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A Study on Energy Saving Rate for Variable Speed Condition of Multistage Centrifugal Pump

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Centrifugal pumps are being widely used in many industrial and commercial applications. Many of these pumps are being operated at constant speed but could provide energy savings through adjustable speed operations. The purpose of this study was to get the energy saving rates of the multistage centrifugal pump with variable speed conditions. For this investigation an experimental set up of variable flow and pressure system was made to get energy saving rates and numerical analyses are applied to validate the pump performance. The energy saving and therefore the cost saving depends on the specific duty cycle of which the machine operates. Duty cycle is the proportion of time during which a component, device and system is operated. The duty cycle segmented into different flow rates and weighting the average value for each segment by the interval time. The system was operated at 50% or less of the pump capacity. The input power of the system was carried out by pump characteristics curve of each operating point. The energy consumption was done by the product of specific duty cycle and the input power of the system for constant speed and variable speed drive operation. The total energy consumed for constant speed drive pump was 75,770 kW.hr and for variable speed drive pump was 31,700 kW.hr. The total energy saving of the system was 44,070 kW.hr or 58.16% annually. So, this paper suggests a method of implementing an energy saving on variable-flow and pressure system of the multistage centrifugal pump.

Keywords: Multistage Centrifugal Pump, Performance Analysis, Energy saving rate, SST Model, Duty Cycle

Introduction

A centrifugal pump is a type of fluid machine which is driven by a prime mover (e.g., an electric motor) used to impart energy to fluids, and continuously feed “the required amount” of such fluid to “an intended height or distance” [1]. Pumps of a rotating wheel called an impeller or rotor driven by an external power source to increase the flow kinetic energy, followed by an element to decelerate the flow, thereby increasing its pressure. The combination is known as stage and these elements are contained within a housing or casing. A multistage pump might consist of several stages within a single housing, depending on the amount of pressure rise required of the machine [2]. Nowadays, multistage centrifugal pumps are being widely used in a variety of applications, such as, water supply and irrigation, power generating utilities, industrial and mining enterprises, marine and air vehicles. One of the most important components of a multistage centrifugal pump is the impeller [3]. Multistage centrifugal pump design method are based on the application of empirical and semi empirical rules along with the

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Nomenclature		Greek letters	
Q	volume flow rate (m^3/hr)	μ	dynamic viscosity (N·s/m^2)
P	pressure (Pa)	ω	rotational speed (rev/min)
H	Head (m)	μ_t	turbulent viscosity
ρ	density (kg/m^3)	**Subscripts**	
η	Efficiency (%)	i, j	vector tensor
Pw	hydraulic power (kW)	1	inlet pressure and velocity
Ps	mechanical power (kW)	2	outlet pressure and velocity
g	gravitational acceleration (m/s^2)	1,2	elevation of inlet and outlet
Z	elevation of the pump		

use of available information in the form of different data which helps to obtain a through performance evaluation of a particular design [4].

Speed variation is the most energy efficient method to respond to variable system demands [5]. Frequency converters are widely used nowadays to control the head and flow rate of circulation pumps. Traditionally a circulation pump was driven by a constant speed induction motor with system demand controlled by a throttling valve [6] Variable speed drives (VSDs) provide the user with a variety of benefits, including potentially significant energy saving and improved reliability in pumping applications. Assessment of the technical and economic advantages gained by using VSDs on circulation pumps have been widely publicized in recent years. According to statistics, pumps consume around 20% of the world total energy [7]. The energy efficiency of a system depends not only on the design of the pump but also, and more so, on its working conditions and system design [6].

The authors have already shown theoretically, centrifugal applications are ideal candidates for realizing energy savings for a given system curve, the output varies proportionally with speed, and power varies with $(\text{speed})^3$. 100 percent power is required at 100 percent speed. It is necessary to analyze potential savings needs required information which are includes the pump model with efficiency data, the system curve, the duty cycle of the system and the control method and efficiency data [8].

On the other hand, computational fluid dynamics (CFD) is being applied in the design of multistage centrifugal pump which can be used for numerical simulation to get the performance of the flow field inside the pump. Numerical simulation makes it possible to visualize the flow condition inside a centrifugal pump, and provides the valuable hydraulic design information of the centrifugal pumps [4]. It is mainly difficult to get the complex geometry of the pump flow domain. For these difficulties many geometries are often considered for simplifications. Many hydrodynamics models are reported in 2D and 3D by using the CFD code and studied impeller diffuser interaction on the pump performance and showed that a strong pressure fluctuation is due to the unsteadiness of the flow shedding from impeller [9-11].

For the purpose of this paper, research attention is focused on the energy saving rates of the multistage centrifugal pump for the variable flow and pressure system. Therefore, the experimental set was made to get energy saving rates and numerical simulation was used to validate the steady hydrodynamics of multistage centrifugal pump model DR 20-60. A system duty cycle was considered for calculating the energy saving rate. Since the centrifugal pump must be peaked to meet system demands but most systems are not operated at 100 percent constantly operated speeds, so, the system must be reduced. Therefore, system duty cycle must be defined so that the percentage of one period and a period of time operated for a complete cycle. The input powers of the variable flow and pressure system for both constant speed and variable speeds were calculated from the pump system characteristics curve. So, for getting the pump system characteristics the experimental data were measured by the electronics flow meter for the constant speed and variable speed condition, and pressure head was calculated by the pressure sensors of inlet and discharge sections of the pump. The energy saving rates of six stages centrifugal pump was measured by the percentage of operation time times input power of the system. Also, this paper describes CFD investigation methodology which predicts to get the performance effect of the parameters such as flow rate, head, efficiency, and power to compare with the experimental data for its effectiveness and reliability of the pump.

Experimental Method

Experimental tests were conducted to analyze the energy saving rate for the multistage centrifugal pump and different rotational speed were applied to use the same operating conditions. The test layout of the variable -flow variable-pressure system and experimental measurement system is shown in Fig.1. The experimental data were measured by electronic flow meter. One is connected with the control computer and another is conventional electromagnetic flow meter which was used for their better accuracy. The equipment used in the tests is shown in Table 1. There have been used two reservoirs one was rectangular type tank and another was circular type reservoir tank. The rectangular tank was used for calculating the pump performance test and circular tank installed for cavitation test and a vacuum pump also used to vacuum air from the circular reservoir tank.

The working temperature was 28.5°C and humidity was 81% and the working fluid temperature was measured at 25°C. Different angular frequencies were applied by frequency inverter and the rotational speed has been measured by a torque meter to get accurate pump shaft power. Pump head has been determined by algebraic difference height of liquid between the discharge and inlet sections. On both suction and discharge side a pressure sensor installed to calculate the suction and discharge pressure. Pump shaft torque was measured by torque meter which was connected to the signal amplifier. The mechanical power was measured as the product of shaft torque and angular velocity. Hydraulic power is given by the rate of mechanical energy input to the fluid. Efficiency was calculated as the ratio of hydraulic power to mechanical power.

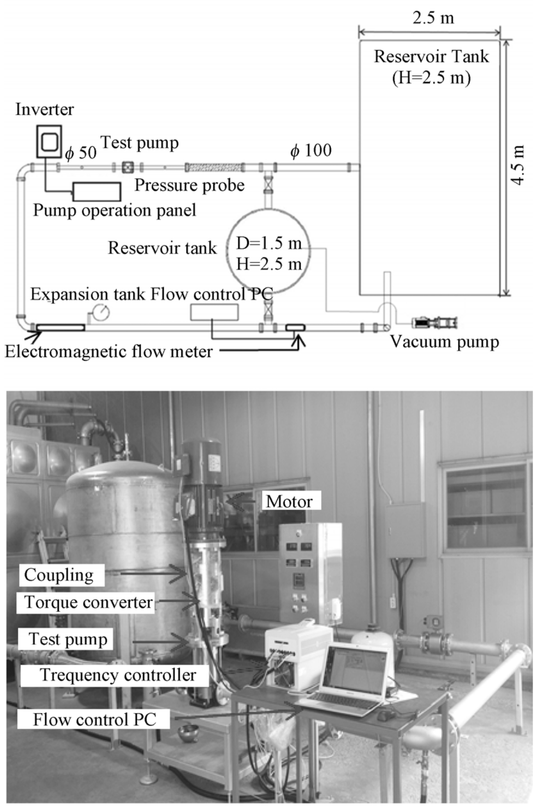

Fig. 1 Experimental layout and performance measuring system of the centrifugal pump

Table 1 Pump experimental setup devices

Equipment	Specification
Pump	DR 20-60
Motor	Vector motor 11 kW
Inverter	SV01101S7-4No-22A
Electromagnetic flow meter	BELIMO BLC100-EPIV
Pressure Tank	PWB-DCP-100LV-4bar
Vacuum pump	30Torr, 1050 L/min, 2.2kW

According to the affinity law for centrifugal pump flow is proportional to speed, pressure is proportional to square of the speed and power is proportional to cube of the speed. Therefore, with the controlled throttled valve with variable speed the system were operated at different flows with different head. The 100 percent operated speed of the pump curve and the system curve was carried out of maximum operating point of the pump flow rates at 36.8 m^3/hr but the system could not be operated at 100 percent speed. So, reduced pump speed operated and with the controlled throttled valve the system were operated at reduced flow rate with reduced head.

The duty cycle is of the key element in determining the energy savings of the systems. Duty cycle is the proportion of time during which a component, device, or system is operated. The duty cycle can be expressed as a ratio or as a percentage. The duty cycle was measured by segmented into ten segments and weighting the average value for each segment by the segment time. The system 40% duty cycle means the system would be operated on 40% time but off 60% of the time. The energy savings was carried out by the product of percentage operating time and input power of the system.

Numerical Method

To validate the experiment, computer simulation is conducted. The geometry of a six stage centrifugal pump impeller and diffuser were used for meshing by ANSYS ICEM-CFX-14.5 (Ansys Inc., 2012, USA). Each of impeller, diffuser, inlet casing, and outlet casing has been meshed with unconstructed tetrahedral cells which are shown in Fig. 2. The impeller has six blades and diffuser has ten blades. With the ANSYS ICEM-CFX surface mesh setup was made, applied tetrahedral method, and computed the mesh volume, checked mesh quality and the total meshing grids were 5,265,401 nodes and 27,529,524 elements. Multistage centrifugal impeller domain were set up as rotating part with y-axis and the rotational speed of the pump were 3600 r/min, 3400 r/min, 3200 r/min, 3050 r/min, 2720 r/min and 2550 r/min and diffuser were considered as stationary domain. The model of a six stages centrifugal pump in this study is shown in Fig. 3. It consists of impeller, diffuser with return vanes and casing. To run the numerical simulation; we accounted the governing equations for centrifugal pump, the following assumptions were made: a three-dimensional incompressible, steady- state flow, and the turbulence flow using the SST model was assumed as Newtonian fluid and the thermo-physical properties were constant with the temperature.

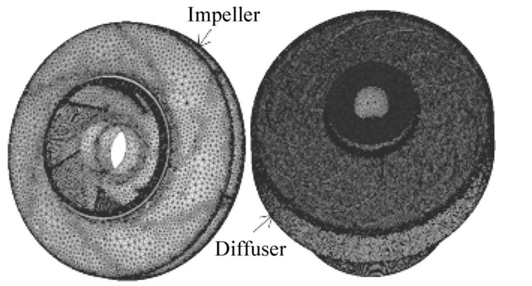

Fig. 2 Meshing of impeller and diffuser of pump

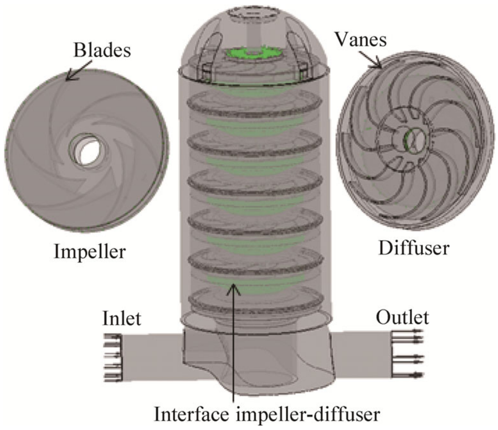

Fig. 3 Centrifugal pump domains of inlet, outlet and interfaces

To account for these assumptions, the theoretical analysis of the fluid flow was based on the continuity and momentum equations [12, 13]. The continuity and momentum equations are expressed as Eq. (1) and Eq. (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 u_i is the velocity vector, p is the pressure scalar, ρ is the density, i and j are the tensor notations, $-\rho\overline{u_i' u_j'}$ is the apparent turbulent stress tensor, μ is the dynamic viscosity.

The k- ω based SST model accounts for the transport of the turbulent shear stress and accurately predict the onset and the amount of flow separation under adverse pressure gradients. The unknown turbulent viscosity μ_t is determined by solving two additional transport equations for the turbulent energy k, and for the turbulence frequency ω. These two equations can be written as Eq. (3) and Eq. (4)

k -equation:

$$\frac{\partial(\rho\kappa)}{\partial t} + \frac{\partial}{\partial x_j}(\rho\kappa u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\kappa}\right)\right]\frac{\partial\kappa}{\partial x_j} + P_k - \beta'\rho\kappa\omega + P_{kb} \tag{3}$$

ω -equation:

$$\frac{\partial(\rho\omega)}{\partial t} + \frac{\partial}{\partial x_j}(\rho\omega u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\omega}\right)\right]\frac{\partial\omega}{\partial x_j} + \alpha\frac{\omega}{\kappa}P_k - \beta\rho\omega^2 + P_{\omega b} \tag{4}$$

Where, P_k is the production rate of turbulence, μ_t is the turbulent viscosity, α, β, β', σ_k and σ_ω are constants.

Pump performance

In order to calculate the pump performance for both experiment and numerical simulation we need to calculate pump head, hydraulic power, mechanical power and hydraulic efficiency. In actual practice, hydraulic method, and thermodynamic methods are applied to calculate the pump performances [14]. Hydraulic method is used in our performance investigation. Therefore, based on the hydraulic method head is formulated as Eq. (5):

$$H = \frac{p_2 - p_1}{\rho g} + \frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1) \tag{5}$$

Where P_1 and P_2 are the pressure at suction and discharge of the pump; V_1 and V_2 are the velocities at the inlet and outlet of the pump; and Z_1 and Z_2 are the elevation of the pump system.

Furthermore, the hydraulic power is given by the rate of mechanical energy input to the fluid as Eq. (6),

$$P_w = \rho Q g H \tag{6}$$

Where Q is the volume flow rate and H is the pump head. In addition, the mechanical power of the pump is expressed as Eq. (7)

$$P_s = \omega T \tag{7}$$

Where ω is the angular velocity and T is impeller torque. From the hydraulic power and mechanical power Eqs.6~7; the pump efficiency is defined as Eq. (8)

$$\eta = \frac{P_w}{P_s} = \frac{\rho Q g H}{\omega T} \tag{8}$$

Model Validation of the experimental study

In order to validate the numerical approach, a comparison of the experimental and the computed data were carried out. Fig. 4 shows the pump head, efficiency and power of the system used for different flow rates with different rotational speed. From this graph it is shown a good agreement between the experiment and numerical data so that the average deviation of the head values was only 5.4%. The highest deviation of 11.06% for the highest flow rate at 3050 rpm and the differences for only three conditions are larger than 8.5%. Along with the decreases of the rotational speed the head is being continuously decreased. The numerical methods predict the pressure rise more accurately compared with the experimental data. After that, the pump efficiency shows an average deviation of only 8.22%, and the highest deviation is 13.12% for the highest flow rate of 3600 rpm. Also, the pump shaft power average deviation was only 6.35% and Fig.4 shows a small deviation of the numerical data and follows the trend of the experimental results.

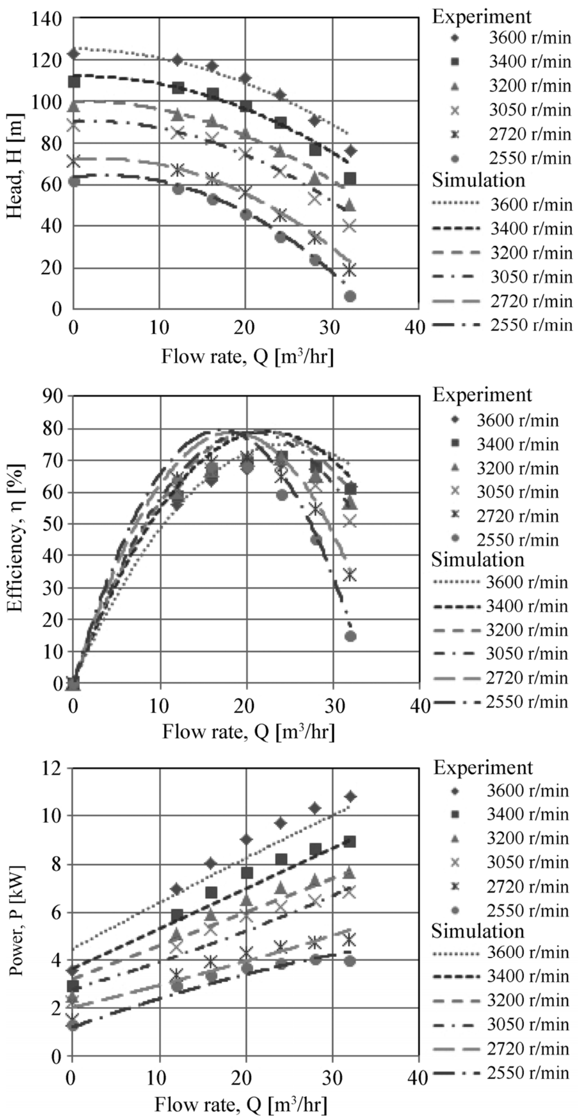

Fig. 4 Comparison of numerical and experiment performance curve (H-Q, η-Q, and P-Q) of the multistage centrifugal pump

Results and Discussion

Pump duty cycle

Duty cycle can estimate the amount of time spend at several pressure and flow points depending on the system. Typically, duty cycle segmenting into different flow rates segments and weighting the average value for each segment by the interval time. Since centrifugal pumps must be sized to meet peak system demands and most systems are not operated at 100 percent output constantly, so the system output must be reduced. Thus it is important to define the duty cycle of the system that is expected operating points and the percent of time operated at those points [8]. Fig. 5 shows the duty cycle of the system and the system operates at 50% capacity or below.

Pump operating characteristics

Basically, a pump system would be required to run at the operating point, the head is rise with the flow rate and required system head. A single pump performance system curve is shown in Fig.6. This system shows the unthrottled system curve (valve full open) and the 100 percent speed pump curve determine the maximum operating point of the pump rates at 36.8 m^3/hr. With the full open controlling valve the system head loss is small. The operating point would be determined by the intersection of the pump curve and the system curve. With the control of throttled valve the pump curve is reduced by reducing the pump speed and the reduced pump speed can be obtained by the centrifugal pump laws. With the control throttled valve the operating point was at 24 m^3/hr at.

103.31 m head and reduced the speed the operating point moves down the system curve with the result that flow and head are reduced. If the flow rate falls below

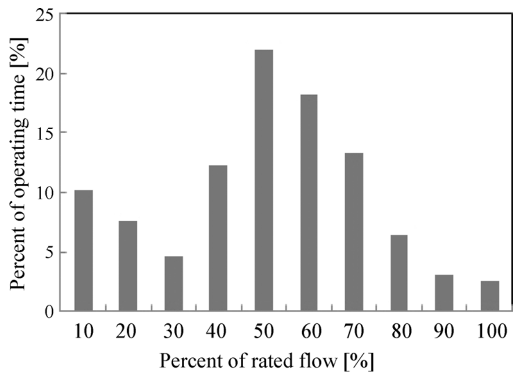

Fig. 5 Mean duty cycle for the centrifugal pump

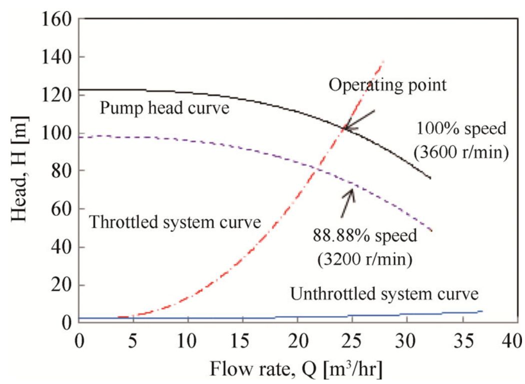

Fig. 6 Variable flow variable pressure system characteristics curve and revolution control operation

the operating flow rate, the pump pressure head rises above the required system head and so, the flow increases back to the operating point. Conversely, if the flow rate increases, the required system head exceeds the head provided by the pump and the flow rate decreases back to the operating point. The point of interaction of operating point is the only condition where the pump and system flow rates are equal and the pump and system head are equal simultaneously. With the variable rotational speed the system curve would be cut the different pump head and the operation point of the pump can be operated below the constant drive pump. The difference in pressure between operating along the 100 percent pump curve and along the system curve represents the potential energy saving because of the valve losses. From the above description, when the variable speed control is subjected by the pump, the shaft power differs depending on the pump system characteristics curve even under the same speed control. So, to determine the input power for different operating points on the 100 percent speed pump curve and on the system curve would be determined. Table 2 shows the power requirements for constant and variable speed drive operation.

Input power of the system

In the centrifugal pump, input power profile for both constant and variable speed drive pump is the most important to calculate the energy saving rates. A variable-flow variable-pressure system is conducted which is shown in Fig.1. This system operates at different fixed flow reference with the controller and this type of system would be required to maintain a constant flow through a changed of characteristics of the system. The input power profile for both constant and variable drive system is shown in Fig. 7. From this figure reduce the different rotational speed with different flow rate the input power of the variable speed drive system is reduced gradually.

Energy saving rates

The energy saving, and therefore the cost saving is depend on the input power and specific duty cycle of the system. Fig.7 and Fig.5 shows the input power profile and mean duty cycle of the centrifugal pump. From Fig.7 input power is carried out using pump performance and specific duty cycle is segmented into ten segments. The input power reduction is afforded by use of variable speed drive is shown in Table 2. From Table 2 at 16.8 m^3/hr, the power input is cut almost 56.25 percent for the variable speed system; the reduction at 2.4 m^3/hr is more than 83 percent. Table 3 illustrates the energy consumption for the constant and variable speed drive pump. The operation involved 365 days × 24 hours per day, or 8760 hours per year. The procedure is illustrated using operation 50 percent flow rates as a reference calculation. A 50 percent flow rate, the pump delivery was 0.5×24 m^3/hr =12 m^3/hr. From Table 2, the pump input power was 8.12 kW at this flow rate for the constant speed drive operation. At this flow rate, the pump operates 21.944 percent of the time or 0.21944×8760 = 1922.33 hours per year. The total energy consumed at this duty point was 12.94× 10^3 kW.hr. The electrical energy consumed was the corresponding cost of electricity [at ₩114.5 ~ \$0.11/(kW.hr)] is

$$C = 12.94 \times 10^3 x \frac{\$0.11}{\text{kW.hr}} = \$1423.4$$

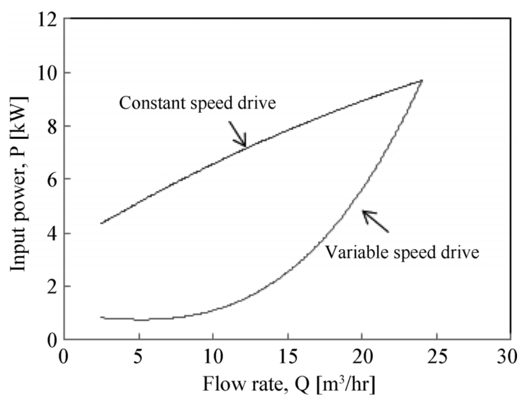

Fig. 7 Variable-flow variable pressure system, input power profile

Table 2 Power requirements for constant and variable drive pump

Throttle valve control with constant speed (3600 rpm) motor and variable speed drive with energy efficient motor				
Flow rate (m^3/h)	Head (m)	Pump efficiency (%)	Shaft power (kW)	Power Input (kW)
24.0	103.31/103.31	70.20/70.20	9.69/9.69	11.03/11.03
21.6	108.47/81.15	69.85/71.02	9.24/6.68	10.55/7.70
19.2	112.63/80.11	68.05/70.97	8.77/5.68	10.02/6.61
16.8	115.90/51.63	64.77/70.15	8.25/3.43	9.44/4.13
14.4	118.38/34.56	60.02/69.31	7.69/1.97	8.80/2.36
12.0	120.19/25.88	53.80/68.45	7.09/1.24	8.12/1.51
9.6	121.44/28.50	46.12/64.65	6.46/1.13	7.39/1.37
7.2	122.22/30.42	36.96/55.90	5.78/1.01	6.60/1.21
4.8	122.65/31.63	26.34/42.21	5.08/0.86	5.77/1.03
2.4	122.83/31.90	14.25/23.57	4.33/0.69	4.88/0.83

Table 3 Energy calculation for constant speed drive variable speed drive pump

Constant speed drive/ variable speed drive, time 8760 hr/yr					
Flow rate (%)	Flow rate (m^3/h)	Operation time (%)	Operation time (hr)	Power Input (kW)	Energy (kW.hr)
100	24.0	2.50	219.00	11.03/11.03	$9.80\times10^3/9.80\text{x}10^3$
90	21.6	3.0556	267.67	10.55/7.70	$7.06\times10^3/5.15\text{x}10^3$
80	19.2	6.389	559.66	10.02/6.61	$4.02\times10^3/2.65\text{x}10^3$
70	16.8	13.333	1168.00	9.44/4.13	$10.10\times10^3/4.42\text{x}10^3$
60	14.4	18.194	1593.83	8.80/2.36	$16.92\times10^3/4.54\text{x}10^3$
50	12.0	21.944	1922.33	8.12/1.51	$12.94\times10^3/2.41\text{x}10^3$
40	9.6	12.222	1070.66	7.39/1.37	$8.63\times10^3/1.60\text{x}10^3$
30	7.2	4.583	401.50	6.60/1.21	$3.69\times10^3/0.67\text{x}10^3$
20	4.8	7.639	669.17	5.77/1.03	$1.54\times10^3/0.28\text{x}10^3$
10	2.4	10.139	888.17	4.88/0.83	$1.07\times10^3/0.18\text{x}10^3$
				Total	$75.77\times10^3/31.70\text{x}10^3$

Table 4 Annual energy savings of variable speed drive pump

Energy saving result of variable speed drive pump	
Cost saving (annually)	$4,847.7
Energy consumption (annually)	44,070 kW.hr
Energy saving rate	58.16%

Similarly, Table 3 was prepared using the different flow rates. Summing the last column of the table shows that for the constant speed drive system the annual energy consumption was 75.77×10^3 kW.hr and the total energy cost of the system was $8,334.7.

On the other hand, the annual energy consumption was 31.70×10^3 kW.hr of the variable speed drive system shown in Table 3. At $0.11/kW.hr, the energy cost for the variable speed drive system was only $3487. Thus in this application, the variable speed drive reduces energy consumption by 44,070 kW.hr (58.16%). The cost saving is $4,847.7 annually.

Conclusion

This study is based on the energy efficient system operation on the variable speed drive multistage centrifugal pump in a closed loop system. The result of this system indicates that the best energy saving can be obtained when the same type of the pumps running at the same speed ratio. The system duty cycle and input power of the system was considered for calculating the energy saving rate. A model DR 20-60 multistage centrifugal pump was installed of our designed layout to get the pump performances for both constant and variable speed drive systems.

The numerical simulation was investigated to compare with the experimental data for its model validation, effectiveness and reliability of the pump. The mean deviation of the head was only 5.4%, efficiency was only 8.22% and the power deviation was only 6.35%. So, the trend of numerical simulation pump performances data was following the experimental results.

The duty cycle was measured by segmented into different flow rates and the average value for each segments by the interval time which were from 24 m^3/hr to 2.4 m^3/hr pump capacity. The maximum duty cycle operated on 50% rated pump capacity and the operation time was 21.944% but in 100% flow rate was only 2.5%. The energy consumption annually was 44,070 kW.hr and saving rate of the system was 58.16% at an operating rate from 24 m^3/hr to 2.4 m^3/hr. According to specific duty cycle, actual energy saving rate would be changed.

In order to obtain operating point of the system we accounted the experimental results of pump curve and system curve. The pump system was measured experimentally which is described as head = actual head + K (flow)^2, the flow rate changed with the changed of different speed. The K (head loss) values were unknown because each of controlled flow rates, the K values changed. Theoretically K values are considered of all minor losses like as reducer losses, gate valve losses, elbow losses and so on. The system head loss was less at full speed operation at 36.8 m^3/hr at 3600 rpm and the system curve and the head curve was intersects at that point of 100 percent operating speed which we found the operating point but the head and efficiency was only 6 m and 7.43% but the system could not be operated at 100 percent speed, so, we reduced the pump speed with different flow rates with the control throttled valve. The best efficiency point of the pump was at 24 m^3/hr at 69.16% at 3600 rpm but the efficiency would be changed when the change of the speed. The operating point changed below the full operated speed at 88.88% speed. The input power was calculated from the each operating point of the system curve of the system.

On the based on results, the application for variable speed drive control can be a numerous in the domestic and industrial sectors and mainly to analyze the energy saving rates and it is necessary to specify to each specific duty cycle on which the system operates. Therefore the energy savings would be improved performance and life cycle of the pump.

Acknowledgement

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