CFD Analysis of PUMP: A Case Study

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Abstract:

There is an ever increasing use of centrifugal pumps in applications which require the pumping of liquids containing solids, and liquids other than water. The application in industry nowadays such as oil and gas with the increasing use of centrifugal pumps for applications of this type, it becomes important to be able to predict pump characteristics and create efficient design. Design changes will be suggested in an attempt to improve the performance of the impeller used in the pump. This project consists of the detailed study of a model of the centrifugal pump and also consists of the detailed to identify, observe and determine the pattern of velocity profile and pressure distribution by using CFD simulation program. This project will definitely provide much helpful information while contributing to the knowledge about the design and characteristics of centrifugal pumps.

1. Introduction:

The growth and improvement of many key industrial processes have always been linked to the improvements in the pumping equipment. Industrial Energy Equipment Pumps play a particularly important role because of their capacity to handle high flows. In fact, Industrial Energy Equipment Pumps constitute more than 85% of the world's production of pumps, as they are frequently used in sewage, food processing, water treatment and manufacturing plants, as well as the chemical and petroleum industries, where they are used for the pumping of all types of low-viscosity fluids. They can also easily handle liquids with high proportions of suspended solids present in them. With the many varieties of available pump configurations, a proper design is the most important requirement for any facility. 20% of the total energy consumed globally is used to run a pump of one sort or another—yet, two-thirds of these pumps use 60% more energy than is required. To ensure energy efficiency and prevent equipment failure, it is important to be able to predict and evaluate the Improved Industrial Energy Equipment Pumps design with CFD Simulation FloEFD pump's performance under different operating conditions. [1-10]

2. Methodology Adopted :

The governing equations of viscous flow are based on conservation of mass, momentum and energy which are Langrangian in nature. The governing equations are expressed using equations (notations used have their usual standard meanings) shown below:

$$\frac{\partial \rho}{\partial t} = -\nabla \cdot (\rho \, \mathbf{u}) \tag{1}$$

$$\rho \left(\frac{\partial}{\partial t} + \mathbf{u} \cdot \nabla\right)\mathbf{u} = -\nabla P + \rho \mathbf{g} + \frac{1}{c}\mathbf{J} \times \mathbf{B} \tag{2}$$

$$\rho \left(\frac{\partial}{\partial t} + \mathbf{u} \cdot \nabla\right)e = -P\nabla \cdot \mathbf{u} + \rho \mathbf{u} \cdot \mathbf{g} + \frac{1}{\sigma}\mathbf{J}^2 \tag{3}$$

The numerical analysis of CFD in Industrial Energy Equipment Pumps consists of incompressible fluid flow that reduces the conservation of mass and momentum to equations shown below

respectively. In addition, the temperature effect is negligible during the analysis. Therefore, conservation of energy is ignored during analysis.

There are many commercial general-purpose CFD programs (Interdisciplinary field of study based on Physics, Engineering, Mechanics, Biology, Material Science supported by both Mathematics and Computer Science) available, e.g. Ansys-Fluent, Ansys-CFX, Star-CD, FLOW 3D, SoldWorks Flow Simulation and Phoenics. A very useful open-source program that can handle CFD problems is OpenFoam. However, the documentation and the user interface are not well developed as those for the commercial codes. Commercial CFD packages contain modules for CAD drawing, meshing, flow simulations, solver and post-processing.

The CFD analysis for the same was carried out using the FLOEFD, for analysing the performance. For very complex systems the results are not very accurate, but CFD can still be very useful saving design engineer's time-cost-effort. Experimental validation verifies the codes to make sure that the numerical solutions are correct and compare the results (making a provision for measurement errors).

The FLOEFD solver solves the Navier Stokes and conservation equations. The equations that we used are not closed, so we need to use Turbulence Modelling to close the equation set and then iterate towards a solution. We used what is called a Reynolds Averaged Navier Stokes (RANS) approach, (or we can use an Eddy Simulation technique which resolves the larger eddies in the flow and is only really required when you have separation or large re-circulating regions). The most commonly used models are the RANS models due to their low cost in terms of compute power and run times. The Eddy Simulation methods can be quite mesh sensitive but will yield much better results for separated and recirculating flow, but takes much longer run times. There are different turbulence models available in FLOEFD as mentioned below: Spalart-Allmaras Model; k-Ɛ (k-Epsilon) Model-widely used; k-ω (k-Omega) Model; v2-f Model; Reynold's Stress Model (RSM); Detached Eddy Simulation Model (DES); Large Eddy Simulation Model (LES) etc.[11-20]

3. Theory and Calculation :

The modified k-ε turbulence model with damping functions proposed by Lam and Bremhorst (1981) describes laminar, turbulent, and transitional flows of homogeneous fluids consisting of the following turbulence conservation laws.

$$\frac{\partial \rho k}{\partial t} + \frac{\partial \rho k u_i}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right] + \tau_{ij}^R \frac{\partial u_i}{\partial x_j} - \rho\varepsilon + \mu_t P_B \,,$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial \rho \varepsilon u_i}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_i}\right] + C_{\varepsilon 1}\frac{\varepsilon}{k}\left[f_i \tau_{ij}^R \frac{\partial u_i}{\partial x_j} + C_B \mu_t P_B\right] - f_2 C_{\varepsilon 2}\frac{\rho \varepsilon^2}{k} \,,$$

$$\tau_{ij} = \mu S_{ij}, \tau_{ij}^R = \mu_t S_{ij} - \frac{2}{3}\rho k \delta_{ij} \,, S_{ij} = \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k} \,,$$

$$P_B = -\frac{g_i}{\sigma_B}\frac{1}{\rho}\frac{\partial \rho}{\partial x_i} \,,$$

Where $C_\mu = 0.09$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_k = 1$, $\sigma_\varepsilon = 1.3$, $\sigma_B = 0.9$, $C_B = 1$

If $P_B > 0$, $C_B = 0$ If $P_B < 0$, *The turbulent viscosity is determined from*

$$\mu_t = f_\mu . \frac{C_\mu \rho k^2}{\varepsilon},$$

Lam and Bremhorst's damping function f_μ *is determined from* :

$$f_u = (1 - e^{-0.025 R_y})^2 \left[1 + \frac{20.5}{R_i} \right],$$

Where

$$R_y = \frac{\rho \sqrt{ky}}{\mu},$$

$$R_t = \frac{\rho k^2}{\mu \varepsilon},$$

y is the distance from point to the wall and Lam and Bremhorst's damping functions $f1$ and $f2$ are determined from:

$$f_1 = 1 + \left[\frac{0.05}{f_\mu} \right]^3, \quad f_2 = 1 - e^{R_t^2}$$

Lam and Bremhorst's damping functions $f\mu$, $f1$, $f2$ decrease turbulent viscosity and turbulence energy and increase the turbulence dissipation rate when the Reynolds number Ry based on the average velocity of fluctuations and distance from the wall becomes too small. When $f\mu$ =1, $f1$ =1 , $f2$ =1 the approach obtains the original k-ε model.[11-20]

4. Result and Discussion :

Comparison between numerical results and experimental data reveals a good agreement. Industrial Energy Equipment pumps are not much accurate at low flow rates due to rotor/bearing drag that decelerates the rotor. Almost 5% of minimum rated flow capacity is required. It should not be run at high velocity because premature bearing wear and/or damage can occur. We need to be careful when measuring fluids that are non-lubricating because bearing wear can cause error. With the help of CFD the rotor driving torque (and power) is calculated on blades by using boundary conditions.

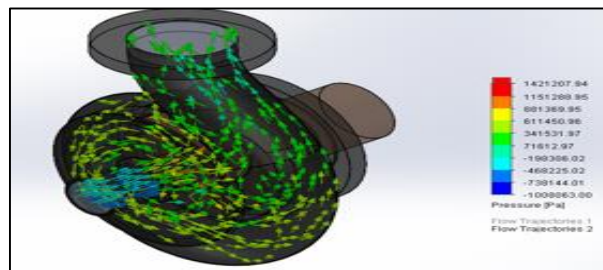**Case-1:** <u>Impeller with 5 number of blades</u>

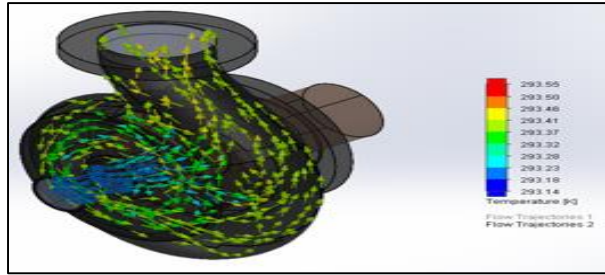Fig. 1. Pressure – CFD Analysis (5 Blades)

Fig. 2. Temperature – CFD Analysis (5 Blades)
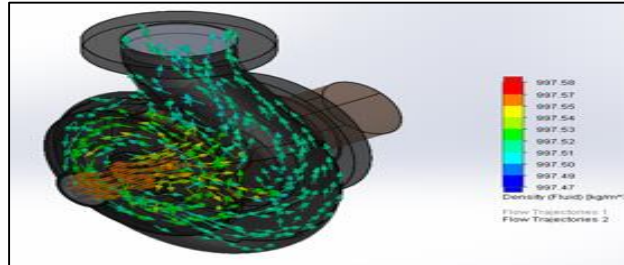

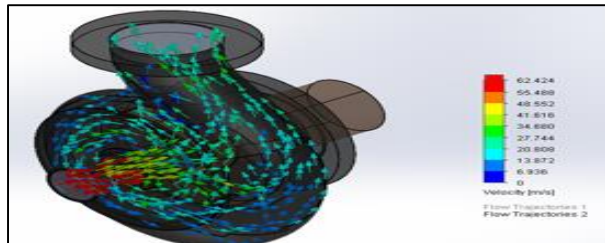Fig. 3. Density – CFD Analysis (5 Blades)
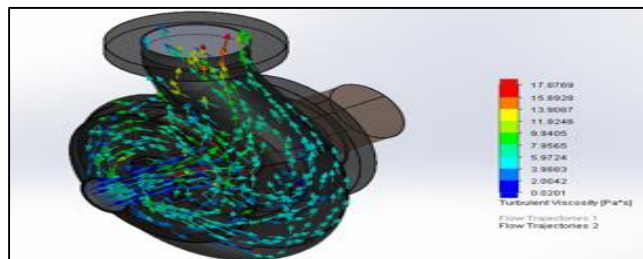

Fig. 4. Velocity – CFD Analysis (5 Blades)

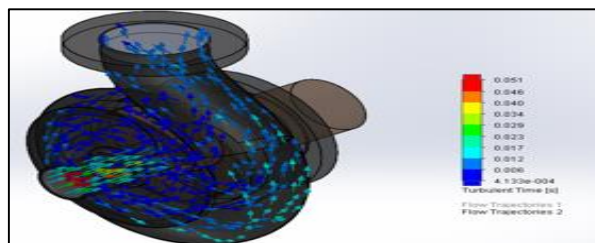Fig. 5. Turbulent viscosity – CFD Analysis (5 Blades)
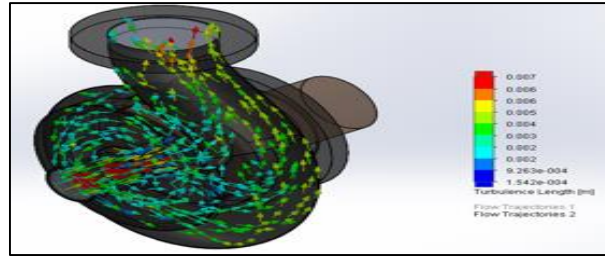

Fig. 6. Turbulent time – CFD Analysis (5 Blades)

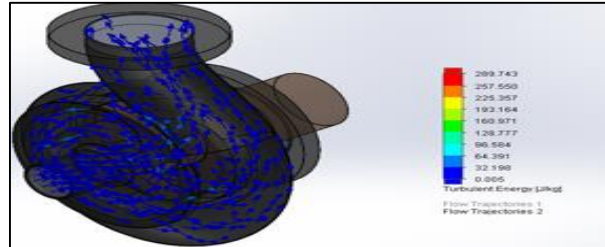Fig. 7. Turbulent Length – CFD Analysis (5 Blades)

Fig. 8. Turbulent Energy – CFD Analysis (5 Blades)
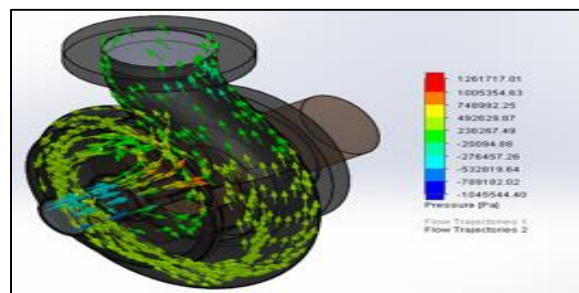
Case-2: Impeller with 7 number of blades

Fig. 9. Pressure – CFD Analysis (7 Blades)

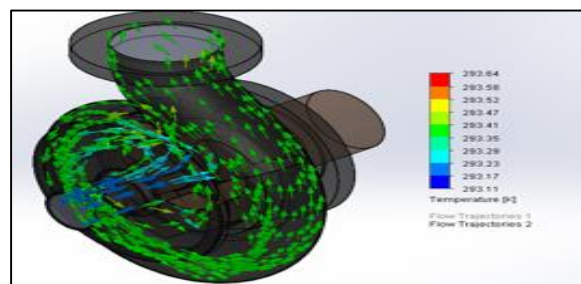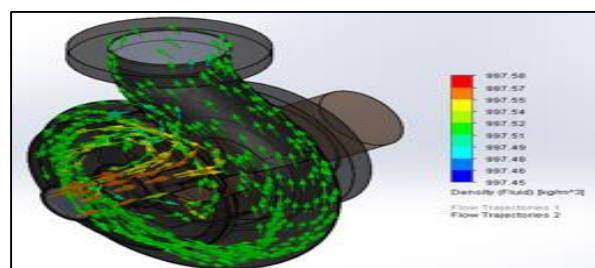Fig. 10. Temperature – CFD Analysis (7 Blades)

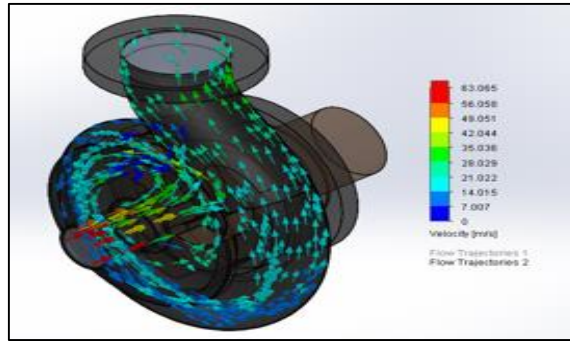Fig. 11.Density – CFD Analysis (7 Blades)

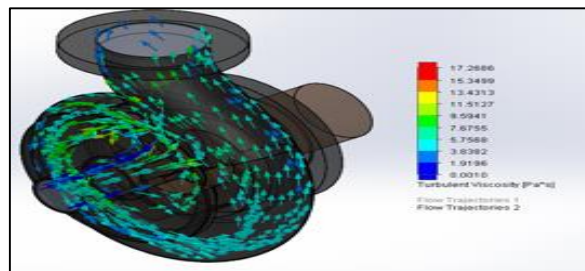Fig. 12. Velocity– CFD Analysis (7 Blades)

Fig. 13. Turbulent Viscosity – CFD Analysis (7 Blades)
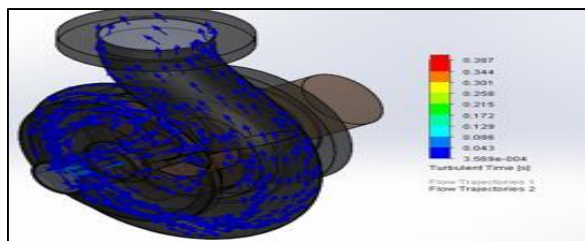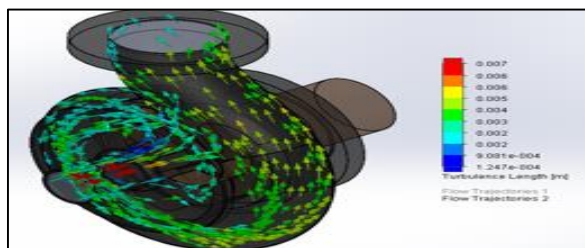

Fig. 14. Turbulent time – CFD Analysis (7 Blades)

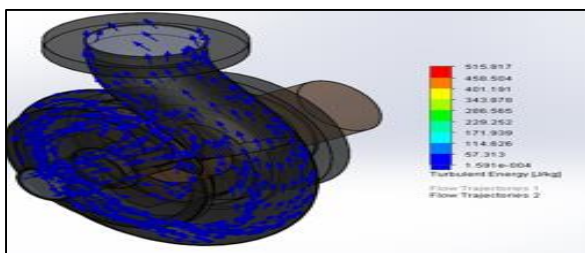Fig. 15. Turbulent Length – CFD Analysis (7 Blades)

Fig. 16. Turbulent Energy – CFD Analysis (7 Blades)

5. Conclusion :

The advanced CFD model used in this research solves the Navier-Stokes equations, which are formulations of mass, momentum and energy conservation laws for fluid flows. This CFD model is able of predicting both laminar and turbulent flows. Most of the fluid flows in engineering practice are turbulent, so this model uses the Reynold-Averaged-Navier-Stokes (RANS) equations, where time-averaged effects of the flow turbulence on the flow parameters are considered. Through this procedure, extra terms known as the Reynolds stresses appear in the equations for which additional information must be provided. To close this system of equations, it employs transport equations for the turbulent kinetic energy and its dissipation rate (k-ε model). This research shows the utility of the CFD numerical simulations as a tool for design and optimization of Industrial Energy Efficient Pump performance and flow behaviour through hydro mechanical devices or hydraulic structures at minimum time-cost-effort.

Acknowledgements :

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