Design,Simulation and Performance on the Effect of Modification on Impeller Tip for Pump as Turbine (PAT)

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Abstract- Pump as Turbine (PAT) is typically used as electromechanical components in microhydro systems, especially by rural communities in developing countries to reduce initial capital cost. The technology is readily available and easily accessible compared to commercially available turbines.This paper presents experimental design ,the results of an experimental test and performance of a centrifugal pump working as turbine (PAT). An end suction centrifugal pump was tested in turbine mode at PAT experimental rig installed in the Mechanical Engineering Laboratory of Mandalay Technology University. The experiment showed that a centrifugal pump can satisfactorily be operated as turbine without any mechanical problems. As compared to pump operation, the pump was found to operate at higher heads and discharge values in turbine mode. Centrifugal volute-pumps are potential alternative solution to be using as hydro turbines, by flowing water in the reverse direction through in the pumps. Since the overall efficiency of these machines is lower than the conventional turbines, their application in larger hydro sites is not economical. Therefore, the efficiency improvement of reverse pumps is essential. In this study, by focusing on a pump impeller, the shape of blades was redesigned to reach a higher efficiency in turbine mode. The aim of this research is to compare the performance of a small centrifugal volute-pump as turbine at the different head of the pump before and after modification on impeller tips of the pump at various capacities. Modification is performed by grinding the inlet ends of the impeller tips of the pump to a sharp-edge, round and bullet nose shape to preclude excessive turbulence for efficiency consideration. After modification, a new impeller was tested in the test rig. Then, testing is carried out by operating the pump as turbine at the maximum head of the pump, 13.5 meter, and at various capacities. The results show that the output-power and efficiency of the modified impeller tips the centrifugal volute-pump as turbine slightly better than before modified one.

Index Terms- Experimental tests, Design, Pump as Turbine,Modification Impeller, Performance

I. Introduction

A Small hydroelectric power stations became attractive for generating electrical energy after the oil crisis of the seventies. However cost per kW energy produced by these stations is higher than the hydroelectric power plants with large capacity. Numerous publications in recent years emphasize the importance of using simple turbine in order to reduce the cost of produced electrical energy. We considered the idea of using pumps as hydraulic turbines an attractive and important alternative. Pumps are relatively simple machine, are easy to maintain and are readily available in most developing countries. From the economical point of view, it is often stated that pumps working as turbines in the range of 1 to 500 kW allow capital payback periods of two years or less which is considerably less than that of a conventional turbine. Small hydropower plants are very attractive for producing energy from water in these days. The main problem for developing such units is the high price of installation as well as the time needed for their construction and maintenance. One solution of this problem is the extensive use of pumps that can operate in the reverse mode as turbines. They are easily found on the market at much lower price compared to a standard turbine. The maintenance of the pump is very low and the pumps can be operated by non-highly trained people. Pumps operated in the reverse mode are usually of the single stage type with low or medium delivery head and the description of flow inside. It is possible of course to use also the multistage pump in the turbine regime in the case when sufficiently high head is available, nevertheless there is lack of information about applications of the multistage pumps in the reverse mode, especially about the flow phenomena occurring inside these pumps when operating as turbines.

Rural electrification plays a significant role in enhancing the quality of life for rural communities by promoting access to modern energy services. This leads to increased economic strength and improves productivity, consequently reducing inequality. The traditional way to supply electricity to rural areas is to extend the national grid into these areas; however, after taking into account financial viability, this approach is usually found to be uneconomical. Using microhydro technologies for off-grid electricity generation is the most suitable method whenever there is an accessible potential site. The use of appropriate technologies to suit local conditions is one of the critical factors affecting the success of an off-grid microhydro electrification system [1, 2]. Economic feasibility is one of the main considerations associated with microhydro systems. Hence, the application of low-cost equipment for the microhydro system has always been the central focus for rural electrification.

The use of Pump as Turbine (PAT) in place of commercial electromechanical components has proven to be a feasible, low-cost solution. Some successful microhydro projects have been reported for rural area sites, especially in developing countries [3-5]. There has been growing interest on utilising the PAT as a substitute for the commercial microhydro turbine, in particular for generating power between the ranges of 10 kW to 25 kW. Even though the PAT has lower efficiency than commercial turbines, its availability covers a wide variety of flows and heads, which makes microhydro projects more economical and practical. Also, the PAT offers benefits such as simple construction, easily-attainable spare parts, readily-available maintenance services, and installation that can be carried out by local laymen. These desirable characteristics of the PAT are indeed important, as they reduce dependence on third-party professional services, which can be very expensive. Even though there are many ways to predict the behaviour of PATs, such as simulation analysis, theoretical frameworks and prediction models, the best approach is to test the pump at a predefined operation range of flow rate and pressure. Each PAT is unique to its application; thus, it needs to be tested in its working environment. There are many prediction and simulation models; however, they are limited to certain applications and cannot definitively predict the actual performance. Therefore, the experimental results will give specific indicators of its operational performance. The outcome of the experimental work will accurately present real operational conditions from the perspective of microhydro applications [6].

Pump manufactures do not normally provide the characteristic curves of their pumps in reverse operation. Therefore, establishing a correlation enabling the passage from the "pump" characteristics to the "turbine" characteristics is the main challenge in using a pump as a turbine. The hydraulic behavior of a pump when rotates as a turbine will be changed. In general a pump will operate in turbine mode with higher head and discharge in the same rotational speed. Many researchers have presented some theoretical and empirical relations for predicting the PAT characteristics in the best efficiency point (BEP). A good literature review has been done by Nautiyal and Anoop Kumar [7]. But the results predicted by hese methods are not reliable for all pumps with different specific speeds and capacities. Most recent attempts to predict performance of PAT, have made using CFD [8]. However, without verifying the CFD results by experimental data, they are not reliable. Besides, also all of these simulations included only hydraulic losses.

II. THEORETICAL DESIGN

The design of centrifugal pumps involves a large number of interdependent variables so there are several possible designs for the same duty. One of the most difficult design problems is to predict the impeller head slip, i.e. the difference between the theoretical head for a number of impeller vanes and the theoretical head deduced from the net horsepower given to the fluid passing through the impeller. Before pump design or selection can be got, specifications need to be established which express several requirements.

A. Specific Speed of Pump

Selection of the specific speed value at the best efficiency point is the first step in the design of centrifugal pumps. This involves a selection of the rotation speed and whether the required head should be produced in one or more stages. Another distinction in impellers is the liquid traverses and leaves the impeller blades. This is called the specific speed. It is another index used by pump designers to describe the geometry of the impeller and to classify impellers according to their design type and application.

The equation for specific speed of pump is;

$$n_{sp} = \frac{N_p \sqrt{Q_p}}{H_p^{3/4}} \quad (1)$$

B. Capacity

This is the volume of liquid per unit time delivered by the pump. The capacity of a pump is the amount of water pumped per unit time and it is also known traditionally as volume flow rate. Capacity means the flow rate which liquid is moved or pushed by the pump to the desired point in the process. It is expressed in English units gallons per minute. In metric units, it is expressed as liters per minute (L/min) or cubic meters per second (m^3/sec). As liquid is essentially incompressible, the capacity is directly related with the velocity of flow in the suction pipe.

This relationship is as follows:

$$Q = AV \quad (2)$$

C.Overall Efficiency

The overall efficiency of the turbine is the ratio of useful power available at the turbine shaft to the power supplied by the water turbine at the entrance.

$$\eta_o = \frac{\text{power available at the shaft}}{\rho_w g Q H} \tag{3}$$

$$\eta_o = \eta_h \times \eta_m \times \eta_v \tag{4}$$

The hydraulic turbines are directly coupled to electric generators to produce electric power. There is loss of power in transmission. Coupling and the generators also have some losses. Therefore, the power output at the generator shaft is a bit less than the power available at the runner outlet.

D. Dimensions of Impeller Diameters

Before the impeller diameters, the rotational speed, design flow rate and the pump head must be known. Electric motor drive prescribes standard motor speeds. So, the maximum size of the impeller diameter is limited. The value of the impeller inlet diameter D_1 is usually selected to minimize the inlet relative velocity. The impeller diameter is one of the principal determinants of the power and head of the pump. So, the inlet and outlet diameter of impeller is calculated by the following equation.

The inlet diameter of impeller is carried out from stepanoff chart,

$$D_1 = \frac{D_1}{D_2} \times D_2 \tag{5}$$

The outlet diameter of impeller is calculated by using equation,

$$D_2 = \frac{U_2 \times 60}{\pi N_p} \tag{6}$$

$$U_2 = K_u \sqrt{2 \times g H_p} \tag{7}$$

E. Inlet and Outlet Width of Impeller

The width of the blades at inlet and outlet of the impeller are calculated from the continuity equation.
The inlet width of impeller, b_1 is calculated by:

$$b_1 = \frac{Q_s'}{\pi D_1 V_{m1}} \tag{8}$$

The volumetric efficiency for pump can be calculated by

$$\eta_v = \frac{1}{1 + \left(\frac{1.124}{\left(n_{sp}\right)^{\frac{2}{3}}} \right)} \tag{9}$$

$$Q_S' = \frac{Q_p}{\eta_v} \tag{10}$$

$$V_{m1} = K_{m1}\sqrt{2 \times gH_p} \tag{11}$$

The outlet width of impeller, b_2 is calculated by:

$$b_2 = \frac{Q_s'}{\pi D_2 V_{m2}} \tag{12}$$

$$V_{m2} = K_{m2}\sqrt{2 \times gH_p} \tag{13}$$

III. SIMULATION

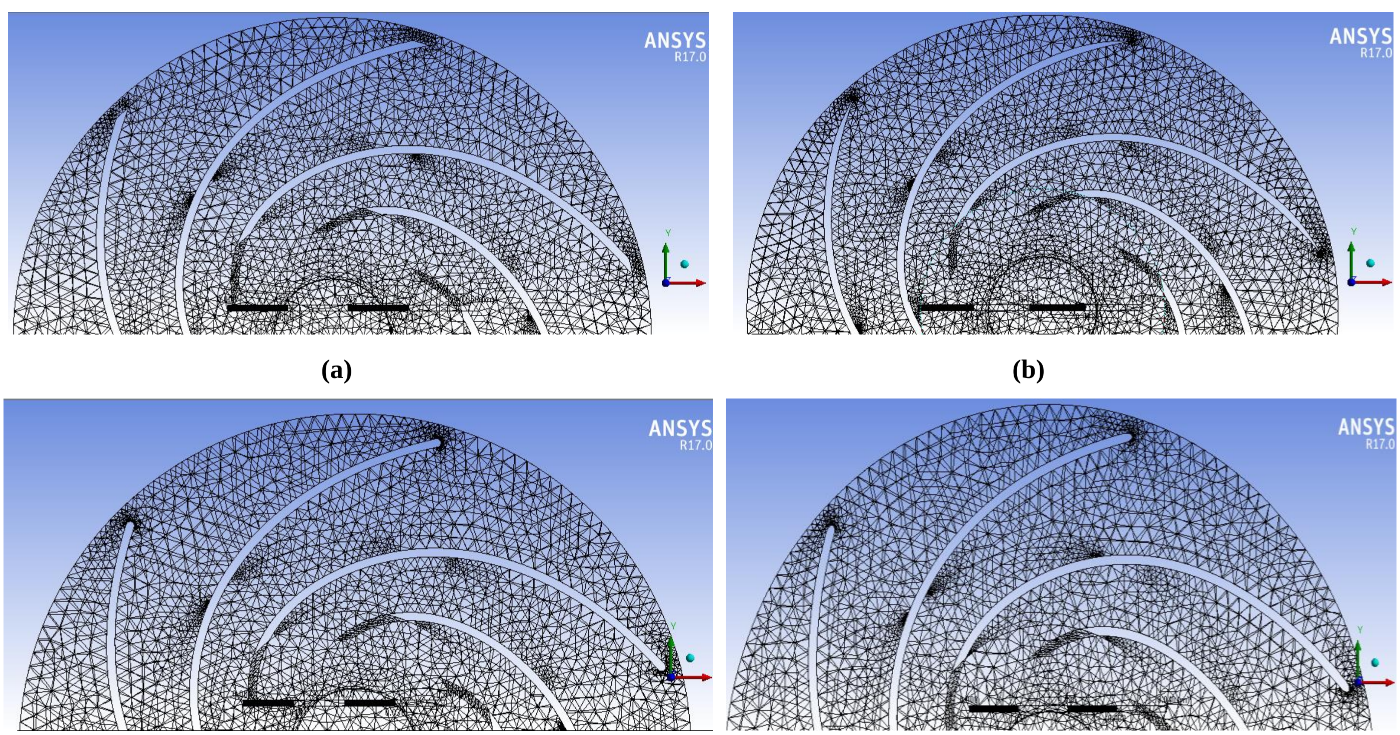
The Computational Fluid Dynamics (or CFD) programs are gaining in importance for the design and analysis of fluid turbomachinery. The CFD analysis of the PAT was carried out inside the premises of ANSYS Workbench 17.0. In this study, all the simulations have been performed in turbine mode and the setups were done for the same condition.

A. Modelling of the Geometry

Modeling of the geometry of a centrifugal pump running in turbine mode involves defining the impeller and volute components. Geometric modeling of draft tube is also considered since it is one of the major components in PAT system. The volute, impeller and draft tube are generated using Solidworks software.

B. Mesh Generation

The entire geometry consisting of casing, impeller, and draft tube is modelled for numerical analysis using an unstructured tetrahedral mesh as shown in the following figures. The geometry and the mesh of a six bladed pump impeller domain is generated using ANSYS Workbench 17.0. An unstructured mesh with tetrahedral cells is also used for the zones of impeller and volute.



(a) **(b)**

(c) (d)

Figure.1. Meshed Model of the Impeller with (a) Flat Tip (b) Round Tip (c) Sharp Tip and (d) Bullet Tip

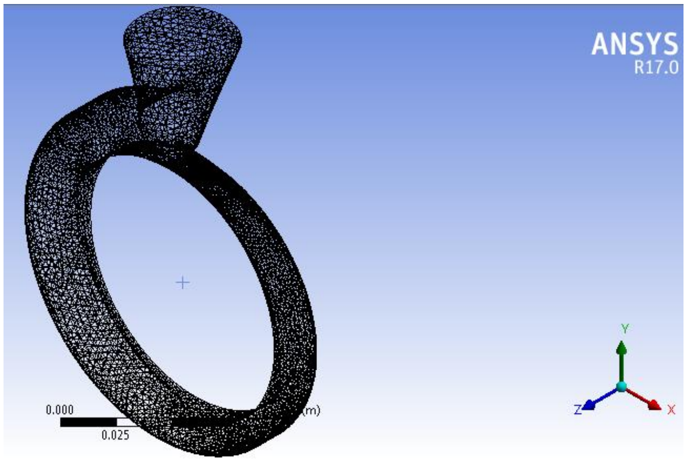


Figure .2. Meshed Model of Volute Casing

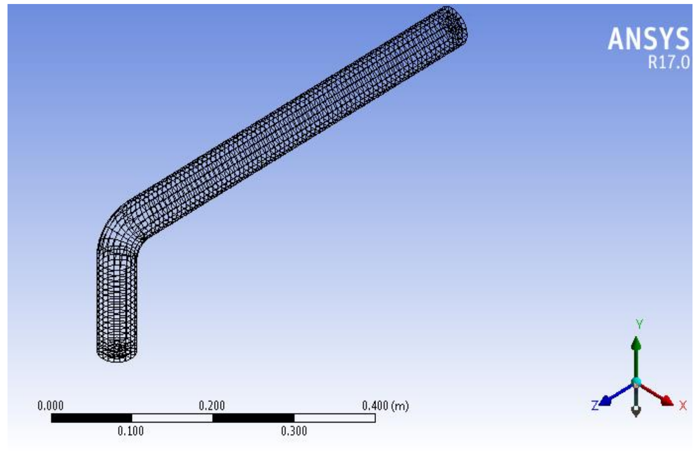


Figure .3. Meshed Model of Draft Tube

C. Boundary Conditions

Centrifugal pump impeller domain is considered as rotating frame of reference with a rotational speed of 1655 rpm. The working fluid through the pump is water at 25˚ C. k-ε turbulence model with turbulence intensity of 5% is considered. The inlet and outlet boundary condition were set by imposing total pressure on the casing inlet surface and variable mass flow rate on the impeller outlet surface respectively.

D. CFX-Pre

CFX Pre is a mode allowing to set up fluid flow simulations such as compressors or turbines. Each components of a machine can be simply defined by selecting a mesh file and some basic parameters and then add the boundary conditions and interfaces between the components. The meshed model are imported to CFX-Pre to analyze the velocity and pressure distribution on the PAT due to the fluid flow.

E. Simulation Results

The pressure contours show a continuous pressure decrease from leading edge to trailing edge of the impeller. It is observed that total pressure on pressure side of the blade is more than that of suction side. The difference of pressure from the pressure side to the suction side of the impeller blade is decreasing from leading edge to trailing edge of the blade. The minimum value of the static pressure inside the impeller is located at the leading edge of the blades on the suction side.

The velocity streamlines and pressure distributions with the above setup is shown in Figure.4. From the runner, the water flows towards the draft tube with high velocity. This velocity head is converted into the pressure head generating more power and efficiency. The diverging passage of the draft tube allows for the decrease in the velocity at the outlet from the equation of continuity.

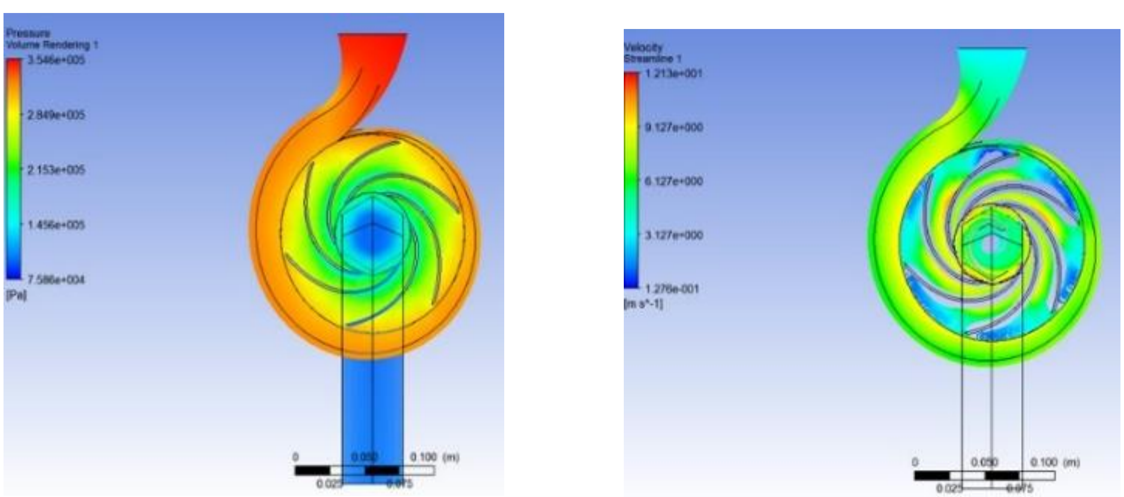


Figure.4(a) . Pressure and Velocity Distribution for the Impeller with Flat Tip

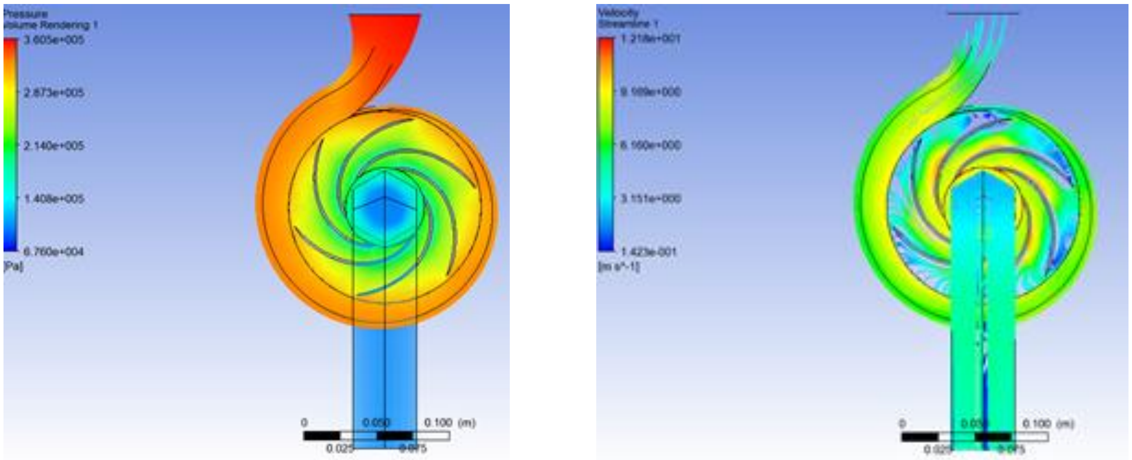


Figure.4(b) . Pressure and Velocity Distribution for the Impeller with Sharp Tip

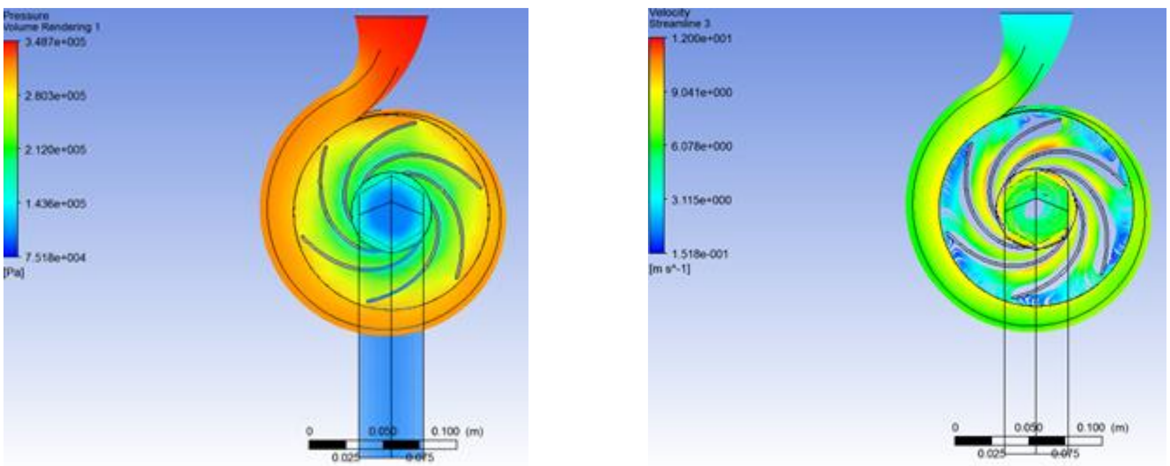


Figure.4(c) . Pressure and Velocity Distribution for the Impeller with Round Tip Modification

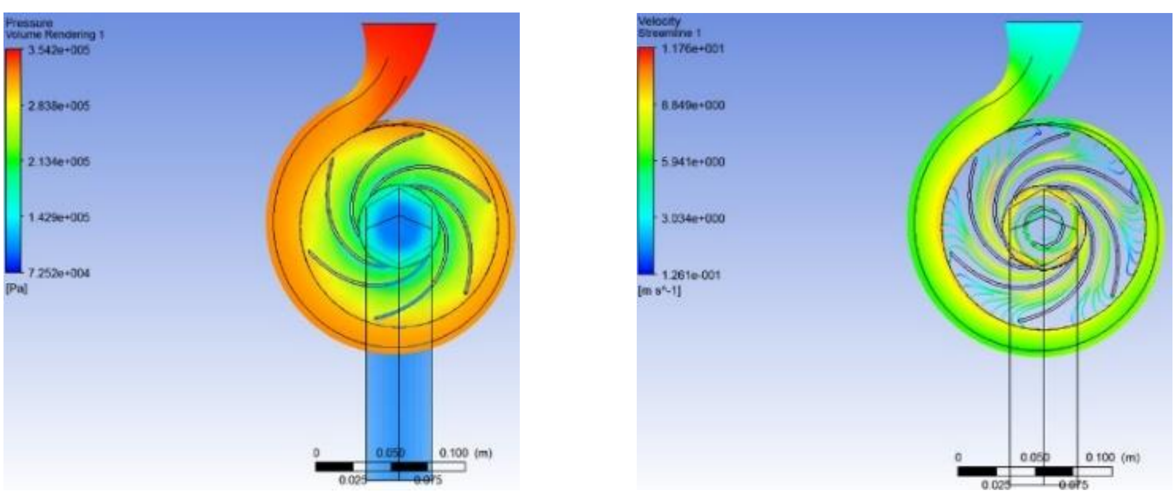


Figure.4(d). Pressure and Velocity Distribution for the Impeller Bullet Tip Modification

The total pressure patterns are varying along the span of the impeller. Low total pressures are observed near hub of the impeller. A low total pressure and high velocity is observed near the trailing edge on suction side of the blade because of the vane thickness. At leading edge of the blade total pressure loss is observed for all span wise locations due to the wake formation at trailing edge of the blade. At trailing edge, pressure drop on both pressure and suction side are observed due to the acceleration of the flow in to the impeller. By seeing the simulation result, the pressure side along the blade after modification is more increased than before modification.

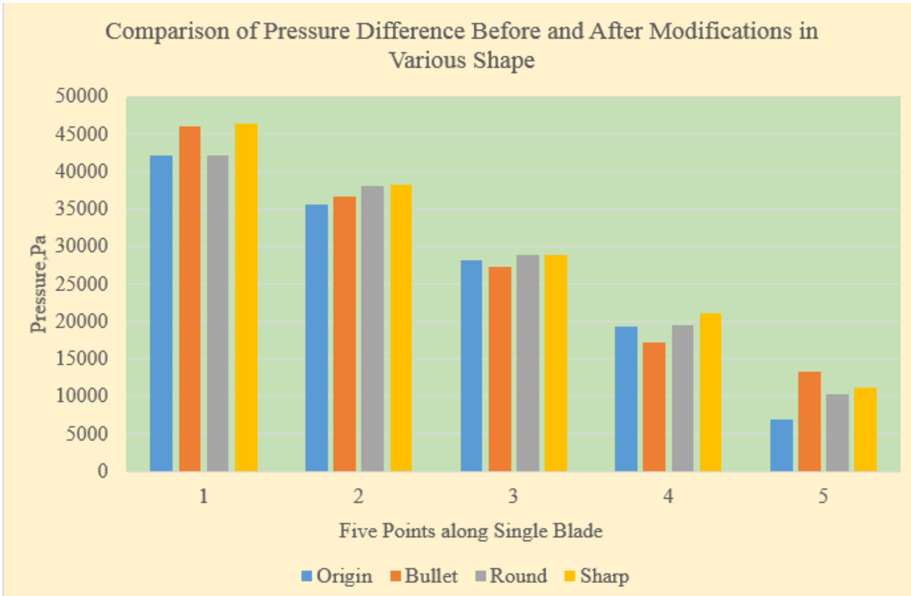


Figure.5. Comparison of Pressure Difference between Before and After Modification

IV. Methodology

Figure.6 shows the schematic of the hydraulic test rig that was adopted for the PAT test. The main components consist of a feed pump, PAT, test bed, Pressure gauges , PVC piping network and control valves. Bends and tees were used as the pipe connectors and fittings. All the pipes and fittings were fitted with flanged joints and connected with bolts and nuts.

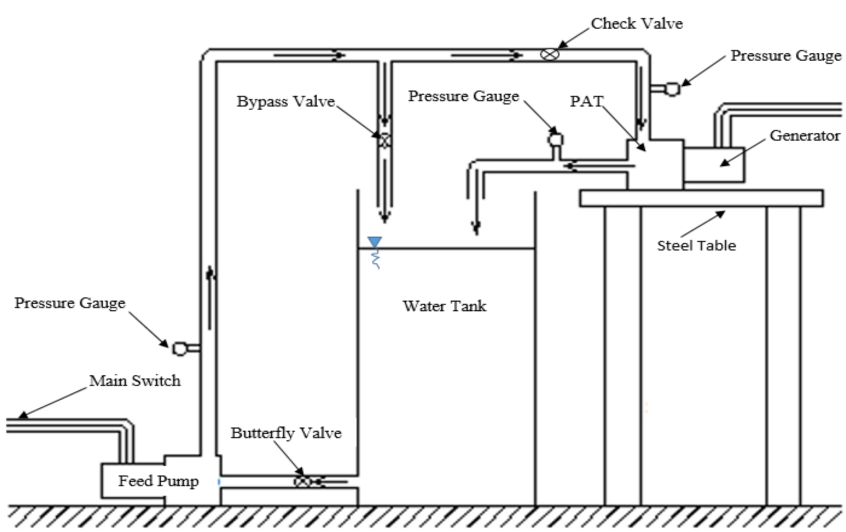


Figure .6. Schematic view of the hydraulic test rig

Two pressure gauges were positioned at each end of the PAT to measure pressure difference across the device. The driving force behind the flow of water across the PAT was recorded as a pressure drop at the corresponding flow rate. The hydraulic test rig was an open-loop system. The inlet and outlet piping systems were joined to water tank that acted as a reservoir. The open-loop system allowed a continuous water supply to the piping system. The inlet of the feed pump was positioned at the lower water tank line to ensure sufficient suction pressure. This served to prevent flow separation and reduce air suction to the feedpump. The pressurised water flowed through the piping network to the PAT before being returned to the water tank.

A. Test Rig Main Components and Measuring Instruments

The hydraulic test rig consists of six main components and measuring instruments as describe below.

Feed pump: The feed pump that was used to supply pressurised water to the PAT was a monoblock pump with a power rating of 2.2 kW and 2900 rpm maximum speed. The pump used an induction generator with a single-phase power input. The feed pump was mounted on a large fixed frame bed and secured by a bolt connection. This kept the pump attached and stabilized thus reducing the vibration generated during the experiment. This pump will produce a maximum water pressure of 22.2 m at a flow rate of 600 l/min. The additional pressure at the pump inlet will be added to the total pressure generated at the pump outlet.

Pump as Turbine: The centrifugal pump used as Pump as Turbine was a low specific speed pump that delivered high pressure at a low flow rate. The impeller had a diameter of 127 mm, and the flange diameter was 50.0 mm at the inlet and 54.5 mm at the outlet. The pump's power rating was 200 W, and the rotational speed was 1650 rpm. The best efficient point was at 65.0% with a corresponding pressure and flow rate of 9.05 m and 240 l/min, respectively.

Piping system: The pipes were made from polyvinyl chloride, PVC. The wall thickness was recorded at 3.0 mm, with the permissible working pressure of 12 bars at 20° Celsius. Reducers and expanders were used to connect the pipes to the pump that had a different diameter. The PVC pipe was fitted with a set of connectors and fittings including tees, bends, and a valve with flange ends. PVC pipe with diameter of 50 mm was used for the experimental test rig .A bypass pipe was connected from the main pipe to regulate excess flow and pressure.

Water tank: The water was re-circulated within the polythene tank having capacity of 80 gallons.The water tank was connected by a pipe at the side wall of the tank. Two control valves were used to adjust water flow to the PAT and to regulate excess flow. The water level was kept at a defined height to generate sufficient suction pressure at the feed pump inlet. The iron pallet held the tank foundation with a iron frame for support.

Control valve: The feed pump did not have a control mechanism to regulate the flow and pressure generated at the pump outlet. Two control valves controlled the flow rate. The flow rate was adjusted by manipulating the opening of the control valve and diverting a portion of the flowing water from the piping system to the water tank, by passing the PAT.

Bulbs: Three pieces of bulbs, 100W each were used as ballast load for the generated power. On/off toggle switches were used to increase the ballast load by switching on the 100W bulbs.

Pressure gauge: The pressure gauges with a measuring range of 0 psi to 30 psi were placed at the outlet of the feed pump and at the inlet and outlet of the PAT to measure the head. The pressure difference between the two points gave the pressure difference across the PAT.

Clamp meter: One of the most basic measurements of a clamp meter is current. Another common function for a clamp meter is measuring voltage.

Tachometer: The speed of the generator can also be measured by using the tachnometer. Figure .8. is the tachometer to measure the speed.

Figure.7. Clamp Meter

Figure .8.Tachometer

Figure .9. Gate Valve

Figure .10. Pressure gauge

B. General Assembly and Operation Procedure

Figure .11. shows the completed hydraulic test rig that was used to test the PAT. The arrangement of rig components was adapted to suit the existing hydraulic lab facilities. The rig was designed to have modular components, which would allow for a simple system that could be upgraded for future research. The hydraulic test rig was fabricated and installed in the Mechanical Laboratory of Mandalay Technological University. The feed pump supplied a continuous flow to the piping system. The water was contained in the reservoir and was open to atmospheric pressure.Two control valves were used for flow rate control between the PAT and feed pump. The flow was adjusted by regulating the opening of the two valves byhand.

As the water entered the PAT, the impeller rotated and produced rotational power. The impeller shaft was coupled generator. The desired rotational speed was measured by applying the tachnometer. Meanwhile, the corresponding pressure across the PAT was recorded with a pressure gauge positioned at the inlet and outlet of the PAT. Each experimental process gave a set of data consisting of flow rate Q, pressure P, current I, voltage V and rotational speed N. The purpose of the experimental work on the PAT was to determine the efficiency, flow rate and power over the operation rotational speed.

Figure.11. Complete hydraulic test rig

Firstly, the water is pumped to the inlet pipe from the open tank. The flow rate to the turbine is controlled by gate valve which has twelve cycles (fully open). The valve was marked at opening one cycle. In this test, the flow was controlled by opening gate valve starting from one cycles to twelve cycles (fully open). In the event of an overflow, there is a by-pass pipe line by opening by-pass valve to the tank.The pressure is measured by the pressure gauge on the suction and discharge pipe of PAT. The runner rotational speed is measured by tachometer. The voltage and current are measured by clamp meter.

C. Manufacturing Impeller with the 3D Printer

All the parts created using a 3D printer need to be designed using some kind of CAD software. This type of production depends mostly on the quality of the CAD design and also the precision of the printer. There are many types of CAD software available. The model to be manufactured is built up a layer at a time. A layer of powder is automatically deposited in the model tray. The print head then applies resin in the shape of the model. The layer dries solid almost immediately.

In case of impeller manufacturing using RP, the initial phase of this procedure is exactly the same as CAD/CAM method. Scanned data prepared and transferred to the software, after analyzing and refining the data it transferred to 3D printing machine and 3D impeller is produced. After data analyze, and drawing 3D shape of impeller it is ready to transfer to RP machine, machine receive the data and produce initial model of impeller layer by layer.The model tray then moves down the distance of a layer and another layer of power is deposited in position, in the model tray. The print head again applies resin in the shape of the model, binding it to the first layer. This sequence occurs one layer at a time until the model is complete.

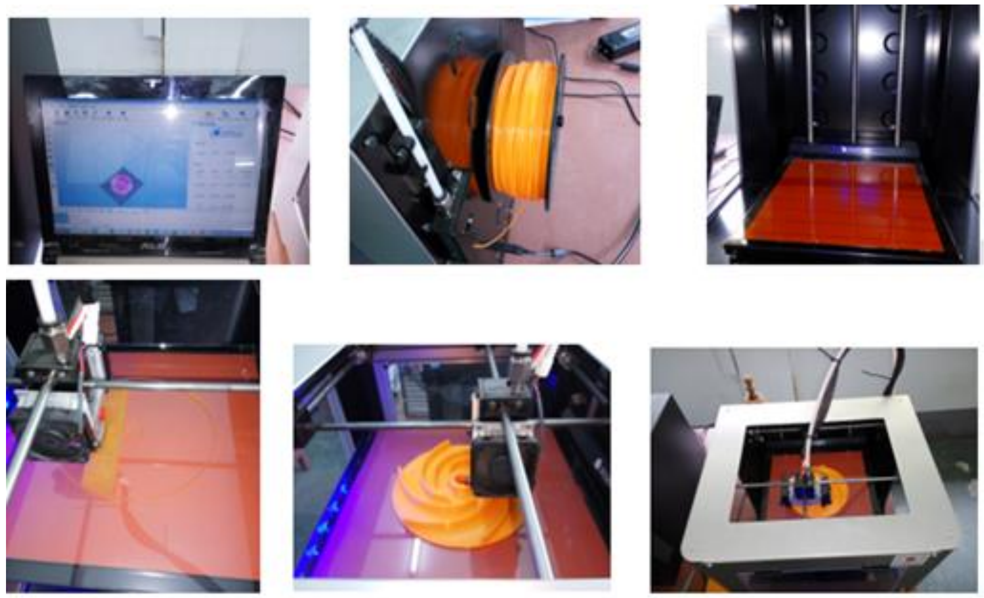


Figure.12. Manufacturing Impeller using Rapid Prototyping

Figure.13.Primary Pattern of Impeller Produced by 3D Printer (Using PLA Material)

The materials that used for printing the impeller are Acrylonitrile Butadiene Styrene (ABS) and Polylactic Acid (PLA) .ABS is made of oil-based resources and has a higher melting-point, it is stronger and harder than PLA plastic. PLA is made of plant-based resources (corn starch or sugar cane) and it is biodegradable. Both ABS and PLA have advantages and limitations, and choosing a printer depending on the used material could be a good idea, as they have different properties: ABS as a longer lifespan and has a higher melting point, while PLA is more malleable, easier to use, looks better and is suitable for creating artistic 3D objects.

D. Impeller manufacturing using conventional sand casting process

In conventional method, manufacturer use casting procedure to produce impeller. It will be taking less cost compare to other methods but it has some other limitation, which compel manufacturer to select and try other methods depending the case, which they want to produce. Such like other casting operation it need to main section such as die, core and core box. That is the traditional core box making method. In this way, the wooden or plastic core box produced by hand via professional model maker. These initial blades are used to making sand core. In this regard, aluminum core box will be casted via this wooden core box and prepared to make main core to use for final casting of impeller.Mentioned method is old and traditional method in reverse engineering of impeller and it is to keeping being not more used method. Because most of the work is done by hand and actually, the mistake of labor transferred to work piece so it wouldn't be so accurate and dependable. But as advantage of this method the cost issue is so lower than compare to another method and it has also done in some traditional workshops. The finished blades joint to shroud part of impeller and finally are used as core box to make sand core.

Figure.14. Pattern Making Process

Figure.15. Metal Casting process

Figure.16. Making Sand Mold

Figure.17. Making Sand Mold

Figure.18.Lathe Machining Process

Figure.19. Bronze Casting Process

Casting technology has been observed and evaluated in order to produce model and core box for our case study. After the aluminum core box has been prepared and also, dimensional control has checked, now, it is ready to produce the sand core of impeller by foundry experts. Special sand is mixed properly with particular glue and poured inside of core box then CO_2 gas is injected to inside of Die for solidification purposes.

E. Modification on the Geometry

The main difference in the pump and turbine design is that, conventional turbines having flow control mechanism to increase its part load efficiency but the standard pumps are not having any flow control mechanism to increase its part load efficiency. Pumps are generally operated with constant speed, head and flow. A pump is therefore designed for one particular of operation for maximum efficiency and it not required any regulating flow control device. Turbine operates under variable head and flow condition. In various seasonal variations of the available water so to adjust the power output and consumer requirement regulate the flow regulator are required. Also in the reverse operation of pump it may be less efficient because the direction of flow is reverse and hydraulic and frictional losses increase sharply. When fluid enters in to impeller because of round edges of impeller at the end flow separation occurs at inlet of PAT due to that separation losses occurs.

Modification is performed by grinding the inlet ends of the impeller tips of the PAT in various shape to preclude excessive turbulence for efficiency consideration. Then, testing is carried out by operating the pump as turbine at the maximum head and at various capacities.The sharp tip, round tip and bullet-nose tip impeller that used in experimental test are shown in Figure.20, 21 and 22.

Figure.20. PAT Impeller Sharp-edge Tip Modification

Figure.21. PAT Impeller Round Tip Modification

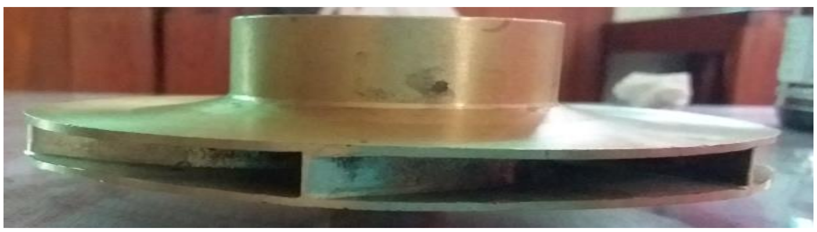


Figure.22. PAT Impeller Bullet Tip Modification

V. Experimental results

The experiments are carried out on the PAT to find out the performance at various flow rate with impeller in modified and non-modified condition. From the measurements and calculation that have been done in the experiment as presented in Table.1.

Table.1: Result Data of Current and Voltage for Different Types of Impeller Tips

I_f	V_f	I_s	V_s	I_r	V_r	I_b	V_b
0.31	3.47	0.46	5.0	0.72	5.2	0.45	4.3
1.126	59.2	1.23	65.9	1.3	69.1	1.2	60.3
1.393	88	1.48	90.4	1.45	94.1	1.41	88.9
1.426	92.67	1.54	99.5	1.57	99.2	1.52	96.93
1.483	99.7	1.59	104.2	1.6	105.5	1.53	101.1
1.51	103.43	1.63	107.8	1.65	108.3	1.59	106.1
1.56	104.47	1.65	109.8	1.66	109.7	1.60	107.53
1.563	105.10	1.67	111.5	1.68	113.0	1.61	108.7
1.566	106.97	1.69	111.9	1.70	111.4	1.61	109.4
1.583	108.67	1.7	112.9	1.71	111.3	1.63	110.33
1.593	108.37	1.68	111.5	1.71	111.5	1.64	111.1
1.6	110.36	1.69	112.2	1.72	112.1	1.66	111.2

Table.2: Result Data of Flow Rate, Head, Speed, Power and Efficiency for Different Types of Impeller Tips

Q	H	N	P_f	P_s	P_r	P_b	η_f	η_s	η_r	η_b
35	2	820	1.08	2.3	3.68	1.935	14.5	13.16	21.06	21.1
240	6.02	1390	66.66	81	89.83	72.96	43.1	48.17	50.89	51
250	9.05	1551	122.58	133.8	136.5	125.34	44.4	50.86	53.39	52.1
254	11.38	1596	132.15	153.2	155.7	147.33	42.5	45.56	46.30	46.7
262	12.17	1632	147.86	165.7	168.8	154.68	42.1	44.68	45.53	45.4
268	12.68	1637	156.18	175.7	178.7	168.69	41.1	44.46	45.22	44.5
270	13.06	1640	162.97	181.2	183.1	275.27	40.8	44.12	44.34	44.5
276	13.25	1643	164.27	186.2	186.9	175.01	40.1	43.72	44.57	43.8
278	13.41	1645	167.52	189.1	189.9	176.14	39.6	43.59	43.65	43.5
282	13.5	1646	172.03	191.9	190.3	179.84	40.2	43.19	42.93	43.1
290	13.53	1649	172.63	187.3	190.7	182.21	38.9	40.93	41.75	41.8
330	13.56	1650	176.58	190	192.81	184.59	34.3	36.39	37.00	41.5

VI. PERFORMANCE CHARACTERISTIC

Performance analysis with different type of impeller tips are shown in the following figures.The graph shows that the head extracted from the turbine is increasing if the flow of water capacity is increased both on before and after modification of pump impeller tips. In addition, the speed, the power and efficiency of the pump as turbine after modification is slightly higher than before modification.

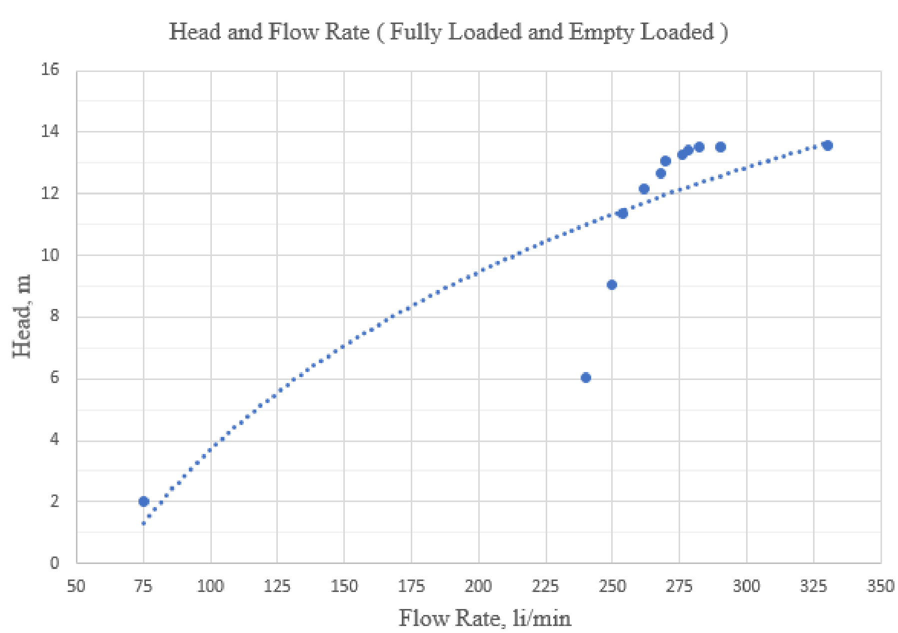


Figure.23. Flow rate verses Head with Different Type of Impeller Tip

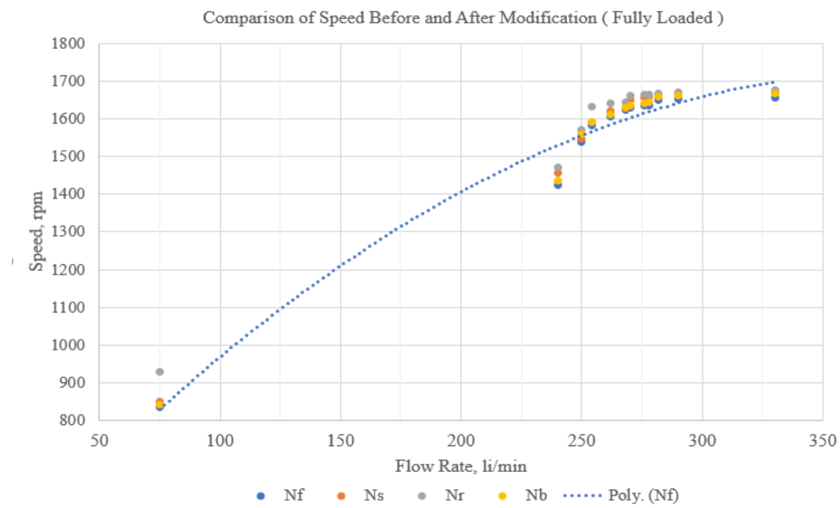


Figure.24. Flow rate verses Speed with Different Type of Impeller Tip

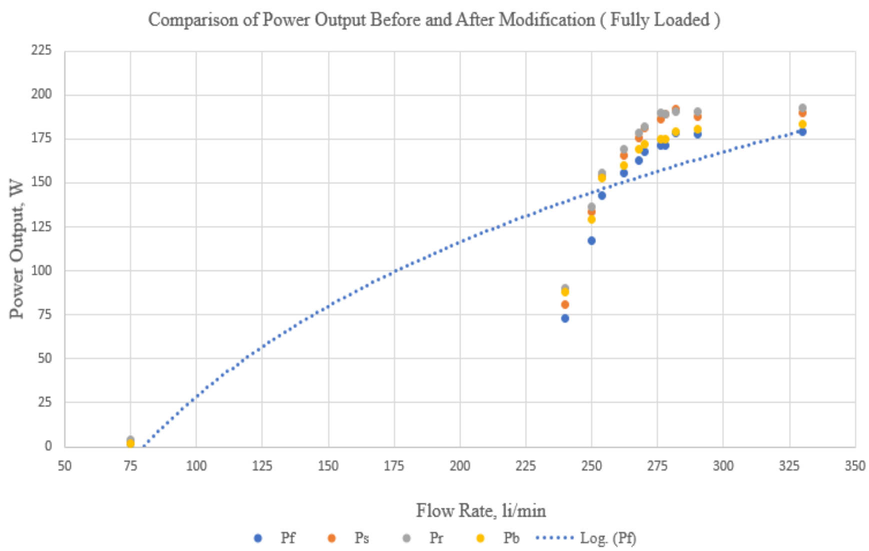


Figure.25. Flow rate verses Power Output with Different Type of Impeller Tip

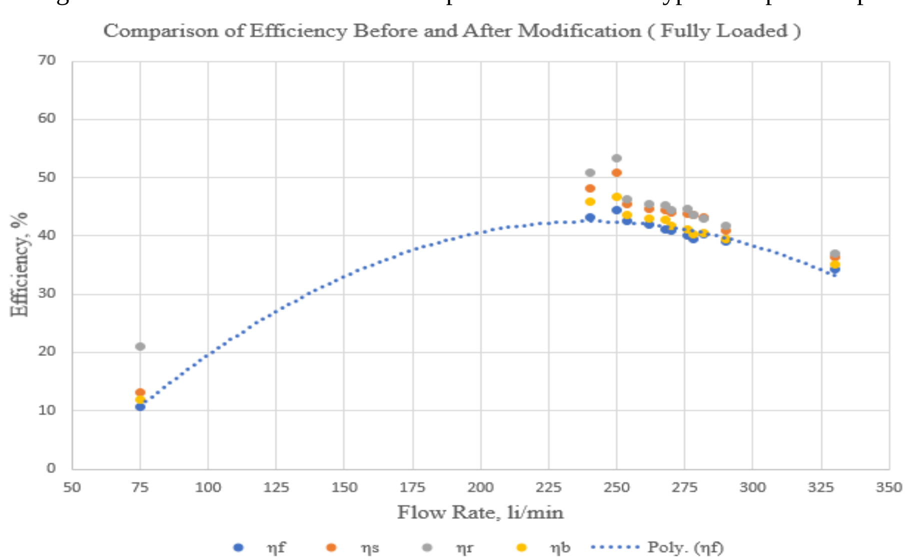


Figure.26. Flow rate verses Efficiency with Different Type of Impeller Tip

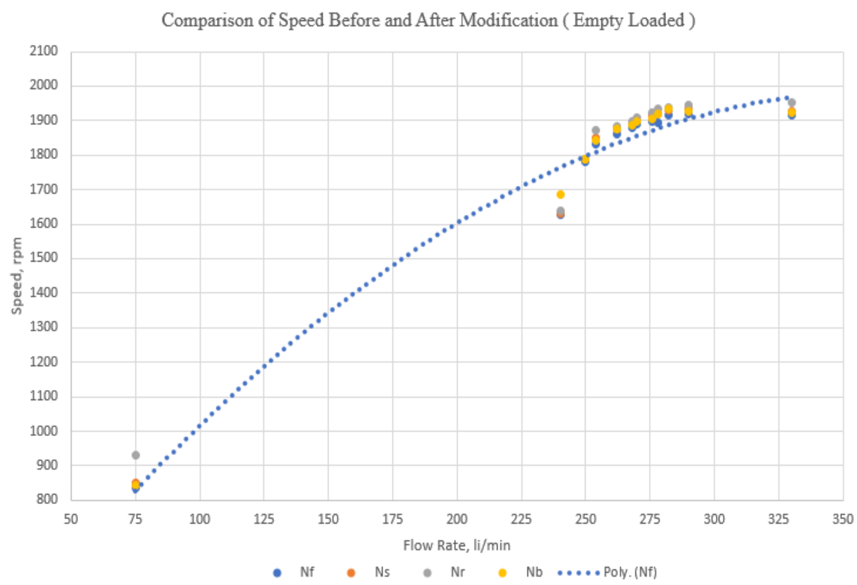


Figure.27. Flow rate verses Speed with Different Type of Impeller Tip

VII. NOMENCLATURE

Symbol	Description	Unit
n_{sp}	Specific speed of pump	-
N_p	Speed of pump	rpm
H_t	Design head	m
P_{out}	Output power	watts
Q	Capacity or volume flow rate ,	m^3/sec
A	Area of pipe	m^2
v	Velocity of flow	m/s
ρ_w	Density of water	kg/m^3
g	Acceleration due to gravity	m/s^2
η_0	Overall efficiency	(%)
η_h	Hydraulic efficiency	(%)
η_m	Mechanical efficiency	(%)
η_v	Volumetric efficiency	(%)
D_1	Inlet diameter of impeller	, (m)
D_2	Outlet diameter of impeller	, (m)
U_2	Absolute tangential velocity of the impeller outlet	, (m^2/s)
K_u	Constant coefficient	-
b_1	Inlet width of impeller	(m)
Q_s'	Leakage flow rate through an impeller	(m^3/s)
V_{m1}	Inlet flow velocity	(m/s)
K_{m1}	Constant parameter	-

P_f	output power for flat tip impeller	W
P_S	output power for sharp tip impeller	W
P_r	output power for round tip impeller	W
P_b	output power for bullet tip impeller	W
η_f	turbine efficiency for flat tip impeller	%
η_s	turbine efficiency for sharp tip impeller	%
η_r	turbine efficiency for round tip impeller	%
η_b	turbine efficiency for bullet tip impeller	%
I_f	Current for flat tip impeller	A
I_s	Current for sharp tip impeller	A
I_r	Current for round tip impeller	A
I_b	Current for bullet tip impeller	A
V_f	Voltage for flat tip impeller	V
V_s	Voltage forsharp tip impeller	V
V_r	Voltage for round tip impeller	V
V_b	Voltage for bullet tip impeller	V

VIII. CONCLUSION

This paper outlined the experimental testing of PAT performance using a hydraulic test rig. The corresponding pressure, current, voltage and flow rate of the PAT system were measured using measuring instruments. From the experimental results, it can be concluded that the centrifugal pump could run in turbine mode with satisfactory operational performance. The ideal flow rate for the pump in turbine mode was higher than in pump mode; however, due to the constraints of the hydraulic test rig, the control parameters were limited to the designed test rig setup. For future research, it is recommended that some of the system components, such as feed pump capacity and diameter of the pipe, are upgraded. This will allow a higher flow rate and greater pressure to be transferred to the pump; consequently, a higher rotational speed that matches the synchronous speed of the induction generator could be achieved, allowing full performance characteristics to be acquired. Efficiency of the volute-pump operating as turbine after modification its impeller tips is slightly better than before modification. Furthermore, both pumps as turbines before and after modification generate high shaft-revolution that is about 1.600 rpm at their maximum efficiency; therefore, it can be coupled directly to the load, a generator for example, without reduction gear. The maximum efficiencies of the pump as turbine both on before and after modification are performed between the rated flow capacity of the pump at its maximum efficiency and the maximum flow capacity of the pump.

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