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The Performance Characteristics of The Low Head Cross Flow Turbine Using Nozzle Roof Curvature Radius Centered on Shaft Axis

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Abstract: The experimental study was intended to investigate the performance characteristics of three cross flow turbine models using nozzle roof curvature radius centered on shaft axis designed on the same flow rate, runner diameter and rotational speed with each model having different runner width as well as its nozzle entry arc. The nozzle and runner width were designed as the fuction of the nozzle entry arc, the shorter pair of runner-nozzle, the larger nozzle entry arc and vise versa. The nozzle entry arcs used in this experimental study were 75°, 90° and 120°. In addition, the three models had equal cross sectional area of nozzle entry. The three turbine nozzles were designed to have roof curvature radius centered on shaft axis. The nozzle roof curvature were expected to be able to deliver water in better direction as well as its flow condition as the water entered the tubine runner. The experimental test rig consisted of three turbine models, pump, piping systems, magnetic flow meter, and tachometer. The Flow rates, that entered the turbine, supplied by the pump, were measured by the magnetic flow meter. The power generated on the turbine shaft was determined by measuring the torsion forces detected by using a spring balance and turbine speeds were detected by a hand held tachometer. The turbine performance characteristics were shown by the relation of efficiency versus flow rate, head, and specific speed; as well as the relation of efficiency versus velocity ratio and speed ratio. The velocity ratio was the ratio of runner pheripheral velocity to water jet velocity that entered the runner; the speed ratio was the ratio of runner speed to water jet speed entering the runner. The results of the study indicated that best efficiency points increased as the nozzle entry arc decreased or on the other hand best effisiency points decreased as the nozzle entry arc increased. The results showed that the cross flow turbine using 75 and 90 degree entry arcs indicated efficiency and power which was higher than that of turbine with 120 degree nozzle entry arc.

Keywords: Cross flow turbine,efficiency, nozzle roof curvature radius,specific speed, velocity ratio

1. Introduction

There are abundant hydro potentials as energy sources that are gently utilized to generate electricity to fulfill the need of electricity supply. In order to convert hydro potentials energy into electrical energy, a turbine generator set is needed, where the turbine converts hydro potentials into mechanical energy and the generator converts the mechanical energy into electrical energy. The cross flow turbines are considered to be the simplest turbine construction consisting of a squirell cage and square cross area of nozzle which resultin the cheapest fabrication. Therefore, the cross flow turbines are usually used as energy converter for a micro hydro power schemes. Such microhydro power schemes would be valuable to be constructed in remote areas that lack of national grade-electricity. Many studiesconcerned with cross flow turbine characteristics have been carried out by researchers such as :Azis and Desai [1], Djoko Sutikno [2,3], Fiuzat and Akerkar [4], Hothersall,R [5], Kosrowpanah,S;et.al [6], Nakase et.all [7], Olgun [8], Ott and Chappell [9] and Varga [10].

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The high efficiency of crossflow turbine is prompted by the energy of water collision on the blades when water flows into the water pressure energy on the blades when the water exits he runner. The flow rate determination and specific rotation on water turbine provides benefit in terms of its high effectivity and simplicity on the water exit system of the runner [11, 12]. The previous studies on the cross-flow turbine have analyzed the optimum characteristic and turbine by experimental [11 – 15] and numerical methods [16 – 20]. The turbine performance can be observed based on the head and the capacity of the water flow towards the power generation of the axis and the turbine efficiency [11]. The turbine performance can also be seen from the velocity, the flow rate comparison, and the specific rotation of the turbine operation [21]. Based on the correlation of these variations towards the turbine performance, it will be obtained the comparison or performance difference of the cross-flow turbine which is designed with the same head, capacity, and turbine wheels diameter but with different angle of jet arc.

In order to experience thedesigning and manufacturing of the cross-flow turbine using nozzle roof curvature radius centered on shaft axis, it is necessary to further study characteristics of the cross-flow turbine based on low head, low capacity and low rotational speed.

In cross-flow turbine, a noozle creates water jet that hits the turbine runner to produce power on the turbine shaft [22]. The noozle roof has to be able to lead the water jet enter the turbine runner. A curved noozle roof has been designed and fabricated in this experimental study to direct water to enter the runner in a certain angle to obtain effective action when the water hits the active runner blades of the turbine. The turbine runner shaft axis is matched with the noozle roof curve center. Hopefully, the turbine can have high efficiency as well as the power generation. In this experimental study, three turbine models with different noozle entry arc were examined.

This paper discusses the characteristics of the cross flow turbine based on the 3 turbine models that have the same runner diameter, different nozzle entry arc and runner width tested at rotational speed of 500 rpm.

2. Experimental Setup

The experiment was to study performance characteristic of three cross flow turbine modelswhich were designed with 197 mm runner diameter of each and had the ratio of runner diameter to runner length of 1:2; 1:2.4 and 1:3.214, operated on various input head, flow rate and rotational speed. In order to guide and deliver the flow of water in better direction as well as its flow condition as the water enters the tubine runner, the nozzle roof curvature radius of the three turbinenozzles were constructed with their centre on their runner shaft axis. The experimental installation consisted of three sets of turbine models, a water reservoir, a centrifugal pump, a piping system, a control valve, a magnetic flow meter, two spring balances, and a tachometer. Water was pumped from the reservoir to the tested turbine model by the centrifugal pump. The water flow rate was controlled by the valve and magnetic flow meter. Torque forces were detected by the two spring balances and the runner rotational speeds were measured by the tachometer. Schematic diagram of the experimental set up is shown in Figure 1.

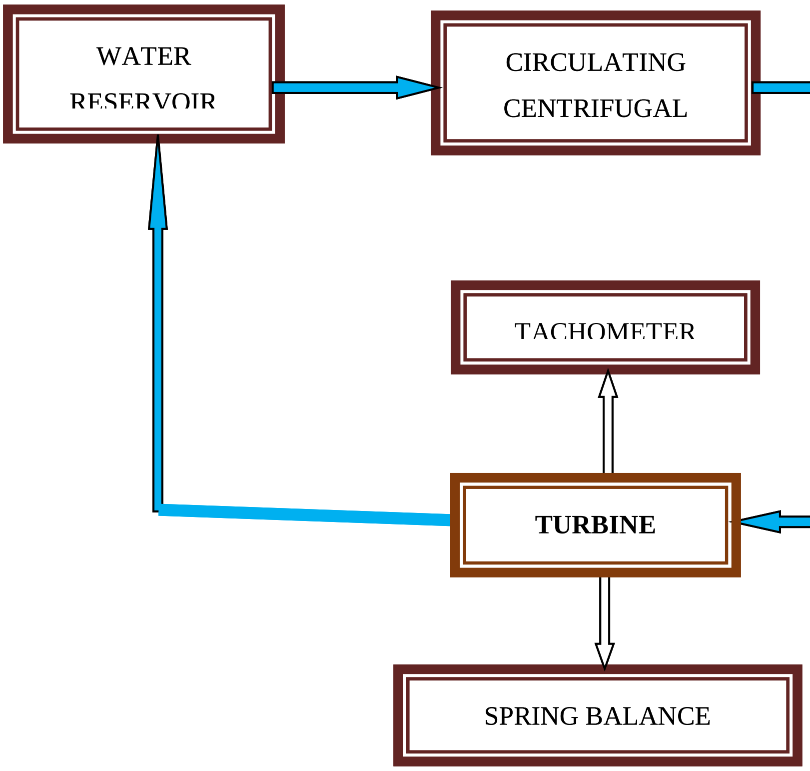


Fig. 1. Schematic diagram of the experimental set up

The nozzle roof of a cross flow turbine is a critical section, where the contour of roof curvature directs the water flow rate toenter and strikethe turbine runner in a certain angle of entry. Principally, the roof maintains water entering all the blade passages of the turbine runner within the jet entry arc of θ in the direction with an angle of α_1.The roof curvature radius of the three nozzles used in this experimental study is presented by figure 2, where the roof curvature radius R_θ was constructed by the following formula below. With the reference of figure 2 and cosine rule, the equation of nozzle roof curvature radius can be determined.

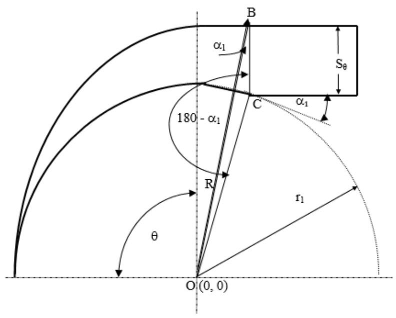


Fig 2. The nozzle roof curvature radius

$$OB = R;\ BC = S_\theta;\ OC = r_1 \quad (1)$$

Based on the cosine's formula

$$(OB)^2 = R^2 = r_1^2 + S_\theta^2 - 2r_1 S_\theta \cos(180 - \alpha) \quad (2)$$

Where,R is curvature radius of nozzle with the origin shaft coordinate (0; 0), R_θ is curvature radius of nozzle at θ with the origin shaft coordinate (0; 0), r_1 is outer runner radius, S_θ is passage height, distance between outer runner diameter to nozzle roof, α_1 is inlet angle.

Since S_θ increases with the increase of θ, then S_θ can be determined by the formula below:

$$S_\theta = \frac{\theta Z}{360} t \sin \alpha_1 \quad (3)$$

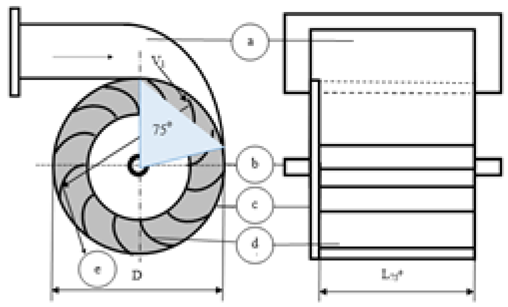
Where, θ is jet entry arc, in degree, Z is number of blade, t is blade distance, in mm, $\frac{\theta Z}{360} =$ number of blade passage, $t \sin \alpha_1 = s_0 =$ water jet thickness entering blade passage, in mm. Inserting $t = \frac{\pi D}{Z}$ into $t \sin \alpha_1 = S_0 =$ The Roof curvature of the nozzles used in this experimental study were constructed by using the equation 4

$$R_\theta = \sqrt{r_1^2 \left\{ 1 + (\theta \sin \alpha_1)^2 + 2\theta \sin \alpha_1 \cos \alpha_1 \right\}} \quad (4)$$

The dimensions of three turbine models are shown in Table 1.

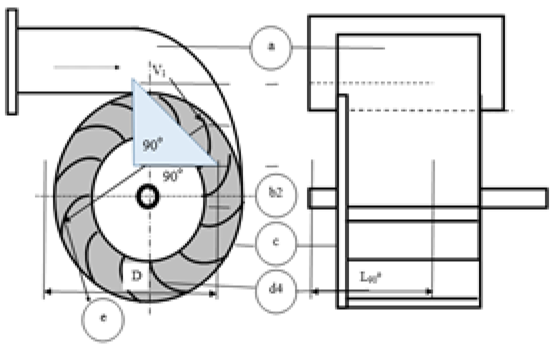
Table 1 - Dimensions of the turbine models

Parameter	Entry arc		
	θ= 75°	**θ = 90°**	**θ = 120°**
Outer Diameter D_1 (mm)	180	180	180
Inner Diameter D_2 (mm)	115.4	115.4	115.4
Turbine Length (mm)	90	75	56
Height of Nozzle Passage So (mm)	31	37,2	49,8
Length of blade arc L_s (mm)	37,6	37,6	37,6
Area of Inlet Nozzle passage A (mm^2)	2790	2790	2790



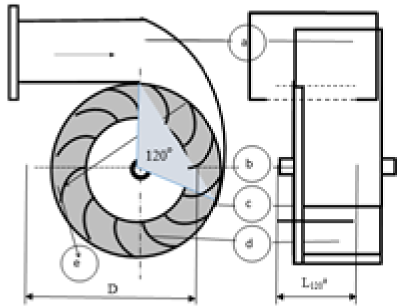
a = nozzle, b = shaft, c = disk , d = blade, e = water jet

Fig. 3a. Turbine model with nozzle entry arc of 75°



a = nozzle, b = shaft, c = disk , d = blade, e = water jet

Fig.3b. Turbine model with nozzle entry arc of 90°



a = nozzle, b = shaft, c = disk , d = blade, e = water jet

Fig. 3c. Turbine model with nozzle entry arc of 120º

In addition, principally, the three models were designed with the same runner diameters and the same area of nozzle throats as well as the area of runner inlets. The models wereexamined on the same flow rate and rotational speed. Thus, the power and the efficiency produced by the model can be compared. As the result, the model with the best efficiency can be determined.

3. The Results and Discussions

The experimental results are presented in Figure 4 up to Figure 10. They indicate the performance characteristics of the cross-flow turbine models tested at 500 rpm.

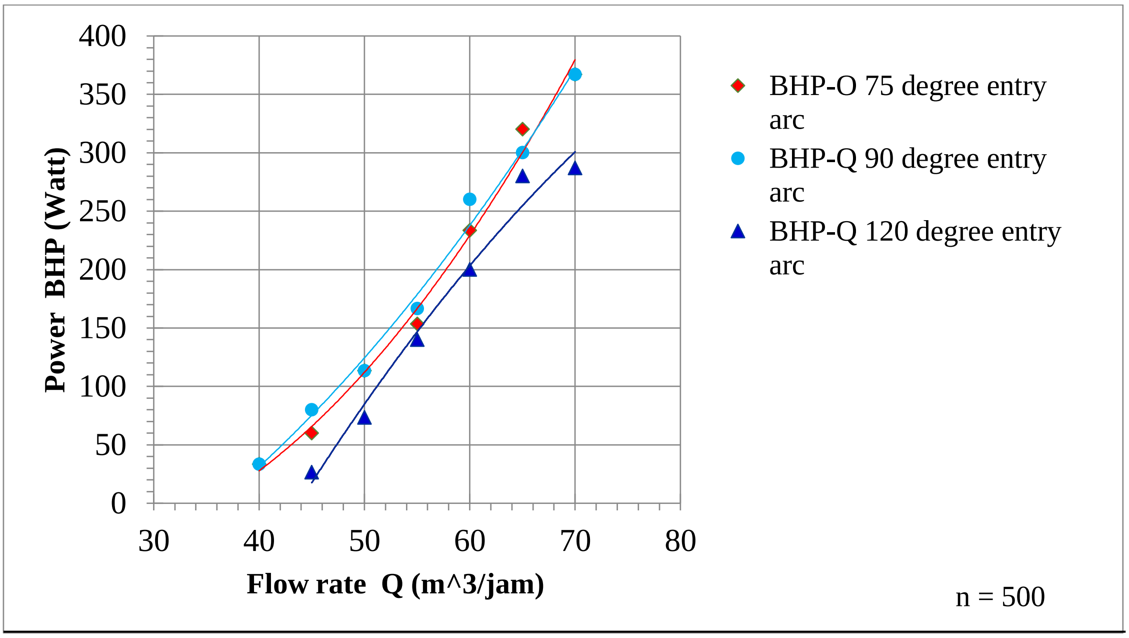


Fig. 4. Power vs Flow rate

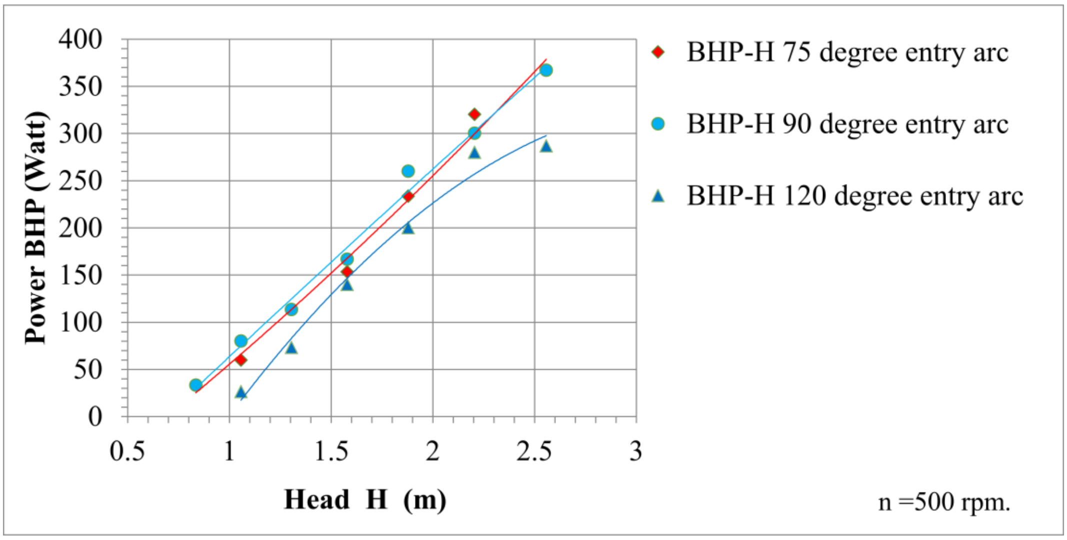


Fig. 5 Power vs Head

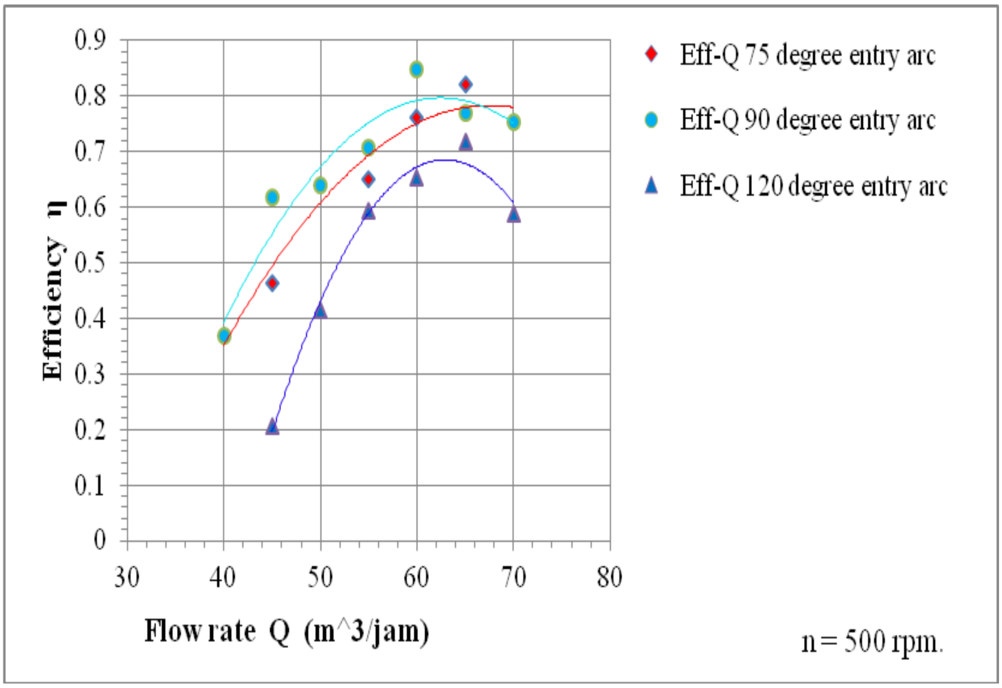


Fig. 6 Efficiency vs Flow rate

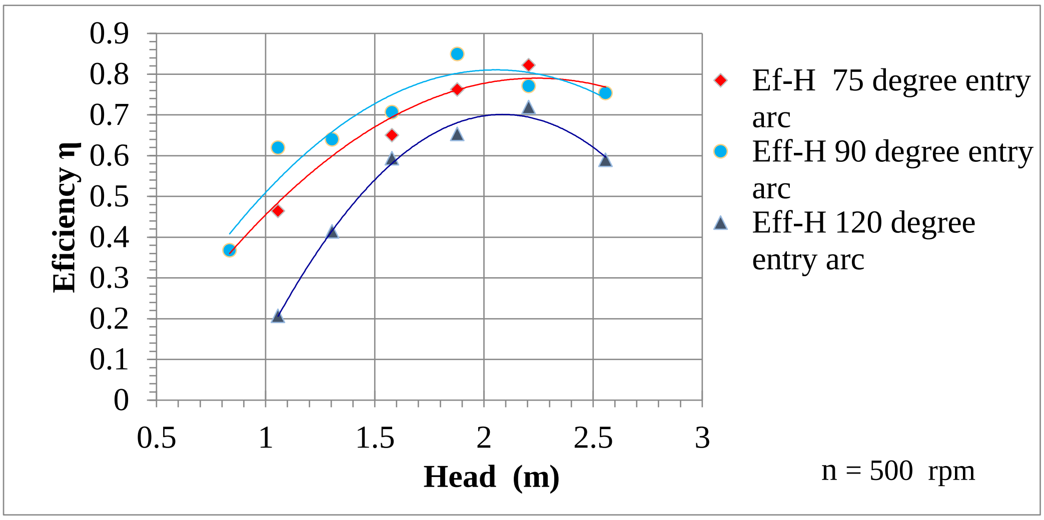


Fig. 7. The Efficiency vs the Head

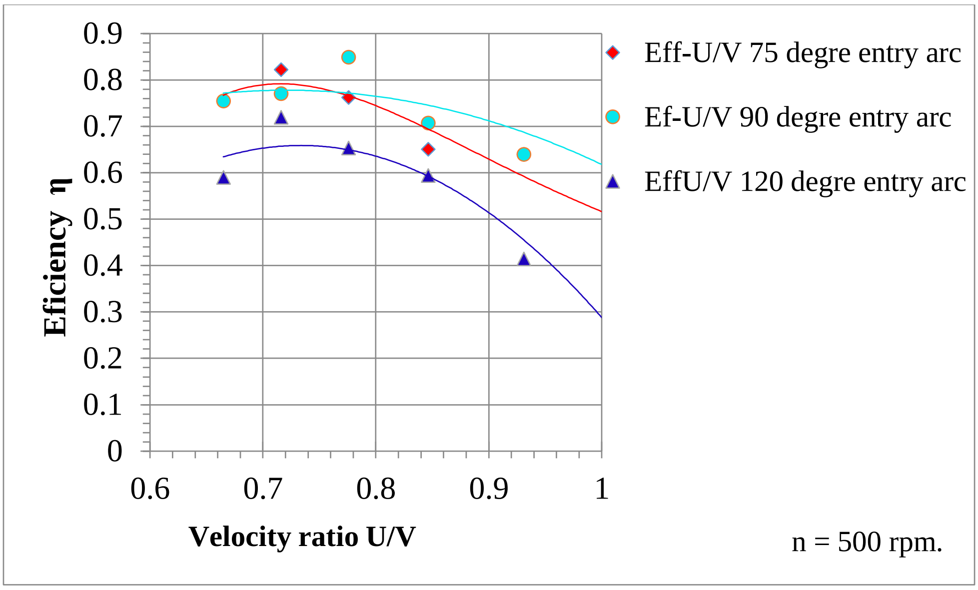


Fig. 8. Efficiency vs Velocity ratio

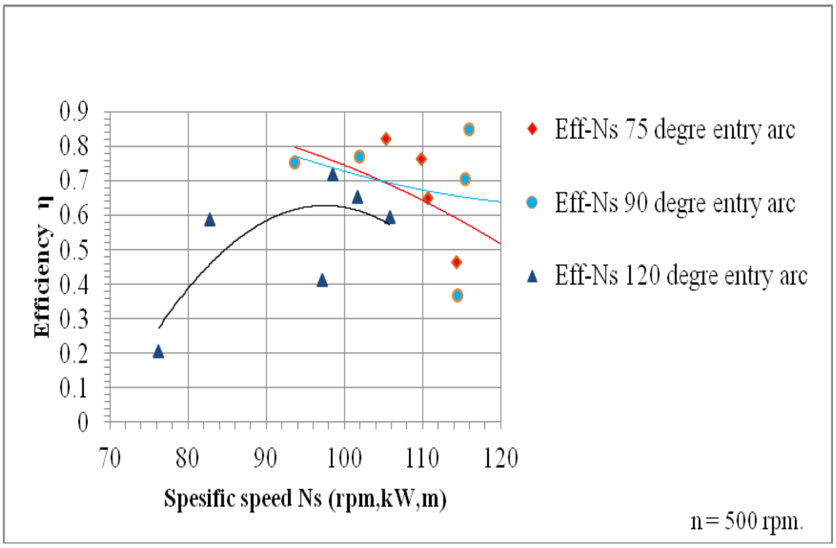


Fig. 9. Efficiency vs Specific speed

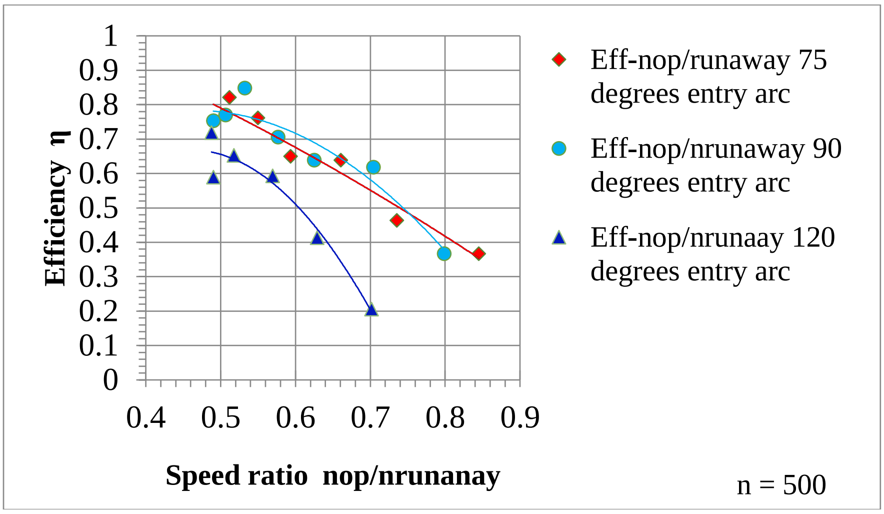


Fig.10 Efficiency vs speed ratio $n_{op}/n_{runaway}$

Figure 4 presents the relation of power versus flow rate for the runner rotational speed of 500 rpm. Figure 4 shows that the power generated by the cross-flow turbine P increases as the flow rate Q increases. The graph generated by the cross-flow turbine with 75^o nozzles entry arc and the graph generated by the cross-flow turbine having 90^o nozzles entry arc are barely connecting. The two graphs intersect at the flow rate of 68 m^3 per hour, while the graph of the cross-flow turbine with 120^o nozzle entry arc is at the lowest position. Since the number of water jet increases as the nozzle arc increases, the model with 120^o nozzle entry arc having the largest number of water jet is compared to those of the model with 90^{o}and 75^{o}nozzle entry arcs. The model with 75^o nozzle entry arches the smallest number of water jet. The more the number of water jet enters the empty space of the cross-flow turbine runner, the more the collision among the water jets is. The collision results in the existence of hydraulic loss and direction changing of water jets towards the second stage. In addition, in this experimental study, thesmall amount of water leakage through the gap between runner disk and nozzle wall is considered negligible [21].Further, in order to harness water power efficiently by minimizing hydraulic loss and water leakage, it is reccomended to use the cross flow turbine with nozzle entry arc of 90^{o}or 75^o rather than using the cross flow turbine with nozzle entry arc of 120^o.

Based on figure 4, in terms of hydro potential with the availability of water during the rainy season and the dry season, itis experiencing much change, it is recommended to exploit the hydro potential for a system of hydro power plant using a cross flow water turbine with a nozzle entry arc of 90^o or 75^o. Next, it can be submitted also that in terms of a system of hydro-electric generation for servicing a constant load where the system uses a cross flow water turbine, it is advised to use a cross flow turbine that has a nozzle entry arc of 90^o or 75^o rather taunting cross flow turbine with a nozzle entry arc of 120^o.

In terms of construction, the runner widthof cross flow turbine with 120^o nozzle entry arc has the shortest runner widthcompared withthe runner widthof cross flow turbine with 90^o and 75^o nozzle entry arcs. The greater the nozzle arc, the shorter the runner width. Thus, the turbine with 75^o nozzle entry arc has the longest runner width.It is further recommended to use the cross flow turbine fabricated with nozzle entry arc of 120^o having apposition-changeable floor so that the magnitude of the nozzle entry arc can be set by means of changing the position of the nozzle floor [23]. Thus, the use of the cross-flow turbine is suitable for a power plant that is built at the site where the hydro potential significantly fluctuates during the rainy season and the dry season. Therefore, the use of a cross flow turbine with a movable floor nozzle is suitable for mikrohydro power system that is built on the location with the hydro potential fluctuates as well as to handle the load fluctuates.

Figure 5 presents the relation of power generated by the turbine P versus turbine headH at 500 rpm runner rotational speed. The power increasessharply as the head increases. The graphs generated by the turbine with 75^o nozzles entry arc and thatby the turbine with 90^o nozzles entry arc are coincide and they intersect at the head of 2.10 m. Before the intersection point, the graph of the turbine with 90^o nozzles entry arc is slightly in higher position compared to the graph of the turbine with 75^o nozzles entry arc, while the position of the graph of the turbine with 120^o nozzle entry arc is lower than that of 75^{o}and 90^o nozzles entry arcs. Further, for low head hydropotensial, it is recommended to use cross flow turbine with 90^{o}and 75^{o}nozzle entry arcs rather than the turbine with 120^o nozzle entry arc.

As flowrates as a function of heads, then itis further recommended trustee cross flow turbine fabricated with a nozzle entry arc of 120^o with position-changeable floor, so that the magnitude of the nozzle entry arc can be set by the means of changing the position of the nozzle floor. Thus, the use of the cross-flow turbine is suitable for a power plant being built at the site where the hydro potential significantly fluctuates during the rainy season and the dry season. So, the use of a cross flow turbine with a movable floor nozzle is suitable for mikrohydro power system that is built at the location with the hydro potential fluctuates as well as to handle the load fluctuates [24].

Figure 6 shows the relation of turbine efficiency η versus flow rate Q curves for the model with nozzle entry arc of 75°, 90°and 120°. The curves show that the efficiency increase as the flowrate increases up to flow rate64 m^3 per hour and then the efficiency decreases as the flowrate increases. Maximumeffisiency for the model with nozzle entry arc of 75° and that of 90° is about 79 % , and for the model with nozzle entry arc of 120°is 68%. The efficiency versus flow rate curves for the model with nozzle entry arc of 75°, and 90°are nearly coincide and both curves intersect at flow rate of 68 m^3 per hour. In order to keep the operation stability, these turbine characteristics should be fully understood by turbine operatorswhen the operators want to change turbine load.

Figure 7 shows the relation of turbine efficiency η versus turbine head H for the model with nozzle entry arc of 75°, 90°and 120°; the efficiency of the turbine models increases with increasing the head up to a value of 2 meters and then the efficiency decreaseswhen the head is over 2 meters.Maximumefficiency for the model with nozzle entry arc of 75° and that of 90° is about 79 % and for the model with nozzle entry arc of 120°is 68%. The efficiency versus headcurves for the model with nozzle entry arc of 75°, and 90°are nearly coincide and both curves intersect at the head of 2.3 meters.

This turbine characteristics must be considered by designers and operators of turbine because of the head changes during the periode of dry season and rainy season. May the very noteworthy by the turbine operators that in doing any additions of the turbine load after reaching the maximum efficiency must be very carefully due to the decline in efficiency.

Figure 8 shows the relation of turbine efficiency η versus velocity ratio U/V when the models running at 500 rpm. The velocity ratio is defined as the ratio of the runner peripheral velocity and the water jet velocity enters the runner. Peripheral velocity is tangential velocity of runner per second and water jet velocity is velocity of water jet leaving nozzle, which is equal to velocity water jet entering runner. The result of this research shows that the efficiency of the three turbine modelsincreased and then decreased as the velocity U/V increased. The maximum efficiency of the turbine models were achieved at the U/V of about 0.73.The highest efficiency of about 0.79 is for the model with 75°and that with 90°nozzle entry arcs, and the highest efficiency of 0.68 is for the model with 120° nozzle entry arc.According to the basic principle of rotating machines derived mathematically, the optimum efficiency occurring at the speed ratio is 0.5. There is a difference in value between the efficiency of theoretical and experimental efficiency, it is presumed that the existence of hydraulic losses when water flow mashing a number of active blades and the condition of the water flow in the empty space of the turbine wheel that collides with one another resulting in changes in the direction of the water flow towards the turbine blades of the second stage, the such conditionsof water flow resulted in the different efficiency of each examinedturbine model [21]. In addition, the larger the nozzle entry arc is, the more the number of water jet enters the runner, and consequently the larger entry arc is, the more the water collisioninside the runner that causes hydraulic losses [21].These losses result in different efficiencies of the models as reported above.

The tested three models had a nozzle with the same cross-sectional area of each. According to the law of mass flow rate, the fluid velocity is equal to the flow capacity divided by the nozzle cross-sectional area; and according to the law of energy conservation of the fluid that the fluid velocity is a function of the turbine head. Since the cross-sectional area remains constant, the fluid velocity increases with the increasing capacity of the flow; and the flow velocity also increases with increasing the head [23]. The three models have the same diameter of the wheel and operate on the same rotational speed. In this case, this means that the three turbine models are operating at the same peripheral velocity. Thus, the speed ratio U/V changes with the capacity of flow and U/V also changes with the turbine head [25]. Figure 8 is certainly very useful forturbine designers in determining the size of the turbine to the achievement of high efficiency with the respect to the potential hydro potential (capacity of flow and head) available [26].

Figure 9 shows the relationship of the turbine efficiency η versus specific speed Ns of the three turbine models rotating on 500 rpm., where the efficiency of the turbine model with nozzle entry arc of 75° and that of 90° decreases with increasing specific speed; whereas the efficiency of the turbine model with nozzle entry arc of 120° increases and then decreases with increasing specific speed. The highest efficiency of approximately 0.79 occurring at a specific speed 95 (rpm, kw, m) is indicated by the model that has a nozzle entry arc of 75o and also by a model that has a nozzle entry arc of 90°; and the highest efficiency of 0.62 occurring at a specific speed 98 (rpm, kw, m) is indicated by the model that has a nozzle entry arc of 120°.

In order to harness hydro potential efficiently for a hydroelectric power station using a crossflow turbine as prime mover, Figure 11 suggests the turbine designers that the crossflow turbine should be designed with specific speed 95 up to 105 (rpm, kW, m) and the figure also recommends to turbine operators that the turbine should be operated at specific speed in between 95 up to 105 (rpm, kW, m). Further, the turbine desainers should be wise to choose nozzle entry arc which determines runner diameter and runner width in accordance with the strength of the material used in constructing the turbine in order to assure the turbine endurance in the operation [27,28].

Figure 10shows the curves of efficiency η versus speed ratio $n_{op}/n_{runaway}$for the turbine model with nozzle entry arc of 75°, 90° and120°operating at 500 rpm rotational speed. The curves present that the efficiency decreases as the speed ratio $n_{op}/n_{runaway}$ increases. The curve for the model with nozzle entry arc of 75° and that of 90°almost connect each other and mutually intersect at the speed ratio of 0.52 and 0.74 that indicate at the efficiency of 0.75 and 0.5. While the position ofthe curve of model with nozzle entry arc of 120°which has a maximum efficiency of 0.65 is lower than theposition of curve ofthe model with nozzle entry arc of 75° and the position of curve of the model with nozzle entry arc of 90°.These turbine characteristics should be fully understood by a turbine operator in order to keep operation stability of a hydroelectric power system when the operator wants to change turbine load.

Fig. 4 to 10 suggest that in order to reach the reasonable efficiency, the turbine should be operated on a speed of (0.48 up to 0.56) runaway speed. Recommended efficiency occurs on the values of speed ratio nop/n runaway 0.5 up to 0.38. Further, turbine operators should be wise to choose operational speed in accordance with runaway speed and strength of material used in constructing the turbine in order to assure the turbine endurance in operation.

4. Conclusion

- The use of nozzle having roof curvature radius centered on shaft axis is recommended in order to improve cross flow turbine efficiency.
- The characteristics of the cross-flow turbines change with the changingof the nozzle entry arc, mass flow rate and head.
- The use of cross flow turbine with nozzle entry arc of 75°, 90° is preferred for constant hydropotensial as well as constantload applied on hydro electric generating system, and the use of cross flow turbine with movable floor nozzle with 120°entry arc is recommended for hydro electric generating system built on fluctuating hydro potential as well asto handle fluctuating load system.
- When the turbines are operated at 500 rpm the best efficiency pointsoccur at the velocity ratio U/V off 0,72, spesific speed Ns of 100 (rpm,kW,m) and speed ratio $n_{op}/n_{runaway}$of 0.5.
- Reasonable efficiency occurrs when the turbine speed ratio $n_{op}/n_{runaway}$is at 0.2 up to 0.6, specific speed is at 40 up to 110 (rpm,kW,m) and velocity ratio U/V is at 0.35 up to 0.75.

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