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A study on multistage centrifugal pump performance characteristics for variable speed drive system

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Abstract

Nowadays centrifugal pumps are being widely used in the commercial, industrial and power plant applications and most of pumps operated by constant speed drive system. Therefore, pump consumes a huge energy of each nation's total energy. But it could be operated in variable speed drive system which would be provided energy saving. The purpose of this study is to investigate the pump performance characteristics of the multistage centrifugal pump with the variable drive system. For this study an experimental set up of the system was constructed to achieve the centrifugal pump performances such as H-Q, η-Q, P-Q curves and operating points which interact between performance and system curves. In the variable speed drive system, a vector controlled inverter driving (variable voltage variable frequency) was installed in the experimental system. A numerical investigation also applied for getting the pump performances for the validation and reliability of the pump design development and also the pressure and velocity effects in internal flows of the pump are analyzed. For the numerical analysis, the Navier-Stokes equations were discretized by the finite volume method and two equations transport turbulence (SST) model accounts for three dimensional steady flows. In the experiment system, we also carried out system head performance of the three pumps in parallel to compare with one pump system head for its validation. In order to get the energy saving rate using the inverter control variable speed drive system instead of the constant speed drive system, it is necessary to identify the specific duty cycle of the pump operation cycle and operating system curve of the pump. Hopefully, this paper will be useful as a guide for identifying a method of implementing a variable speed drive system with inverter control in the variable flow and pressure system.

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Keywords: Multistage centrifugal pump; Variable speed drive system; Inverter; Performance analysis; System curve; SST turbulence model;

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1. 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]. The combination of rotating impeller and diffuser is called the stage. A multistage centrifugal pump might consist of several stages within a single housing, depending on the amount of pressure rise required of the pump [2].

According to statistics, pumps consume around 20% of the world total energy [3]. 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 [4,5]. The point of interaction is the only condition where the pump and system flow rates are equal and the pump and system heads are equal simultaneously. For the same type of pump operation, for one pump when the valve is fully open and flow rate compared with two or three pumps in parallel operation resulting to obtain the system curve. There are two driving systems. One is constant speed drive system which is installed with the pump and motor for transferring fluid. The other is variable speed drive system which is installed with pump, motor and inverter for changing speed. Inverter is a revolution control device which is used in the variable speed drive system [6, 7] and this control method is effectively used for pumps where operation time is long and output is high. Also energy savings are likely due to surface elevation and usually lower pumping rates and lower pipe friction losses can be obtained using variable speed drive system [5]. Assessment of the technical and economic advantages gained by using VSDs on centrifugal pumps have been a limited publicized in recent years. But with the performance and system curve we can't estimate satisfactory efficiency improvement using activated pump system. So, we have to find alternative method to evaluate the performance characteristics driving system.

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. CFD has proven to be a very useful tool in the analysis of the flow inside pumps, both in design and performance prediction. Much research has been carried out in the last years. Croba et al. [8] give an updated list of general selected papers while Denuset al. [9] give a more extended and specific bibliography. However, due to the difficulties of the task, most of these studies have been carried out with strong simplifications of the problem either in the geometry or in the flow characteristics. Research is slowly tending toward more complete simulations and approach developed follows the trend [10]. 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. For the numerical simulation in the pump, the main difficulty is to better reproduce the complex geometry configuration of the flow domain [11]. For these difficulties many of geometry are often considered for simplifications. In the literature, many hydrodynamics models are reported in 2D and 3D by using the CFD code and studied impeller diffuser interaction on the pump performance showed that a strong pressure fluctuation is due to the unsteadiness of the flow shedding from impeller [12, 13]. Both experimental and numerical approaches have been reported and have contributed to the understanding of the highly complex flow interactions that occur in a centrifugal pump [14].

In this paper, the study is focused on the pump performance characteristics of the multistage centrifugal pump with variable speed drive system. Therefore an experimental set up is constructed for both the one pump and three pumps in parallel. The pump performances were calculated by the electronic flow meter for the variable speed condition controlled by inverter. The system head curve calculated from both of one pump and three pumps in parallel. Also, we investigated the numerical simulation which predicts to get the pump performances and effect on pressure and velocity inside the pump. Thus these results compare with the experimental data for its effectiveness and reliability of the pump model DR 20-60.

2. Experimental Method

Fig. 1 shows the experiment layout of model pump and Fig. 2 represents the experiment layout of three pumps. The test layout is of the variable flow and pressure system. A fully computerized pump test facility is designed and built to obtain more accurate pump performance test data. Two electronic flow meters are used in the experiment system for better accuracy.

The working temperature and humidity were 28.5^{0}C and 81%. For each test, the rotational speed was set by the frequency inverter and the speed was measured by a torque meter showing in the pump operation panel. Pump head was measured by algebraic difference height of liquid between the discharge and inlet sections. Pump shaft torque measured by torque meter which is 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.

In order to calculate the system head of one pump, the pump operated at constant speed with different flow rates and head data are taken. Then flow rate was fixed with the design flow rate and operated at full rotational speed that was 3600 rpm. Then gradually decreased the speed and flow rates and head were taken with the reduced speed. For the three pumps in parallel at first the one pump was operated with the design flow rate at constant speed and head data was taken. With the changing speed, flow rates and system head data were taken. Accordingly two pumps and three pumps were operated at the design flow rates and measured the system head. Pump head were measured with constant speed at different flow rates.

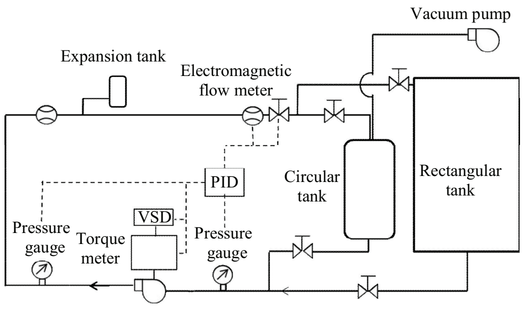

Fig. 1 Experimental layout of the model pump

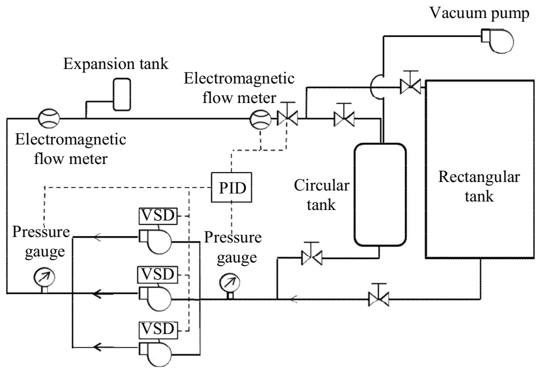

Fig. 2 Experimental layout of the three pumps in parallel

2.1. Pump performance evaluation

In order to evaluate 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 [15]. Hydraulic method was used in our performance investigation.

3. 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 meshed with unconstructed tetrahedral cells which are shown in Fig. 3(a). The impeller had six blades and diffuser had ten blades. With the ANSYS ICEM-CFX surface mesh setup made, applied tetrahedral method, and computed mesh volume, and checked mesh quality. The total meshing grids were 5,265,401 nodes and 27,529,524 elements. Multistage centrifugal impeller domain was set up as rotating part and diffuser was considered as stationary domain. The model of a six stages multistage centrifugal pump considered in this study is shown in Fig. 3(b). 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 (shear stress transport turbulence) model was assumed as Newtonian fluid and the thermo-physical properties were constant with the temperature. To account for these assumptions, the theoretical analysis of the fluid flow was based on the continuity and momentum equations [16, 17]. The continuity and momentum equations are expressed as Eq. (1) and Eq. (2)

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

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

Where u_i is the velocity vector, p is the pressure scalar, ρ is the density, i and j is 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 highly accurate predictions of 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)

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

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

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.

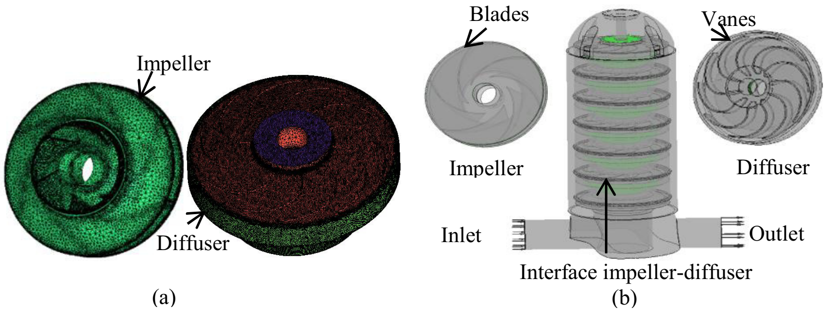

Fig. 3 (a) Meshing of impeller and diffuser; (b) impeller-diffuser domain interface

4. Results and discussion

4.1. 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 of different flow rates with different rotational speed. From this graph its being 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 found 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 in 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%. The Fig. 4 shows a small deviation of the numerical data and follows the trend of the experimental results.

4.2. 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

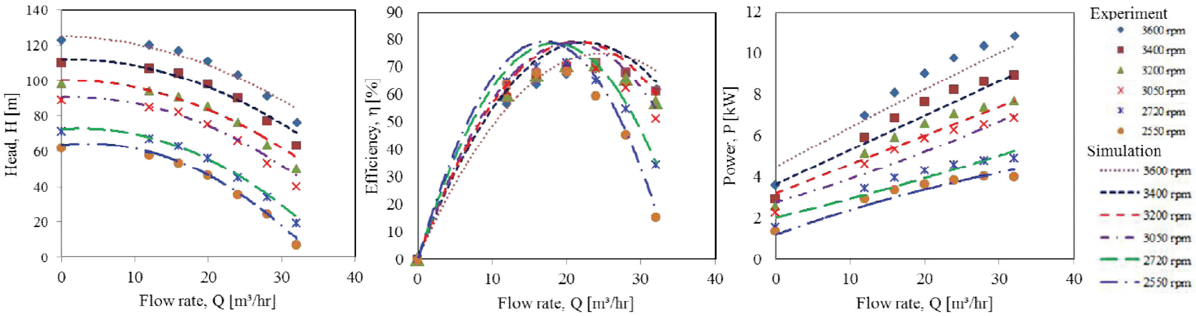

Fig. 4 Comparisons of experiment versus numerical performance curve of the multistage centrifugal pump (a) head (H) vs flow rate (Q); (b) efficiency (η) vs flow rate (Q); and (c) power (P) vs flow rate (Q)

required system head match. A single pump performance system curve is shown in Fig. 5(a). This system shows the system curve at the rated flow rate. For the rated flow rate pump curve determine the maximum operating point of the pump rate at 36.8 m^3/hr. With the full open controlled valve the system head loss is small and H-Q curve not intersect the system curve. The operating point would be determined by the intersection of the pump curve and the system curve. With the controlled valve the pump curve is reduced by reducing the pump speed and the reduced pump speed can be obtained by the centrifugal pump laws. At that operating point the designed flow rate was 24 m^3/hr at 103.31m head. The operating point moves down the system curve with the result that flow and head are reduced accordingly controlled speed.

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 rated flow pump curve and the system curve represents the potential energy saving because of the valve losses Fig. 5(b) represents the operating characteristics of pump head curve and system curve of three pumps in parallel. In this figure, with the one pump the operating point is at the best efficiency point at the design flow rate at maximum rotational speed (3600 rpm). The system head is done by varying the different speeds. With the two pumps in operation at full speed the system head is increases as well as the pump head and flow rate increased. The pump head and the system head matched at the 2Q flow rates. For three pumps, the operating point matched at 3Q flow rates and head is slightly increases. The system curve is intersecting at just above the second pumps head. Fig. 5(c) shows the inverter drive control system curve of model pump. Fig. 5(d) represents the comparison of one pump and three pumps in parallel for the model pump system curve for validation. The system head for model pump is almost same of the three pumps system head in parallel.

5. Conclusion

The study is based on the inverter controlled variable speed drive multistage centrifugal pump in a closed loop variable flow and pressure system. A model pump was installed in designed layout to achieve the performance for the constant and variable speed drive conditions. The result of this system indicates that the energy saving could be obtained when the similar type of pumps would be running at same speed ratio. For variable speed drive pump indicates, the operating point moves down the system curve with the result that flow and head are reduced accordingly controlled speed. But in case of constant drive pump, the operating point moves forward to the head curve as a result that flow is reduced and head is increased. 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.

Three pumps system curve are considered for model pump system curve validation. The potential applications for variable speed drive with inverter control system represents a potential energy saving and it can be used a numerous in the domestic and industrial sectors. The system would be improved performance and life cycle of the pump because of low pressure.

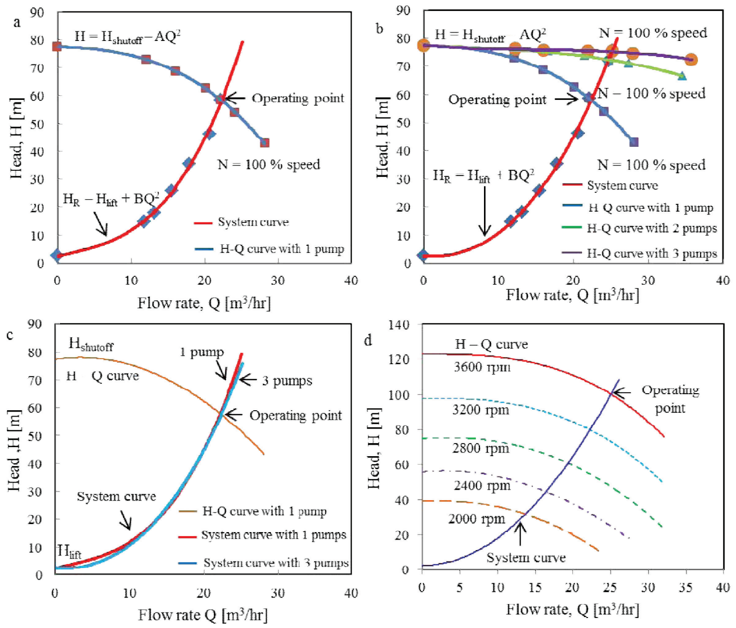

Fig. 5 (a) inverter controlled variable flow and pressure system characteristics; (b) operating characteristics of three pumps; (c) System head for model pump (d) system head for one pump and three pumps.

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