NUMERICAL STUDY OF FLOW IN SIDE CHAMBERS OF A CENTRIFUGAL PUMP AND ITS EFFECT ON DISK FRICTION LOSS

[1]MOHAMMADREZA DAQIQSHIRAZI, [2]ALIREZA RIASI, [3]AHMAD NOURBAKHSH

[1,2,3]University of Tehran

Abstract— Centrifugal pumps are ubiquitous in industry. Huge attempts for loss reduction in this gamut of instruments have been made; however, due to complexity of fluid flow in Turbomachinery, most previous works were confined to experimental works. In this paper we try to elucidate the flow in a real pump side chambers with the use of Computational Fluid Dynamics. A real pump chamber has been modeled and a high quality mesh has been developed, thus the modeling error has been reduced. Navier-Stokes equations have been solved for this domain. In order to predict the turbulence effects the SST model has been employed. This model has shown its capability in capturing physical phenomena occurring in swirling flows. The influence of three different parameters, namely pre-rotation, leakage and viscosity, on disk friction has been investigated. The latter is especially important when the pump is subjected to different temperatures.

Keywords—Numerical Study, Centrifugal pumps, Disk friction loss

I. INTRODUCTION

Disk friction can cause a huge loss of energy in turbo machineries. This loss is majorly caused by the interaction between the fluid and the moving bodies, say, impeller; as well as the interaction between the rotating fluid and the stationary components. This sort of losses becomes very important in the case of a low N_S where, sometimes, the disk friction loss may account up to 50 percent of the total power delivered to the fluid for a pump with $N_S = 10$. As mentioned above, such importance has initiated many works in this area. A gamut of reference can be reviewed in. Fluid in a rotating chamber due to centrifugal forces on the rotating surface pumps out, and due to continuity it has to return inward. Initial works were mostly confined to experiments with a disk rotating in a cylindrical casing. For example in the influence of geometrical parameters of a chamber has been investigated and some correlation as a function of Reynolds Number and roughness has been presented. In later works, the importance of leakage flow effect through the room between the impeller and the casing pressure distribution and disk friction loss has been shown [6]. What is obvious from these works is that there is an influence on the disk friction loss with the change of parameters like Reynolds Number, surface Roughness, leakage flow rate, leakage absolute velocity and geometry.

Flow on both front and rear shrouds of one stage centrifugal pumps enters the side chambers radially. Fluid on shroud in this domain moves outward due to centrifugal forces. Then, continuity causes the fluid to return the pump eye. This rotation has a profound role on the flow in these chambers, which is usually expressed as a ratio of radial velocity of the fluid c_u to the rotation velocity $u = r_2\ \omega$ and denoted by k. Without leakage, rotation becomes independent of its radius and equals to β/ω. In such circumstances the rotation is only dependent to impeller geometrical parameters. Correlations have been introduced for this in [7]. Leakage brought in a momentum with itself to the side gaps. The momentum can be calculated using a simple analytical equation

$$M_{in} = \rho Q_l\ r_i\ c_{u,i} \tag{1}$$

There will be a pre-rotation of 0.2 to 0.5, as the seal length differs [1]. Power consumed by a disk rotating in a cylindrical casing without leakage flow has been measured enormously. Although a rotating impeller in a cylindrical casing is a simplified model; it can be used as a reference for further studies. In most works core rotation has been used as a measure for power loss. For pre-rotation and core rotation many correlations have been introduced. These correlations for four different flow regimes from laminar with attached boundary layer to turbulent flow with detached boundary layer have been presented. Pantel and linneken extracted correlations for all flow regimes. to consider roughness and leakage, Gulich has modified linneken correlation. In other experiments, correlation for rough impellers in sooth casing and vice versa has been introduced. As the flow patterns are quite different in a real pump, others investigated rotating disk power loss in a spiral casing. Through these studies a 40 to 70 percent higher losses were seen. Hregt and Prager examined a volute with separated shroud which was moving by a different motor. By this scheme they could measure the exact amount of disk friction. For studying the influence of the net flow on the disk friction Karakowa solved governing differential equations and illustrated the results in several graphs. Zieling and Mohring used an integration method on the basis of momentum balance in Equation 1 and drag forces on rotating and stationary walls. Gulich used the same method, but he accounted the second term of the turbulence shear stress ò$_{\tau}$; by such method, he had to solve the equations numerically.

In this work, we analyzed a three dimensional domain of a real pump, thus our domain harbor less simplification errors and leads us to a better understanding of the fluid flow .Also, SST model with a very fine mesh near walls used . In this way, near wall modeling of fluid excels. Viscosity, leakage magnitude, and leakage direction were studied and results were presented in result section.

II. METHODOLOGY

In this study, we have analyzed our domain by solving equation of fluids for Newtonian fluids, i.e. Navier Stokes Equations.

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho U) = 0 \qquad (2)$$

Momentum equation

$$\frac{\partial(\rho U)}{\partial t} + \nabla.(\rho U \otimes U) = -\nabla p + \nabla.\tau + S_M \qquad (3)$$

In which, the stress tensor, τ , is related to strain rate with the use of following formula

$$\tau = \mu(\nabla U + (\nabla U)^T - \frac{2}{3}\delta \nabla.U) \qquad (4)$$

In the SST turbulence model, the wall distance from the modified wall scale equation is used to compute the two blending factors. In this method, the turbulence eddy viscosity is computed from the turbulent kinetic energy and turbulent frequency

$$\mu_T = \frac{\rho k}{\omega} \qquad (5)$$

III. NUMERICAL SIMULATION

In our study a real pump model side wall gap has been modeled. The pump has a specific speed of $N_S = 13$. As mentioned above, in such ranges of specific speed the disk friction losses becomes a major part of our losses. In addition to this, the Reynolds Number is $\mathrm{Re} = 151*10^4$ .It worth mentioning that the Reynolds number is defined as follows:

$$\mathrm{Re} = \frac{u_2 r_2}{\nu} \qquad (6)$$

Figure 1 the three dimensional fluid domain

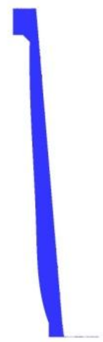

Figure 2 A two dimensional overview of domain

A high quality structured mesh was developed. One of the obstacles in this modeling was producing good meshes in wear ring area where the gap width is about 100 microns.

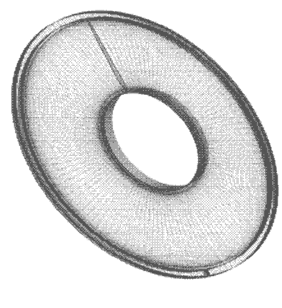

Figure 3 the three dimensional mesh

IV. GRID INDEPENDENCY

In order to assure the quality of the mesh, the model was meshed three times with cell counts of 317640, 451440, and 847440. Torque on rotating body used as the controlling parameter, and the order of 1.644 has been reached. With having this in mind that all

equation has been discretized to second order. It seems satisfactory.
Torque changes in these three meshes have been illustrated in the diagram below.

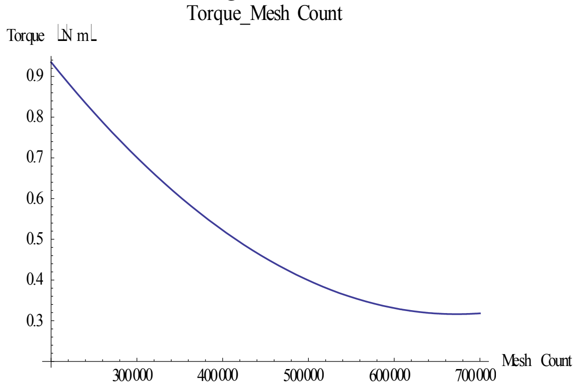

Figure 4 torque mesh relationship

To get a good near wall modeling, the mesh needs an extremely fine mesh in these areas. The mesh with 847440 high quality cells was used for our study. In addition to this, y^+ is much smaller than equity in most of the domain area.

The velocity inlet boundary condition and a static pressure outlet with the data of best efficiency design point of the pump have been used. Leakage as a function of the normal component of velocity on the inlet of side wall gap, and pre-rotation with the consideration of radial velocity component has been imposed

V. RESULTS AND DISCUSSION

In our study we solved fluid governing equations for side chamber of a centrifugal pump. Three influencing parameters, pre-rotation, leakage flow magnitude, and viscosity were investigated.

There are several different configurations for diffuser at outlet in a centrifugal pump. In our type B configuration, the flow enters the space between the shroud and casing and through a bend it reaches the side gap. Here, a small eddy forms. Then, fluid enters the gap mostly far from the rotating surface as the fluid is being pumped out by the rotating surface which shapes a large eddy (Figure 4). This rotation is essential part of disk friction loss origin as discussed in introduction.

In high Reynolds Numbers the separated flow regime occurs, the boundary layers becomes totally separated (Figure 10)

As viscosity rises, the torque imposed on the front shroud grows slightly (Figure 5). This could be inferred by intuition that as the fluid becomes more viscous, the dissipation of energy as the fluid comes in contact with surfaces and also during the rotation when the fluid particles hit each other.

The magnitude of leakage enters side wall chambers can be investigated by a study on the C_m. This component of the entered velocity represents the magnitude of leakage. Dissimilar to most previous studies, we impose velocity components adjacent to the pump outlet. As a consequence, it gives us a better control on inlet boundary condition where on this place we extracted the data from the pump outlet.

As we have more leakage flow, the imposed torque dwindles (Figure 7). This is the consequence of the fact that as there is more mass to absorb energy, as well as, a limited time to convey this energy (the flow exit faster). The offset between the current work and the Nece and Daily work illustrated on this figure is a result of the different inlet position. In their work, leakage flow was radially imposed on the chamber.

Pre-rotation is the ratio of Cu or the radial velocity component to the tangential velocity of the impeller at outlet, thus in Figure 3 the connection with this parameter with torque has been illustrated. Torque by itself denotes the disk friction imposed on rotating body as it is the ratio of absorbed power to the angular velocity ω.
As pre-rotation increases, the torque enforced to the rotating body decreases (Figure 8). This phenomenon is caused by the fact that the inlet flow has more angular momentum entailed with itself to this domain as discussed in Equation 1; therefore it needs less energy to be conveyed to it from the rotating body. In this way, the equal force imposed on fluid lessens.

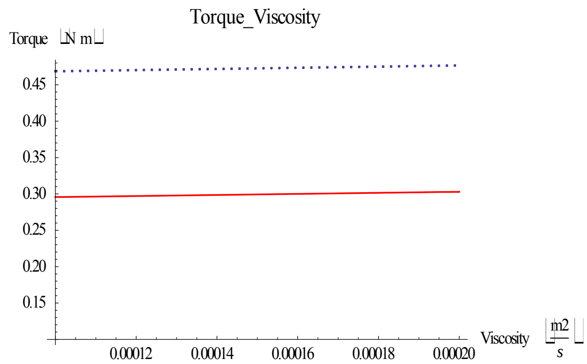

Figure 5. Torque and viscosity connection, This study results are presented with a red line and based on equation Daily and Nece with a dotted line

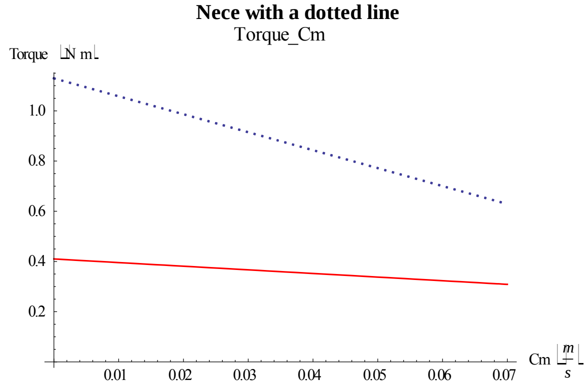

Figure 6. Torque versus leakage magnitude , ,This study results are presented with a red line and based on equation by

Daily and Nece with a dotted line

In figures 9 and 10 the flow rotation has been depicted. There, we have two swirls one at the entering area where fluids moves to the side chambers. Subsequently, a large rotation in secondary flows occurs in this part.

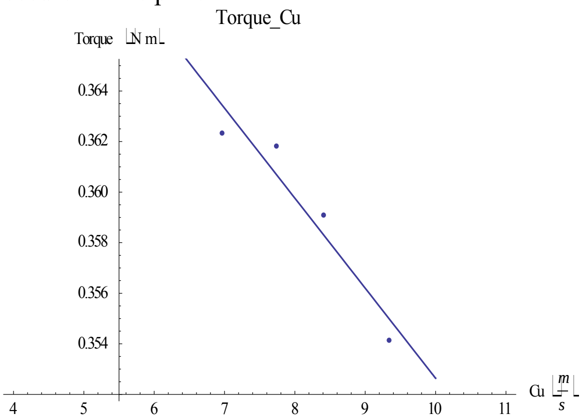

Figure 7. Prerotation influence on torque

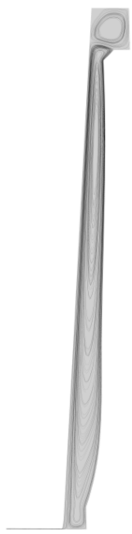

Figure 8 streamlines in sidewall chamber

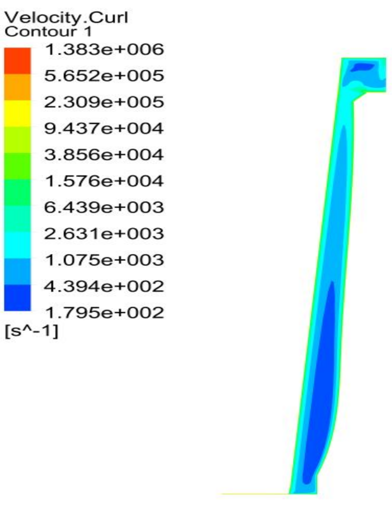

Figure 9 Velocity.Curl Contour in the side wall chamber

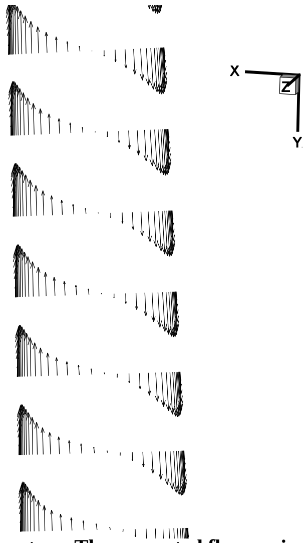

Figure 10.Velocity vectors. The separated flow regime as well as core flow is obvious in this picture

CONCLUSION

The disk friction loss of centrifugal pumps is being investigated in this study. Unlike previous works, we used a real pump side chamber; a high quality mesh has been developed for this study, subsequently fluid characteristics were investigated by solving governing equations numerically. Leakage magnitude and prerotation in fluid entering the domain cause less torque imposed on rotating bodies thus reducing the friction loss. In contrary, viscosity intensifies disk friction loss. Therefore, cooling or temperature control and in pumps with low specific speed is suggested.

REFERENCES

[1] Gülich, J., *Disk friction losses of closed turbomachine impellers.* Forschung im Ingenieurwesen, 2003. **68**(2): p. 87-95.

[2] Gulich, J.F., *Centrifugal pumps*. 2007: Springer.

[3] Kurokawa, J. and M. Sakuma, *Flow in a narrow gap along an enclosed rotating disk with through-flow.* 1988.

[4] Daily, J. and R. Nece, *Chamber dimension effects on induced flow and frictional resistance of enclosed rotating disks.* Journal of Basic Engineering, 1960. **82**: p. 217.

[5] Pantell , K., *Versuche uber Scheibenreibung.* Forsch Ingenieurwes, 1949/50. **16**: p. 97-108.

[6] Hergt, P. and S. Prager, *Influence of different parameters on the disc friction losses of a centrifugal pump*, in *Conf on Hydraulic Machinery*1991: Budapest. p. 172-179.

[7] Zilling, H., *Untersuchung des Axialschubes und der Strömungsvorgänge im Radseitenraum einer einstufigen Radialpumpe.* Strömungsmech Strömungsmasch, 1973.

[8] Linneken, H., *Der Radreibungsverlust, insbesondere bei Turbomaschinen.* AEG Mitt, 1957. **47**(1/2): p. 49-55.

[9] Gülich, J.F., *Kreiselpumpen: Handbuch Fur Entwicklung, Anlagenplanung und Betrieb*. 2004: Springer DE.

[10] Geis, H., *Experimentelle Untersuchungen der Radseitenverluste von Hochdruck-Wasserturbinen radialer Bauart.* 1986.

[11] Osterwalder, J. and C. Ettig. *Determination of individual losses and scale effect by model tests with a radial pump*. in *Instnt. Mech. Engrs. Conf. Stirling, paper C*. 1977.

[12] Kurokawa, J. and T. Toyokura, *Axial thrust, disk friction torque and leakage loss of radial flow turbomachinery.* 1976.

[13] Möhring, U.K., *Untersuchung des radialen Druckverlaufes und des übertragenen Drehmomentes im Radseitenraum von Kreiselpumpen bei glatter, ebener Radseitenwand und bei Anwendung von Rückenschaufeln*, 1976.

[14] Childs, D.W., *Dynamic analysis of turbulent annular seals based on Hirs' lubrication equation.* Journal of lubrication technology, 1983. **105**: p. 429.

[15] Nece, R. and J. Daily, *Roughness effects on frictional resistance of enclosed rotating disks.* Journal of Basic Engineering, 1960. **82**: p. 553.

★★★