ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

SECOND STAGE CROSS FLOW TURBINE PERFORMANCE

Rudy Soenoko, Purnami and Fransisca Gayuh Utami Dewi
Mechanical Engineering Department, Engineering Faculty, Brawijaya University, Malang Indonesia
E-Mail: rudysoen@ub.ac.id

ABSTRACT

Indonesia has the potential energy of water around 75000-76000 MW. But the utilization is still about 3,783 MW for large-scale and 220 MW for small scale power generation. There are still many people living in rural areas where electricity supply is very low. To answer the social problems in rural areas, it is clearly that looking for alternative energy sources is important to increase people revenue. To solve these problems the exact answer is developing an alternative energy, namely micro hydro. Many theories and research on cross flow has been done up to now aims to enhance the turbine performance by concentrating on the turbine internal flow behavior and preventing the water flow chaotic. Previous research suggests that by putting several guides on the turbine nozzle could increase the water pressure entering the turbine blade. The following study will elaborate on the effect of nozzle angle attack and the water flow rate on the turbine second level toward the turbine performance. By positioning the nozzle direct to the turbine second level it is expected a maximum cross flow turbine second level performance. This study uses a true experimental research. In this case the experimental device is made under a laboratory scale. For this study, the independent variable is the water flow rate and the dependent variable is the turbine efficiency. The turbine rotation angle variation was observed under a nozzle angle attack of 30°, 45°, 60° and 70°. The highest turbine performance was found on a 30° nozzle angle attack, under a 20 Newton loading and on a maximum valve opening. Through this laboratory testing it could be concluded that the nozzle angle attack would greatly affect the cross flow turbine performance.

Keywords: renewable energy, micro hydro, cross flow turbine, second stage.

INTRODUCTION

Lately, micro hydro attracts attention as a source of clean energy, renewable and a quit good future development. However, this type of turbine must be in accordance with the conditions of the place where the turbines are placed and also need to be assessed against the turbine type. Besides, high production cost relative to the complex turbine structure is the biggest obstacle to developing micro hydro. Cross flow turbine was adopted because it has a relatively simple structure. Although micro hydro is a small plant but to note that any source of water that flows into a river system must be controlled or regulated in accordance with the designation. Due to the use of water will affect many aspects of life and environmental system. Such as for irrigation, flood control and water for the necessities of life. Rispiningtati [1] make an observation and neighbor regulation of water usage in the Sutami dam especially for the sake of optimum electrical generation. The study also developed, as a comparison for irrigation, the operation of the reservoir nodes which connected with the irrigation allocation [2, 3]. So it can be concluded that the use of water for any purpose must be adapted to the local regulation. Because if water use is not in accordance with the local regulation, many problems would occur, for example, interfere with agriculture (lack of water) or even flooding due to the excessive water consumption to drive the turbine during the rainy season.

Micro hydro by developing kinetic turbines has been developed to balance the needs of small electrical power generators even portable generators. Various ways has been done by several researchers. Monintja [4] conducts a research by utilizing the kinetic turbine optimization response surface methodology to obtain the appropriate turbine design and hopefully has a good performance. Monintja [5] is also conducting a research about a cup bladed kinetic turbine for the turbine optimal performance through the variation of the water incoming flow steering angle. While Lempoy [6] conducts a research on optimization a bowl bladed turbine utilizing the response surface methodology, to obtain the bowl bladed kinetic turbine optimum performance. The goal of all of these researches is to optimize the micro hydro turbines performance.

In another further research is a research conducted by Boedi [7], which is observing a new design that the kinetic turbine blades could close and open as required, so that the obstacles in kinetic turbines occurs as small as possible. Once again the purpose of this study is the optimization of the micro hydro operation system. In addition Saroinsong [8] develop a micro hydro turbine screw type (Archimedes turbine) to add a variant of small turbines for a micro hydro power generation.

For a cross flow turbine, Soenoko [9] perform a cross flow design optimization. This design optimization is done by utilizing the triangle velocity theorem as the basic flow velocity analysis to get the closes phenomenon water flow passing through a cross flow turbine disc.

As it is known, that the cross flow turbine power generated take place at two stages, namely at the first stage and the second stage. Based on the cross flow turbine review, Soenoko [10] conducted a study of cross flow performance in the first stage turbine power generation. This study was conducted to determine the variables that influence the turbine performance. So from this research, the best turbine performance at the first turbine stage was found.

Some more previous studies have been conducted by researchers for cross flow turbines with the aim to determine the optimal configuration of the turbine with experimental and numerical methods. A research using the CFD to analyze the effect of inlet nozzle shape and the internal flow behavior toward the cross flow turbine performance. The draft tube shape effect was also investigated and found that a relatively narrow and converging inlet nozzle shape gives a better turbine performance [11, 12]. Mockmore, has been using one-dimensional method of theoretical analysis and experimentation [13]. Another research has tried experimentally to improve the turbine performance by changing the output flow shape [14, 15, 16]. Nakase, has demonstrated the effects on a turbine component parts with a series of studies using experimental and numerical methods. In particular, they have demonstrated that there is an effect of nozzle shape on a large turbine performance [17]. According to Fukutomi the internal flow in a cross flow turbine is not uniform, because water flows are partly from the turbine runner. Thus, the fluid force works unstable on the blade. Experimental and theoretical studies were carried out to determine the fluid force on a cross flow turbine blade. In the experiment, the tangential and radial force on the blade was measured by using strain gauges and a slip ring, while on the theoretical study the tangential and radial force was calculated numerically using the unsteady momentum theory [18]. Kyoshi doing a research on the effect of the nozzle angle and the inlet guide vane length toward the cross flow turbine performance. By installing a special guide vane on the turbine nozzle, the water pressure increase and the turbine efficiency become higher compared with a turbine without guide vane [19, 20]. Research done for cross flow turbines are always making some observations and modifications on turbine mechanical parts, such as to modifying the turbine nozzle and blade radius. While studies, that has not been done is the influence of water jet which is directed straight forward to the turbine second level.

From the explanation above the research that would done is “The effect of the nozzle angle variation and nozzle water flow rate toward the second level cross flow turbine performance”. According to the continuity law, the water flow rate through a cross-sectional are would always remain constant, which means that the amount of incoming water is equal to the amount of water output. It is valid too for the water passes through the cross flow turbine blades.

This study will observe on the second turbine stage, in order to get the best turbine performance. So that, by utilizing the optimized parameters on the turbine first stage and by combining with the optimized parameters on the second stage, hopefully a complete optimized cross flow turbine could be reach.

MATERIAL AND METHOD

Before doing the research it is necessary to clarify some terms that will be used in this study. The cross flow first stage in this study is part of the cross flow turbine that gets the first water jet from the turbine nozzle. While the cross flow turbine second stage is the turbine part that get the further impetus after the water jet coming out from the first stage. The first stage and the second stage of the cross flow turbine are illustrated on the Figure-1 below.

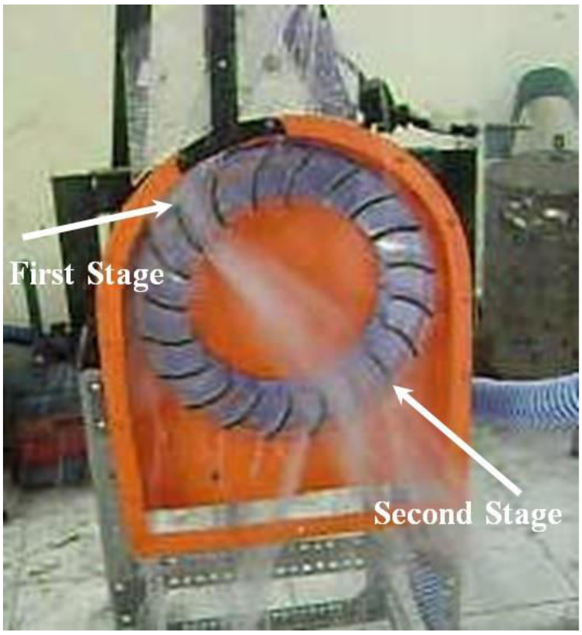

Figure-1. Cross-flow turbine model.

As mentioned above the research on the optimal performance of cross flow turbine in the first stage has been investigated by Soenoko [10]. It is known that a cross flow turbine is an intact turbine that has a first stage and second stage part. To get an optimum turbine performance it is necessary to observe the second stage cross flow turbine performance. So with the optimal performance of the first stage cross flow turbine supported by the optimal performance of the second stage cross flow turbine, would result an optimal overall cross flow turbine.

The research equipment dimension

In this study the cross flow turbine used has a specification as seen in Figure-2. The cross flow runner; includes three main parts, namely the turbine shaft, the turbine disc which has an outer diameter of 36 cm and an inner diameter of 24 cm in diameter made from steel plate with a thickness of 5 mm, and lastly is the turbine blade which has 20 blades, with a steel plate installed around the disc, while the turbine cover plate is made from a 3 mm thick transparent material.

VOL. 12, NO. 6, MARCH 2017 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

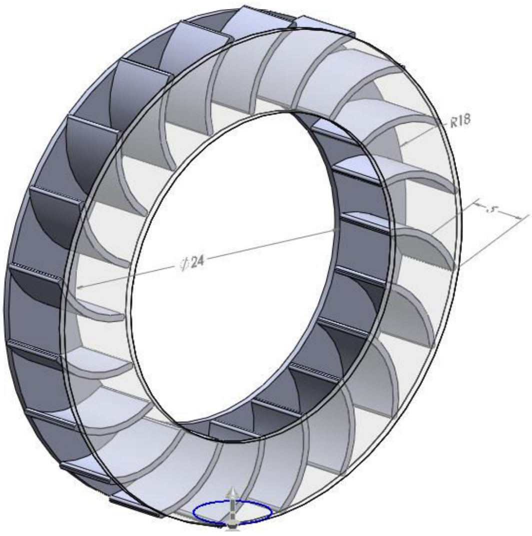

Figure-2. Cross-flow turbine dimensions.

To support this research, a special nozzle was design and attached in such a way, so that the water jet could hit straight forward to the second stage part of the cross-flow turbine. The nozzle design could be seen in Figure-2.

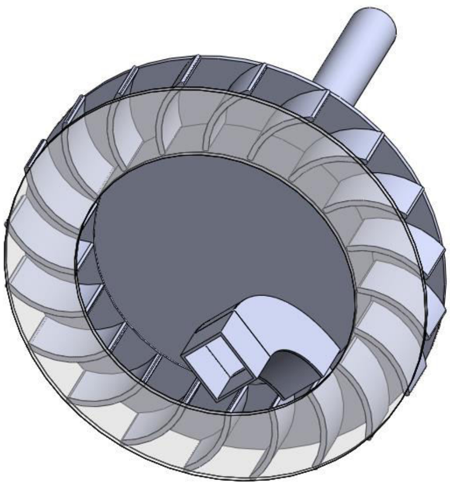

Figure-3. Water jet nozzle design on the 2nd stage.

As mentioned above that the research setup was installed in such a way so that the water coming out from the first level would deflect immediately entering the second stage cross-flow turbine. The objective of this study is to determine the best nozzle angle on the cross flow turbine first stage to obtain the highest turbine energy generated.

This research was done experimentally, where the research devices was made under a laboratory-scale research. Three variables are declared on this research. First is the independent variable, which is he water flow rate under a valve opening variation from 25% till 100%. Second is the dependent variable, which is the turbine efficiency and lastly is the controlled variable which is the nozzle angle which is set from 30° till 70°.

Figure-4. Cross flow turbine runner.

The cross flow turbine performance

The cross-flow turbine performance is the properties that involving the independent variable and the dependent variable. In this case the independent variable is the operational indicator which is the water flow rate and the dependent variable is the turbine power indicator, which is pointed at the turbine power and the turbine efficiency. The cross-flow performance formulae are as follows:

Brake Horse Power (BHP)

$$BHP = \frac{2.\pi.n.T}{60}\ (\text{Watt}) \quad (1)$$

where:

n = Turbine rotation (rpm)
T = Torque (Nm)

Water Horse Power (WHP)

$$WHP = \frac{\rho.g.Q.H}{3600}\ (\text{Watt}) \quad (2)$$

Where:

ρ = Water specific weight (kg/m^3)
g = gravitational acceleration (m/s^2)
Q = Water flow rate (m^3/s)
H = Head drop (m)

Efficiency

Bernoulli equation is an ideal fluid equation which is declared for the fluid energy conservation law. In implementing the Bernoulli equation, many assumptions should be considered include the condition where the fluid should be a steady flow, the fluid ideally should not a viscosity (frictionless flow), the fluid density (ρ) is constant (incompressible), so that there is no energy loss during the fluid flow.

$$\frac{P}{\rho.g} + \frac{v}{2g} + z = Total\ head\ (H) = constant \quad (3)$$

where:

P = Fluid static pressure (N/m^2)
V = Fluid speed (m/s)
g = gravitational acceleration (m/s^2)
z = elevation of the same datum (m)
ρ = fluid specific weight (kg/m^3)

The continuity equation

The continuity equation is a mathematical equation on the net amount of fluid mass flow on a limited surface which is equal to the mass on the fluid flow surface. The fluid volume entering the input entry is equal to the fluid volume on the output entry.

$$m = \rho_1.A_1.V_1 = \rho_2.A_2.V_2 \quad (4)$$

where:

m = Mass flow (kg/s)
ρ = fluid specific weight (kg/m^3)
A = cross-sectional area (m^2)
V = fluid velocity (m/s)

RESULT AND DISCUSSIONS

The data processing on the water jet angle toward the water flow rate on the second stage cross flow turbine, on the first turbine loading, could be seen on Figure-5 below.

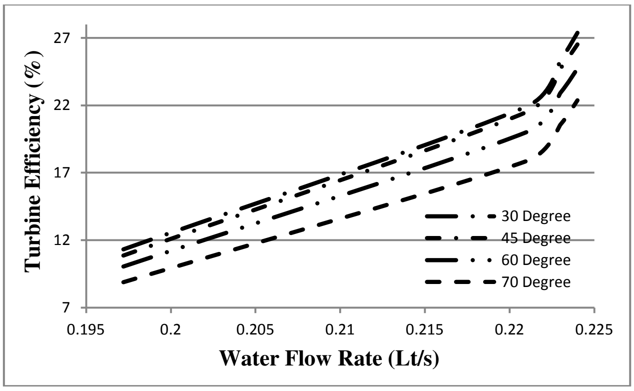

Figure-5. Water flow rate vs turbine efficiency (1st load).

From Figure-5 it can be seen that the water jet directed to the turbine second stage could produce a torque from the turbine second stage would affect the flow pattern inside runner, so that the torque produced by the second stage will be increase.

The testing is done with the nozzle water jet angle variation toward the water flow rate. These observations were repeated under a 25%, 50% to 75% and 100% valve opening, and of course each valve opening would give a different water flow rate. From each minimum to maximum valve opening there is an efficiency increase. Increased efficiency is also proportional to the increase in water flow rate variations. At the maximum valve opening in all likelihood increase efficiency at each test is due to the nozzle angle variation causes a change in water jet direction. As a result, the momentum will increase and the torque given on the second level will also increase, resulting in an increase in turbine power. Where the greater the water flow rate the higher the load that must be given, so that the turbine efficiency will increase.

From Figure-5 it could be seen that the 30° water jet at the maximum valve opening, the turbine efficiency is increased compared to a greater or smaller water jet angle of 30°. This is due to the effective water kinetic energy water getting in the blade runner because the water strikes the turbine right at the blade front part. So that the turbine torque increase and would resulted an efficiency increase. Therefore, by directing the water jet nozzle properly on the turbine second stage will produce a more stable torque and an increase of cross flow turbine efficiency. The turbine rotation stability is needed for the electricity generated quality.

The data processing on the water jet angle toward the water flow rate on the second stage cross flow turbine, on the second turbine loading, could be seen on Figure-6 below:

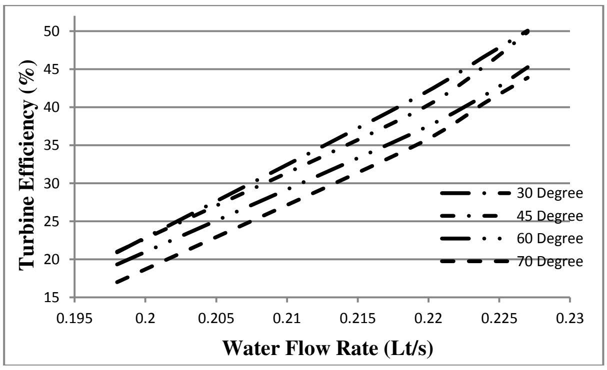

Figure-6. Water flow rate vs turbine efficiency (2nd load).

From Figure-6 it can be seen that the water jet directed to the turbine second stage could produce a torque from the turbine second stage would affect the flow pattern inside runner, so that the torque produced by the second stage will be increase. The explanation is analogous with the second stage turbine prime moving on the first turbine loading. On the second turbine loading there is also an increase on the turbine torque that result a turbine efficiency increase. The same objective as on the first turbine loading is the turbine rotation stability is needed for the electricity generated quality.

VOL. 12, NO. 6, MARCH 2017 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

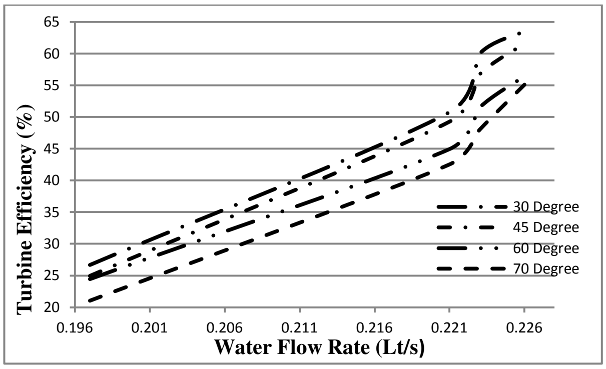

Figure-7. Water flow rate vs turbine efficiency (3rd load).

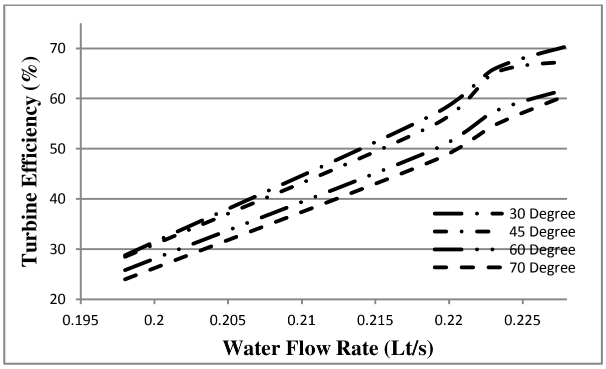

Figure-8. Water flow rate vs turbine efficiency (4th load).

Figure-8 point out the relation between the water flow rate and turbine efficiency under nozzle angle variation on the second stage cross flow turbine. From the graph above it can be seen that by directing the nozzle straight to the second level, it would affect the low pattern inside the turbine runner. The torque produce on the turbine second stage will increased the overall turbine torque so that the overall efficiency would be higher. The experiment settings are for four valve opening variation, namely the opening of 25%, 50% to 75% and 100%. That, of course at every opening of the valve from opening up to a maximum opening size of minimum the water flow rate would be different. For every valve opening in this observation there will be different efficiency values. Increased efficiency is also proportional to the increase in water flow rate variations. At the maximum valve opening, the efficiency obtained will achieve the highest value. It is caused by the nozzle angle variation which causes a change in water direction on the second level. As a result, the momentum will increase, and the torque given on the second level will also increase, resulting in an increase in turbine power. The greater the water flow rate, the higher the turbine rotation will be, so as for the turbine rotation intended the turbine load should be higher, and of course the turbine efficiency will increase.

It can also be seen that on the water inlet blade angle of 30° produces the highest efficiency when compared with the blade water inlet angle greater or smaller than 30°. This is due to the water inlet velocity energy getting into the blade runner were well or optimally utilized. This is due to the water inlet velocity energy blade runner more utilized because the water strikes the right front part of the blade as a result of torque which is given on the second level will also be increased so that the efficiency will increase. Therefore, by directing the nozzle straight to the second stage it will produce a more stable torque and higher cross flow turbine efficiency. The turbine rotation stability is a must for the electricity generated quality.

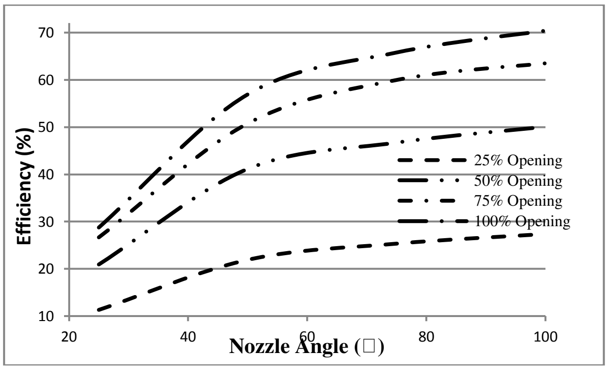

Figure-9. Nozzle angle vs turbine efficiency (1st load).

From Figure-9 it can be seen that with the supply of water that is pumped passes through the nozzle resulted a water jet that hit the turbine blades will arise a thrust/collision. The water jets will hit and drives the turbine so that the turbine rotates. The water flow directions will turn, thus change the amount of water moving. The water jet hit the turbine blades would drive the cross flow turbine disc with a speed of u. So that on a similar nozzle angle and on varied valve opening there would be a trend of an efficiency increase at each test with variable angle nozzle. The turbine rotation and the water flow rate Q (l/s) is also increase but the pressure decreases. This is due the angle nozzle variation causes a change in direction of the water outflow from the turbine first stage (triangle velocity). The force acting on the working fluid causing a change of water speed and momentum will increase. The torque given on the second stage will also increase and result an increase of turbine power. The greater the water flow rate, the higher the torque that should be given, so that the turbine efficiency will increase.

It can also be seen that the water jet inlet angle of 30° produces the highest efficiency when compared with the water jet angle greater or smaller than 30°. This is due to the water kinetic energy entering the turbine blade runner is more effective utilized. But here there's been a change in the nozzle angle of 45° at the first turbine load, where the water flow rate decreases and the turbine efficiency also decreased but not significant. On a nozzle angle of 60° and 70° variation the efficiency result is the smallest compared with other jet angle variation. This is because on the water jet entry angle of 60° and 70° would decrease the water flow rate and of course decreased the turbine torque. It is because the water flow, on this condition, did not hit the blade on the front part but hit the

VOL. 12, NO. 6, MARCH 2017 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

back part of the blade. This is thought to be one cause of the instability of the turbine rotation.

The effect of water jet angle variation toward the cross flow turbine on the second stage can be explained through the following Figure.

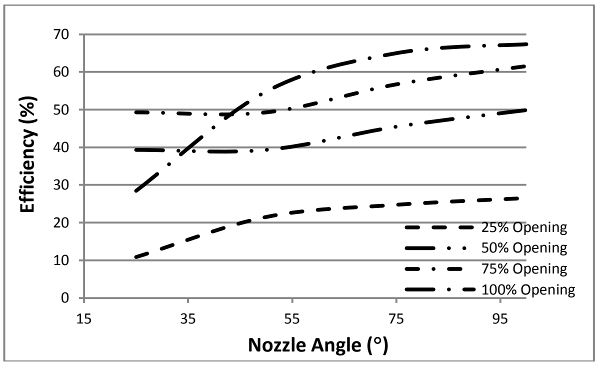

Figure-10. Nozzle angle vs turbine efficiency (2nd load).

From Figure-10 it can be seen that on the same nozzle angle on a varied valve opening there is a trend of increased efficiency at each test variable-angle nozzle, also an increase of turbine rotation and water flow Q (l/s) but resulted a pressure decrease. This was due to the variation of the angle nozzle causes a water out direction change on the first level (can be explained by a triangle speed), because the force acting on the working fluid thus causing a change of water speed and momentum will increase and the torque given to the second level would also be increase. Thereby it would increase the turbine power. Where the greater the water flow rate supplied for the turbine rotation, the higher the torque would be produced so that the turbine efficiency will increase.

It can also be seen that on the water inlet blade angle of 30° produces the highest efficiency when compared with the blade water inlet angle greater or smaller than 30°. This is due to the water inlet velocity energy getting into the blade runner were well or optimally utilized. On the nozzle angle of 45° at the load F2 (N) the turbine rotation and the water flow rate decreases and the efficiency also decreased, but the efficiency decrease was not significant. On a nozzle angle of 60° and 70° the turbine performance reaches the smallest efficiency compared with other nozzle variation angle. This is due to on nozzle angle of 60° and 70° the water flow rate (l/s) decreased so that the turbine torque produced was small. The effect of an inaccurate nozzle angle would result a water jet hit on the blade back side and this would resulted a negative power. This is thought to be one cause of the instability of the turbine rotation.

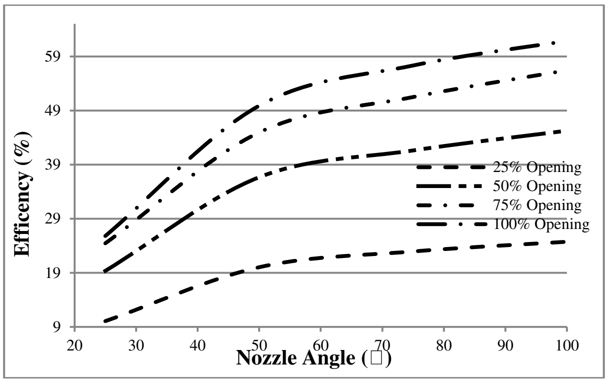

Figure-11. Nozzle angle vs turbine efficiency (3rd load).

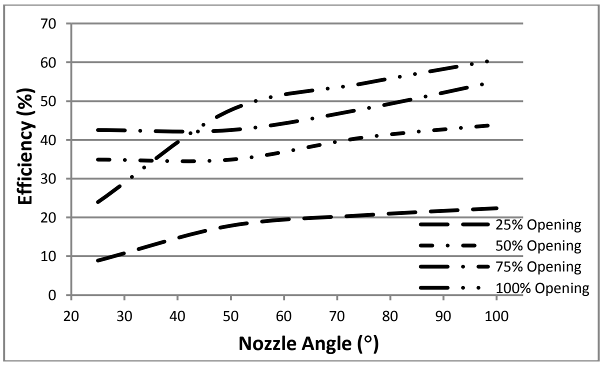

Figure-12. Nozzle angle vs turbine efficiency (4th load).

From Figure-11 and Figure-12 it can be seen that from each water inlet blade angle variation with the supply of water that is pumped past the nozzle resulted a water jet on the turbine blades and pushes the turbine blade. The water jet drives the turbine to rotate. The water jet directions will turn, thus change the amount of water moving. Water jet with a speed of u, hits the turbine blades and would drive the cross flow turbine wheel. So that on a same nozzle angle and varied valve opening there is a trend of increased efficiency. The turbine rotation increase too, but the resulting pressure decreases. This is due to the nozzle angle variation that causes a water direction change out from the first stage (can be explained by the triangle velocity theorm). The water direction change would increase the water speed and the increase momentum and the torque given on the second level will also increase and finnaly would increase the turbine power. The greater the water flow supplied to the turbine blades the higher the torque that should be given, so that the turbine efficiency would increase.

The graph also shows that on the 30° nozzle angle the turbine efficiency reach the highest value when compared with the greater or smaller than 30°. This is due to the water inlet velocity into the blade runner energy is more utilized. But on the nozzle angle of 45° and at the first turbine load variation the water flow decreases and the turbine efficiency also decreases but the decrease in efficiency is not significant. The lowest turbine efficiency is under the nozzle angle of 60° and 70°. This is because at the 60° and 70° nozzle angle the water flow rate (l/s) decreased so that the torque produced would reduce. The water flow entering the blades does not give a torque addition, because the water jets strikes the back part of the blade. This is what is thought to be one of the turbine rotation instability.

CONCLUSIONS

From the investigation of the influence of nozzle angle variation and the water flow rate toward the second stage cross flow turbine performance, it can be concluded that the large nozzle angle would affect the cross flow turbine performance (torque, power and efficiency).

- The water jet coming out from the nozzle would hit the turbine blade and push the turbine wheel to rotate.
- The torque given on the turbine second level would increase the turbine efficiency and the efficiency increase would be proportional with the water flow rate increase.
- The greater the water flow rate supplied the higher the turbine rotation. The efficiency increased could be seen by increasing the turbine load. The highest turbine efficiency on the second stage was reach on a nozzle angle of 30° and on a load of 20 Newton and water flow rate of 0.228 (1 l/s).
- The smallest turbine efficiency occurs on a nozzle angle of 45°, 60° and 70°. This is due to the less water flow rate entering the turbine blade accurately. Some of the water jet hitting the back side of the turbine blade. This has been thought to be one of the causes of the turbine rotation instability.

REFERENCES

[1] Rispiningtati, Soenoko R. 2015. Regulation of Sutami Reservoir to have a Maximal Electrical Energy. International Journal of Applied Engineering Research. 10(12): 31641-31648.

[2] Rispiningtati, Soenoko R. 2015. Optimization Operation of Bening Reservoir to Maximize Irrigation Allocation. International Journal of Applied Engineering Research

[3] Rispiningtati. 2014. Irrigation Allocation Optimization of Wonorejo Reservoir. International Journal of Applied Engineering Research. 9(22): 12871-12878

[4] Monintja N.C.V., Soenoko R., Wahyudi S., Irawan Y. S. 2014. The Vertical Shaft Kinetic Turbine Optimization using Response Surface Methodology. International Journal of Applied Engineering Research. 9(21): 8841-8856.

[5] Monintja N.C.V., Soenoko R., Wahyudi S., Irawan Y. S. 2014. The Influence of Flow Steering Angle on the Performance of a Cup-bladed Kinetic Turbine. International Journal of Applied Engineering Research. 20(9): 7481-7489.

[6] K.A. Lempoy, K.A., Soenoko, R., Wahyudi, S., Choiron M.A. 2015. Bowl Bladed Vertical Shaft Kinetic Turbine Performance using Response Surface Methodology (RSM). International Journal of Applied Engineering Research. 10(7): 16399-16407.

[7] Boedi S., Soenoko R., Wahyudi S., Choiron M.A. 2015. An Outer Movable Blade Vertical Shaft Kinetic Turbine Performance. International Journal of Applied Engineering Research. 10(4): 8565-8573.

[8] Saroinsong T., Soenoko R., Wahyudi S., Sasongko M.N. 2015. The Effect of Head Inflow and Turbine Axis Angle towards the Three Row Bladed Screw Turbine Efficiency. International Journal of Applied Engineering Research.

[9] R. Soenoko. 2015. Design Optimization to Increase a Cross Flow Turbine Performance, a Review. International Journal of Applied Engineering Research. 10(18): 38885-38890.

[10] Soenoko R. 2016. First Stage Cross Flow Turbine Performance. International Journal of Applied Engineering Research. 2(11): 938-943

[11] Choi Y.D., Son S.W. 2012. Shape Effect of Inlet Nozzle and Draft Tube on the Performance and Internal Flow of Cross-Flow Hydro Turbine, Journal of the Korean Society of Marine Engineering. 36(3): 351~357.

[12] Fukutomi J., Senoo Y., Nakase Y. 1991. A Numerical Method of Flow through a Cross-Flow Runner. JSME International Journal, Series II. 34(1): 44-51.

[13] Mockmore, C.A., Merryfield, F. 1949. The Banki Water Turbine. Engineering Experiment Station, Oregon State Colleg, Corvallis, Oregon, No. 25

[14] Khosrowpanah S., Fiuzat A.A., Albertson M.L. 1988. Experimental Study of Cross-Flow Turbine. Journal of Hydraulic Engineering. 114(3): 299-314.

[15] Fiuzat A.A., Akerkar B.P. 1991. Power Outputs of Two Stages of Cross-Flow Turbine. Journal of Energy Engineering. 117(2): 57-70.

[16] Desai V.R., Aziz N.N. 1944. An Experimental Investigation of Cross-Flow Turbine Efficiency. Journal of Fluids Engineering. 116: 545-550.

ARPN Journal of Engineering and Applied Sciences

[17] Nakase Y., Fukutomi J., Watanabe T. Suetsugu T., Kubota T., Kushimoto S. 1982. A Study of Cross-Flow Turbine (Effects of Nozzle Shape on Its Performance), ASME Small Hydro Power Fluid Machinery. pp. 13-18.

[18] Fukutomi J., Nakase Y., Watanabe T. 1985. A Numerical Method of Free Jet from a Cross-flow Turbine Nozzle, Bulletin of JSME. 28(241): 1436-1440.

[19] Kiyoshi K., Toshiaki K., Yamasaki K. 2013. Guide Vane with Current Plate to Improve Efficiency of Cross Flow Turbine. Open Journal of Fluid Dynamics. 3, 28-35.

[20] Fukutomi J., Nakase Y., Ichimiya M., Ebisu H. 1995. Unsteady Fluid Forces on a Blade in a Cross-Flow Turbine, JSME International Journal, Series B. 38(3): 404-410.