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Performance of a Centrifugal Pump with Variation of Impeller Speed

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***Abstract* - This research is addressed to investigate the characteristics of a centrifugal pump. In this work, the experiments were carried out with variations in impeller speed and at several valve openings. The measurement results show that head have the same tendency for all variations of inlet valve opening. Meanwhile for discharge and break horse power have almost the same values at low speed for both theoretical and measurement; however at higher speed there is a significant difference. Large inlet valve openings produce large discharges as well as break horse power and motor horse power. Meanwhile for the head, the inlet valve opening has no crucial effect because it produces almost the same value.**

Keywords: Centrifugal pump, impeller, speed, discharge.

I. INTRODUCTION

The use of centrifugal pumps has been found in various sectors ranging from agriculture, residential and commercial buildings to industrial sectors such as the water processing industry, chemical industry, oil and gas, and the food and beverage processing industry. In the chemical industry, pumps are used to convey corrosive fluids such as acids and alkalis, flammable fluids such as oil and gasoline, and are also used for fluids with high viscosity such as mud. In the power generation sector, centrifugal pumps are used to circulate water in cooling systems, circulate fuel and lubricants, and pump condensate from the condenser to the boiler.

One of the main problems that affects pump performance is the formation of losses. Losses in centrifugal pumps can be caused by friction between fluids due to viscosity or by changes in flow direction. Perissinotto et al [7] in their study showed that losses can be caused by mechanical components in piping installations such as elbows, valves, and bearings. Losses in the pump cause a decrease in performance which will affect pump efficiency. Therefore, the pump must have sufficient energy to resist pressure losses caused by the flow and mechanical components of the pumping system.

Capuroso et al [1] introduced a new generation of centrifugal pumps. In particular, the impeller has been redesigned improving the flow guidance inside the vanes and simultaneously reducing the number of blades, with the aim of increasing the blade loading, reducing manufacturing costs, and enhancing the conversion efficiency with respect to conventional configurations. Then, Foslie [2] verified the calculation of pump characteristics and velocity profiles at the impeller outlet through testing. He presented a detail description of the relevant theory regarding pump design, and examined different calculation models for the pump characteristics.

Li et al [5] made a detailed review of theoretical guidance for the design and operation of pump-turbines, in terms of the stability of the pumped-storage powers plants. including flow characteristics, and influence parameters, related to positive slopes. The generating mechanism, accompanying unsteady phenomenon, and corresponding hysteresis phenomenon. Investigations on the influence of geometric and operating unit parameters are presented

Another important issue that usually arises with centrifugal pumps is the appearance of cavitation. Cavitation is the process of forming steam bubbles when the static pressure of water drops below its saturated vapor pressure. Cavitation can damage pump parts, especially the impeller, which can reduce pump performance. Li et al [6] tested the formation of cavitation in the pump causing vibration intensification when the pump was working which would then reduce the NPSHa value of the pump. Therefore, a pump is expected to work as well as possible to avoid the formation of cavitation during pump operation. Meanwhile, Suhane [8]used simultaneous measurements of vibration and sound for variable speed pump to detect cavitation. The results of the study show that the occurrence of cavitation depends on the rotational speed, and the sound signals in both no cavitation and cavitation conditions appear in random manner. Meanwhile, surveying the vibration and sound spectrums at the second, third, and fourth blade passing frequencies reveals no indications or phenomenon associated with the cavitation at variable speeds.

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Zhang et al [9] made various efforts to improve and increase the performance of centrifugal pumps such as head, efficiency, NPSHr, or vibration performance for high pressure pumps. Juckelandt et al [3] investigated the performance of pumps to compare wall roughness conditions, where pumps with higher surface roughness will reduce the efficiency of the pump. Meanwhile, Kim et al [4] in their work found a result in increasing pump efficiency to 97.1% by designing the impeller and carrying out numerical analysis using CFD.

Zhang et al [10] used the fluid-structure interaction (FSI) to investigate the transient vibrations performance of centrifugal pumps with various blade shapes. They found a good consistency for the comparison results of Kriging surrogate model, FSI simulation, and experimental test.

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

II. METHODOLOGY

2.1 Test Model of Pump

The pump used in this test is a centrifugal pump made by EBARA type JESX/I 6 IE3 as seen in Figure 1. This pump has specifications for maximum delivery head= 36 m, maximum flow rate = 45 L/min, and rotation = 2,820 RPM. This pump is driven by an asynchronous motor with power = 0.44 kW. For pressure measurements, there are two manometers installed on the intake and delivery sides with a height difference between the two manometers = 0.18 m. In this pump system there is a valve on the delivery side to regulate the fluid flow rate coming out of the pump.

Figure 1: Centrifugal pump test model

2.2 Basic Module for Measurement

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

Figure 2: Basic module for measurement

2.3 Instrumentation

Some installed instruments for this measurement: first intake side manometer: this manometer is used to measure the pump inlet pressure. This type of manometer is Bourdon tube RKG 63, with a measurement range from -1 bar to 1.5 bar. The accuracy of the manometer is based on EN 837-1. The diameter of this manometer is 63 mm with a connection size of G ¼”.secondly, delivery side manometer: this manometer is used to measure the outflow pressure from the pump. This type of manometer is Bourdon tube RKG 63, with a measurement range from 0 bar to 2.5 bar. The accuracy of the manometer is based on EN 837-1. The diameter of the manometer is 63 mm with a connection size of G ¼”.Then, remote sight gauge: this tool is a scaled pipe column connected to a volumetric tank. This tool is used to measure the volume produced by the pump per unit time. Stopwatch: this tool is used to measure the time required for the water flow out of the pump to fill the volumetric tank to the specified volume.

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III. RESULTS AND DISCUSSIONS

3.1 Basic Characteristics

3.1.1 Flow rate

The results of the discharge measurement as a function of rotation are shown in Figure 3. From this plot it can be seen that the volumetric flow rate increases as the pump rotation increases. The minimum flow rate obtained at the minimum rotation is 16.67 rps with a flow rate value = 0.399 × 10^{-3} m^3/s. This discharge value continues to increase linearly with impeller rotation until rotation = 33.33 rps. At rotations of more than 33.33 rps, the discharge tends to be constant, namely around 0.823 × 10^{-3} m^3/s. From this curve, it can also be seen that there are differences between measurement results and theoretical calculations, where in the initial conditions, the measurement data quite agrees with the theoretical calculations. However, from the 16.67 rps rotation onwards, the measured discharge value tends to be constant, this is different from the theoretical calculation results which continue to increase.

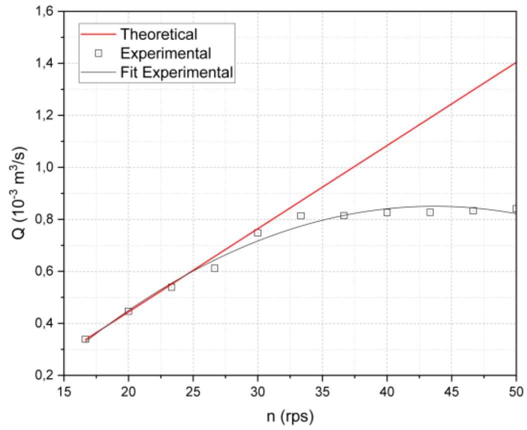

Figure 3: Comparison of flow rate as a function of impeller speed using theoretical and measurement

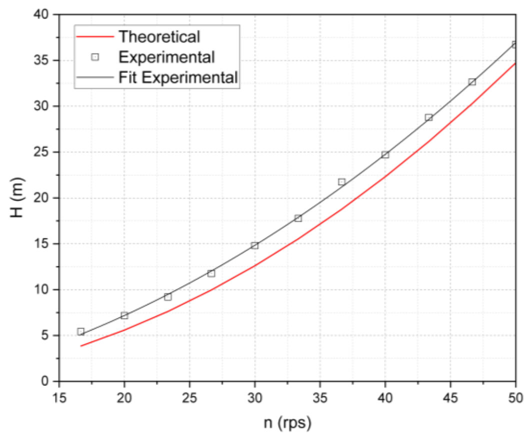

Figure 4: Comparison of flow rate as a function of impeller speed using theoretical and measurement

3.1.2 Head

Figure 4 is the result of head measurements as a function of impeller rotation which is compared with the results of theoretical calculations. From this plot, it is found that the ratio of head to rotation values in the measurement results and theoretical calculations is quite in agreement where the curves formed always coincide. Apart from that, from this plot it can also be seen that the fitting results of the test data obtained have a curve shape that is similar to the curve resulting from theoretical calculations. Based on the law of affinity, as the rotation increases, the head value will increase parabolically. This is in accordance with the measurement results where the minimum head is at the lowest rotation of 16.67 rps with a head value = 5.45 m, while the maximum head value is obtained at a rotation of 50 rpm with a head value = 36.83 m.

3.1.3 Break Horse Power (BHP)

A theoretical and experimental comparison of the BHP from the test results is presented in Figure 5. Based on this plot it can be found that in the initial conditions, the scatter point is around the theoretical curve until the pump speed = 33.33 rps. At the top of this circle the scatter point deviates far from the curve. Based on the law of affinity, BHP will increase with an increase in the third power of the impeller rotation. The BHP increases as the impeller rotation increases, but the value is not as high as the theoretical calculation results, especially at impeller speeds above 33.33 rps. This quite significant difference can be seen at a maximum rotation of 50 rps with impeller BHP = 299 W. Meanwhile, at the same rotation, theoretically the impeller will use a BHP of 746.8 W.

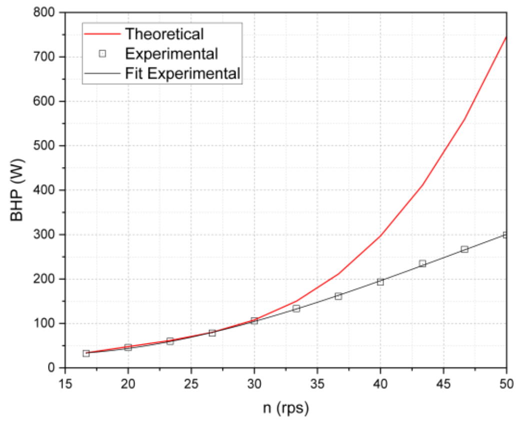

Figure 5: Comparison of flow rate as a function of impeller speed using theoretical and measurement

3.2 Performance of Pump

3.2.1 Flow rate

The plot in Figure 6 is the result of testing the flow rate as a function of rotation at three inlet valve openings.

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Variations in inlet valve openings are divided into three, namely Partially Closing (PC)-1 is the largest opening and Partially Closing (PC) -3 is the smallest opening. Based on this plot it can be seen that the discharge changes as the impeller rotational speed increases. PC-1 produces a higher discharge than the other two conditions PC-2 and PC-3. This is because the valve position on PC-1 is open to a greater extent, thus allowing more water to enter the pump. Meanwhile, the trend of the discharge curve for each opening condition is quite the same as the full opening condition where the discharge tends to increase linearly with increasing rotation up to 33.33 rps. The increase in discharge is not very significant until maximum rotation. The pump does not experience a significant increase in flow rate above 33.33 rps in all variations of inlet openings.

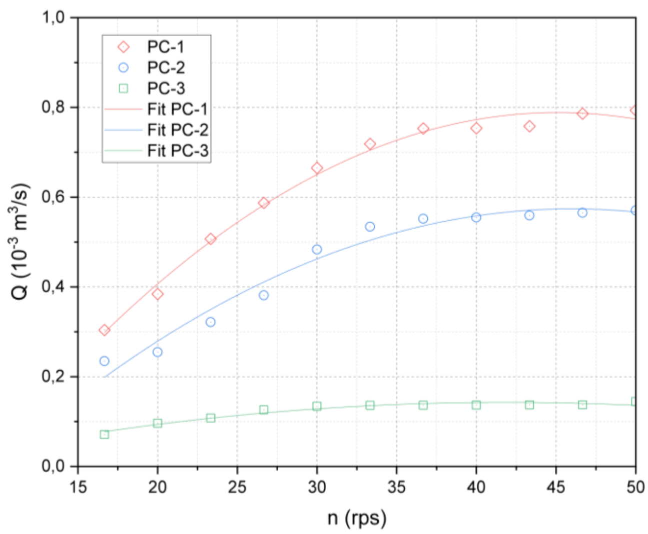

Figure 6: Flow rate as a function of impeller speed at three variations of inlet valve

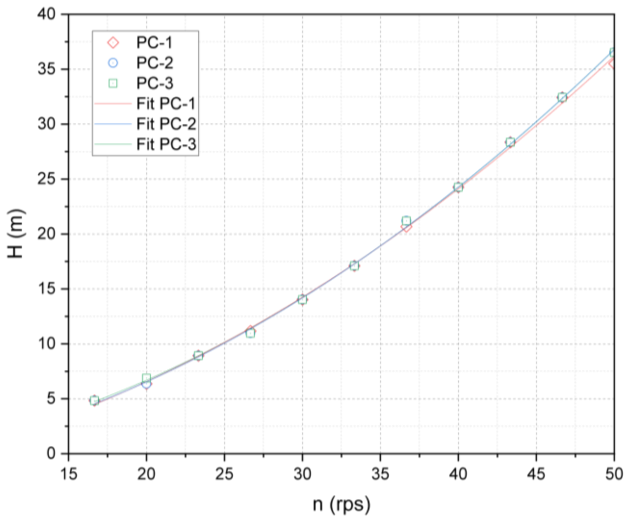

Figure 7: Head as a function of impeller speed at three variations of inlet valve

3.2.2 Head

The results of the head test as a function of rotation for three inlet valve openings are presented in Figure 7. From this plot, under PC-1 conditions with the lowest rotation of 16.67 rps, the pump can flow water with a flow rate of 0.304×10^{-3} m^3/s. The discharge generated at a maximum rotation of 50 rps is 0.794×10^{-3} m^3/s. On PC-2, the minimum discharge produced is 0.235×10^{-3} m^3/s and the maximum is 0.570×10^{-3} m^3/s, respectively at 16.67 rps and 50 rps rotation. Meanwhile, at the smallest opening PC-3, the pump produces a much smaller discharge than the two previous opening variations PC-2 and PC-3. The minimum discharge produced is 0.071×10^{-3} m^3/s and the maximum discharge is 0.145×10^{-3} m^3/s. At a rotation of 16.67 rps, three variations of inlet openings produce the same head, namely 4.83 m. Meanwhile, at a maximum rotation of 50 rps, the resulting head is different, although not too significant for each PC-1, PC-2, and PC-3 respectively, namely = 35.51, 36.53, and 36.53 m.

3.2.3 Break Horse Power (BHP)

The BHP as a function of rotation for three variations of inlet valve opening are as shown in Figure 8. Based on this plot it can be shown that BHP has a curve shape that is similar to the MHP curve even though it has different values. This is because BHP is obtained from the results of multiplying MHP with the efficiency value of the asynchronous motor used to rotate the shaft, which is 0.92. At a rotation of around 16.67 rps at each inlet opening variation, the BHP is 35 W. Meanwhile at a rotation of 50 rps, the pump BHP decreases for each inlet opening, namely 322.5 W, 307.5 W and 275 W respectively. for PC-1, PC-2, and PC-3 respectively.

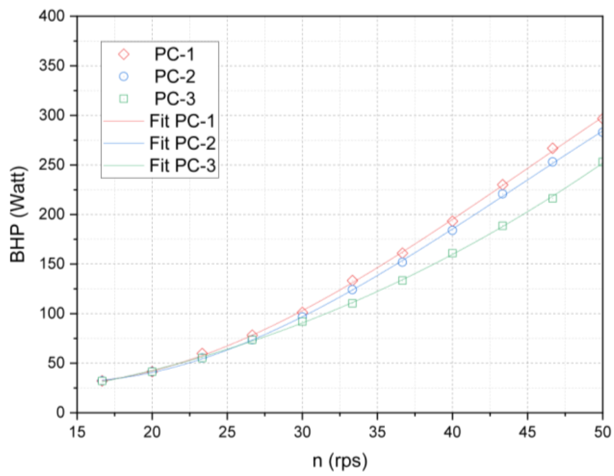

Figure 8: BHP as a function of impeller speed at three variations of inlet valve

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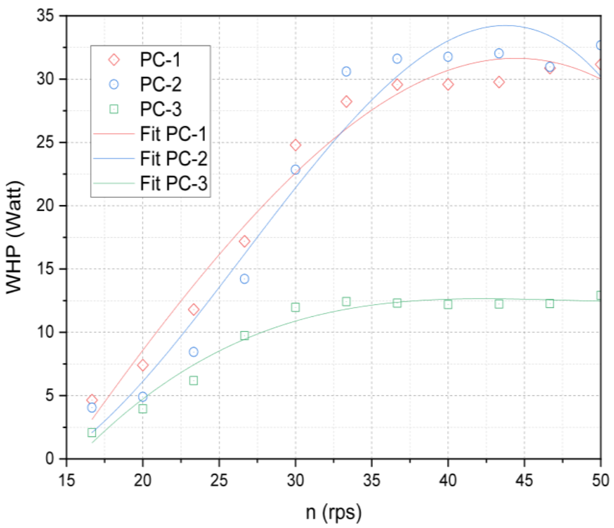

Figure 9: WHP as a function of impeller speed at three variations of inlet valve

3.2.4 Water Horse Power (WHP)

Figure 9 is the measurement result of WHP as a function of rotation for three variations of inlet opening. From this plot it can be seen that WHP produces an increase that is not as consistent as the BHP or MHP curves. Starting from 33.33 rps onwards, the increase in WHP produced by the pump is not very significant or tends to be constant. In general, the WHP produced in the PC-1 variant is greater than the other two opening variations. However, at round = 33.33 rps and so on, the PC-2 condition has a larger WHP than PC-1. At a maximum rotation of 50 rps, the PC-2 condition produces a WHP of 32.667 W while the PC-1 condition produces a WHP = 31.163 W.

3.2.5 Motor Horse Power (MHP)

The MHP test results as a function of rotation for three variations of inlet valve opening are plotted in Figure 10. The results of this plot show that the motor power used increases as the impeller rotational speed increases. This indicates that the pump requires more motor power to increase the impeller rotation speed. Apart from that, it can also be seen the influence of variations in inlet openings on the resulting plot. PC-1 produces plots with the highest range of MHP values compared to other aperture variations. At a speed of 16.67 rps at each inlet opening variation, the MHP provided by the motor is 38.043 W. Meanwhile at 50 rps rotation, the MHP decreases at each inlet opening, namely 350.543 W, 334.239 W and 298.913 W respectively for PC-1, PC-2, and PC-3.

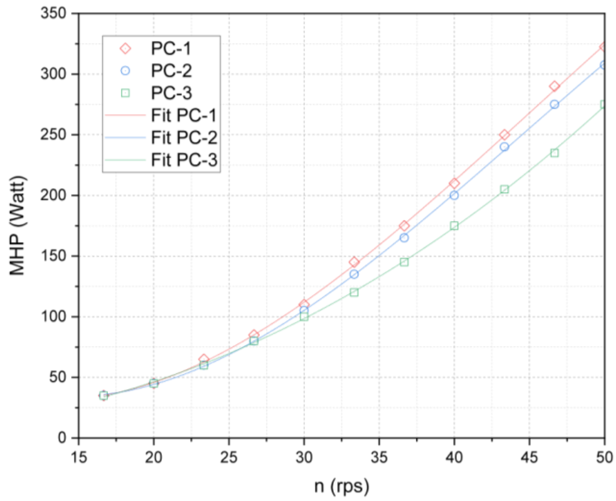

Figure 10: MHP as a function of impeller speed at three variations of inlet valve

3.2.6 Efficiency

The pump efficiency as a function of rotation for three variations of inlet opening can be seen in Figure 11. From this plot it can be seen that the pump efficiency has the same trend. In all PC conditions. The maximum pump efficiency is around 30 rps with an efficiency value of 22.54%. In PC-2 conditions, the pump efficiency is around 33.33 rps with an efficiency value of 22.66%. Meanwhile, in PC-3 conditions, the pump efficiency is at 26.67 rps with an efficiency of 12.17%. Apart from that, it can also be seen that PC-2 is always greater than PC-1 when the impeller rotation is above 30 rps. Another thing that can be obtained from the plot is that there is data instability on PC-2 in the 20 rps round which does not follow the trend of other data, where the value is smaller than the previous data. This might happen due to instability of the rotor blade of the impeller produced by the pump when the rotation speed is 20 rps.

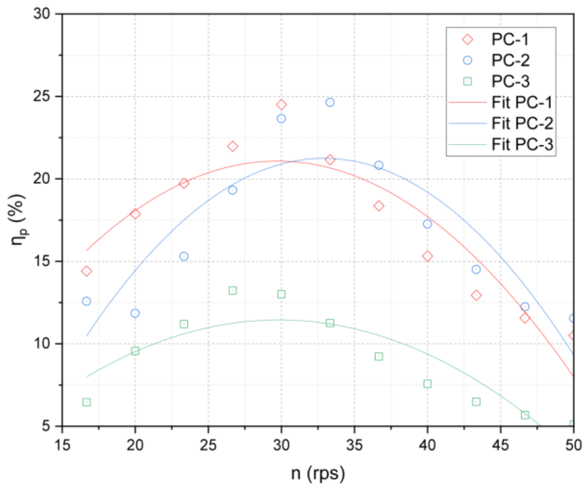

Figure 11: Efficiency as a function of impeller speed at three variations of inlet valve

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IV. CONCLUSION

From the results of these experimental investigations, several important information related to the performance of centrifugal pumps can be noted as follows; For head as a function of rotation, the test results and calculations theoretically have the same tendency even though they have different values with an average difference of 2.11 m. Meanwhile for discharge and BHP, the test results and calculations theoretically have almost the same value at low revs (16.66 rps to 33.33 rps). However, at rounds of more than 33.33 rps, the test results and theoretical calculations have significant differences. PC-1 produces discharge with the largest range of values. Meanwhile for the head, the inlet valve opening has no effect because it produces almost the same value at each valve opening. For BHP and MHP, the largest range of values is produced at PC-1. Meanwhile for WHP the largest value was produced on PC-2. For efficiency, PC-2 has the greatest efficiency. Meanwhile, losses data on PC-3 has the largest losses value for this work.

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