

ANALYSIS OF PERFORMANCE CHARACTERISTICS OF MULTISTAGE CENTRIFUGAL PUMPS USING VARIABLE SPEED DRIVE

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Abstract —Centrifugal pumps are used to increase the energy of the fluid by raising it to a specified pressure level. Centrifugal pump performance at a certain rotor speed is given by rate of flow delivered, the pressure rise achieved, the power absorbed at coupling NPSH and efficiency. The purpose of this study is to investigate and study the pump performance characteristics of using multi stage centrifugal pump using variable speed drive conducted by “Rakibuzzaman, San Ho Suh and others” in there journal on “A study on multistage centrifugal pump performance characteristics for variable speed drive system” . For this reviews there study which is based on experimental setup of the system was constructed to achieve the centrifugal pump performance such as H-Q, η -Q, P-Q curves and operating points which interacts between system curves and performance. In variable speed drive system, a vector controlled inverter driving (variable voltage variable frequency) was installed in experimental setup. A numerical analysis was also applied to get the pump performances for validation and reliability of the pump design development and also the pressure and velocity effects in internal flows of the pump are analyzed. For numerical analysis Navier-stokes equation was used in finite volume method and two equations of transport turbulence (SST) model accounts for 3D steady flow.

Keywords-multi stage centrifugal pump, variable speed drive system, performance analysis, SST turbulence model.

I. INTRODUCTION

The centrifugal pump is the most used pump type in the world. The principle is simple, well-described and thoroughly tested, and the pump is robust, effective and relatively inexpensive to produce. There is a wide range of variations based on the principle of the centrifugal pump and consisting of the same basic hydraulic parts.

An increase in the fluid pressure from the pump inlet to its outlet is created when the pump is in operation. This pressure difference operates the fluid through the system or plant. The centrifugal pump creates an increase in pressure by transferring mechanical energy from the motor to the fluid through which has to be pumped to the rotating impeller. The fluid flows from the inlet of the impeller center called “eye” and moves out along its blades. The centrifugal force here increases the fluid velocity and as a result also the kinetic energy is transformed to pressure. Figure 1.1 shows an example of the fluid path through the centrifugal pump [1]. Computational fluid dynamics is being used in the design of centrifugal multistage pump which can be used for numerical simulations to obtain the performance of the flow field inside the pump. In this paper, the study is mainly focused on the pump performance characteristics of centrifugal multistage pump with variable speed drive. Also observed the numerical simulations which predicts to get the pump performance, effect of pressure and velocity inside the pump. [1-6]

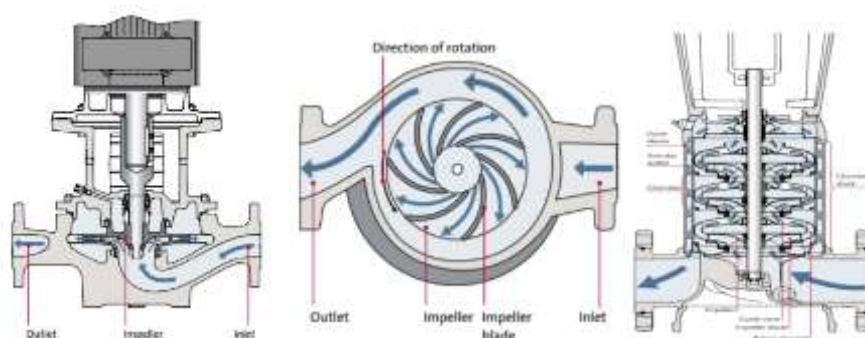


Figure 1. Fluid path through the Inline multistage centrifugal pump. [1]

II. LITERATURE REVIEW

Figure 2 shows the experimental layout of model of pump and figure 3 shows the experimental layout of the 3 parallel pumps [1] as setup by “Rakibuzzaman, San Ho Suh and others” in there journal on “A study on multistage centrifugal pump performance characteristics for variable speed drive system”. There test layout is of variable flow and pressure system. In order to calculate the system head of one pump, what they have done is the pump was operated at constant

speed with different flow rates and head data were taken and then operated at full rotational speed 3600rpm, then gradually reduce the speed and flow rates and head were taken at reduced speed.

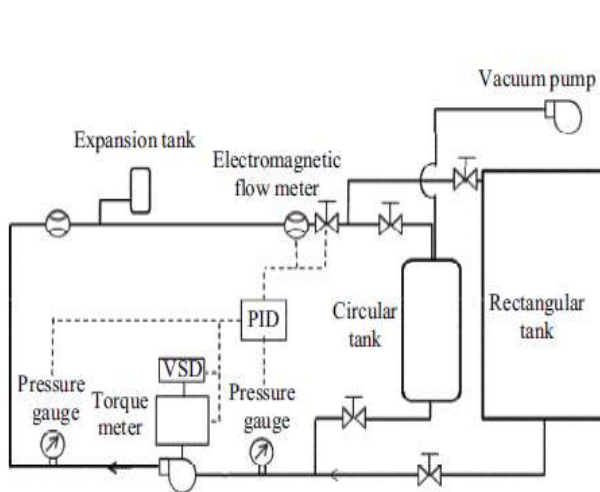


Figure 2. Experimental layout of pump model [1]

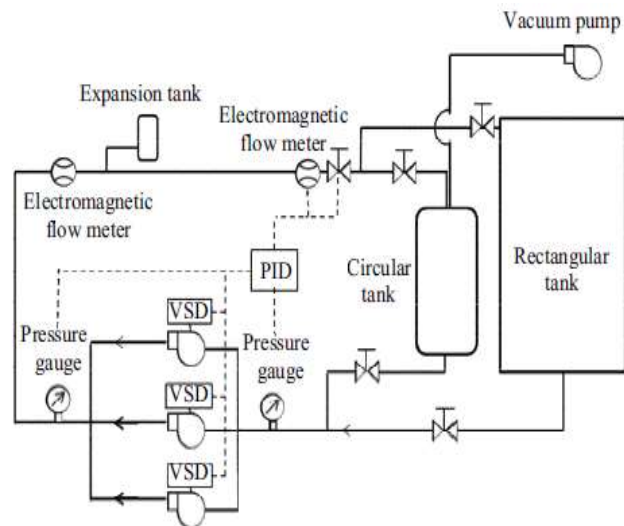


Figure 3. Experimental layout of 3 pumps in parallel [1]

3.1. Performance evaluation- Numerical Method

To validate the experiment, there team have used the computer simulation. The geometry of the 6 stage centrifugal impeller and diffuser was here used for meshing by using the ANSYS ICEM-CFX-14.5. Each of the impeller, diffuser, inlet and outlet casing was meshed with unconstructed tetrahedral cells which are shown in Fig. 4. [1]

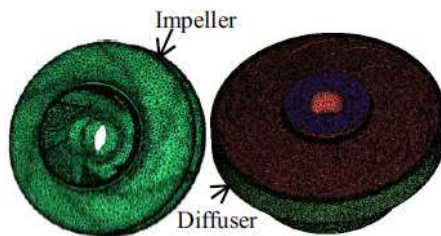


Figure 4 (a). Meshing of impeller and diffuser

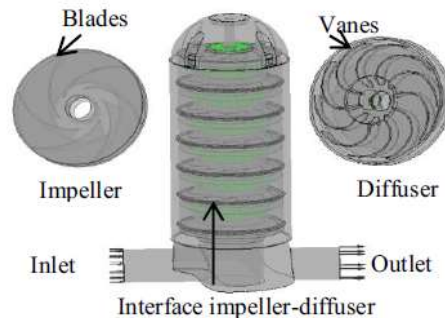


Figure 4 (b). Impeller- Diffuser domain interface

The theoretical analysis was based on well-known continuity and momentum equations. The continuity and momentum equations are expressed by equations (1) and (2), [3]

$$\frac{\partial u_i}{\partial x_i} = 0 \quad \dots\dots (1)$$

$$\rho \left(\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} \right) = - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu \frac{\partial u_i}{\partial x_j} - \overline{\rho u_i u_j} \right) \quad \dots\dots (2)$$

Where u_i is the velocity vector, p is the pressure scalar, i and j are the tensor notations, $-\overline{\rho u_i u_j}$ is the apparent turbulent stress tensor, μ is dynamic viscosity. [1]

They have used the $k-\omega$ based SST model which accounts for the turbulent shear stress and highly accurate of the prediction of onset and the amount of flow separation under adverse pressure gradient case and the unknown turbulent viscosity μ was determined by them by solving two addition transport of turbulent energy k , and for the turbulence frequency ω . These two equations can be written as equation (3) and (4) [1,3]

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

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

Where, P_κ is the production rate of turbulence, μ_t is the turbulent viscosity, α , β , β' , σ_κ , σ_ω are constants. [1,4]

4. RESULTS AND DISCUSSION

Rakibuzzaman, San Ho Suh and others in there journal on “A study on multistage centrifugal pump performance characteristics for variable speed drive system” [1] provided the figure 5 which shows the pump head, efficiency and power of the system used at different flow rates and at different rotational speed. From there study below graph was obtained, it is clear that the deviation in experimental and numerical data is about 5.4%. [1] The highest deviation found is of 11.06% at the highest flow rate at 3050rpm and the differences for only three conditions are larger than 8.5%. With the decrease in the rotational speed, the head is being continuously decreased. [1]

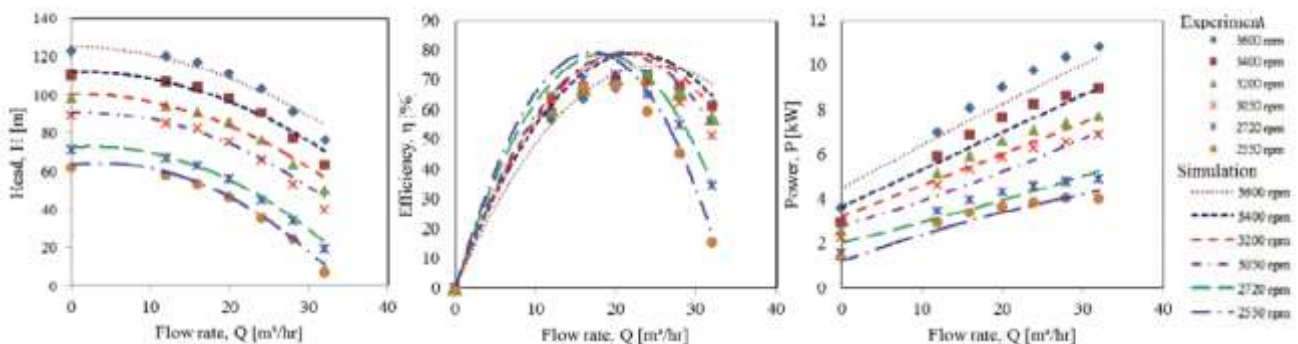


Figure 5. Comparison of the experimental and numerical performance curves of centrifugal multistage pump (a) Head Vs Flow rate (Q), (b) efficiency (η) Vs Flow rate (Q), (c) power (P) Vs Flow rate (Q) [1]

For variable speed drives pumps indicates that the operating points moves down the system curve with the result that head and flow are reduced accordingly controlled speed. But in case of constant speed drive system, the operating point moves forward to the head curve which results in reduced flow but increased head. Thus the result of this system indicates that the energy saving could be obtained when the pump is running by variable speed drive system.

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