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# Energy saving rates for a multistage centrifugal pump with variable speed drive

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## Abstract

Multistage centrifugal pumps with variable speed drives are currently widely used in a variety of industrial and commercial applications. However, there are limitations to defining the efficiency of variable speed drive pumps. As an alternative method, energy saving rates can be evaluated with flow patterns and mean duty cycles. Computational fluid dynamics (CFD) is being used as a good tool to understand this and is less time consuming in terms of analyzing performances the experimental method. Research attention was focused on the energy saving rates of a multistage centrifugal pump for variable flow with variable speed drive through numerical and experiment methods. For this investigation Reynolds-averaged Navier-Stokes (RANS) equations were discretized by the finite volume method and a two equations SST model was used to account for three dimensional steady state flows. In the experimental system, an experimental set-up of a variable flow system was made to obtain energy saving rates and computational results were validated. The energy saving rates of the pumps depend on the flow patterns and specific mean duty cycles on which the machine or system operates. Mean duty cycles were divided into different flow operating conditions and a weighting for the mean value was given for each segment according to interval time. The pump system was operated at 50~70% of maximum flow rates. The energy saving rates were obtained from input power through CFD simulation and experimentally, and the mean duty cycle was obtained from flow patterns in the field of the pump. Energy saving rates were evaluated as a function of mean duty cycle and input power of the system operation. The total energy consumed for the constant speed drive was 25,922 kWh and for the variable speed drive pump was 17,687 kWh through CFD. The total annual energy saving rates were annually 33.81% through computational and 31.77% through experimental method with the variable speed drive system when compared to the constant speed drive system.

*Keywords:* centrifugal pumps; variable speed drive; energy saving rates; mean duty cycle; CFD simulation

## 1. Introduction

Nowadays, multistage centrifugal pumps are widely used in a variety of applications, such as water supply, industrial, and power supply utilities [1]. Driving the pumps consumes a lot of energy. According to statistics, pumps consume around 20% of the world's total energy [2]. Governments and enterprises now face the problem of how to generate additional energy against a background of carbon dioxide emissions regulations and the cost of alternative energy businesses. Centrifugal pumps have been continually studied to increase the efficiency of fluid machines commonly used in industry. Previous research has been mainly focused on impeller geometry simulation analysis [3–5] and further changes to the shape of high-efficiency pumps worldwide [6, 7]. The efficient operation and improvement of sets of two or three pumps can increase total efficiency. Variable speed drive centrifugal pumps are controlled by rotational speed which can increase the efficiency, energy savings and reliability of the pumps [2]. The variable speed driving method using an inverter has been studied and shown to reduce energy by about 15 to 30% compared to operation of constant speed, which controls the flow solely through the control valve at the discharge side without drive [8–10]. The variable speed driving method can increase energy efficiency but there is no evaluation index for variable speed pumps similar to the 'best efficiency point' of a constant speed pump due to the continuous changes in rotational speed [11]. Energy saving rates can be obtained from energy consumptions relative to constant speed operation and they can be used as an evaluation index. Indeed such studies have been carried out [12, 13]. Thus, researchers do study energy saving rates by looking at pump usage patterns [14–16]. However, these previous studies were limited in that they calculated energy saving rates according to the

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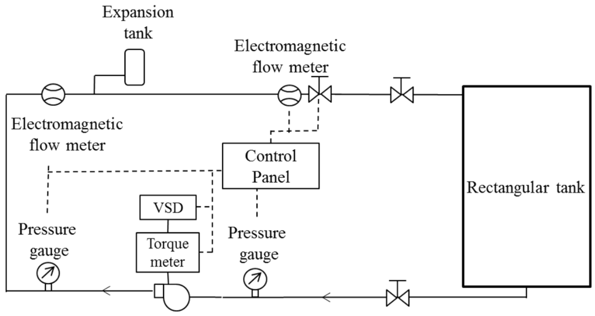


Figure 1: Pump performance facility

duty cycle and experimental data of pump performance.

On the other hand, computational fluid dynamics (CFD), when applied to the design of a multistage centrifugal pump, can be used to create numerical simulations to obtain data about performance and flow fields inside the pump. But owing to often complex geometry it remains difficult to get information about the performance of the pump flow domain. To overcome this, difficulty geometry is often simplified [17]. The evaluation index for a variable speed drive pump can also be measured by the computational method, but to do this duty cycle data is needed.

The present study is focused on the energy saving rates of feed water multistage centrifugal pumps for variable flow and pressure systems. Therefore, CFD simulation was carried out and an experimental construction was made to obtain energy saving rates to validate the model pump. To evaluate the energy saving rates, the mean duty cycle of the system was considered. In pump operating systems, it is expected that the centrifugal pump must peak to meet system demands, but most operating systems are not run at 100 percent operating speed. Therefore, system demands must be reduced. A complete flow pattern is required to define the mean duty cycle of the system, which is expressed as a percentage of one time period which is continued operation for a complete cycle. The input power of the system was calculated from the pump system characteristics curve for both CFD simulation and the experimental method. Experimental data were obtained from the ultrasonic and electromagnetic flow meter to obtain the pump system characteristics. The energy saving rates of the multistage operated centrifugal pump was evaluated as the percentage of the operation time multiplied by the input power of the system. CFD investigation was carried out to compare with the experimental results for validation of the pump.

## 2. Pump performance test

### 2.1. Pump performance facility

An experiment test facility was conducted to evaluate pump performance during variable and constant speed drive operations. The test layout of the measurement system is shown in Fig. 1. The installed pump was a vertical multistage centrifugal type 11 kW, the water tank length was 2.5 m long, 4.5 m wide, and 2.5 m high. Pipe diameter from the water tank was 100 mm, pipe diameter to the centrifugal pump was 50 mm, and pressure gauges were set up before and after 5 D distance.

Table 1: Specifications of experiment equipment

| Equipment type | Specification |
|---|---|
| Pump | DR 20-60 |
| Motor | Vector motor 11 kW |
| Inverter | SV01101S7-4No-22A |
| Electromagnetic flow meter | BELIMO BLC100-EPIV |

Table 2: Physics setup of CFD simulation

| | Setting | Option |
|---|---|---|
| 1. | Physics | Constant Density<br>Liquid—$H_2O$ (Water)<br>Steady state<br>Shear Stress Transport Turbulence<br>Three Dimensional |
| 2. | Domain | Rotating(R), Stationary(S) |
| 3. | Inlet | Velocity Inlet |
| 4. | Outlet | Pressure outlet |
| 5. | Rotational speed, rpm | 3600, 3200, 2800, 2400, 2000 |
| 6. | Inlet Flow rate, $Q$, $m^3/h$ | 0, 12, 16, 20, 24, 28, 32 |
| 7. | Outlet condition | Static pressure |
| 8. | Interface | Frozen rotor |
| 9. | Max. Iterations | 1000 |
| 10. | Total mesh elements | 27,529,524 |

The equipment used in the tests is shown in Table 1. The inlet side pressure gauge measurement range is -1∼1 bar and the outlet side pressure gauge measurement range is 0∼15 bar. The pump head was measured by the algebraic difference in height of the liquid between the outlet and inlet sections. A torque meter was used to measure shaft power between the pump and the motor. The motor was controlled by a frequency inverter. An electromagnetic flow meter was set up to control and calculate the flow rate. Gate valves were installed at the inlet side and after the electromagnetic flow meter at the outlet of the pump. The gate valve after the electromagnetic flow meter was used to control the flow rate at a given constant speed during pump operation.

### 2.2. Constant pressure operation with variable speed drive

The feed water pump generally drives the ejection head of the pump to greater than a certain value in order to deliver a flow rate in high-rise buildings. Constant pressure operation drives the flow rate control through the gate valve on the discharge side, and is set to operate on at least the discharge head. To vary the speed operation driving the pump, the number of revolutions can be set to correspond to the required flow rate, so that during operation the flow rate and discharge head correspond to the minimum lifting height.

## 3. Numerical method

To obtain pump performance using the numerical method, CFD simulations were conducted. The geometry of the centrifugal pump impeller and diffuser were used for meshing by ANSYS ICEM-CFX-14.5. the impeller, diffuser, inlet casing,

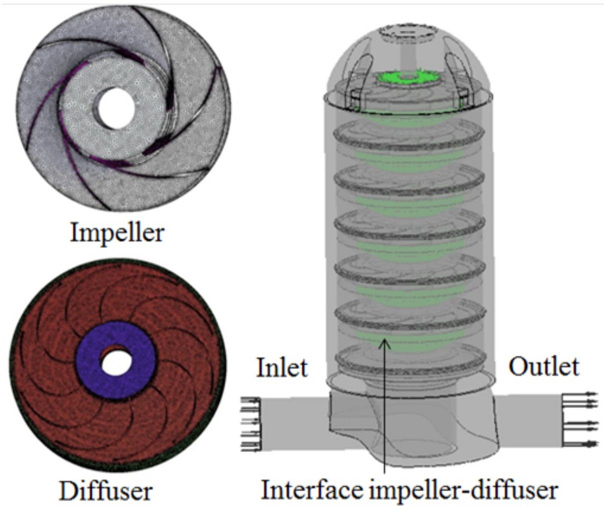


Figure 2: Centrifugal pump model for CFD simulation

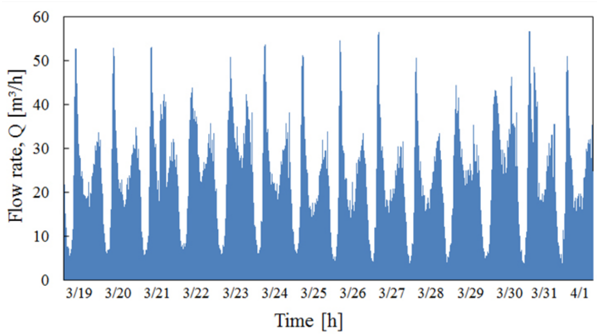


Figure 3: Flow rate pattern of feed water pump

and outlet casing are shown in Fig. 2. Domain meshed was unconstructed tetrahedral cells. The continuity and momentum equations are expressed as Eq. (1) and (2)

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

$$\rho\left(\frac{\partial u_i}{\partial t} + u_j\frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j}\left(\mu\frac{\partial u_i}{\partial x_j} - \rho\overline{u_i' u_j'}\right) \tag{2}$$

Where $\rho$ and $\mu$ are density and dynamic viscosity, $p$ is the pressure scalar $-\rho\overline{u_i' u_j'}$, is the apparent turbulent stress tensor. The $k-\omega$ based SST model accounts for the transport of the turbulent shear stress and provides highly accurate predictions of the onset and amount of flow separation under adverse pressure gradients. In the pump physics setup as shown in Table 2, a frozen rotor was used for interface condition.

## 4. Flow patterns and duty cycle

### *4.1. Flow patterns*

To calculate the energy saving rates of a centrifugal pump with constant and variable speed drive, flow rate patterns

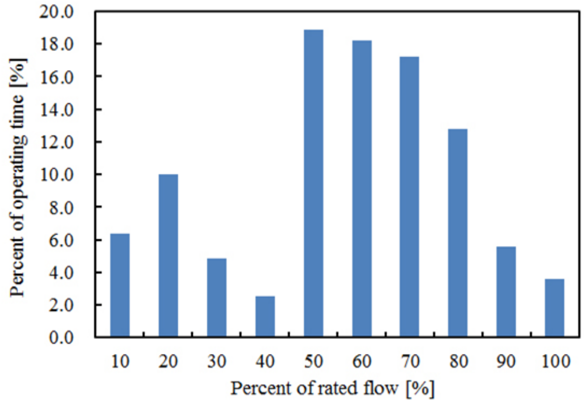


Figure 4: Mean duty cycle of feed water pump

should be obtained. Therefore, an ultrasonic flowmeter was installed in the building from 19 May 2014 to 5 February 2015. The ultrasonic flowmeter model was a FLUXUS. Flowmeter uncertainty was less than 0.2% and the available temperature range was -10~60°C, Flow measure time was once every two minutes. Flow rate patterns obtained through the ultrasonic flowmeter are shown in Fig. 3. Flow rate patterns were classified by each flow rate throughout one year.

### *4.2. Mean Duty cycle*

The mean duty cycle is the key element in determining the energy savings of the systems. It is the proportion of time during which a component, device and system is operated. The duty cycle was segmented into different flow rates and the average value for each segment was weighted by the interval time. The mean duty cycle can be expressed as a ratio or as a percentage [18]. The operating time corresponding to each flow rate was calculated. The mean duty cycle of the feed water pump is shown in Fig. 4. It shows that the mean duty cycle of the system operates at 50~70% of maximum capacity.

## 5. Energy saving rate

The energy saving rate can be defined as the ratio of energy consumption of variable speed operation and constant speed operations. To measure the energy saving rate, pump performance and duty cycle are required. At both constant and variable speed operation, the change in pump flow rate is the same but the head would have variations. The duty cycle can explain the load for each flow rate. At first, at constant and variable speed operation for each flow rate, pump input power should be obtained. Energy consumption of each flow rate is calculated by multiplying the input power and operating time for each flow rate. The energy saving rate can be calculated through comparing energy consumption between variable and constant speed operation.

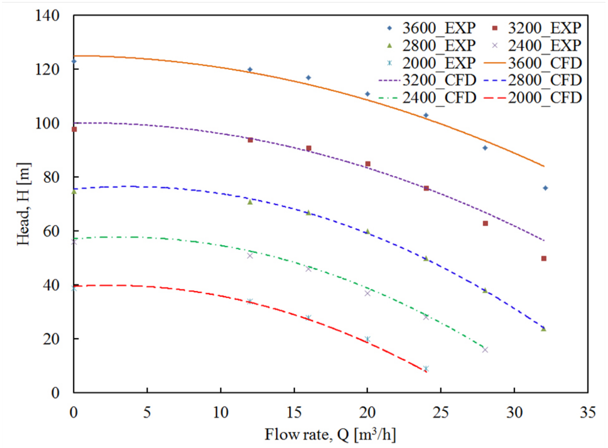


Figure 5: $H$-$Q$ curves of experiment and CFD simulation

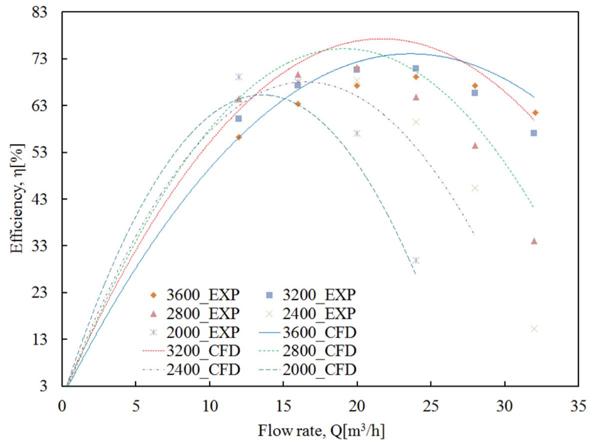


Figure 7: $\eta$-$Q$ curves of experiment and CFD simulation

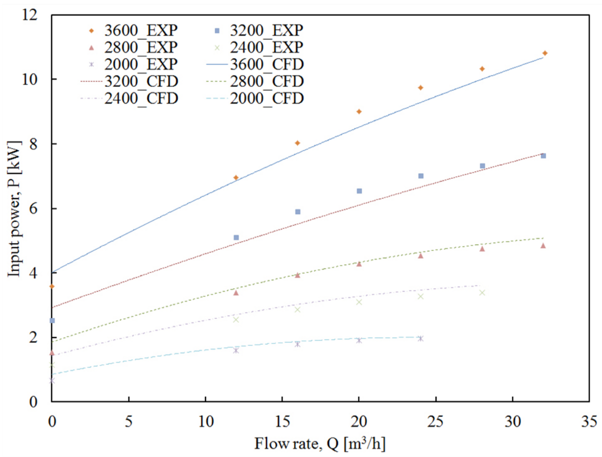


Figure 6: $P$-$Q$ curves of experiment and CFD simulation

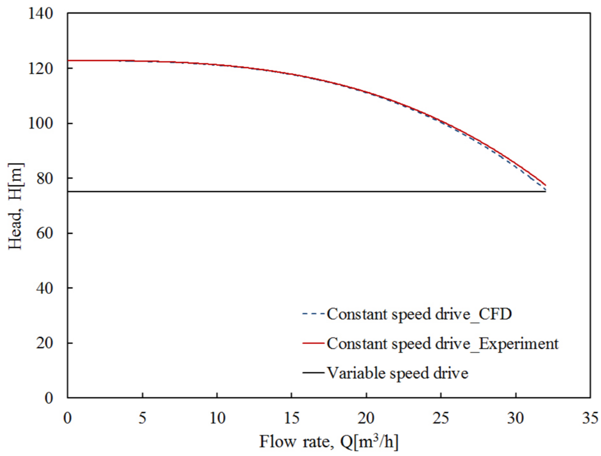


Figure 8: $H$-$Q$ curves with constant and variable speed drive

## 6. Results and discussion

### *6.1. Comparison of experiment and numerical results*

In order to calculate energy saving rates from pump performances by CFD simulation, CFD simulation should be validated with experimental data. Centrifugal pump performances with constant and variable speed operation by experimental and CFD simulation are shown in Fig. 5–7. These figures demonstrate a good agreement between the experimental and computational data.

The highest value of head was 9.7% for the highest flow rate at 3600 rpm and the average value was 5.4%. The highest value of input power was 9.06% and the average value was 3.8%. Average input power of the experimental and CFD simulation had only a 5% deviation, as was expected

### *6.2. Pump input power with constant and variable speed drive*

In buildings, variable speed drive controls pump rotational speed to ensure that the head is fixed to a minimum value for lifting. $H$-$Q$ curves and $P$-$Q$ curves with constant and variable speed drives are shown in Fig. 8–9. Variable speed pumps operate at constant head with changing rotational speed. While the pump was operating with a fixed head, pump input power, with constant and variable speed drive, deviated. This can influence calculation of energy saving rates. In Fig. 9, only 60% of input power with variable speed drive at constant speed is required for around half of the rated flow rate.

### *6.3. Calculation of energy saving rates*

Energy saving depends on the input power and mean duty cycle of the system. Fig. 9 shows pump input power curves with constant and variable speed drives, and Fig. 4 shows the mean duty cycle of the system. From Fig. 9 input power was determined using pump performances from both experimental and computational data. Mean duty cycle was segmented into ten segments, as shown in Fig. (4). Energy consumption and energy saving rates were calculated, as shown in Tables 3–4. As shown in Table 3, at 17.6 $m^3/h$, power input is cut almost 35.76 percent for the variable speed system; the reduction at 1.6 $m^3/h$ is more than 50 percent. On the other hand, as shown in Table 4 at 17.6 $m^3/h$, power input is

Table 3: Energy saving rate with variable speed pump by CFD simulation

| Average flow rate, $m^3/h$ | Input power of constant speed, kW | Input power of variable speed, kW | Operation times, h | Energy of constant speed, kW | Energy of variable speed, kW |
|---|---|---|---|---|---|
| 30.4 | 10.14 | 10.26 | 187 | 1891 | 1914 |
| 27.2 | 9.57 | 8.70 | 292 | 2795 | 2539 |
| 24 | 9.00 | 7.30 | 142 | 1277 | 1037 |
| 20.8 | 8.42 | 6.08 | 73 | 614 | 444 |
| 17.6 | 7.83 | 5.03 | 552 | 4318 | 2774 |
| 14.4 | 7.24 | 4.15 | 531 | 3844 | 2203 |
| 11.2 | 6.64 | 3.44 | 503 | 3337 | 1728 |
| 8 | 6.03 | 2.89 | 373 | 2250 | 1079 |
| 4.8 | 5.42 | 2.52 | 162 | 879 | 409 |
| 1.6 | 4.80 | 2.32 | 105 | 506 | 244 |
| | | | Total | 21713 | 14372 |
| | | | Energy saving rate | 33.81% | |

Table 4: Energy saving rate with variable speed pump by experiment

| Average flow rate, $m^3/h$ | Input power of constant speed, kW | Input power of variable speed, kW | Operation times, h | Energy of constant speed, kW | Energy of variable speed, kW |
|---|---|---|---|---|---|
| 30.4 | 12.08 | 12.29 | 187 | 2254 | 2292 |
| 27.2 | 11.61 | 9.74 | 292 | 3390 | 2843 |
| 24 | 11.04 | 8.30 | 142 | 1568 | 1178 |
| 20.8 | 10.39 | 7.13 | 73 | 759 | 521 |
| 17.6 | 9.65 | 6.17 | 552 | 5321 | 3404 |
| 14.4 | 8.81 | 5.36 | 531 | 4682 | 2849 |
| 11.2 | 7.89 | 4.65 | 503 | 3967 | 2336 |
| 8 | 6.87 | 3.96 | 373 | 2565 | 1478 |
| 4.8 | 5.77 | 3.25 | 162 | 936 | 527 |
| 1.6 | 4.57 | 2.45 | 105 | 482 | 258 |
| | | | Total | 25922 | 17687 |
| | | | Energy saving rate | 31.77% | |

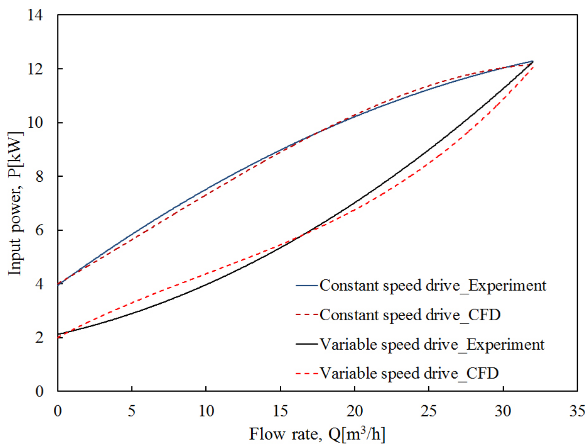


Figure 9: *P*-*Q* curves with constant and variable speed drive

cut almost 36.06 percent for the variable speed system; the reduction at 1.6 $m^3/h$ is more than 45 percent. Table 3 illustrates the energy consumption for the constant and variable speed drive pump by CFD simulation. The operation continued for 365 days and 24 hours a day, or 8760 hours per year. The procedure is illustrated using 47.37 percent flow rate (14.40 $m^3/h$) as a reference calculation. At a 47.37 percent flow rate, the pump delivery was 0.4737 x 30.4 $m^3/h$ = 14.40 $m^3/h$. Again, as can be seen in Table 3, pump input power was 7.24 kW at this flow rate for the constant speed drive operation, but 4.15 kW at the variable speed drive operation. At this flow rate, the pump operates for 531 hours per year. The total energy consumed at this duty point was 3844 kWh, but for the variable speed operation it was only 2203 kWh.

Similarly, the figures in Table 4 were calculated by the experimental method, using a different flow rate. Energy saving rates from CFD simulation and the experimental method were 33.81% and 31.77% respectively. The average deviation of input power with constant and variable speed drive was 3.8%. However, the deviation of calculated energy saving rates was only 2%. From these values, it can be deduced that calculated energy saving rates from CFD simulation could replace the energy savings rate from experimental results.

## 7. Conclusion

The research investigated the energy efficient operation of a variable speed drive centrifugal pump. Energy saving rates were evaluated from numerical simulation and experimental results.

The results indicate that energy saving rates can be obtained when the pumps are running at the same speed ratio. A multistage centrifugal model pump was simulated to obtain performances for both constant and variable speed drives, and a designed layout was installed to obtain pump performances by experimentation for validation of the model pump. From the performance results it was found that average deviation of input power with constant and variable speed drive was less than 5%. The mean duty cycle and input power were the two main elements used for calculating energy saving rates by CFD simulation and the experimental method. The mean duty cycle was segmented into ten different flow rates from 30.4 $m^3$/h to 1.6 $m^3$/h. The mean duty cycle was measured from the pump flow rate pattern, which was taken from the building. The average mean duty cycle operated at 50∼70% of the maximum flow rate. In the building, the pump with variable speed drive operated at a constant head. While the pump was operating with a fixed head, pump input power with constant and variable speed drive had deviation. Pump input power decreased by up to 60% with variable speed drive at half of the rated flow rate.

Calculated energy saving rates by CFD simulation and experiment were 33.81% and 31.77% respectively with the variable speed operation when compared to the constant speed drive system. Deviation of calculated energy saving rates was only 2%. It could be regarded that the calculated energy saving rates from CFD simulation had replaced those from the experimental results. Computer simulation can be used to predict input power prior to manufacturing of the pump. Therefore, energy saving rates can be obtained quickly by the CFD method. If CFD simulation is used to increase pump energy saving rates, less time would be required than experimentally. However, the specific flow rate pattern is required to calculate energy saving rates. This means that pump flow rate patterns should be collected for at least one year. From the results above it can be ascertained that energy saving rates would improve the performance and longevity of the pump.

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