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Turbine Driven Pump CFD Modelling and Simulation- a Centrifugal Pump Optimization for Irrigation

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Abstract— **The primary purpose of this work is to design and investigate the optimum performance of a turbine driven centrifugal pump for small scale irrigation using computational fluid dynamics modelling and simulation that can deliver water to a head of 10m consuming less than 1kW. The centrifugal pump is driven directly by the turbine and the turbine performance simulation is coupled to that of the pump and in this part of the work only the performance optimization of centrifugal pump is reported. Blade numbers, flow rates, outlet angles are the variables used in design optimization. The computed Computational Fluid Dynamics results show that too many blades creates crowding out effect occurrences at the impeller resulting in severe hydraulic losses while too few a blade number causes higher diffuser losses. Increasing the flow rate increases pumping power, but further pumping power is decreased because the change in pressure at impeller inlet and outlet is reduced which decreases pump efficiency and head delivered. Small and large outlet angle of the impeller increases hydraulic loss and decreases pump efficiency. Optimum performance of centrifugal pump is obtained at a flow rate of 25m^3/h, rotational speed of 1125 RPM, and a blade number of six to deliver water to a head of 10.31m consuming a 740W of power.**

Index Terms— **Computational fluid dynamics, impeller, modeling, optimum performance, simulation**

I. INTRODUCTION

Agriculture is the core driver for Ethiopia's growth and long-term food security. The stakes are high: 15 to 17% of government of Ethiopia's expenditures is committed to the sector; agriculture directly supports 85 of the population's livelihoods, 43 % [1] of gross domestic product (GDP), and over 80% of export value [1]-[3]. Agriculture is the mainstay of the Ethiopian economy in terms of income, employment and generation of export revenue [4].

Agriculture in Ethiopia is dominated by smallholder rain-fed systems but, low and erratic rainfall limits productivity and food security. Consequently, investment in small-scale irrigation has been identified as a key poverty reduction strategy [4] Recent estimates indicate that the total irrigated area under small-scale irrigation in Ethiopia has reached to 853,000 hectares during the last implementation period of PASDEP-2009/10 and the plan set for development of small scale irrigation is 1850,000 hectares, which is planned to be achieved by the end of the five years growth and transformation plan (GTP) of 2015 [6],[7].

Even though agriculture plays great role in the country's economy, it suffers from frequent drought and p o or cultivation practices [1]. Therefore, there should be continuous agricultural production through irrigation which is independent of seasonal rain water. For sustainable economic and social development, overcoming food security and increasing annual income of farmers multiple productions in a year is unquestionable and should be adapted in community [1], [8], [9].

Most irrigation schemes, which professionals design approach starting from the site selection, are considerably less demand driven and they are costly for small scale irrigation systems. Small scale irrigation which has high potential in increasing income of farmers and ensures food security faces the following problems. 1) Cost of civil works is too high to design, construct diversion and distribution lines for small scale irrigation. 2) Large part of the rural areas is not connected to on/off grid power supplies. Absence electricity in the remote rural areas to run electric motor pumps. 3) Cost of fuel is increasing from time to time. It is difficult for farmers to buy fuel for fueled motor pumps. 4) Most of small scale water pumping technologies requires human power to operate and their operation is intermittent due to unavailability of operators. For example; treadle pump, and rope pump are lifting water only when human power is applied.

To overcome the specified problems renewable energy source small scale irrigation are necessary to solve rural communities to overcome rainfall dependence and increase productivity. This work focused on CFD modelling and design optimization of a turbine driven centrifugal pump which consumes less power from renewable energy resources.

A turbine driven centrifugal pump is an innovative pumping technology. The flowing water from very low head will exert a pressure on one side and suction the other side of all turbine blades. This pressure difference from pressure side and suction side creates a torque on the whole turbine blades and, hence makes the turbine to rotate.
Power produced by turbine is transferred to centrifugal pump through mechanical coupling (gear). Due to existence of loses in turbine, pump, gear and other auxiliary components power generated by turbine should be greater than power required by pump to lift water to overcome losses. Axial flow turbine designed for high flow rate and low head while

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centrifugal pump design requirement is low flow rate and high head to deliver water from the river.

II. Literature Review

Centrifugal pumps are widely used in many applications like industries to provide cooling and lubrication services, to transfer fluids for processing, and to provide the motive force in hydraulic systems, irrigation [10]-[12] and the like, and pump system may operate over wide flow rate and head depending on their application.

CFD simulation is a power full tool and low cost to predict flow inside the pump, and optimize the design for best efficiency. 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 [13].

Various researchers have considerably contributed to revealing the flow mechanisms inside centrifugal pump, impeller, impeller and volute. Knowledge about off design of centrifugal pump is essential. Few research reports are available that had compared flow and pressure fields among different types of pump and there is still a lot of work to be done in this field [14]. With the increase of the blade number, the limitation between blade and flow stream gets more, and mixture loss is reduced in centrifugal pump [15]. Increasing blade number increases suction side pressure at inlet grows continuously [14], [16]. Pump head increases with a greater blade number because the decrease in the liquid pressure drop in the flow passage with an increased impeller blade number, keeping the same total volume flow rate the pump brake horsepower increases relatively with the increased blade number.

The effect of outlet predicted is using CFD simulation [17], [18]. The flow in the centrifugal pump impeller with a large exit blade angle was subject to separation near the blade pressure side, but, a large exit blade angle helped improve the pump performance [15], [19]. The CFD results were in qualitative agreement with experiment and thus that CFD is able to handle the effect of the exit blade angle on the performance and flow of a pump, regardless of fluid viscosity [19]. A large exit blade angle results in an increase in the hydraulic loss over the entire flow rate range in the volute. But, it caused an increase in hydraulic loss at a low flow rate and reduced the loss at a high flow rate in the impeller [15], [19]. As the outlet blade angle from 200 to 500 increases the gain in the head is more than 6% [18]. At designed or more than designed mass flow rates, the fluid flows smoothly along the blade walls. The blade curvature exhibits a weak vortex at the pressure side of the blade and low-pressure regions [11].

3-D modeling, simulation, design optimization and performance analysis of a centrifugal pump impeller is the primary objective of this paper. Flow rate, blade number and outlet angle are taken as design variables for optimization of the centrifugal pump impeller.

III. Methodology

CFD modeling and simulation of centrifugal pump was done using ANSYS 14.5 CFX package. Design optimization of centrifugal pump was done at a flow rate of 5, 10, 15, 20, 25, 30, 35, 40, 45, 50, 60, and 65m3/h and blade number is varied 4 to 10 with one step size. The performance of centrifugal pump also, analyzed at three different impeller outlet angles (β2) i.e. 11.250, 22.50 and 33.750. The goal of the optimization is to pump water up to 10m from the river for small scale irrigation consuming a power less than 1kW and at high efficiency.

A. Governing Equations

a. Continuity equation

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

b. *Momentum equation*: for the rotor rotating in a fixed frame of reference momentum equation is defined by

$$\frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} + \frac{1}{\rho}\frac{\partial P}{\partial x_j} - \frac{\partial}{\partial x_j}\left[(v+v_t)\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\right] = 2\varepsilon_{ijk}\Omega_j U_k + \Omega_i \Omega_j x_j - \Omega_i \Omega_j x_i \tag{2}$$

The turbulent viscosity is obtained by

$$v_t = c_\mu \frac{k^2}{\varepsilon} \tag{3}$$

c. *The turbulent kinetic energy (k) and its dissipation rate (ε):* are calculated from the transport equations (4) and (5) respectively [12].

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} - \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] = G - \varepsilon \tag{4}$$

$$\frac{\partial \varepsilon}{\partial t} + U_j \frac{\partial \varepsilon}{\partial x_j} - \frac{\partial}{\partial x_j}\left[\left(v + \frac{v_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] = c_1 \frac{\varepsilon}{k} G - c_2 \frac{\varepsilon^2}{k} \tag{5}$$

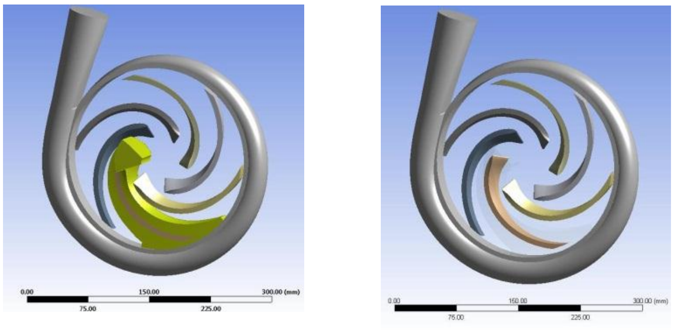
Where G is given by

$$G = v_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j} \tag{6}$$

The model equation constants values are $c_\mu = 0.09$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $c_1 = 1.44$ and $c_2 = 1.92$

B. Centrifugal pump modeling

CFD simulation of centrifugal pump begins with geometry creation of pump impeller and pump volute. The geometry is created using Vista CPD and BladeGen. First geometry is created in Vista CPD and the exported BladeGen. Fluid and Solid domains are defined in ANSYS Geometry window. The solid domain is suppressed from the geometry, because ANSYS CFX used only for fluid physics.



(a) (b)

Fig. 1 Centrifugal pump: (a) solid domain only, (b) both fluid and solid domain

C. Meshing of centrifugal pump

The impellers and volute has an interface and they are discretized using Generalized Grid Interface. The numerical algorithms employed, as well as the control surface treatment of the numerical fluxes across the interface, are designed and implemented in such a way as to provide for maximum robustness and accuracy. The treatment of the interface fluxes is fully implicit and fully conservative in mass, momentum, energy, scalars, etc. This means that the multi grid solver can be applied directly, without any penalty in terms of robustness or convergence rate, to problems involving generalized grid interface (GGI) conditions.

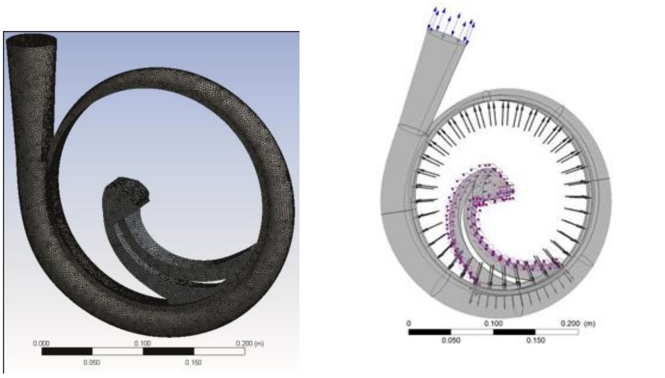


Fig. 2 Meshing, defining boundary condition and pre processing

Only one impeller blade is used in modeling and simulation to minimize computational memory and time. Using interface boundary condition the result of all blades is obtained at end of the solution. Design optimization is done only for impellers not for volute due to the difficulty of creating different volute geometry.

Table 1: Centrifugal Pump Mesh statistics

Mesh category	Number of mesh
Global Number of Nodes	208261
Global Number of Elements	776607
Total Number of Tetrahedrons	5952112
Total Number of prisms	181395
Global Number of Faces	66035

D. Defining boundary conditions

Centrifugal pump boundary condition is divided in to impeller boundary condition and volute boundary condition, they are defined independently. The out let of the impeller is the inlet of the centrifugal pump. Table 1 and 2 shows the boundary condition of the impellers and volute.

Table 2: Boundary conditions

Impeller boundary	
Boundary name	Type of boundary condition
Inlet	Mass flow rate
Outlet	Mass flow rate
Periodic 1	Interface
Periodic 2	Interface
Shroud	Wall
Blade	Wall
Hub	Wall
Volute boundary condition	
Inlet	Mass flow arte
Out let	Pressure out let
Surface	wall

IV. RESULTS AND DISCUSSION

Centrifugal pump performance which is modeled and simulated at different design variable is discussed below.

a. Performance evaluation at different flow

Figure 3 shows that pumping power increases with increasing flow rate and reaches maximum and then decreases with increasing flow rate because as the mass flow rate increases the change in pressure at impeller inlet and outlet decreases which cause a decrease in efficiency and head delivered.

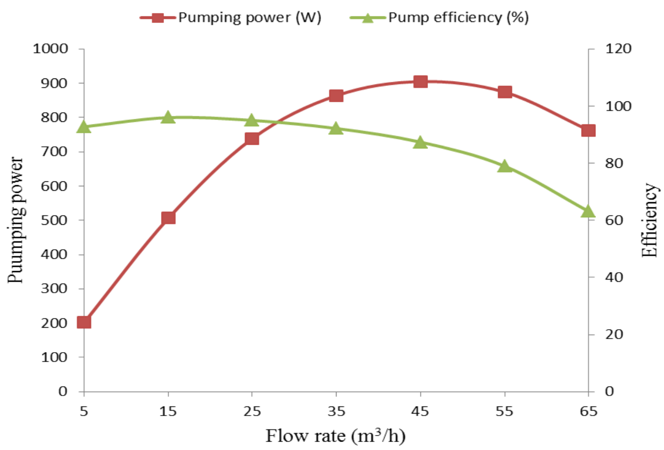


Fig. 3 Effect flow rate on pumping power and efficiency

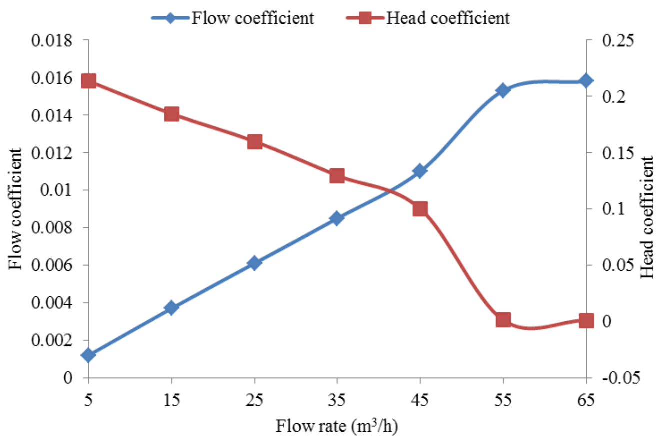


Fig. 4 Non-dimensional performance parameters vs flow rate

Increasing flow rate decreases head coefficient because as flow rate increases change pressure from inlet to outlet decreases. Efficiency increases to maximum point 95.9745% and then decreases while head continuously decrease with increasing flow rate.

a. Performance evaluation at different blade number

The blade number of impeller is an important design parameter of pumps, which affects the characteristics of pump heavily. Maximum head is obtained at maximum power consumption (figure 5) at blade number of 7 but, maximum efficiency is obtained for 6 (figure 6) because as blade number increases crowding effect of blade number increases flow velocity of water which increase flow velocity, increasing flow velocity means friction head loss increases.

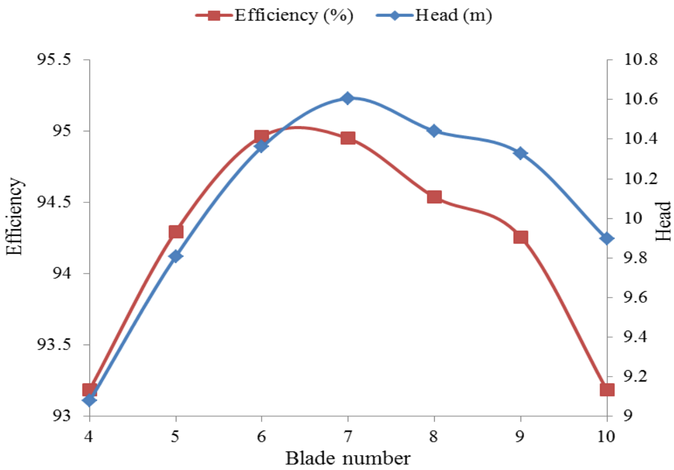


Fig. 5 Effect of blade number on centrifugal pump performance

Figure 6 shows the variation of pump efficiency and head delivered as a function of impeller blade number. Increasing blade number causes crowding out effect at the impeller is serious and causes for increase of flow velocity. The interface between the fluid stream and blade will increase as blade number increases. If blade number is too few, the diffuse loss will increase with the increase of the diffuse extent of the flow passage.

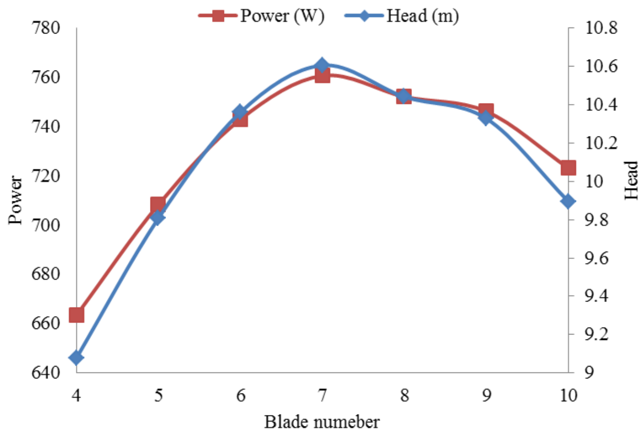


Fig. 6 Effect of blade number on centrifugal pump performance

b. Performance comparison at different out let angle

Increasing exit impeller increases the head delivered and efficiency of a centrifugal pump (figure 7), but, it is not always due to the fact that increasing the exit blade angle increases total hydraulic loss in the impeller and volute. Increasing losses in the impeller and volute increases pump power consumption, decreases head delivered and efficiency.

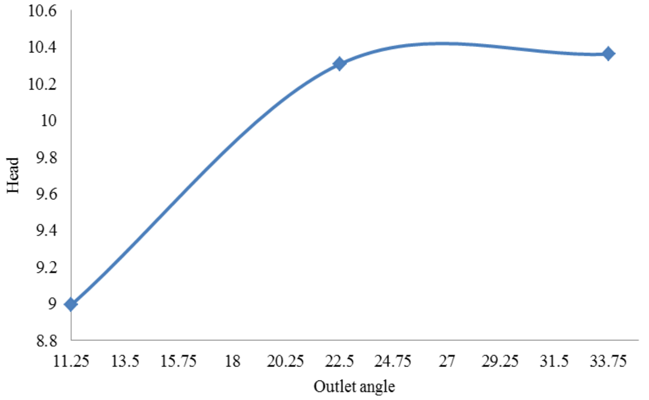


Fig.7 Effect of pump outlet angle on head delivered by pump

Pump performance was analyzed at different blade outlet angles to analyze how pumping power, head delivered and pump efficiencies. Figure 8 to 10 show the effect of impeller outlet angle on pump performances.

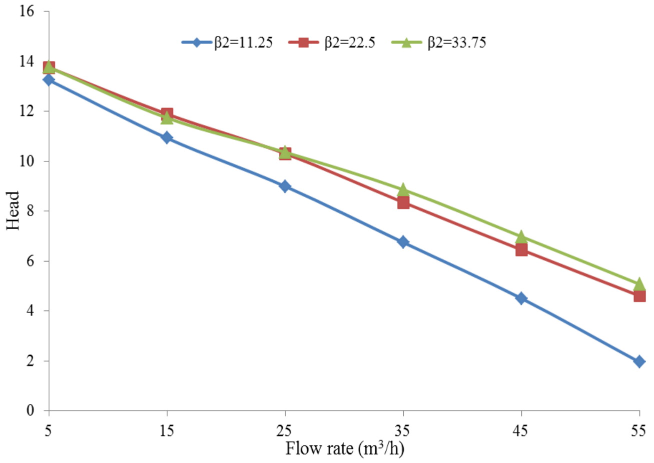


Fig. 8 Effect of outlet angle on head delivered

Head delivered and pump efficiency decreases with increasing impeller exit angle while power consumed by pump increases due to increase of hydraulic losses impeller and volute.

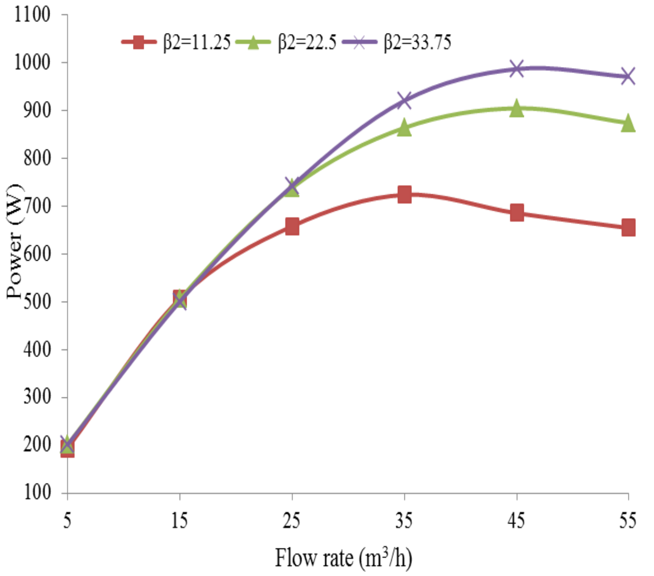


Fig.9 Dependence of pumping power on flow rate for different outlet angle

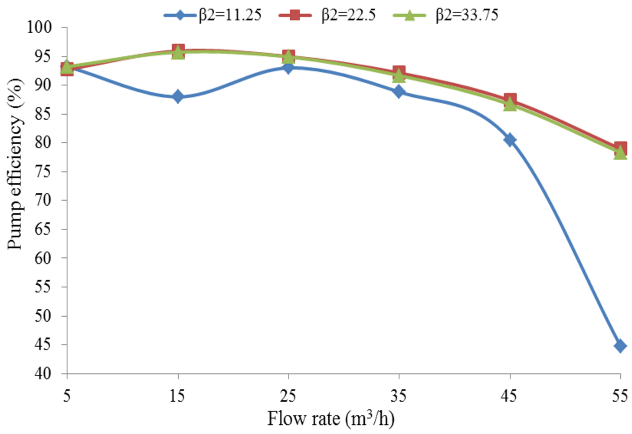


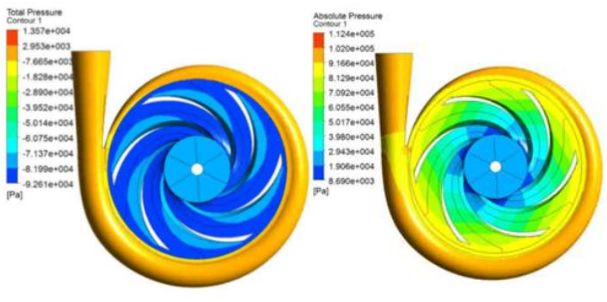
Fig.10 Effect of outlet angle on pump efficiency

c. Pressure and velocity distribution contours

i. Pressure distribution contour

Figure 11 shows that both absolute and total pressures increases from impeller inlet to impeller exit, because the kinetic energy of the impeller is converted to pressure energy inside the pump volute. Pressure inside the volute is increasing in the direction of impeller rotation to volute exit. Absolute pressure has a maximum value of 1.124*105 Pa.

Figure 11 (a) shows the variation of total pressure from impeller inlet to volute outlet. Both absolute pressure and total pressure increase from impeller inlet to exit, this is due to power transferred from the impeller to the fluid inside the pump.

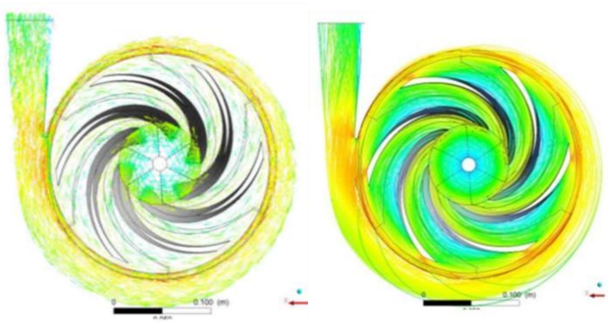


(a) (b)

Fig. 11 Pressure distribution in centrifugal pumps a) total pressure b) absolute pressure

ii. Stream wise velocity vector

Streamline and vector line distribution of velocities of centrifugal pump is shown in figure 12 a) and b). Both figures clearly show that there is no back flow of water inside impeller and volute of centrifugal pump.



(a) (b)

Fig.12 (a) Velocity vector and (b) stream line velocity

V. CONCLUSION

Performance simulation of a centrifugal pump was obtained and analysis of the result had been made. Pressure distribution inside the pump impeller and volute shows the conversion of kinetic energy of the impeller is converted to the pressure energy which lifts water to the required head. From the result obtained it can be concluded that too many and too few blade number increases the losses. Outlet angle of the impeller also have a significant effect on pump performance, too small and too large outlet angle also increases the hydraulic losses and hence pump efficiency decreases. From fixed design variables optimum performance of centrifugal pumps is obtained at a flow rate of $25m^3/h$, rotational speed of 1125 RPM and blade number of 6. At this optimum performance pump lift water to a head of 10.3068m consuming 740W power at, total to total efficiency of 95.0031%.

LIST OF SYMBOLS

U_i	Mean velocity components
G	Production term in k-and ε- equations
c_1, c_2, c_μ	Coefficients in the k- and ε-equations\
k	Turbulent kinetic energy
t	Time
x_i	Cartesian coordinates
Ω_i	Rotation vector
ε	Turbulence dissipation rate
ε_{ijk}	Cross-product tensor
v_t	Kinematic viscosity
$\sigma_k, \sigma_\varepsilon$	Prandtl/Smith number for k- and ε-equations

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