

CFD ANALYSIS OF CENTRIFUGAL PUMP IMPELLER HAVING DIFFERENT EXIT BLADE WIDTH, EXIT DIAMETER AND TRAILING EDGE BLADE ANGLE TO ENHANCE PERFORMANCE

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Abstract- A computational fluid dynamics (CFD) three dimensional simulation is done to optimize the pressure head, efficiency and power required of centrifugal pump by changing in impeller exit blade width, exit diameter, trailing edge blade angle and at different RPM.

In this study initially the geometry of centrifugal pump impeller is created in Ansys blade gen design modeler and further mesh in Ansys turbo grid meshing tool and finally CFD analysis done in Ansys CFX.

From the analysis it is found that the performance of centrifugal pump changes significantly by changing the parameters discussed above. Following are the effects discussed below

- As the blade width at exit of the impeller increases the performance of centrifugal pump increases.
- As the exit diameter of the impeller decreases the performance of centrifugal pump increases.
- As the trailing edge blade angle of the impeller increases the performance of centrifugal pump increases.

The results obtained from all these changing parameters a final optimized model is made and through which it is found that its increases the performance of centrifugal pump significantly

Keywords: Computational Fluid Dynamic (CFD), Bladegen Design Modeler, Ansys, Blade width, Trailing edge blade angle (β_2).

1. INTRODUCTION

Centrifugal pumps are the most common type of pump used in industry, agriculture, municipal (water and wastewater plants), power generation plants, petroleum and many other industries.

Centrifugal pumps are the primary pump type in the class of pumps called "kinetic" pumps and are distinctly different than "positive displacement" pumps shown in Fig1.1 All centrifugal pumps include a shaft-

driven impeller that rotates (usually at 1750 or 3500 RPM) inside a casing. Liquid flows into the suction port (inlet) of the casing and is thrown to the outside of the casing and then exits the discharge port. The velocity imparted to the liquid by the impeller is converted to pressure energy or "head".

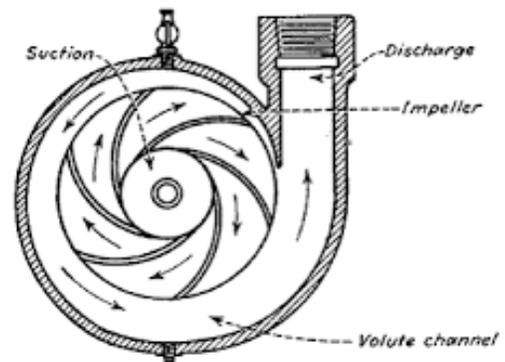


Fig1.1: Centrifugal Pump.

Centrifugal pumps are unique because they can provide high or very high flow rates (much higher than most positive displacement pumps) and because their flow rate varies considerably with changes in the Total Dynamic Head (TDH) of the particular piping system. This allows the flow rate from centrifugal pumps to be "throttled" considerably with a simple valve placed into the discharge piping, without causing excessive pressure buildup in the piping or requiring a pressure relief valve. Therefore, centrifugal pumps can cover a very wide range of liquid pumping applications.

1.1 MAIN PARTS OF CENTRIFUGAL PUMP

A centrifugal pump is a machine that uses rotation to impart velocity to a liquid, it then converts that velocity into flow.

In that definition it can see that there are two things going on:

First, energy is being imparted to a liquid in the form of velocity. Second, that energy is converted into a steady flow.

This two-part definition mirrors the two-part nature of centrifugal pumps. Centrifugal pumps are made up of two primary components: an impeller and a casing.

(I) The Impeller and Centrifugal Force

The rotating part of the centrifugal pump is called the 'impeller'. The impeller is a rotating disc with curved blades standing out vertically from the face of the disc.

The tips of the blades in the impeller are sometimes covered by another flat disc to give shrouded blades, otherwise the blade tips are left open and the casing of the pump itself forms the solid outer wall of the blade passage. The advantage of the shrouded blade is that flow is prevented from leaking across the blade tips from one passage to another.

The impeller is mounted on a shaft connected to the shaft of an electric motor. As the impeller rotates, the fluid that is drawn into the blade passages at the impeller inlet is accelerated as it is forced radially outwards. In this way, the static pressure of fluid is raised.

(II) Casing

It is an air-tight passage surrounding the impeller which converts the K.E of water leaves the impeller into pressure energy before the water leaves the casing and enters the delivery pipe. The three commonly used casings are

(a) Volute casing:- Casing that surrounds the impeller, is of spiral type in which flow area increases gradually. The increase in area of flow, decrease the velocity of flow and thus increase the pressure of water. The efficiency of centrifugal pump having this casing is reduced due to the formation of eddies.

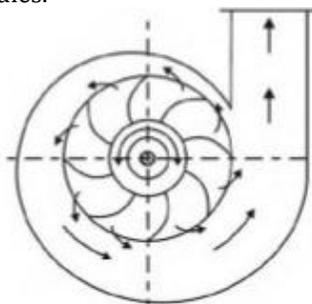


Fig1.2: Volute Casing

(b) Vortex casing:- If a circular chamber is introduced between the casing and the impeller, then that casing is known as vortex casing. This considerably reduces the loss of energy due to the formation of eddies. Thus, the efficiency of the pump is more than the efficiency of volute casing centrifugal pump.

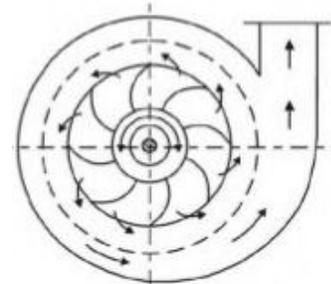


Fig 1.3:Vortex Casing

2. WORKING PRINCIPLE OF CENTRIFUGAL PUMP

As the impeller rotates, the fluid is drawn into the blade passage at the impeller eye, the center of the impeller. The inlet pipe is axial and therefore fluid enters the impeller with very little whirl or tangential component of velocity and flows outwards in the direction of the blades. Fig 2.1 shows the velocity triangles at the inlet and outlet tips of the vane fixed to an impeller. The blades are curved between the inlet radius r_1 and the outlet radius r_2 . β_1 is the angle subtended by the blade at the inlet, measured from the tangent to the inlet radius, while β_2 is the blade angle measured from the tangent at the outlet.

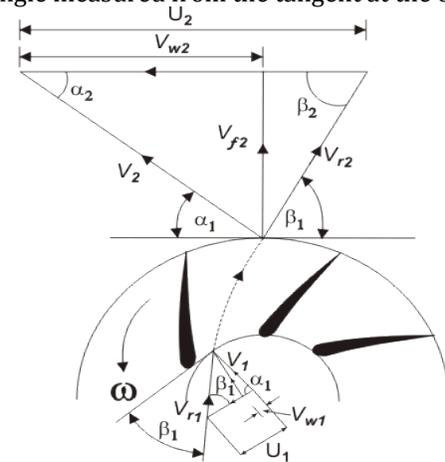


Fig2.1Velocity triangles for centrifugal pump Impeller

The fluid enters the blade passage with an absolute velocity V_1 and at an angle α_1 to the impeller inlet the tangential velocity vector $U_1 = \omega r_1$, where ω is the angular velocity of the impeller. The resultant relative velocity of the flow into the blade passage is V_{r1} at an angle β_1 tangent to the inlet. Similarly, at outlet the relative velocity vector is V_{r2} at an angle tangent to the blade. The absolute velocity vector V_2 is obtained by adding the impeller outlet tangential velocity vector U_2 and relative velocity vector

V_{w2} and V_2 is set at an angle α_2 from the tangent to the blade.

From Euler's pump equation the work done per second on the fluid per unit weight of fluid flowing is

$$\text{Work done on the fluid per unit weight} = \frac{(V_{w2}U_2 - V_{w1}U_1)}{g} \quad (1)$$

A centrifugal pump rarely has any sort of guide vanes at inlet. The fluid therefore approaches the impeller without appreciable whirl and so the inlet angle of the blades is designed to produce a right-angled velocity triangle at inlet (as shown in Fig. 2.1). At conditions other than those for which the impeller was designed, the direction of relative velocity V_r does not coincide with that of a blade. Consequently, the fluid changes direction abruptly on entering the impeller. In addition, the eddies give rise to some back flow into the inlet pipe, thus causing fluid to have some whirl before entering the impeller. However, considering the operation under design conditions, the inlet whirl velocity and accordingly the inlet angular momentum of the fluid entering the impeller is set to zero. Therefore, Eq. (1) can be written as

$$\text{Work done on the fluid per unit weight} = \frac{V_{w2}U_2}{g} \quad (2)$$

3. OBJECTIVE OF STUDY

As discussed in above Centrifugal pumps are extensively used in the oil and gas industries, and the impeller design parameters have Significant effect on the pump performance.

So in this study a impeller of centrifugal pump has been taken and by changing certain parameters of the geometry of impeller the performance of centrifugal pump is being studied.

Firstly the impeller model from base paper has been taken and check out the performance by analyzing the parameters like Head, power imparted by impeller to water and efficiency of pump.

Following are the changes which has been made in impeller model of centrifugal pump

1. Changing the blade width by +-2.5 and +-5 % in both way with reference to the width provided in base paper at the exit of the impeller.
2. Changing the blade outer radius by +-2.5 and +-5 % in both way with reference to the width provided in base paper at the exit of the impeller

3. Changing the trailing edge blade angle (β_2) ranges from 50° to 90° at the increment of 10°.

4.METHODOLOGY

The model of all the cases of impeller has been modeled in a module available in Ansys workbench called BLADEGEN. After completing the model in BladeGen the models are transfer to TURBOGRID module where the meshing of model has been done and after completing the meshing of models that meshed model is transferred to ANSYS CFX module.

CFX-Blade Gen is an advanced, interactive blade design tool for turbo machinery modeling developed by AEA Technology Engineering Software Inc. Blade Gen incorporates years of turbo-machinery design and analysis expertise into an easy to use graphical environment utilizing terminology familiar to engineers involved in the design of rotating machinery. With Blade Gen, the user can re-design existing blades to achieve new design goals or create completely new blade designs from scratch. When either re-designing or evaluating an existing blade design, Blade Gen facilitates the import of blade geometry interactively or though user supplied files.

4.1: Physical models.

In this study the blade of impeller has been designed by using the bladegen module the dimensions of the blade of impeller has been taken from base paper, below table showing the dimensions of the impeller blade and figure showing the Meridional configuration for impeller.

Features of impeller	
PARAMETER	DIMENSION
Shaft diameter, D_s	40 mm
Eye diameter, D_o	182 mm
Hub diameter, D_h	55 mm
Inlet diameter, D_1	160 mm
Inlet blade width, b_1	54 mm
Outlet blade width, b_2	30 mm
Inlet blade angle, b_1	23
Outlet blade angle, b_2	70
Blade number, z	7
Blade Outlet diameter, D_2	365 mm
Shroud diameter	500 mm
Blade thickness	5 mm

Table 4.1 Dimension of impeller

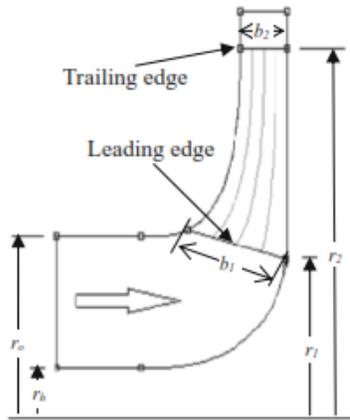


Fig 4.1 Meridional configuration for impeller

The dimensions above in the table is taken from [1] and the performance of pump is being analyzed by parameters like Head (H), Power imparted by impeller in water (P) and efficiency (η).

The performance of impeller has been analyzed by changing some parameter of impeller blade and then a final optimized model has been made by taking the best results obtained from each different cases considered.

Below table shows the different cases of impeller blade of centrifugal pump which has been studied.

Case 1	mm	mm	mm	mm	mm
Blade width at exit	28.5	29.25	30.00	30.75	31.5

Case 2	mm	mm	mm	mm	mm
Outer blade diameter	450	475	500	525	550

Case 3	degree	degree	degree	degree	degree
Trailing edge Beta angle	50	60	70	80	90

The tables above showing the modification which has been done in the existing blade of impeller. In first three cases only single parameter of base model is changed and in case 4 by combining the best results from all the three cases above an optimize model of blade impeller is modeled.

The formulae through which performance of pump has been analyzed are discussed below.

Backward swept ($\beta_2 < 90^\circ$) blades are used in pumps and blowers. The blade angles have influence on the total head and efficiency of pump. As per Euler's equation, head generated by an impeller blade is given by

$$H = \frac{Cu2u2 - cu1u1}{g} \quad (7)$$

Normal entry at impeller inlet implies $C_{u1} = 0$ and from velocity triangle shown in Fig. C_{u2} can be given as:

$$Cu2 = u2 - \frac{Qc \cot \beta_2}{\pi D b_2} \quad (8)$$

From Fig. 1b it is clear that as β_2 increases, the absolute exit

Velocity C_2 also increases. Increase in C_2 causes C_{u2} to increase which results in increase in head.

Input power in terms of cu_2 is given by

$$P = \frac{\rho g Q u_2 C_{u2}}{g} \quad (9)$$

The Eqs. (8) and (9) indicate that the increase in exit angle results in increase in C_{u2} which increases the input power.

Hydraulic efficiency is given by

$$\eta = \frac{Ha}{Hth} = \frac{Hth - \Delta H}{Hth} \quad (10)$$

The hydraulic losses and theoretical head (H_{th}) at design point differ from the off-design conditions. The head and the corresponding efficiency are higher for a larger exit angle.

4.2 Material properties

For analyzing the performance of centrifugal pump in this research the water is used in fluid domain. The fluid properties of water has been given in table below from base paper.

No.	PROPERTIES	VALUE
1	viscosity (μ) N s/m ²	1.002e ⁻³
2	Specific heat capacity (Cp) j/kg k	4181.7
3	Density (ρ) kg/m ³	997
4	Thermal conductivity (k) W/mk	0.6069

4.3 Boundary condition

Here in this research fluid-fluid interface is used to analyze the performance of impeller pump. Water has been used as the working fluid. The direction of the flow was defined normal to the boundary. Inlet velocity and

temperature were specified. Outflow boundary condition was used at pressure-outlet boundary.

Pressure at inlet section of water is considered as 101353 Pa with inlet temperature of 288.15K and at the outlet of the domain the mass flow rate has been changed ranges from 100-140 kg/s with the increment of 10 kg/s.

1	Inlet condition	101353 Pa
2	Outlet condition for inlet fluid mass flow rate (Kg/s)	100-140 kg/s with the increment of 10 kg/s.
3	Outlet condition for inlet fluid Discharge (Q_0m^3/s)	0.1-0.14
4	Inlet temperature (T_{in})	288.15 K
5	RPM of impeller (rpm)	1470

4.4 Image of model and meshing model

The model of all the cases of impeller has been modelled in a module available in Ansys workbench called BLADEGEN. After completing the model in BladeGen the models are transfer to TURBOGRID module where the meshing of model has been done and after completing the meshing of models that meshed model is transferred to ANSYS CFX module.

The schematic view is shown in the Fig 4.2, Fig 4.3 and Fig 4.4

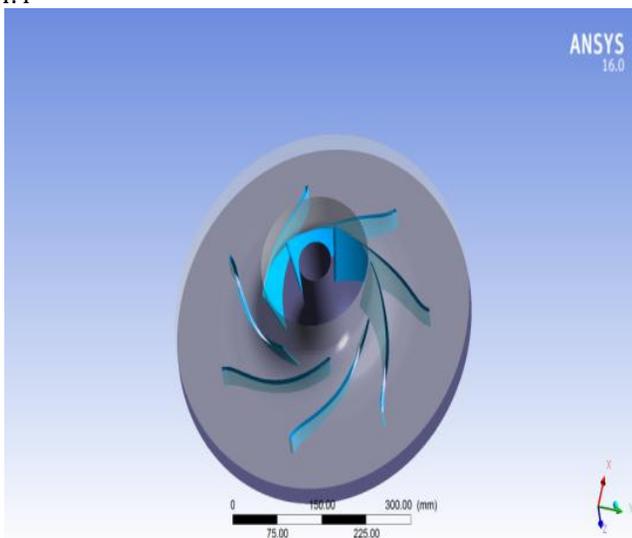


Fig 4.2: Complete model of centrifugal impeller.

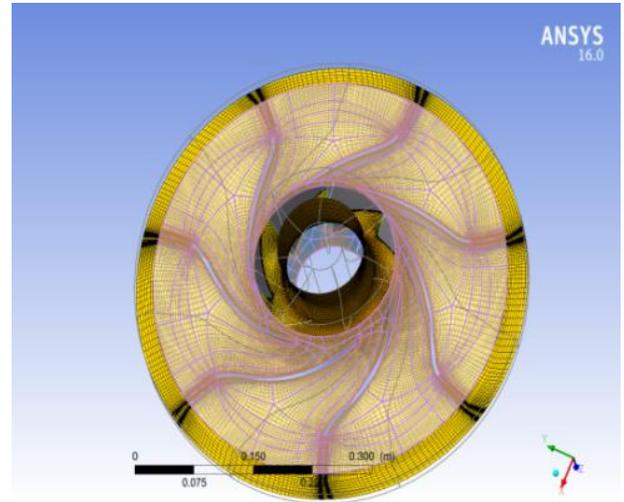


Fig 4.3: The meshing model of impeller.

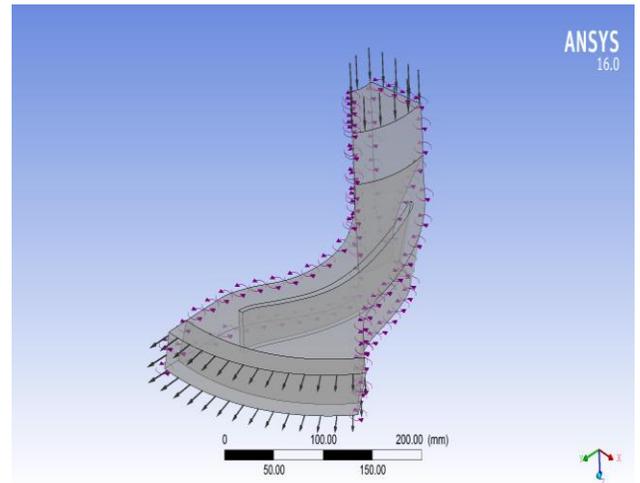


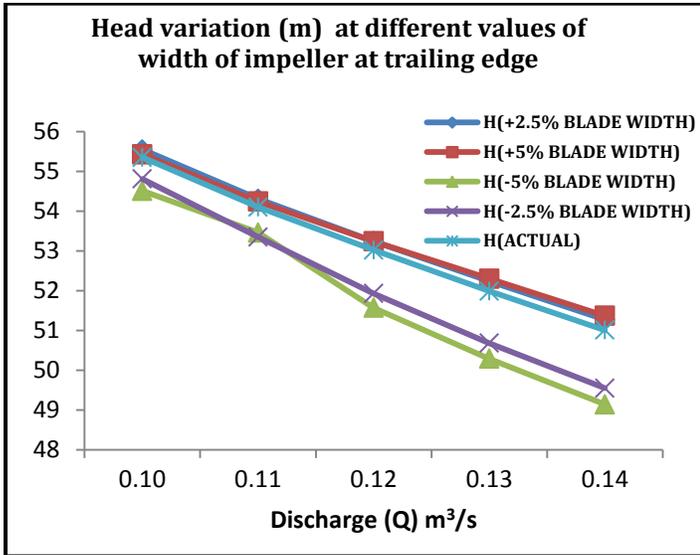
Fig 4.4: The model of impeller with applying boundary conditions.

6. RESULTS AND DISCUSSION

In this section the performance of all the cases have been discussed by parameters such as Head, power imparted by impeller to water and efficiency.

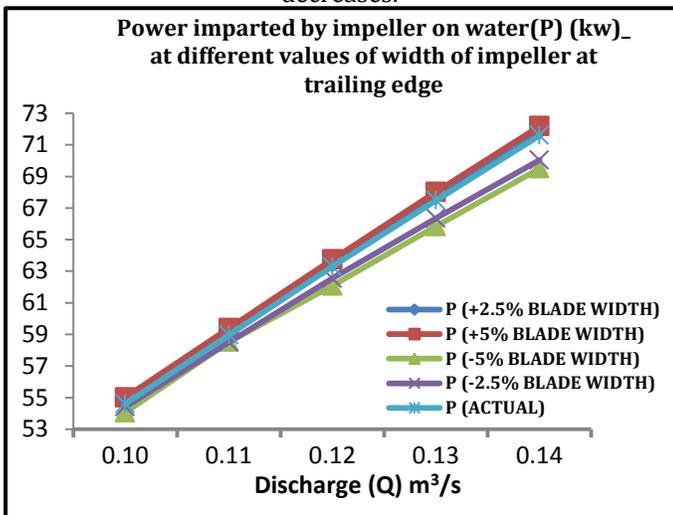
6.1Variation of Head, power and efficiency at different values of width at exit of impeller

In this section the variation of head, power and efficiency had been studied with respect to different values of discharge by changing the blade width at trailing edge of blade (variation of -5%,- 2.5%,+2.5% and +5% from actual width is considered)



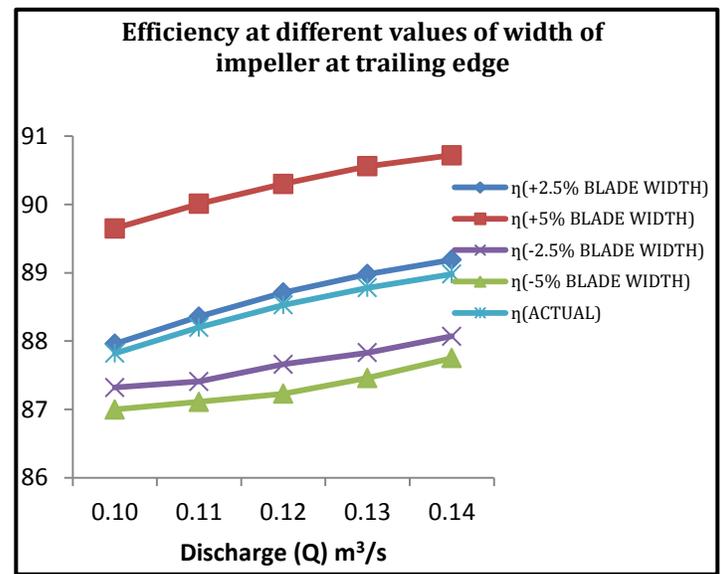
Graph 6.1 head variation at different blade width

From graph 6.1 it is found that as the blade width of impeller increases the Head also increases for every values of discharge and it is also found that after increasing the value of width up to certain level the heads start to remain constant and it is also found that as the value of discharge increases on any value of width of impeller the Head decreases.



Graph 6.2 Power variation at different blade width

From graph 6.2 it is seen that as the blade width of impeller increases the Power imparted by impeller on water (P) also increases for every values of discharge and it is also found that as the value of discharge increases on any value of width of impeller the power increases.

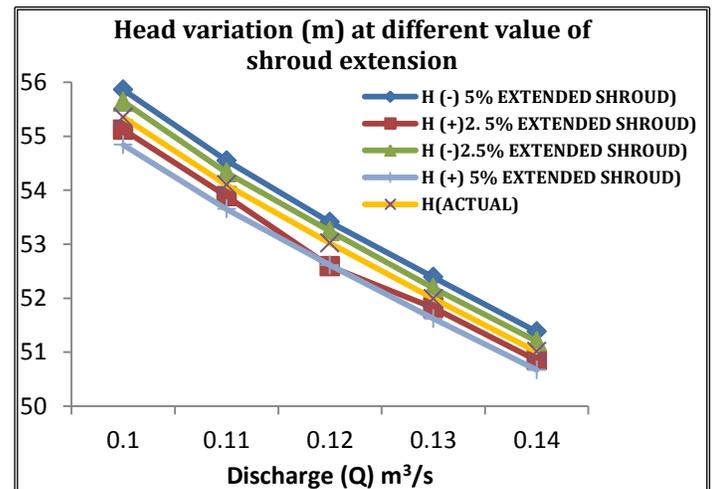


Graph 6.3 Efficiency variation at different blade width.

From above graph and table it is seen that as the width of impeller increases the efficiency of impeller pump also increases for every values of discharge and it is also found that as the value of discharge increases on any value of RPM of impeller the efficiency increases.

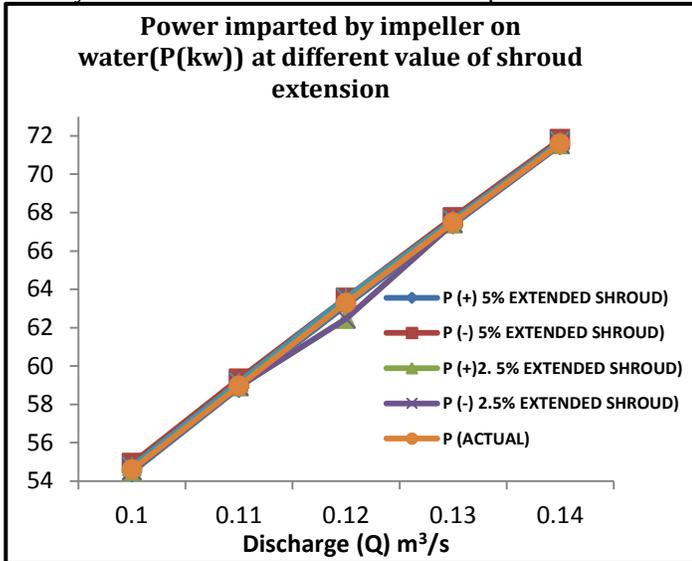
6.2: Effect of increasing and decreasing the value of shroud extension of impeller on Head, Power and Efficiency

In this section the variation of head, power and efficiency had been studied with respect to different values of shroud extension of impeller (variation of -5%, - 2.5%, +2.5% and 5% from actual shroud is considered) here also discharge is considered as constant parameter.



Graph 6.4 Head variation at different value of shroud extension

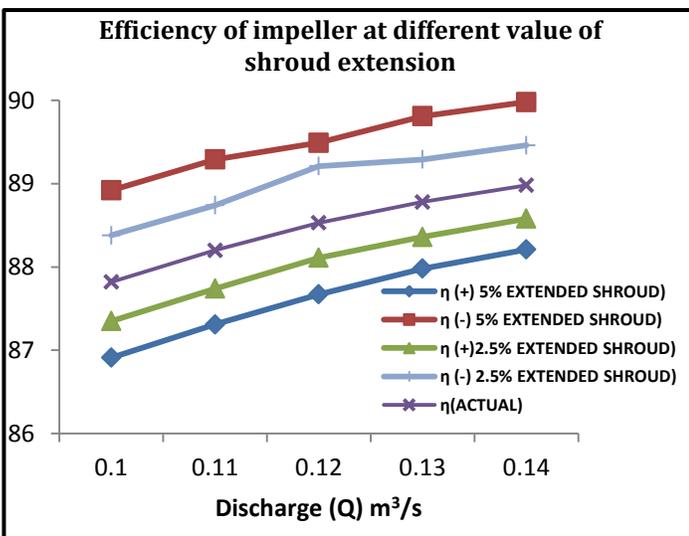
Graph 6.4 shows that as the shroud extension of impeller decreases the Head increases for every values of discharge, and it is also found that as the value of discharge increases on any value of shroud extension of impeller the Head



decreases.

Graph 6.5 Power variation at different value of shroud extension

From above graph 6.5 shows that as the extension of shroud of impeller decreases the Power imparted by impeller on water (P) increases for every values of discharge and it is also found that as the value of discharge increases on any value of shroud extension of impeller the



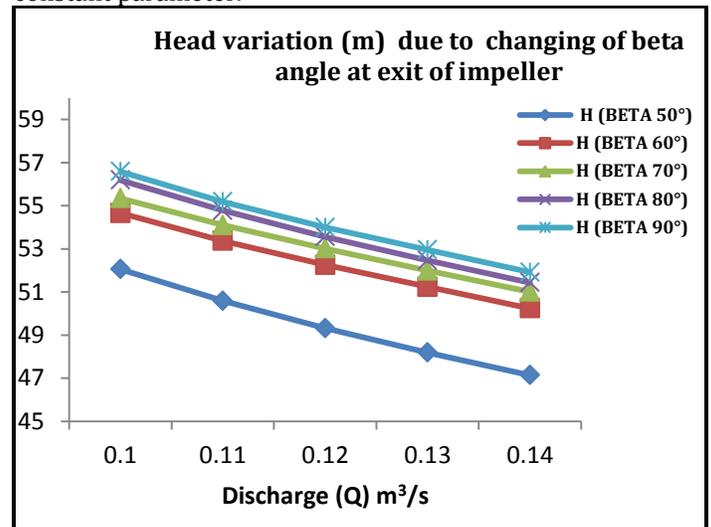
power also increases.

Graph 6.6 Efficiency variation at different value of shroud extension

Graph 6.6 shows that as the shroud extension of impeller decreases the efficiency of impeller pump increases for every values of discharge and it is also found that as the value of discharge increases on any value of shroud extension of impeller the efficiency increases

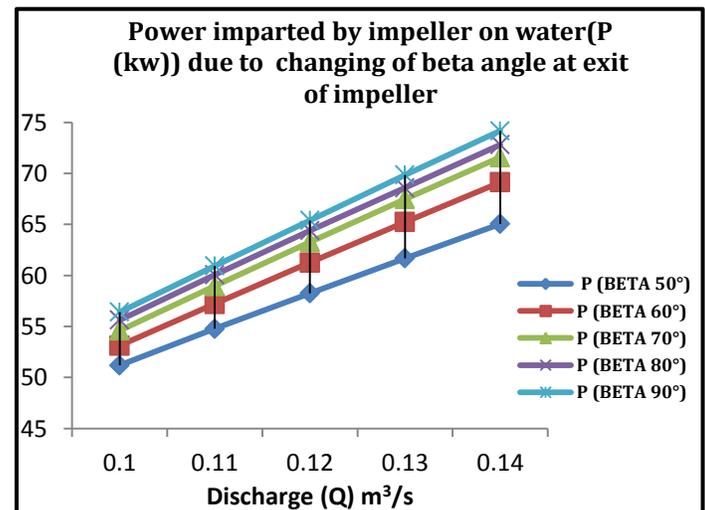
6.3: Effect of changing beta angle at exit of impeller on head, power and efficiency.

In this section the variation of head, power and efficiency had been studied with respect to different values of beta angle at exit of the impeller. (beta angle at exit is taken as 50°, 60°, 70°, 80°, 90°) here also discharge is considered as constant parameter.



Graph 6.7 Head variation at different value of beta angle at exit

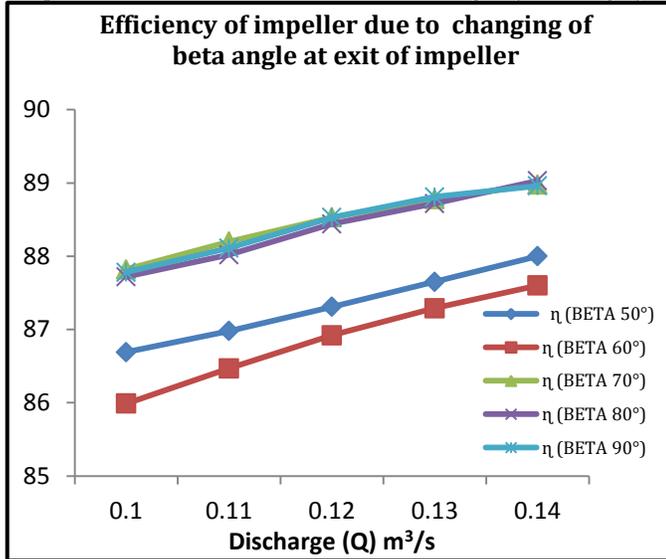
From graph 6.7 it is found that as the beta angle at exit of impeller increases, the Head increases for every values of discharge, and it is also found that as the value of discharge



increases on any value of exit beta angle of impeller the Head decreases.

Graph 6.8 Power variation at different value of beta angle at exit

Graph 6.8 shows that as the exit blade angle (beta angle) of



impeller increases the Power imparted by impeller on water (P) increases for every values of discharge and it is also found that as the value of discharge increases on any value of impeller exit angle the power also increases.

Graph 6.9 Efficiency variation at different value of beta angle at exit

Graph 6.9 shows that as the exit angle (beta angle) of impeller increases the efficiency of impeller pump increases for every values of discharge, it is also noticed that after 70 degree beta angle at exit the efficiency of impeller is almost same and it is also found that as the value of discharge increases on any value of exit angle of impeller the efficiency increases.

7. CONCLUSION

Followings are the conclusions which have been achieved from all the above cases.

- As the value of blade width at exit of the impeller is increases the performance of pump impeller increase in all the parameters discussed above.
- As the extension of shroud of the impeller decreases the performance of pump impeller increases in all the parameters.

- If the Blade angle (beta) at the trailing edge of the impeller blade increases the performance of pump impeller increases.

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