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Experimental Study on Performance of a Hydraulic Francis Turbine with Variations of Runner Speed

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***Abstract* - This investigation was carried out to evaluate the characteristics of a Francis water turbine. Several variations of inlet valve openings of the turbine were tested to determine the effects of changes in rotational speed of runner on turbine performance. The method used in this study was experimental measurement using a test rig installation in Diponegoro University. The results found that discharge was not change significantly with increasing speed. The torque decreases as the turbine rotational speed increases. The break horse power (BHP) and efficiency curves are nearly identical. Both BHP and efficiency increase when speed are increased until the maximum condition is reached, then from this maximum condition, as speed increase, the BHP and efficiency gradually decrease.**

Keywords: Discharge, efficiency, Francis turbine, runner speed.

I. INTRODUCTION

The flow conditions entering the runner blade of the Francis turbine greatly influence the turbine's performance. Changes in speed and pressure will create flow instability in the turbine. If the instability is strong enough then this will create quite large pressure fluctuations. Apart from that, drafttube is also a main aspect in determining turbine performance. The emergence of cavitation is also an important aspect that will affect turbine performance. As the turbine operates, it is inevitable that there will be deposits and sedimentation from particles that are carried along with the flow when they enter the turbine. Abbas and Khumar [1] tested turbine models under various flow conditions and found that the efficiency measurement uncertainty under optimal efficiency conditions was ± 0.1411%. An interesting trend can be seen in each energy coefficient where the uncertainty at the part-load point is greater than at the overload point. Altimemy et al [2] created a vortex flow simulation to predict flow behavior and pressure fluctuations in a turbine. The results of the study showed that central injection with a rate of > 4% was very effective in reducing pressure spikes caused by flow. The level of fluctuation decreased by more than 40% and 75% at the probe closer to the drafttube entrance with central injection flow rates of 4% and 6% during partial load with water injection not being effective in stabilizing the system operation at this load operation.

Arispe et al [3] numerically parameterized the drafttube elbow. The results found that the drafttube in hyperbolic-logarithmic spiral format had the highest efficiency and the logarithmic spiral drafttube had the lowest loss coefficient. Chitrakar et al [4] developed a test tool containing three inlet guide vane (IGV) blades with the flow field adjusted according to actual turbine conditions. The results of the study show that the use of three IGV blades provides optimal capacity and has minimum influence on the walls surrounding the IGV blades. The effect of blockage of runner blades in turbines was studied by Kim et al [5]. Here the flow analysis for off-design conditions is also made with various blade thicknesses. The results show that the power and efficiency gradually decrease as the blockage ratio increases. Efficiency decreases by approximately 3.4% as the blockage ratio increases by 12.5%.

The vulnerability to erosion of the IGV blades of the Francis turbine was studied by Khullar et al [6]. The results found an increase in thickness reduction along with longer operating time. The maximum thickness reduction position shifts towards the edge of the tailings for the guide vane and towards the center for the faceplate. Increasing the cleaning distance causes higher cross-flow across the cleaning gap and so the erosion rates feedback. Erosion of the guide vane and face plate is symmetrical at BEP but asymmetrical at part-load. Xu et al [15] tested the torque characteristics of IGV blade protection devices. This study covers the operating mechanism and torque distribution mode of the IGV blade protection device. The results show that the torque distribution continues to decrease as the load increases, whereas the shear pin distribution ratio increases.

Sun et al [8] examined the spatiotemporal evolution of turbine blade inter-blade vortices and their contribution to the generation of pressure fluctuations at low heads. The results show that the steam volume associated with the interblader

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vortex experiences periodic pulsations, with a frequency close to the runner rotation frequency. The instability of pressure fluctuations in a hydraulic turbine during steep slopes in a high-head turbine was studied by Trevedi et al [11]. The results reveal that the characteristic frequency amplitude, especially the rotor-stator interaction, is small during steep climbs, however, at the end of the transient cycle, the amplitude quickly increases 30-fold. During steep slopes, blade displacement frequency appears in the runer. Furthermore, Trivedi et al [12] measured the pressure in a low specific speed Francis turbine with a vertical axis turbine model and a horizontal axis type. The results show that, in the vertical axis turbine, the amplitude of the asynchronous pressure pulse is 20 times greater than that of the synchronous component; whereas in horizontal axis turbines, the amplitude of asynchronous pressure pulsations is two times smaller than that of synchronous components. Then, Trivedi et al [13] studied unsteady pressure loading on a turbine model at varying speeds. The measurement results show that, in the space without blades and runners, the amplitude of unsteady pressure fluctuations increases with the angular velocity of the runner. The pressure field at the trailing edge of the blade is strongly influenced by the draft tube flow at partial load and low load.

Yu et al [16] made simulations with a shear stress turbulence model for turbines at various flow separation conditions. The results show that the draft tube is the main part that produces the largest energy losses compared to other parts. Yu et al [17] analyzed flow losses in turbines using the entropy production method. The results of his study found that turbine hydraulic losses are closely related to flow separation and vortex movement and backflow. Meanwhile the draft tube contributes the largest proportion of entropy production followed by the runner. Yu et al [18] tested the effect of rotation speed with low head. The results found that varying speed can increase efficiency by up to 10%. For deep load conditions, while providing the same power output, it can save up to 14% discharge thanks to increased efficiency.

Stable and unstable cavitation flows in Francis turbines, based on a two-phase mixed model were tested by Laouari and Ghenaiet [7]. Part-load operation gives rise to undesirable pressure pulsation phenomena due to vortices generated at the runner outlet. In addition, pressure fluctuations and torque oscillations appear to be more pronounced under part-load and over-load operating conditions. Trivedi [14] examined the characteristics of cavitation and unstable pressure fluctuations when the turbine ramps up, to meet energy requirements. The investigated Francis turbine consists of 15 blades and 15 splitters, and the reference diameter is 0.349 m. At a certain moment, the cavitation effect is very dominant, and due to the rupture of the cavitation bubble, the pressure fluctuation becomes very high.

The velocity conditions at the runner inlet of the turbine due to an increase in the clearance gap were characterized by Thapa et al [9] by measuring the pressure and velocity in the guide vane (GV) cascade. At a clearance gap (CG) of 2 mm, the relative velocity approaching the runner inlet casing was found to increase up to three times the nominal value. Vortex sheds were found to develop due to the mixing of the leaking flow with the main flow. Thapa et al [10] investigated the flow conditions around low specific speed turbine IGV blades. It can be seen that the simplification made by replacing the spiral casing with a flat plate changes the tangential velocity at the Guide Vane inlet position by 16%. However, as the flow progresses and approaches the runner inlet position, the average tangential speed approaches that of the prototype turbine, with a difference of less than 4%.

Many studies have been carried out both experimentally and using numerical approaches. However, there are still many parameter aspects that have the potential to increase pump efficiency. So further studies need to be carried out to obtain more detailed information regarding improving pump performance. Therefore, this study was created to determine the performance of turbine. Experimentally, the effects of runner speed on the flow rate, head, power and efficiency were measured.

II. METHODOLOGY

2.1 Francis Turbine Model

In this experiment, a Francis turbine made by GUNT Germany was tested as in Figure 3.1. This turbine model has dimensions of 4 x 4 x 6 meters, weight ± 17 kg, power = 6 W and rotation = 100/min. The flow rate for this turbine is 27 L/min with a head of 5 meters. This turbine has six vane units. Water enters through the inlet pressure tube and then flows into the spiral duct to form a vortex and rotate the impeller blades. The impeller is connected to a shaft which produces torque.

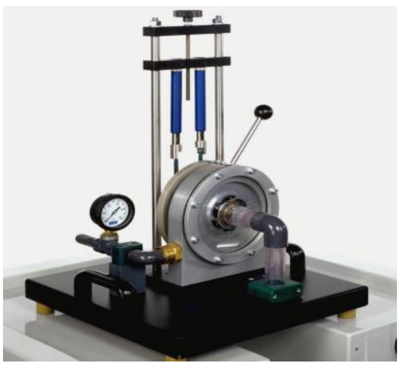

Figure 1: Francis turbine test model

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2.2 Module for Data Measurement

This francis turbine model was tested by using the basic module as shown in Figure 2. In this unit there are two types of tanks, namely storage and volumetric tanks. The storage tank has a capacity of 180 L while the volumetric tank has a maximum capacity of 40 L. In this module there is a submersible pump in the storage tank. This is used as an inducement so that fluid flows from the storage tank to the tested pump above. In the submersible pump delivery pipe, there is a control valve to regulate the flow rate to the tested pump. This base module is also used to measure the flow rate delivered by the pump to the volumetric tank using a remote sight gauge as shown in Figure 2.

Figure 2: Basic module for experimental measurement

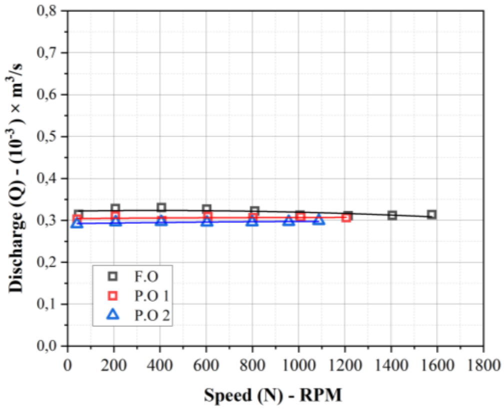
2.3 Instrumentation

Some instruments for this measurement: First, Manometer: this manometer is used to measure the inlet pressure. This type of manometer is Bourdon tube RKG 63, with a measurement range from -1 bar to 1.5 bar. The accuracy of the manometer is based on EN 837-1. The diameter of this manometer is 63 mm with a connection size of G ¼”.Then, remote sight gauge: this tool is a scaled pipe column connected to a volumetric tank. This tool is used to measure the volume produced by the pump per unit time. Stopwatch: this tool is used to measure the time required for the water flow out of the pump to fill the volumetric tank to the specified volume. Tachometer, this device is used to measure the rotational speed of the turbine shaft. The specification of tachometer used has an accuracy of ±0.05% + 1 digits with a measurement range of 2 to 20,000 RPM. The measurement distance from the end shaft that can be detected by this tachometer is 50 – 500 mm.

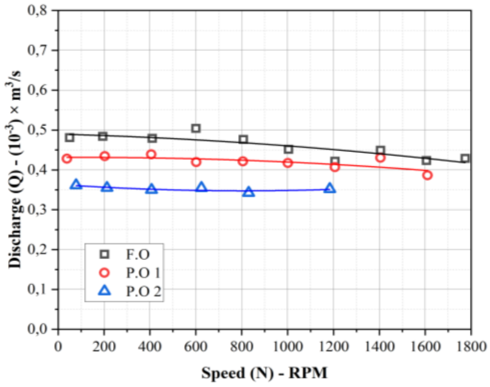
III. RESULTS AND DISCUSSIONS

3.1 Flow rate

Figures 3(a) and (b) show the discharge measurements as a function of rotational speed at two variations of inlet guide vane (IGV) position = 4, and 8. Generally, it can be seen that the amount of discharge produced is directly proportional to the opening position of the IGV and inlet valve opening. The three curves at both plots show that the largest discharge in this measurement is achieved at fully opening condition (F.O). In addition it can be noticed that the inlet opening valve at PO 2 condition results lowest discharge compared with two others due to opening of pipe from the water source becomes reduced.

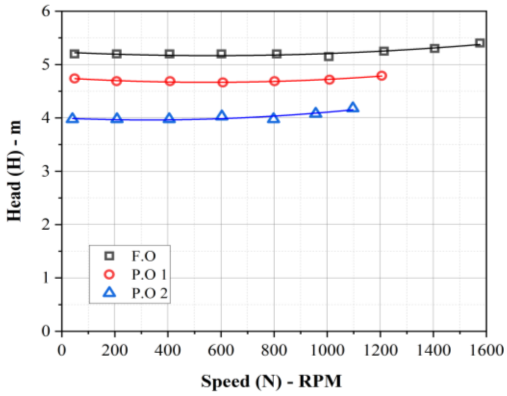


(a)



(b)

Figure 3: Flow rate as a function of runner speed at position (a) IGV = 4 and (b) IGV = 8



(a)

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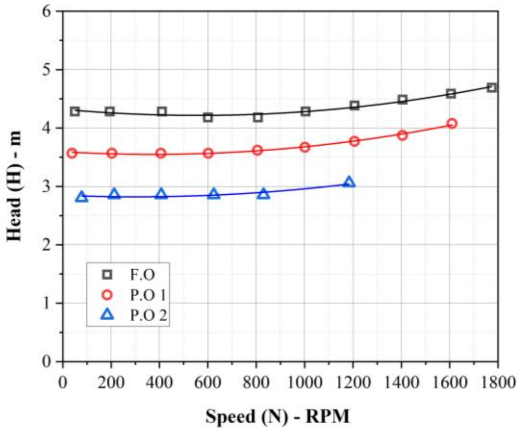


(b)

Figure 4: Head as a function of runner speed at position (a) IGV = 4 and (b) IGV = 8

3.2 Head

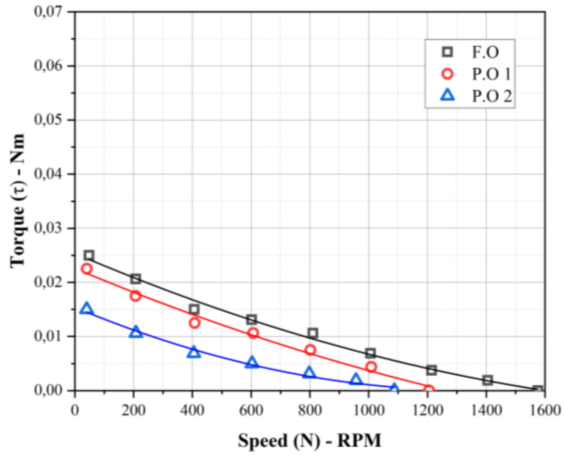
The measurement results on head characteristics as a function of rotational speed of runner at the three inlet valves for two opening positions of IGV = 4 and 8 are represented in Figures 4(a) and (b). Based on the plots, it can be noted that the head value decreased slightly before finally increasing to the optimum efficiency condition. The highest head value was achieved at fully opening of inlet valve (F.O) for IGV = 4 at head approximation 5.40 m and at rotational speed of 1600 RPM. In addition, it is also found that the size of the inlet valve opening is directly proportional to the resulting head value, this is because when the inlet valve is full opening the incoming head is the maximum head of the pump, when the inlet valve is partially closed there is a head loss so that the head received by the turbine is reduced.

Furthermore, it can be seen that the larger the IGV opening, the smaller the incoming head, due to an increase in large losses due to friction which reduces the efficiency of converting hydraulic energy into mechanical energy by the turbine. In addition, if the IGV opening position is increased, the water flow entering the turbine becomes wider, this can cause hydraulic energy to be spread throughout the flow cross section, not just in the central area of efficient flow. As a result, there is a waste of hydraulic energy as most of the energy is wasted in inefficient water flow.

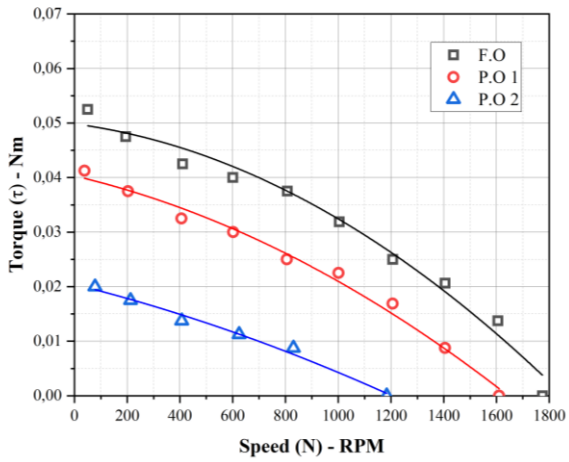
3.3 Torque

Figures 5(a) and (b) are plots of experimental data for torque as a function of speed for two variations in opening position IGV = 4 and 8. In general, based on these plots it can be seen that the torque value decreases along with increasing turbine speed for all IGV position variations were tested also for the three inlet valve openings being compared. The greatest reduction was achieved in the fully opening of inlet valve (FO) in the three variations of IGV opening as seen in the three plots produced. At position of IGV = 4 there was a relatively smaller decrease in torque values at all rotational speed tested compared to the decrease in torque at IGV position = 8 for the three valve openings tested.

Additionally, it is also shown that the largest torque value that can be achieved by the Francis turbine just before the hydraulic power is no longer able to rotate the shaft occurs at fully opening of inlet valve (F.O) for variations in the opening positions of IGV = 8 of T = 0.0525 Nm at rotational speed of 50 RPM. Meanwhile, the torque value to produce the highest speed occurs at fully opening valve (F.O) for an opening position of IGV = 8 of T = 0 Nm producing a rotational speed of 1774 RPM. The tendency for torque to decrease with an increase in turbine speed is caused by the presence of quite strong friction on the turbine shaft as a result of the increase in discharge and speed of water flow entering the runner blades of Francis turbine.



(a)



(b)

Figure 5: Torque as a function of runner speed at position (a) IGV = 4 and (b) IGV = 8

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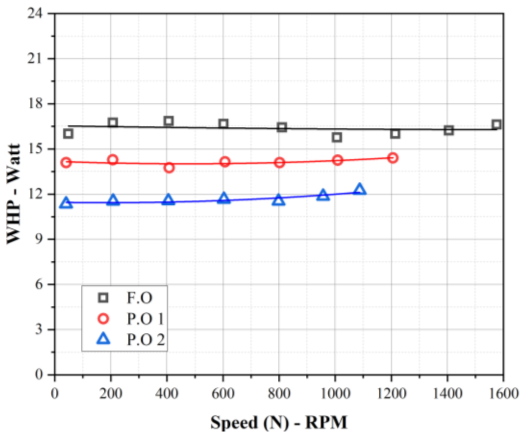
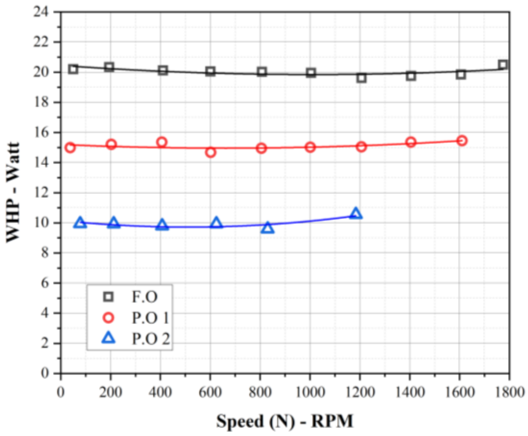


(a)



(b)

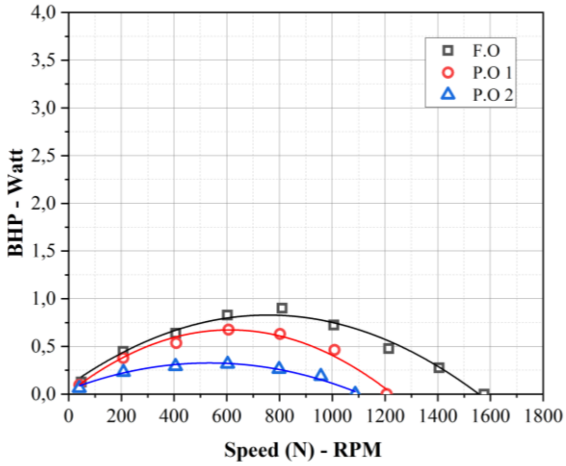
Figure 6: Water horse power as a function of runner speed at position (a) IGV = 4 and (b) IGV = 8

3.4 Water Horse Power (WHP)

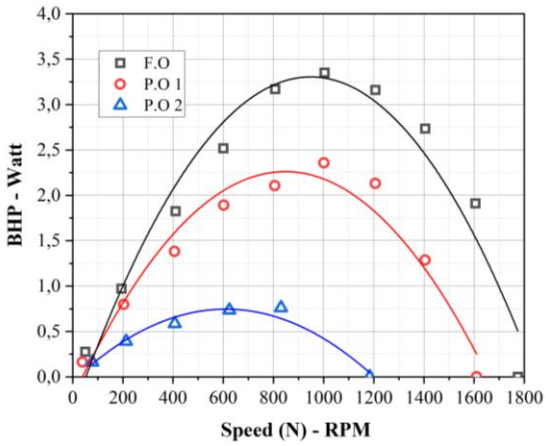
The measurement results for the water horse power (WHP) as a function of rotational speed of turbine runner at both variations in opening position of IGV = 4 and 8 are shown in Figures 6(a) and (b). From these plots it was found that the larger the IGV and inlet valve openings, the greater the WHP value, however, an IGV opening angle that is too large causes a lot of wasted power. The point where the largest WHP value is produced is at the Fully Open (F.O). Meanwhile, when the highest speed value was produced, namely at rotational speed N=1774 RPM, the WHP value was 20.10Watts where the opening position of IGV = 8 for fully opening of inlet valve (FO).

3.5 Break Horse Power (BHP)

Figures 7(a) and (b) are curves of BHP test results as a function of speed for two variations of opening position IGV = 4 and 8. In these curves it can be observed how the output power on the shaft changes along with changes in rotation speed. From these BHP curve plots, it can be seen that in general an increase in speed will cause an increase in BHP up to a certain rotation, then the BHP will decrease with additional turbine rotation for the three IGV openings tested. Furthermore, it can be seen that at opening position of IGV = 4 has a relatively lower maximum BHP value when compared to the BHP values for IGV = 8.



(a)



(b)

Figure 7: Break horse power as a function of runner speed at position (a) IGV = 4 and IGV = 8

From these plots it can also be noted that at the PO 2 inlet valve opening, the BHP value is lower at all three openings. Tested IGV. Meanwhile, the FO inlet valve opening has the largest BHP value for each increase in turbine rotation. There is a point where the output power on the shaft reaches the maximum achieved at Fully Opening for IGV 8 angle of BHP = 3,35Watt at 1004 RPM and then decreases again. along with a further increase in rotation speed.

3.6 Efficiency

The results of testing turbine efficiency as a function of speed at both variations in opening position IGV = 4 and 8 are plotted in Figures 8(a) and (b). Based on this efficiency plot, it

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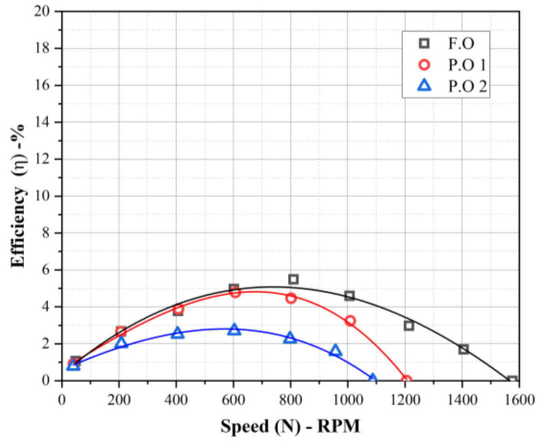
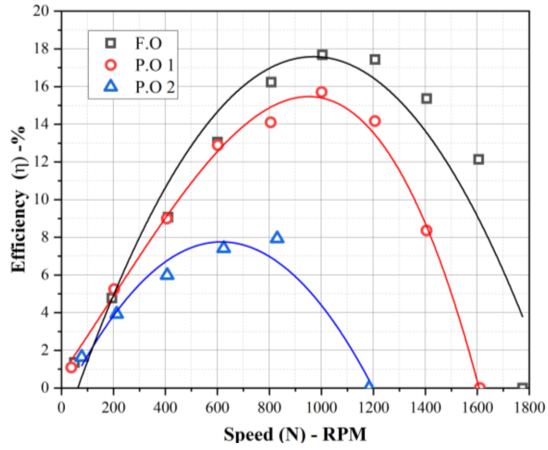
can be noted that increasing the rotational speed will result in an increase in efficiency up to a certain rotational speed range. Then adding further rotational speed will make the turbine efficiency decrease for all IGVs. From this efficiency plot it can also be inferred that the greatest efficiency is produced by the FO inlet valve opening conditions for the three IGVs at all speed ranges. Meanwhile, the lowest efficiency value is produced when the PO 2 inlet valve is open for the both position of IGV variation. Each valve opening condition FO, PO 1, and PO 2 produces maximum efficiency at different speeds as seen in the following plots.



(a)



(b)

Figure 8: Efficiency as a function of runner speed at position (a) IGV = 4 and IGV = 8

IV. CONCLUSION

In general discharge is constant or does not change significantly with increasing speed. The torque decreases as the turbine rotation increases, but this torque continues to increase consistently with increasing load. The BHP and efficiency curves are almost identical. Both BHP and efficiency values increase when speed and load are increased until the maximum condition is reached, then from this maximum value condition, as speed and load increase, the BHP value and efficiency gradually decrease.

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