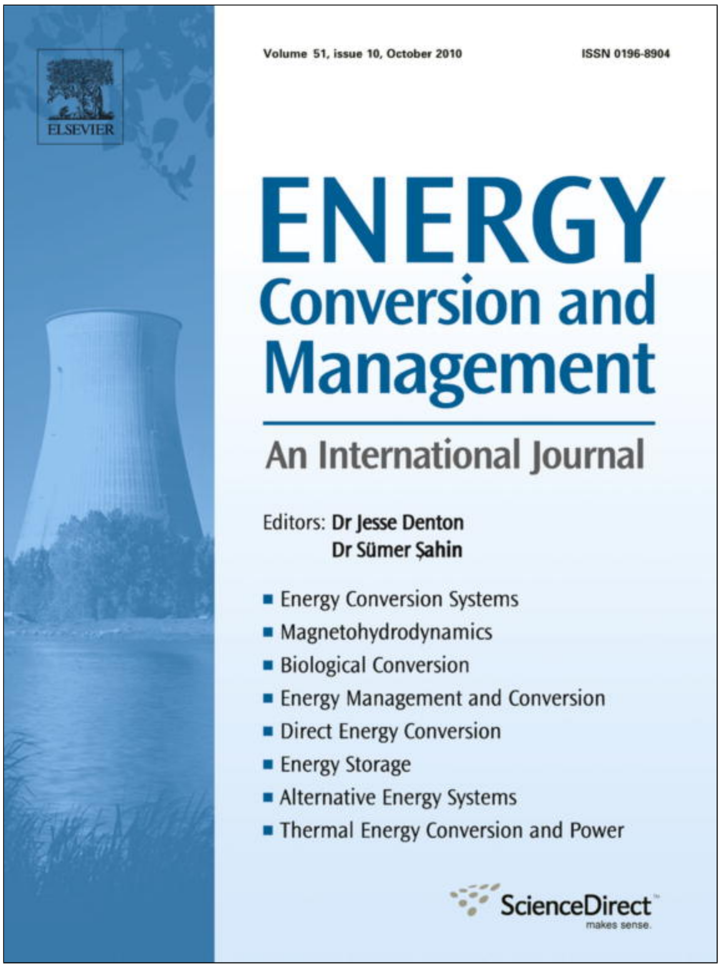

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Energy Conversion and Management 51 (2010) 1970–1975

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Improvement of the efficiency of the Agnew micro hydro turbine at part loads due to installing guide vanes mechanism

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A R T I C L E I N F O

Article history:
Received 23 June 2009
Received in revised form 7 February 2010
Accepted 27 February 2010
Available online 3 April 2010

Keywords:
Agnew turbine
Guide vanes mechanism
Second support
Main shaft
Second rest point
Part-load efficiency

A B S T R A C T

Agnew turbine is an axial micro hydro of Kaplan type, with its main shaft subtending 45° to the line of horizon. Due to governmental power generating programs for limited hydro potentials in Iran and on the basis of a joint project between the University of Glasgow and the Iranian Research Organization for Science and Technology (IROST), the turbine was developed to operate under low heads and limited flow potentials in Iran. However the turbine was originally designed to operate without any guide vanes and test results showed that the turbine achieved an efficiency of 62% at its best point of performance. Later a modified Agnew turbine, consisting of a guide blades mechanism was designed and manufactured, at (IROST). The mechanism was so designed that it was also used as a second support for the turbine's main shaft. The standard turbine tests proved that; the modified version of the Agnew turbine had 23% higher efficiency at its best performance point; compared to the original design. Then the effects of the improvements on the turbine were studied on the performance of the turbine at part loads. However, efficiency improvements were observed under all part load conditions.

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1. Introduction

Application of micro hydros has gained world wide attention during the last decades of the 20th century. Although micro hydros are by no means comparable to large hydro plants in so far as their power production capacity is concerned yet their simple design and relatively simple manufacturing processes, low price per kilo watts, easy installation without requiring heavy construction activities, cheap and easy maintenance, and their minimal or rather non-riverine impacts have made them attractive especially to the countries with ample micro hydro sites (13,100). A pre-requisite to obtain power from large hydro potentials is construction of large scale dams as well as all other back up buildings and installations for the purpose of generation, regulation and transmission of power. They require heavy mid term and long term investments, as well as involving lengthy and time consuming constructional processes. On the other hand building a large dam on the path of a river might have adverse effects on the echo systems before and after the point where the dam is erected. The lake behind the dam can flood endless valuable agricultural lands and also deprive the agricultural activities from the water they were used to for centuries. These parameters would therefore make large hydros unfavorable for application [13]. A viable alternative to large hydros is application of micro hydros in extensive dimensions.

Iranian government's programs for tackling the energy crisis, guided the ministry of power and the ministry of Jihad towards scenarios on decentralization of power generating schemes which in turn lead to programs for establishing small and micro hydro power plants all over the country [3]. Ministry of Jihad embarked upon a joint venture with Chinese government for establishing micro hydro plants, a number of which are at present operating at different points of the country [10]. Since Chinese have the leading role in the field of micro hydros the program was quite successful. They started their work on micro hydros in early 1950s with developing four different types of micro hydros [12], and they managed to establish more than 60,000 micro hydro power stations around China during 1990s [11,12].

As the largest reference research center in Iran, IROST started a program for undertaking research projects to gain the most possible hydro energy of the micro hydro potentials, by designing and developing suitable micro hydro power stations for thousands of micro hydro sites scattered all over the country [2,9]. For this purpose a joint program was undertaken between (IROST) and the University of Glasgow for developing a micro hydro suitable for micro hydro potentials in Iran [5]. Further studies of the results of the field researches by the ministry of Jihad along with other works such as the chart presented by Ramos et al.; showing operating range of various types of hydro turbines, which could help selection of turbines for given hydro potentials helped selecting the Agnew turbine as the main subject of this program [9,10,14,15]. Although a number of these turbines had already

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doi:10.1016/j.enconman.2010.02.029

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been installed at various hydro sites in Scotland yet, due to lack of governmental financial supports no research work was performed on the turbine [1,2].

An important point to be considered when designing a micro hydro turbine for a country like Iran is the water required to run the turbine and the effects of flow variations on its behavior. The importance of this; rises; when noticing the fact that micro hydros mostly do not require a water reservoir(or a dam)upstream or at their operation site, and instead they need a permanent water flow to guarantee their power generation capacity.

Generally in Iran, the raining season begins around the middle of autumn and continues into winter. Then begins the snow fall, especially at high lands and again towards the end of the winter the raining season continues through to the middle of the spring. Then the hot and dry season begins almost all over the country and continues until half way through the fall. Therefore tangible fluctuations at water levels of rivers or even man made canals may be observed from a season to another or even from one month to another. Due to these fluctuations and in the absence of an upstream reservoir at micro hydro sites which can guarantee a constant water flow only in good raining years, these turbines can not operate under constant full load all year round. So it would be essential to study the turbine performance under the effects of variable operating conditions such as part loads.

More over, hydroelectric power plants are also subject to off-design operation in order to follow the demand which would naturally require higher performances at part loads [24]. Therefore for the purpose of improving the performance of different hydro turbines under part load conditions research works have been under taken by different research bodies [19–22].

In general part-load efficiency may be defined as an indication of the efficiency of the turbine when operating at a load a percentage loading below the turbine's rated power output. Since turbines operate at or near their highest efficiency when fully loaded, the part-load efficiency is useful to get an idea of operating efficiency for a turbine producing less than its rated power output [16].

Agnew micro hydro is a 45° Kaplan turbine which inherits some of the characteristics of the larger vertical shaft Kaplan design such as moderate to high operating points [23]. Although Kaplan turbines operate at relatively high efficiencies under part load conditions [17,18], but whether the Agnew turbine follows the same pattern required further research work.

In this work the behavior of the Agnew turbine has been studied at part loads of 25%, 50%, 75% and 100% of the full load, before and after improving the turbine. Consequently a comparative appraisal has been established between the two cases, in order to represent the effects of the turbine improvements on its performance at part loads. The outcomes of this study can represent the degree of suitability of the turbine for the wide variety of micro hydro potentials scattered throughout the country since the turbine may have to operate at off- design conditions due to deficiency of resources or variable power rig demand. They can also show whether the behavior of the Agnew turbine at part loads confirms with that of the Kaplan turbines.

2. Agnew turbine

The turbine is a 45° axial flow Kaplan type micro hydro which originally was designed and manufactured in the UK. The turbine was designed by Mr. Pat Agnew an ex-lecturer of the department of Mechanical engineering of the Glasgow University. It was tested at the British national laboratories and the test results maintained that, 62%, was the highest performance attained by the turbine [2]. Therefore due to relatively poor performance of the turbine it was decided to perform a joint research project on improving the turbine at IROST. However, the joint program consisted of; manufacturing the turbine according to its original design, and testing it according to the current standard turbine test procedures [7]. Then, at the later stage an improvement program was defined for the turbine which consisted of design, manufacture and installation of a guide vane system using its support as a second rest point for the main shaft of the turbine. The main shaft was originally designed to operate as a cantilever; supported by some bearings installed in housing connected to the turbine [4]. The same standard tests were then performed on the improved model of the turbine, results of which showed a 23% improvement on the highest performance point of the turbine [5]. Figs. 1 and 4b show schematic views of the Agnew turbine.

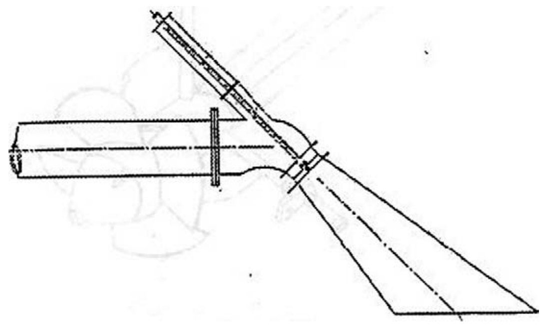

Fig. 1. Cross sectional side view of the Agnew turbine.

Obviously due to the variations of seasonal rainfall and snow in Iran the hydro potentials vary in terms of their flow rates [9]. This would in turn mean that micro hydros can not operate under full load all year round. This is due to the fact that, micro hydros are not usually supported by a dam on their upstream, and without such reservoir obtaining full required flow rate all year round is difficult to achieve. Therefore the turbine may have to operate under different part loads at different times of year. This would advocate the idea of studying the behavior of the turbine under part loads so that seasonal regulations could be anticipated for obtaining the best possible performance of the turbine. Therefore, this study was undertaken for the Agnew turbine before and after improvements, the results of which are compared, appraised and presented in this article.

3. Apparatus

Obviously any changes and improvements on the turbine required essential apparatus to test the turbine in order to determine its performance and compare them with those of the original design. The test circuit design and selection of its elements, and their characteristics such as circularity and cylindricality of pipes, free lengths before and after installation points of measuring devices, required straight run before the entrance to the turbine etc. were all determined according to the ISO 5167-1980 (E) standards [15]. However, the details of the test rig and its measuring devices are as follows:

A 20 m^3 reservoir provides the water requirements of the test rig. The piping is of 25 cm diameter all through. The collector into which the water is pumped from the reservoir is a 4 m long cylinder of 50 cm diameter, which is connected to the main piping via a 2:1 semi-conical reducer. Six 11 kW electro pumps provide the required water flow for the test rig. Pumps are installed parallel to one another, generating heads up to 24 m and an overall flow rate of 420 m^3/h.

The flow is measured by means of an ultrasonic flow meter in (l/s) with a precision range of 0.5% for linearity, 0.0015% m/s of sen-

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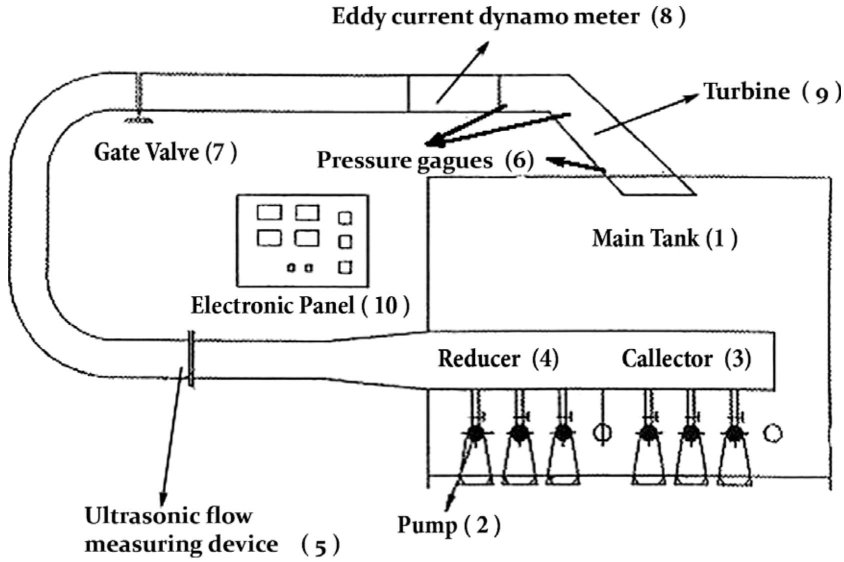

Fig. 2. A schematic view of the test rig.

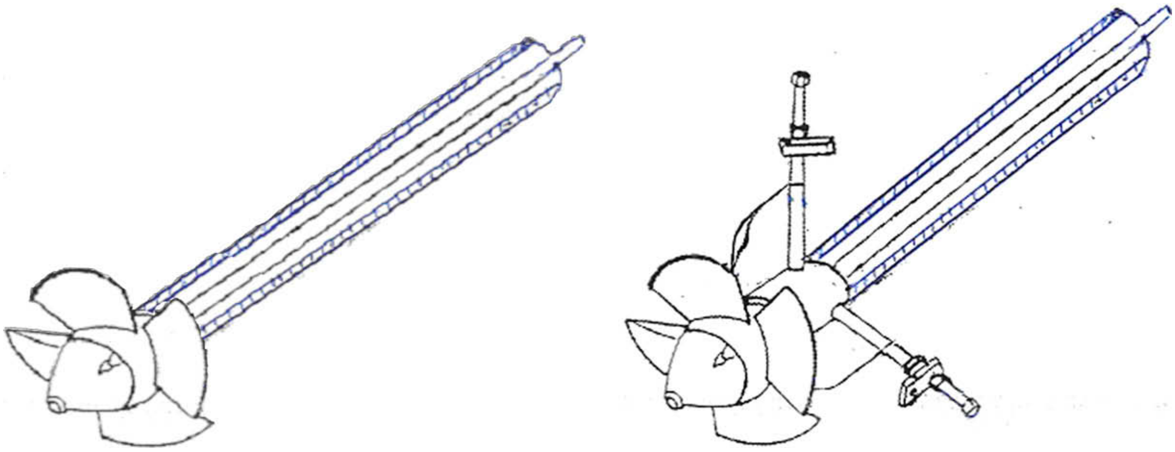

Fig. 3. (a) The main shaft before improvement. (b) The main shaft after improvement.

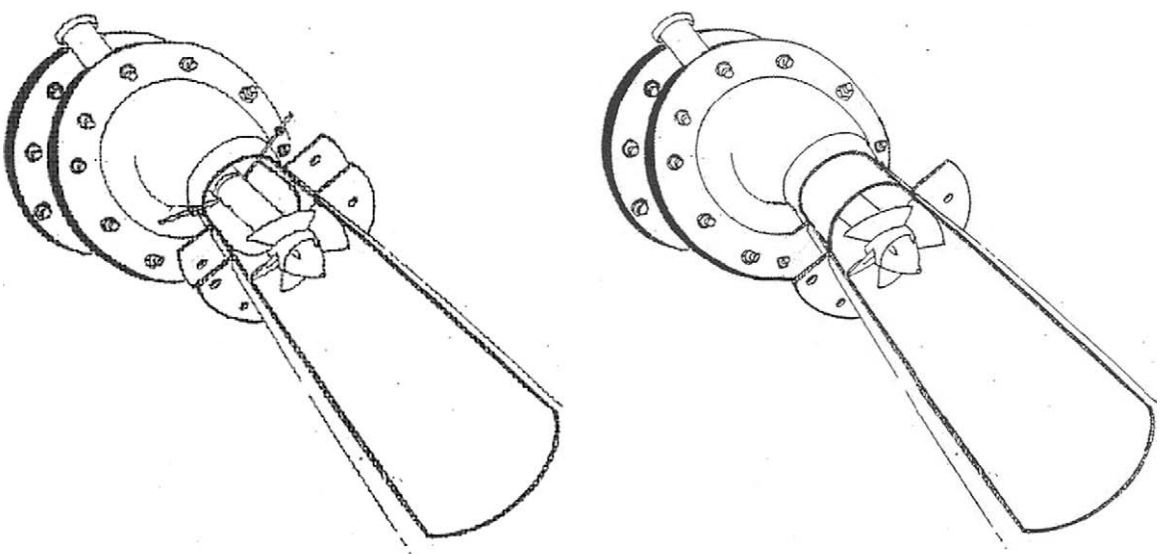

Fig. 4. (a) Cut through schematic view of the turbine after installation of the guide vanes. (b) Cut through schematic view of the turbine before installation of the guide vanes.

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sitivity, 0.25% repeatability, and overall flow measuring accuracy range of 1–3%, manufactured by The Iranian Farasanj Afzar company. The pressure gauges were manufactured by the American Honeywell Co. which were installed at the inlets and outlets of the turbine and the draft tube to measure the effective head on the turbine and the suction head of the draft tube respectively, in (mbar). Accuracy range of these gauges is ±2% given by the manufacturer. A 20 kW Eddy current dynamometer was applied to measure the brake force of the turbine in terms of (*N*). The dynamometer was designed and manufactured at the mechanical engineering research center of IROST. The accuracy of the load cell measuring the brake force of the dynamometer is ±1%, manufactured by the Korean Bong Shin Company. The angular velocity of the turbine is measured in terms of (rpm) by means of a tachometer, using a magnetic induction sensor of ±0.5 accuracy range manufactured by The Iranian Tabriz Pajooh Co. with an overall accuracy range of <0.001%. Fig. 2 shows a schematic view of the test rig. However, the accuracy of the devices and their precision range and their over all effects on the precision of results of the test rig were taken into account according to current procedures when writing the computer software for analyzing the measurements of the measuring devices. Fig. 2 represents the schematic view of the test rig [6].

4. The main shaft

As indicated earlier the main shaft of the turbine is supported by three bearings, installed inside housing outside the main turbine casing and connected to it. The main shaft subtends 45° to the line of horizon, and holds the hub and the runner assembly. An operating rod passes through the main shaft, which operates the angle of runner blades for changing their angle of attack between 0° and 30° around the radius perpendicular to the main shaft. The 30° blade angle condition is called "fully open blades" and 15° blade angle conditions is called "semi open blades". However, the unsupported part of the main shaft is left without any rest point inside the turbine casing. It should also support the hub, the runner, the blades and the mechanism for changing the blade angle installed inside the hub. Also it has to withstand effects of large volumes of water passing through the runner. All in all, these factors tend to escalate a lack of stability in turbine's operation, and therefore increase the vibrations at the free end of the main shaft. This would in turn escalate the danger of collision of the runner blades with the turbine casing, causing efficiency drop, noise, and vibrations. Therefore it seemed appropriate to improve the design of the main shaft so that it could rest on a second support inside the turbine casing, while supporting the guide vane mechanism [4,15].

5. Guide vanes mechanism

For the purpose of installing the guide vanes a suitable support was designed. It was so designed to hold the guide vanes and their operating mechanism as well as serving the main shaft as its second rest point. However, to avoid redesigning and remanufacturing the main shaft of the turbine, an extension to the main shaft was designed and manufactured. This extension was installed at the free end of the main shaft in place of the runner. Instead the runner was installed at the free end of the extension. Fig. 3a and b show the shaft before and after improvement.

The extension consists of three parts. The outer casing which is stationary and acts as a support for the guide blade rods, in fact the outer casing is held in its place by means of these three rods, which connect the outer casing to the turbine casing and hold the guide vanes at the same time. Each rod is separated by 120° from another and is able to turn between 0 and 10°, in order to change the guide blades direction with respect to the flow direction. Apart from holding the guide blades they also support the main shaft, performing as its second rest point. The inner part is actually screwed into the free end of the main shaft and rotates with it. A needle bearing protected by two watertight seals operate between these two parts. Fig. 4a and b represent a cut through schematic view

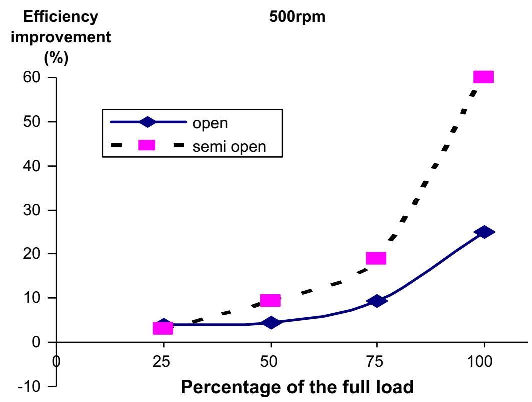

Fig. 5. Percentage variations in the values of the efficiency of the turbine at different part loads for the speed of 500 rpm.

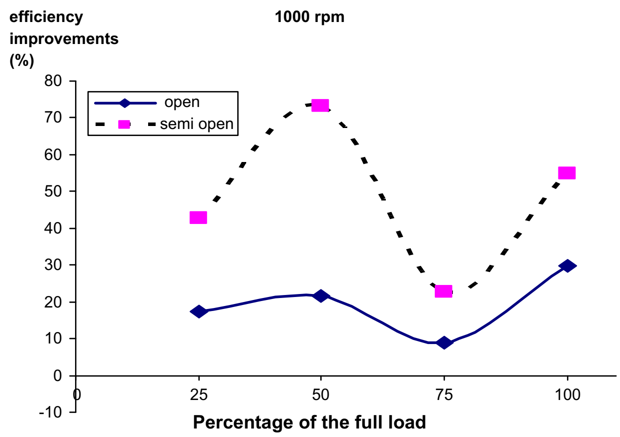

Fig. 6. Percentage variations in the values of the efficiency of the turbine at different part loads for the speed of 1000 rpm.

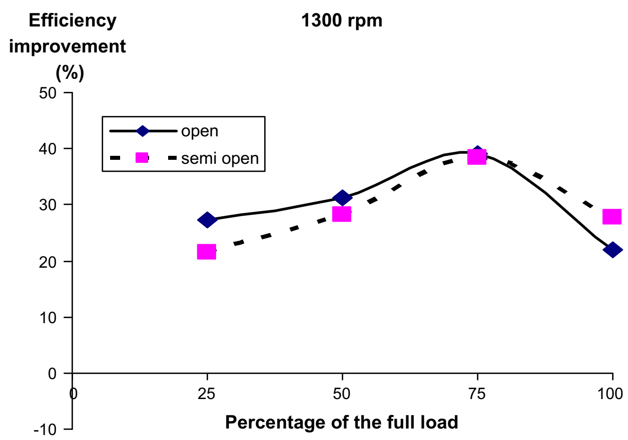

Fig. 7. Percentage variations in the values of the efficiency of the turbine at different part loads for the speed of 1300 rpm.

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of the turbine after and before installation of the guide vanes mechanism inside the turbine respectively.

6. Calculation procedure & formulas

To obtain the values of efficiency for various operating conditions of the turbine the following formulae were used [7,8]:

$$Po = \omega \cdot T \cdot \pi/30 \quad \text{(W)}$$

where ω = Angular velocity (rpm), T = Torque (Nm).

Also, $N = 2 \cdot \pi \cdot \omega/60$.where N = Turbine speed (r/s).

And

$$Pi = \rho g Q H$$

Pi = Input power (W), g = 9.81 (m/s^2), Q = volumetric flow rate (m^3/s), H = Available head (m).

Also, the overall efficiency is given by:

$$\eta t = Pi/Po$$

7. Testing procedure

Turbines in the field must be supplied with natural flow of water (Q) under some head (H). However in a laboratory both Q as well as H are obtained by a suitable pumping unit. The pump draws water from a sump of sufficient capacity, and delivers it under a specified pressure (H) to the turbine.

The pump is started and the delivery valve of the pump is opened fully in order to supply water to the turbine. The inlet butterfly valve to the turbine is then opened. The turbine is run for fifteen to twenty minutes before the readings are recorded. The brake cooling water tap is opened, so that over heating of the braking parts of the turbine is avoided. The turbine is set for a number of inlet openings, at least four, say 25%, 50%, 75% and 100%. For each inlet opening, the turbine is first run and the break is applied with various loadings and torque readings along with all other necessary readings are taken each time until the turbine is at standstill. Not less than six different loadings for a single inlet opening, are to be recorded .The head operating the turbine is kept constant as far as possible during testing by adjusting the delivery or regulating valve of the pomp or by varying the speed of the driving motor [7].

However, tests were taken for different settings of the runner blades, namely fully open and semi open, for various flow rates (represented by the number of pumps in operation) and different heads, according to the standard turbine test procedures [7].

In order to have reasonable bases for comparison, both versions of the main shaft were installed on the turbine and were tested, under the same conditions.

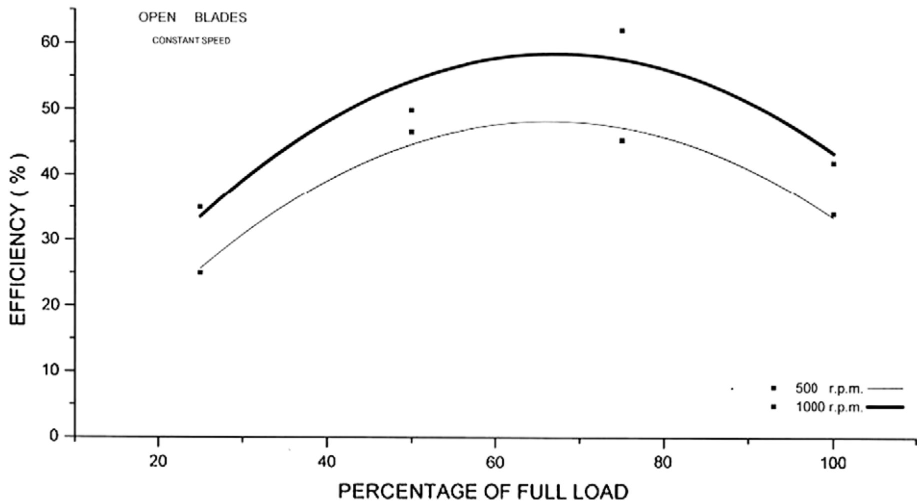

Fig. 8. Curves of variations of efficiency against percentage of the full load, before installing guide blade mechanism for open blades.

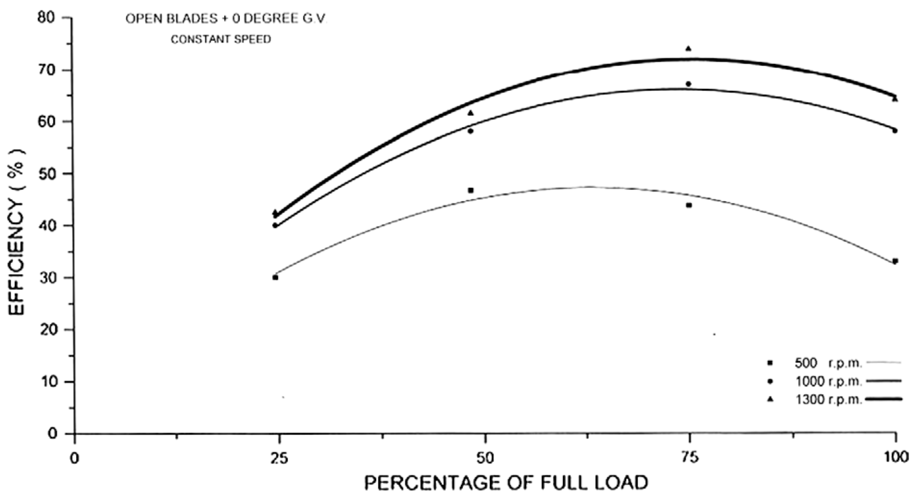

Fig. 9. Curves of variations of efficiency against percentage of the full load, after installing guide blade mechanism for open blades, guide blades at 0° opening.

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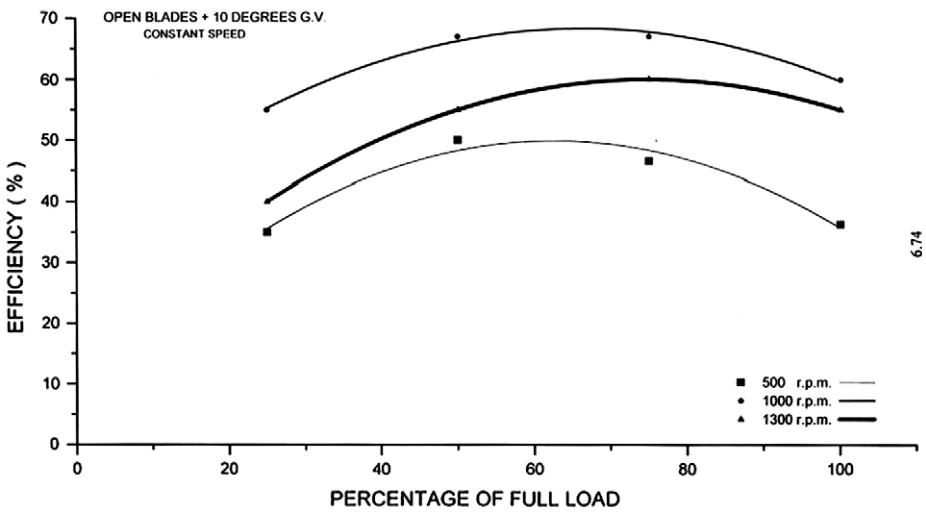

Fig. 10. Curves of variations of efficiency against percentage of the full load, after installing guide blade mechanism for open blades, guide blades at 10° opening.

8. Results

For the purpose of studying the effects of installing the guide vanes mechanism; serving also as a second support for the main shaft of the turbine; on the efficiency of the turbine at part loads, values of efficiency for different blade settings, namely; open, and semi open, were calculated at constant speeds of 500 rpm,1000 rpm, and 1300 rpm. Then comparative analysis was established between the efficiency results obtained for the same blades conditions, at different above mentioned speeds and various part load conditions. The appropriate curves represented by Figs. 5–7 were then plotted. These curves actually show the percentage variations in the values of the efficiency of the turbine at different part loads. The formula used to calculate the percentage efficiency variations is as follows:

$$\tilde{I} = \{(\eta A - \eta B)/\eta B\} \times 100$$

where $\tilde{I}$ = Percentage of the efficiency variations, ηA = Efficiency after improvement of the main shaft, ηB = Efficiency before improvement of the main shaft.

9. Conclusions

As observed from the Figs. 5–7 in all cases the values of variations in efficiency are of positive magnitude and therefore it could be deducted that, improving the Agnew turbine by adding the guide vanes and a second rest point to its main shaft can result in improving the performance of the turbine under all part load conditions, regardless of the amount of the load.

Also, Figs. 8–10 represent samples of curves of variations of efficiency against percentage of the full load, before and after installing guide blade mechanism, for different settings of the guide vanes.

It may also be concluded that, although the behavior of the Agnew turbine at part loads is close to that of the Kaplan turbine but the flatness of the curves is not as expected and they tend to follow a downward trend especially towards the higher loads [18,25]. However, this effect improves as the turbine speed increases towards the nominal required speed for electric generators (see Figs. 8–10). Therefore it may be conclude that at higher turbine speeds the turbine behavior at part loads is very close to that of the Kaplan turbine.

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