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Investigation of the Performance of a Centrifugal Pump Impeller Design with the Addition of a Junction Disc Plate

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Abstract
This research analyzes the impeller design performance that has been modified based on previous impeller designs. The previous impeller design used high engine power consumption due to the total head, so the modification of the impeller design is expected to reduce the engine power consumption. The existing design and the modified impeller design with the addition of the junction disc plate are used by this research. This research used experiment methods and theoretical methods to compare both of impeller design performances. The experiment method measures total head, fluid capacity, engine speed, and engine power consumption. The theoretical method analyzes actual fluid velocity, specific velocity, total suction head, NPSH, and pump efficiency. The results showed that the fluid flow rate was able to increase the efficiency of the centrifugal pump by 2.8%. The conclusion explains that the addition of a junction disc plate produces energy from a steady fluid flow rate to reduce the engine power consumption and escalation of pump efficiency.

Keywords
centrifugal pump, efficiency, fluid capacity, impeller design, junction disc plate, power consumption

1 Introduction

Centrifugal pumps have a relationship with fluid dynamics phenomena due to their function as a fluid transport device based on pressure differences [1]. However, the pressure difference often results in a low-pressure fluid flow before entering the centrifugal pump. This condition is indicated by vibration, noise, and material damage, which is called the cavitation effect [2]. Cavitation has a negative effect on the rotational speed of the impeller, head loss, and increase in power consumption [3–6]. The function of the impeller is to manage the high-pressure fluid flow in the suction section of the centrifugal pump. Optimal impeller performance is measured by analysis of the fluid flow characteristics, including low viscosity, high pressure, and fixed flow conditions [7, 8] An improvement in centrifugal pump performance, especially the impeller design as a runner, is needed to reduce the pressure fluctuations that produce an unsteady fluid flow [9].

Some previous research on improving impeller design have conducted to reduce the cavitation effect. Chen et al. [10] developed a drainage gap technology with the addition of a vice blade overlapping the existing impeller position. It produced a steady fluid flow in the suction section. Skrzypacz and Bieganowski [11] applied the microgroove concept to the impeller design, whose utilization could increase the impeller's ability to transmit power into the fluid flow and improve the centrifugal pump efficiency. Bozorgasareh et al. [12] applied innovatively designed shrouds on the impeller of a semi-open centrifugal pump that can influence the pressure head and pump efficiency. The innovatively designed shrouds create the losses reduced due to the secondary flow and improve the hydraulic performance. It could also raise the pressure head and pump efficiency. Shim and Kim [13] improved the width of the impeller outlet section with Reynolds Average Navier Stokes (RANS) analysis and the Response Surface Approximation (RSA). The rise of the outlet dimension section could reduce the unstable fluid flow through the impeller outlet section and increase the hydraulic efficiency of the impeller. Capurso et al. [14] improved the impeller design by adding some blades at the impeller outlet section. The new design allowed the channel outlets that come from two sides to have circumferentially

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arranged, and therefore it can increase the fluid flow. The new design also creates the homogeneity of fluid flow on the outlet section and reduces the slip factor from the previous impeller design. Doshi et al. [15] changed the impeller rotation position and entry angle to lessen fluid flow instability and pump losses by 5–10%. Impeller design is also influenced by the blade, which is its main component. Blade designs with different angles can create blade geometry differences to obstruct the cavitation rate in the impeller inlet due to the pressurized fluid flow diffusion phenomenon [16]. The angle of the blade inlet on the impeller suction section, which is small, is also capable of producing a positive impact on the fluid flow characteristics, especially total hydraulic pump performance, with an angle range of 10° to 30° [17].

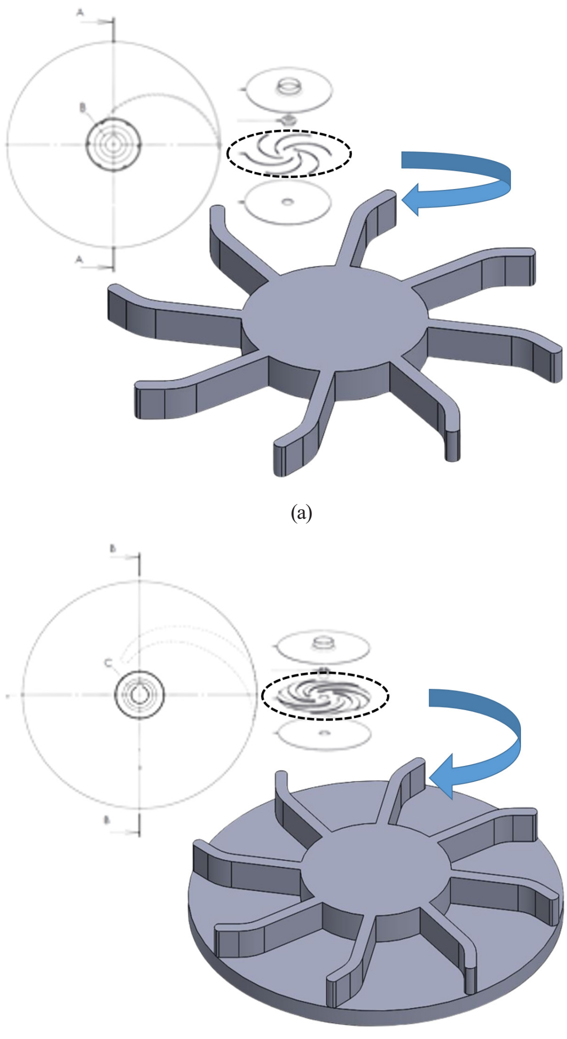
The reduction in pump losses and the cavitation effect by modifying the impeller design, both in geometrical terms and switches in blade position and angle, can increase fluid flow capacity and centrifugal pump efficiency, as shown in some previous research. Based on this, this research develops a pump impeller design by adding a junction disc plate arranged between the two parts of the existing impeller disc plate. The methods have conducted on two impeller design, an existing design and a modified one, to determine the impact of design changes on impeller performance. The addition of the junction disc plate aimed to increase the fluid flow distribution area, thus influencing the discharge and pressure inside the impeller. This condition can reduce pump losses and the cavitation effect of fluid flow friction. The addition of the junction disc plate can make fluid flow produce energy as a driving force and reduce engine power consumption. The method parameters are focused on the total head, fluid velocity, fluid capacity, blade angle, Net Positive Suction Head (NPSH), engine speed, engine power consumption, and pump efficiency.

2 Methodology

Investigation into centrifugal pump impeller performance is necessary to determine the impeller characteristics, which have relationship to the rise of fluid flow rate. The research investigated five pump performance characteristics. They were the requirement of a peak differential head at a maximum fluid flow rate; the Net Positive Suction Head (NPSH); fluid flow flexibility requirements; fluid composition-temperature fluctuations; and pump impeller transient conditions [18]. It involved two steps, the experiment and theoretical methods. The experiment measured the variety of impeller designs that have relationship with the suction head and discharge head, with the results relating to fluid capacity, motor speed, and motor power consumption. The theoretical method measured the pump impeller performance based on the experiment results by focusing on the calculation of actual fluid velocity, specific velocity, total head, NPSH, and pump efficiency. Parameter analysis is a primary analysis conducted in some previous research [19].

2.1 Experimental method of centrifugal pump impeller design

The experiment method measured pump impeller performance classified into two models: Fig. 1 (a) an existing impeller with eight blades gripped by two-disc plates; and Fig. 1 (b) a modified impeller with the addition of a junction disc plate as the location for eight blades gripped by two disc plates. The two impeller models can be seen in Fig. 1.



(a)

(b)

Fig. 1 (a) Existing model and (b) modified model of a centrifugal pump impeller

Both impeller models were placed on a *Pedrollo* HFm51A 1 phase centrifugal pump with the specification shown in Table 1.

The centrifugal pump used for the experiment method has a power of 746 W. The utilization of centrifugal pumps in this research due to the percentage of centrifugal pump applications in the industrial sector has reached 75%. Moreover, the centrifugal pump also has stable specifications based on pressure head, centrifugal pump efficiency, NPSH, and fluid flow rate [20]. The experiment method scheme explains that the fluid flow from a storage tank will pass through a centrifugal pump, which has two impeller models (as seen in Fig. 1) to drive the fluid flow to the pipe. The fluid flow through the pipeline and centrifugal pump is controlled by the valve, which determines the discharge head and suction head. The suction head influences the fluid flow pressure, while the discharge head determines the fluid flow rate [21]. The fluid flow through the test pipe returns to the storage tank. The experiment method schemes can be seen in Fig. 2.

The experiment method considered the discharge head in three positions (0.51 m, 0.52 m, and 0.53 m). The discharge head position that depends on the optimal total head condition of the centrifugal pump system could determine the fluid velocity [22]. The engine used in the experiment method met the specification of motor conditions presented in Table 1. These are the motor speed of 2,730 rpm and fluid capacity of 126.5 l/min. The experiment results explained the measurements of total head, fluid capacity, engine speed, and engine power consumption.

Table 1 Specification of centrifugal pump (*Pedrollo* HFm51A 1 phase)

No	Parameter	Value
1	Power (W)	746.0
2	Head Optimum Total (m)	25.0
3	Capacity Maximum (l/min)	126.5
4	Discharge (l/min)	70.6
5	Current (A)	5.1
6	Engine Speed (rpm)	2730.0
7	Power Consumption (W)	1160.0

2.2 Analysis of centrifugal pump impeller design

Analysis of pump impeller performance based on the results of the experiment method. Analysis of the pump performance was divided into actual fluid velocity, specific velocity, total suction head, NPSH, and pump efficiency, as shown in the flowchart in Fig. 3.

2.2.1 Actual fluid velocity

Actual fluid velocity measured the fluid velocity that enters the centrifugal pump impeller at a steady fluid flow and constant impeller rotational speed [23]. The fluid flow condition is determined by the inlet/outlet angle from the blade impeller, which is calculated by the velocity triangle equation. The speed triangle equation is shown in Table 2 [24].

Generally, the inlet blade angle is smaller than the outlet blade angle. The function of the inlet blade is to convert the fluid flow energy into the centrifugal pump. Meanwhile, the outlet blade's role is to increase the flow rate [24]. The rise of flow rate occurred in a steady total head condition [25]. The addition of the junction disc plate can change the impeller blade angle based on the enhancement of the impeller surface area; the angle conditions are seen in Table 3. Changes in the blade angle can influence the flow rate and pressure in the production of fluid kinetic energy [5].

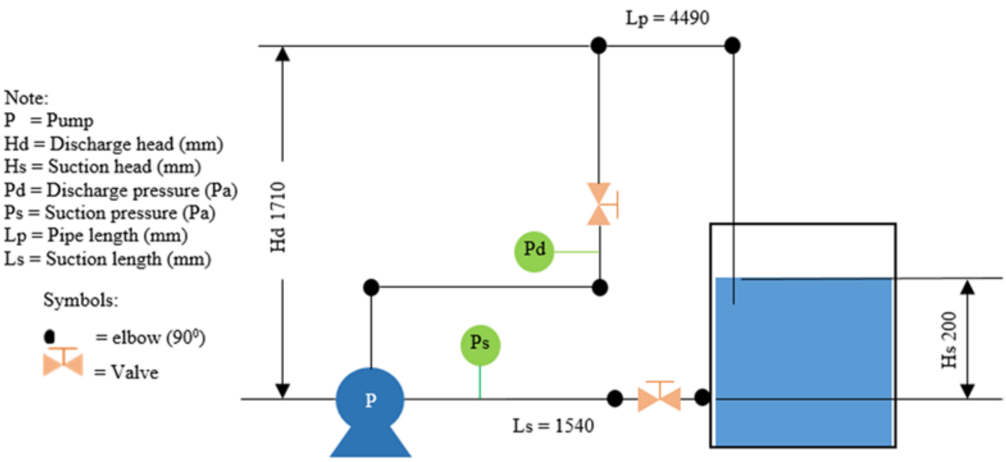


Fig. 2 Experiment method scheme

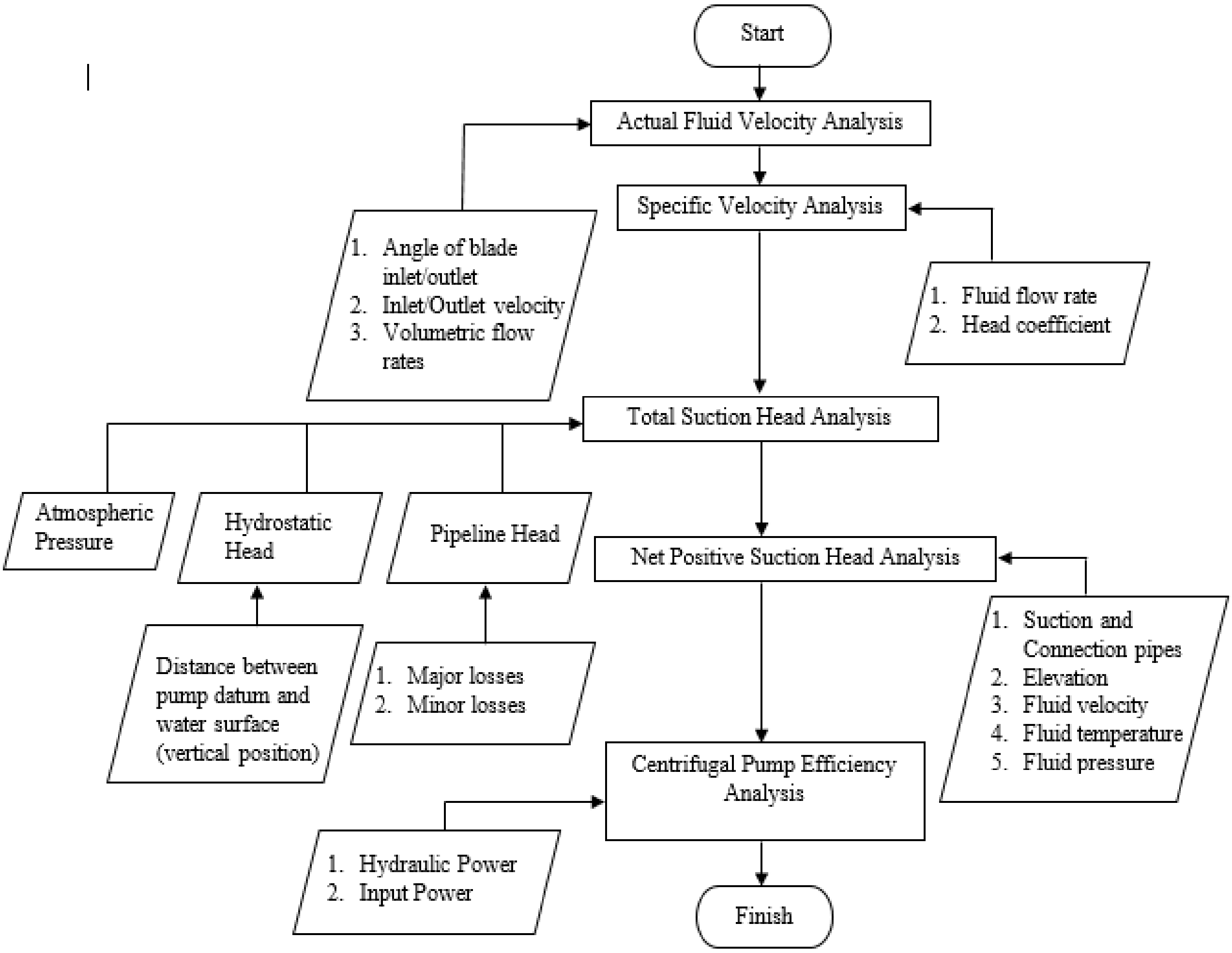


Fig. 3 Flowchart analysis scheme of centrifugal impeller design

Steady fluid flow conditions are indicated by volumetric flow rates (V), Average Flow Velocity (G_V), and suction pipe dimensions (A), which are calculated by Eq. (1) [25]:

$$V = \frac{G_V}{A} . \tag{1}$$

2.2.2 Specific velocity

Specific velocity is a measure of motor rotation speed for the centrifugal pump operation. Specific velocity is influenced by the fluid flow rate and head coefficient, which are calculated by Eq. (2) [26]:

$$n_s = n \cdot \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} . \tag{2}$$

2.2.3 Total suction head

The total suction head occurred in the centrifugal pump suction side is influenced by atmospheric pressure, the hydrostatic head, and the pipeline head (major and minor loss heads). The total suction head can be measured by the Bernoulli equation, shown as Eq. (3) [27]:

$$H_{\text{total}} = \frac{P_{atm}}{\gamma} + H_{\text{static}} - H_{\text{mayor}} - H_{\text{minor}} , \tag{3}$$

where atmospheric pressure is a measure calculated on the specific weight of ideal water conditions. Meanwhile, the hydrostatic head is the magnitude of the distance between the pump datum and the water surface in a vertical position. It can be calculated by Eq. (4) [28]:

$$H_{\text{static}} = g \cdot \Delta Z . \tag{4}$$

Fluid flow through the pipeline usually has friction on the pipeline surface, which is known as head pipe losses [29]. These can be classified into two forms: major and minor losses. The major losses are influenced by the diameter and length of the pipe, which could contribute to 20% of head losses [29]. Measurement of major losses is made by the *Darcy-Weisbach* equation, shown as Eq. (5) [30]:

Table 2 Angle of blade inlet and outlet [24]

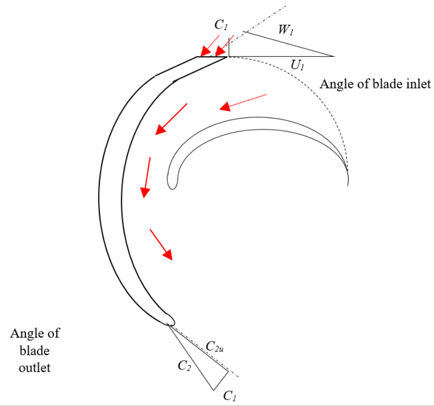
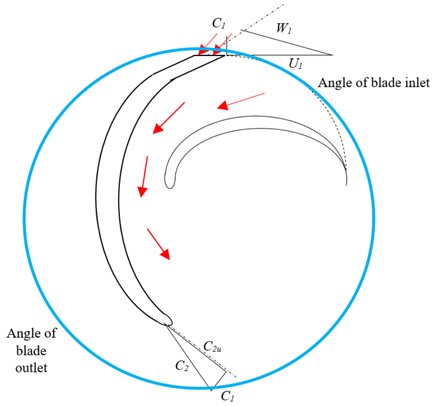
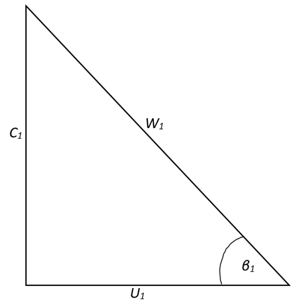
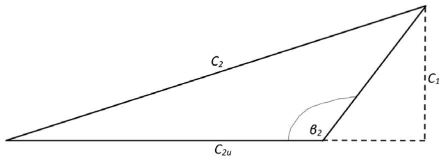
Existing model		Modified model	
Parameter	 Angle of blade inlet	Parameter	 Angle of blade outlet
U_1	$U_1 = D_1 \cdot \pi \cdot \frac{n}{60}$	U_2	$U_2 = D_2 \cdot \pi \cdot \frac{n}{60}$
W_1	$w_1 = \sqrt{C_1^2 + U_1^2}$	C_{2u}	$C_{2u} = \frac{H \cdot g}{\eta_2 \cdot U_2 \cdot k}$ where the k value is 0.78 and the η_2 value is 0.74
β_1	$\beta_1 = \arctan \frac{C_1}{U_1}$		
Inlet velocity	$C_1 = \frac{Q}{D_1 \cdot \pi \cdot b_1}$	Outlet velocity	$C_2 = \sqrt{C_1^2 + C_{2u}^2}$

Table 3 Blade angle condition

No	Parameters	Existing design	Modified design
1	Angle of inlet blade (degree)	45.6	46.1
2	Angle of outlet blade (degree)	94.4	93.9

$$H_{\text{mayor}} = f \cdot \frac{L}{D} \cdot \frac{v^2}{2g}, \tag{5}$$

where f is the friction coefficient, which is influenced by the Reynold Numbers. It classified into laminar flow (Re < 2000) and turbulent flow (Re > 4000). The correlation of friction coefficient and Reynold Numbers is indicated by the *Moody Diagram*. *Moody Diagram* is a diagram that explains the correlation between the Reynolds number factor and the pipe friction factor. The Reynolds number, which has a high value, causes the pipe friction factor can be smaller. The pipe friction factor can affect minor pipe losses caused by the connection between the pipe and the elbow or gate valve.

Pipe minor losses are influenced by valve and pipe connections, which have a different coefficient of friction (K) (as shown in Table 4). Valve and pipe connections contribute 13% of head losses [31].

Minor pipe losses are calculated using Eq. (6) [30]:

$$H_{\text{minor}} = K \cdot \frac{v^2}{2g}. \tag{6}$$

The effect of internal corrosion on the pipe system also influences the fluid flow rate [32], even though it has a minor result and is unnoticed. The pipe system characteristics that depend on major and minor losses influence

Table 4 Friction coefficient of pipe fitting

No	Component	Component number	Friction coefficient (K)	Total of friction coefficient component (K)
1	Elbow 90°	6	0.12	0.72
2	Gate valve	3	0.08	0.24
Total				0.96

the velocity distribution of fluid flow in the pipeline system [33], which is accumulated by another loss. It could affect the energy convenience for driving the fluid flow into the centrifugal pump. This condition has an influence on the Net Positive Suction Head (NPSH).

2.2.4 Net Positive Suction Head (NPSH)

NPSH is a measure of fluid flow energy into a centrifugal pump and is classified into two forms: actual NPSH ($NPSH_a$) and required NPSH ($NPSH_r$). It is influenced by suction and connection pipes, elevation, fluid velocity, temperature, and pressure. The ideal NPSH condition exists when the inlet fluid flow rate is higher than the outlet fluid [34]. However, this condition can be disrupted by slight vapor bubbles. It was the result of vapor pressure reduction into liquid pressure. Total suction head measurements can reduce the vapor bubble effects [35]. The total suction head measurement uses the available energy to push fluid flow into the centrifugal pump is called $NPSH_a$. $NPSH_a$ can be calculated by Eq. (7) [27]:

$$\mathrm{NPSH}_a = \frac{P_{atm}}{\gamma} + H_{\mathrm{static}} - H_{\mathrm{mayor}} - H_{\mathrm{minor}} - \frac{P_{\mathrm{vapor}}}{\gamma} . \tag{7}$$

$NPSH_r$ is a manufacturing standard parameter used to prevent cavitation. $NPSH_r$ is influenced by an escalation of fluid flow (n_s) and cavitation coefficient (σ); this condition is shown in the graph of cavitation-specific velocity.

$NPSH_r$ evaluation usually uses $NPSH_{r3\%}$ because it has a working operational standard with a total head of 1 m until 3 m, measured by the pump casting process [27, 36]. $NPSH_{r3\%}$ can be calculated by Eq. (8) [27]:

$$\mathrm{NPSH}_r = \sigma \cdot \mathrm{NPSH}_a . \tag{8}$$

The graph of cavitation-specific velocity also explains the "Specific Suction Speed" (S) used as a replacement parameter for σ. Meanwhile, the definition of a "Specific Suction Speed" equation showed in Eq. (9) [35]:

$$S = n \cdot \frac{Q_n^{\frac{1}{2}}}{H_{SVN}^{\frac{3}{4}}} . \tag{9}$$

If Q_n is expressed in m^3/s; H_{SVN} (m), and n (rpm); then the value of S for general pumps is 1200. There is a correlation between S and σ which can be seen in Eq. (10) [35]:

$$S = \frac{n_s}{\sigma^{\frac{3}{4}}} . \tag{10}$$

Equation (10) applies to the pump with the highest efficiency. If the working point is beyond the best efficiency of the pump, then the inlet angle and blade angle are no longer compatible. Optimal centrifugal pump performance can be seen from the higher $NPSH_a$ value compared to the $NPSH_r$ value.

2.2.5 Centrifugal pump efficiency

Centrifugal pump efficiency is the ratio between hydraulic power (P) and input power. Input power is classified into fluid density (ρ), fluid discharge (Q), and the total pump head (H) [37]. Efficiency can determine the effectiveness of the centrifugal pump for energy convention activities. Generally, the efficiency of centrifugal pumps has percentage of 75% to 85% [38]. The efficiency calculation of centrifugal pumps can be made using Eq. (11) [39]:

$$\eta_p = \frac{\rho g Q H}{P_{in}} \cdot 100\% . \tag{11}$$

3 Results and discussion

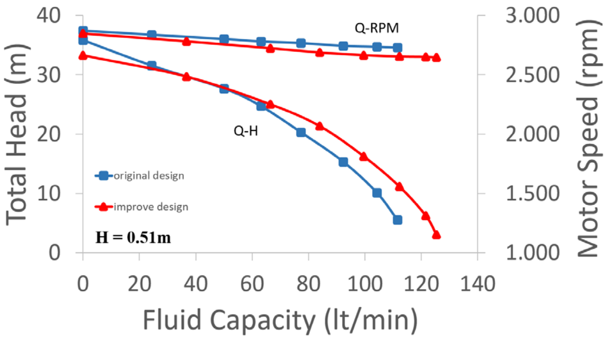
A centrifugal pump is used to transport fluids based on Bernoulli's Law. This law concerns atmospheric and fluid pressure, which has a relationship to the total head of a centrifugal pump. Fluid displacement required high pressure and a steady flow to reduce the total head. However, fluids have low pressure, and unsteady flow can increase the total head. This loss condition can affect the performance of centrifugal pumps, especially the impeller, which manages the fluid flow. The junction disc plate applied to the impeller design is expected to reduce this negative impact.

The modified and existing impellers performance were evaluated by experiment and theoretical method. The experiment method used a discharge head as a variable parameter divided into three positions (0.51 m, 0.52 m, and 0.53 m). The discharge head influences fluid capacity (Q), motor speed (rpm), and total head (H). The results of the experiment method are shown in Table 5 and Fig. 4.

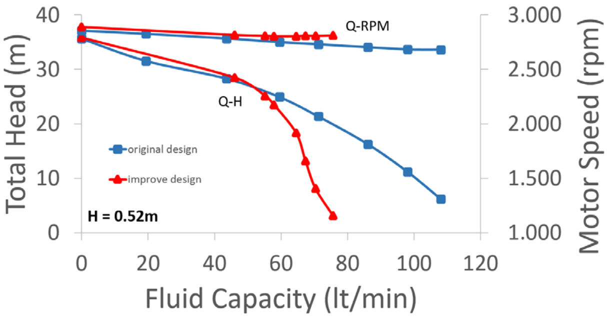
The three parameters related to each other [fluid capacity (Q), motor speed (rpm), and total head (H)] can determine motor power consumption, which influences the efficiency of the centrifugal pump. The experiment method involved eight experiments on the junction disc plate's performance with differences in the discharge head

Table 5 Average impeller parameters in the experiment method

No	Parameter	Existing design	Modified design
1	Total head (m)	35.8	33.6
2	Capacity (l/min)	110.1	111.5
3	Current (A)	4.9	5.2
4	Motor speed (rpm)	2,772.0	2,757.0
5	Power consumption (W)	1,107.0	1,147.0



(a)



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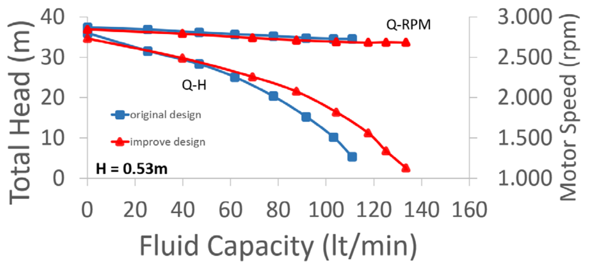

(c)

Fig. 4 Performance graphs comparing the existing and modified impeller designs by various discharge head (a) 0.51 m; (b) 0.52 m, and (c) 0.53 m

condition; the results of the average impeller parameters are shown in Table 5. Table 5 shows that the performance results of the modified pump impeller are better than those of the existing pump impeller. The modification version with the addition of the junction disc plate has a low total head, which can increase fluid capacity, reduce motor speed and increase motor power consumption. However, this increase in motor power consumption comes from two sources: electrical energy and junction disc plate energy. The energy source of the junction disc plate from the fluid flow is high due to the decreased distance between the suction pipe and the centrifugal pump, which influences the impeller rotation in driving fluid flow. Even though the reduction of impeller power to encourage the fluid flow into the impeller centrifugal pump can occur if the total head condition has the same value as the existing condition. The junction disc plate energy has advantages in reducing the cavitation effect caused by impeller disc plate rotation and fluid flow friction, which are reasons for pump losses. The energy portion component of the junction disc plate is higher than the electrical energy portion. The fluid flow rate condition in both impeller models shown in Table 5 has increased as the junction disc plate can increase the impeller area and be a transitory location for fluid flow. The fluid flow rate conditions in Table 5 also compared with the fluid flow capacity in Table 1, which can still be classified into the fluid flow optimum range of the centrifugal pump specifications used in the research. The friction effect of fluid flow inside the impeller results in differences between the actual and design conditions of fluid flow rates, resulting in little losses that are neglected.

Fig. 4 shows the performance results of the modified and existing impellers, with discharge head parameters divided into Fig. 4 (a) 0.51 m, Fig. 4 (b) 0.52 m, and Fig. 4 (c) 0.53 m in the experimental method. It explained that the discharge head rise could influence the total head reduction from the centrifugal pump and the increase in fluid capacity. The discharge head requires a high motor speed with steady fluid flow and high fluid pressure. However, utilization of a high motor speed has a negative effect, with a rise in motor power consumption and a reduction in centrifugal pump efficiency. The junction disc plate applied to the modified pump impeller design (as seen in Fig. 4) provides a decrease in motor power consumption. The plate could generate its own energy from the high-pressure fluid flow rate to drive the fluid flow. The energy available for the driving force of the fluid flow is also supported by the inlet/outlet angle of the impeller blade. With a discharge head condition of 0.52 m, Fig. 4 (b) shows different results from the 0.51 and 0.53 m discharge head conditions. The results explain the modified pump impeller's performance is lower than that of the existing pump impeller. This situation occurs due to a fracture of the junction disc plate surface, which reduces fluid pressure and disrupts energy availability at the plate. This fractured condition can be remedied by minor repairs (welding and grinding), followed by a performance test of the plate. The experiment method results used for the basic parameters of the theoretical method with the total pump system head condition of 25 m; the results of the theoretical method shown in Table 6.

Table 6 explained that the modified design could improve pump performance rather than the existing design with the same total head condition. The junction disc plate applied to the modified impeller design can increase the energy

Table 6 The results comparison of the centrifugal pump impeller from the theoretical method

No	Parameter	Existing design	Modified design
1	Fluid actual velocity (m/s)	0.5325	0.5325
2	Total suction head loss energy (J/kg)	10.0	10.0
3	Rotational speed motor (rpm)	423.6	444.8
4	$NPSH_r$	13.9	6.6
5	Efficiency (%)	80.4	83.2

availability of a steady fluid flow due to the enlargement of the disc plate area to receive more fluid flow. The increase in energy availability influences the rise in engine-specific speed and rotational speed. The Net Positive Suction Head Requirement ($NPSH_r$) parameter also provides high pressure to the steady fluid flow on the centrifugal pump suction side. In addition, the inlet/outlet angle of the blade impeller also increases the available energy, which comes from the high-pressure fluid flow produced by the junction disc plate. All these conditions can increase centrifugal pump efficiency. The efficiency results shown in Table 6 approach those of centrifugal pump efficiency obtained in other previous research on impeller design modification; for example [10], whose study obtained a centrifugal pump efficiency of 65% [11] with an efficiency percentage of 43%, and [40] who obtained an efficiency of 61%. It showed that the addition of the junction disc plate became an effective method to improve centrifugal pump performance. The fluid flow rate was able to increase the efficiency of the centrifugal pump by 2.8%.

4 Conclusion

The research results conclude that improving impeller design achieves optimum performance, as shown by the results of the experimental and theoretical methods. The modified design can reduce motor power consumption, increase fluid capacity, and increase centrifugal pump efficiency. It occurred due to the junction disc plate applied to the impeller design, which can provide energy availability to manage the steady fluid flow. Moreover, the inlet/outlet angles of the blade impeller can increase fluid flow pressure. The results show that a fluid flow rate of 111.50 l/min could reduce electric engine power consumption and employ the fluid flow rate as an alternative power to increase the efficiency of the centrifugal pump by 2.8%. Cracks on the junction disc plate will be investigate in future research.

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