# SIMULATION OF OPTIMUM LOSS MODELS FOR RADIAL CENTRIFUGAL PUMP

Thesis submitted in partial fulfilment of the requirements for the award of degree of

**Master of Engineering**
in
**CAD/CAM & ROBOTICS**

**Thapar University, Patiala**

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# CERTIFICATE

I hereby certify that the work which is being presented in the thesis entitled, "**SIMULATION OF OPTIMUM LOSS MODELS FOR RADIAL CENTRIFUGAL PUMP** ", in partial fulfilment of the requirements for the award of degree of Master of Engineering in Mechanical Engineering with specialization in **CAD/CAM & ROBOTICS** submitted in Mechanical Engineering Department of Thapar University, Patiala, is an authentic record of my own work carried out under the supervision of **Mr. Satish Kumar** and refers other researcher's works which are duly listed in the reference section.

The matter presented in this thesis has not been submitted for the award of any other degree of this or any other university.

**Sushil Mittal**

This is to certify that the above statement made by the candidate is correct and true to the best of my knowledge.

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# ACKNOWLEDGMENT

Working without proper guidance and expecting success is just like making castles in the air, so whenever one wants to start any work, he requires guidance from experts. I express my sincere gratitude to my guide ,Mr. Satish Kumar, lecturer , Department of Mechanical Engineering, Thapar University**,** for acting as supervisor and giving valuable guidance during the course of this investigation, for his ever encouraging and timely moral support. His enormous knowledge and intelligence always helped me unconditionally to solve various problems.

I am greatly thankful to Dr. S. K. Mohapatra, Professor and Head, Mechanical Engineering Department, Thapar University for his encouragement and inspiration for execution of the thesis work. I do not find enough words with which I can express my feeling of thanks to entire faculty and staff of Department of Mechanical Engineering, Thapar University, Patiala for their help, inspiration and moral support, which went a long way in successfully completion of my thesis.

**Sushil Mittal**

# ABSTRACT


Conventional design methods of centrifugal pump are largely based on the application of empirical and semi-empirical rules along with the use of available information in the form of different types of charts and graphs in the existing literature. The program developed in this present work is the best suitable for low specific speed radial centrifugal pump. Same program is also suitable for the design of high specific speed and multistage centrifugal pump with few modifications.

As the design of centrifugal pump involve a large number of interdependent variables, several other alternative designs are possible for same duty. Hence theoretical investigation supported by accurate experimental studies of the flow through the pump.

In conventional method for calculating the pump efficiency is the function of specific speed, which is available in form of graph, empirical correlation in various text books and references .But in practical, efficiency has direct influence due to change of flow pattern, Renoldnumber, relative eddies in the impeller blade passage.

Present work is aimed to calculate optimum set of loss models for low specific speed radial centrifugal pump .These optimum set of loss models are directly correlated with the geometrical and hydraulic parameters of centrifugal pump. These allow the study of the variation of performance with geometry. For the simulation computer program code has been developed which permits wide range of variables to be investigated.

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# NOMENCLATURE

| SYMBOLS | STANDS FOR |
|---|---|
| a | Constant used in determination of the shaft power |
| $A_{th}$ | Throat area at volute in $m^2$ |
| $B_1$ | Impeller width at inlet in m |
| B2 | Impeller width at outlet in m |
| $B_3$ | Inlet width of volute in m |
| Cp | Specific heat |
| D | Diameter in m |
| $D_e$ | Impeller eye diameter in m |
| $D_h$ | Hub diameter in m |
| $D_{sh}$ | Shaft diameter in m |
| dp/r | Mean blade loading |
| f | Weighting factor |
| $f_s$ | Shear stress $N/m^2$ |
| g | Gravitational acceleration in $m/s^2$ |
| H | Head in m |
| $h_1$ | Suction head loss in m |
| $h_2$ | Incidence loss in m |
| $h_3$ | Blade loading loss in m |
| $h_5$ | Mixing loss in m |

| | |
|---|---|
| $h_6$ | Disk friction loss in m |
| $h_7$ | Recirculation loss in m |
| $h_9$ | Internal loss in m |
| $h_L$ | Total loss in head in m |
| $H_d$ | Depression head in m |
| $K_m$ | Capacity constant |
| $N_s$ | Specific speed of pump in rpm |
| N | Speed in rpm |
| $P_{in}$ | Input power in HP |
| $P_{sh}$ | Shaft power in HP |
| Z | Number of blades |
| r | Radius in m |
| $R_t$ | Tongue radius in m |
| $R_e$ | Reynolds number |
| t | Blade thickness in m |
| U | Peripheral velocity in m/s |
| $U_e$ | Peripheral velocity at the eye diameter |
| $V_r$ | Relative velocity in m/s |
| Q | Flow rate in m3/s |
| A | Angle at which the water leaves the impeller |
| β | Blade angle |
| η | Overall efficiency |
| $\theta_A$ | Maximum total angle between the side |

| | |
|---|---|
| | of the volute |
| $\nu$ | Kinematic viscosity $m^2/s$ |
| $\sigma$ | Slip factor |
| $\Psi$ | Head coefficient |
| $\Omega$ | Angular velocity in rad/s |
| $\theta$ | Volute angle |
| $\theta_t$ | Tongue angle |
| $\tau$ | Permissible stress $N/m^2$ |
| $\Phi$ | Flow coefficient |
| $\rho$ | Density of the fluid |
| $g$ | Acceleration due to gravity |

**Subscripts:**

| | |
|---|---|
| 0 | Eye of the Impeller |
| 1 | Inlet to the Impeller |
| 2 | Outlet of the Impeller |
| 3 | Inlet to the Volute |

# CHAPTER 1
# INTRODUCTION

## 1.1 INTRODUCTION

A pump is one such device that expends energy to raise, transport, or compress liquids**.** Pumps are used in a wide range of industrial and residential applications. Pumping equipment is extremely diverse, varying in type, size, and materials of construction. There have been significant new developments in the area of pumping equipment since the early 1980s.

Large amount of slurry is pumped in industries. The application, which involves the largest quantities, is the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines and winning material for landfill and construction purposes. Dredging is one of the most common and ancient processes involving slurry flows; the dredged materials contain a wide range of particles, tree debris, rocks, etc. Mining has employed the concept of slurry flows in pipelines since the mid-nineteenth century, when the technique was used to reclaim gold from placers in California.

## 1.2 PUMP

Pump is a mechanical device, which is used to increase the pressure of a liquid. It is also used for rising fluid from a lower level to a higher level.

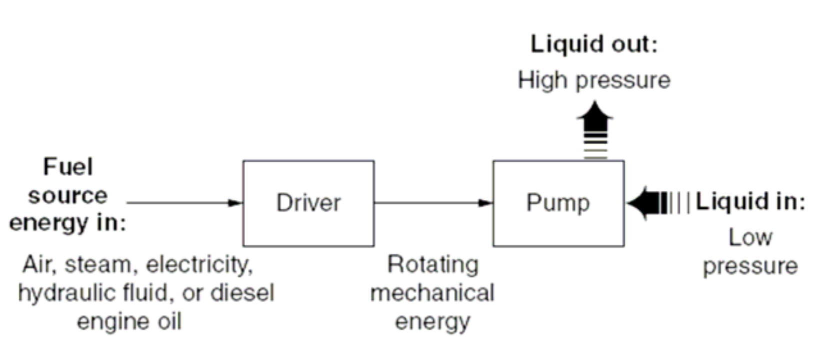



Fig1.1Working principle of Pump

### 1.2.1 CLASSIFICATION OF PUMP:

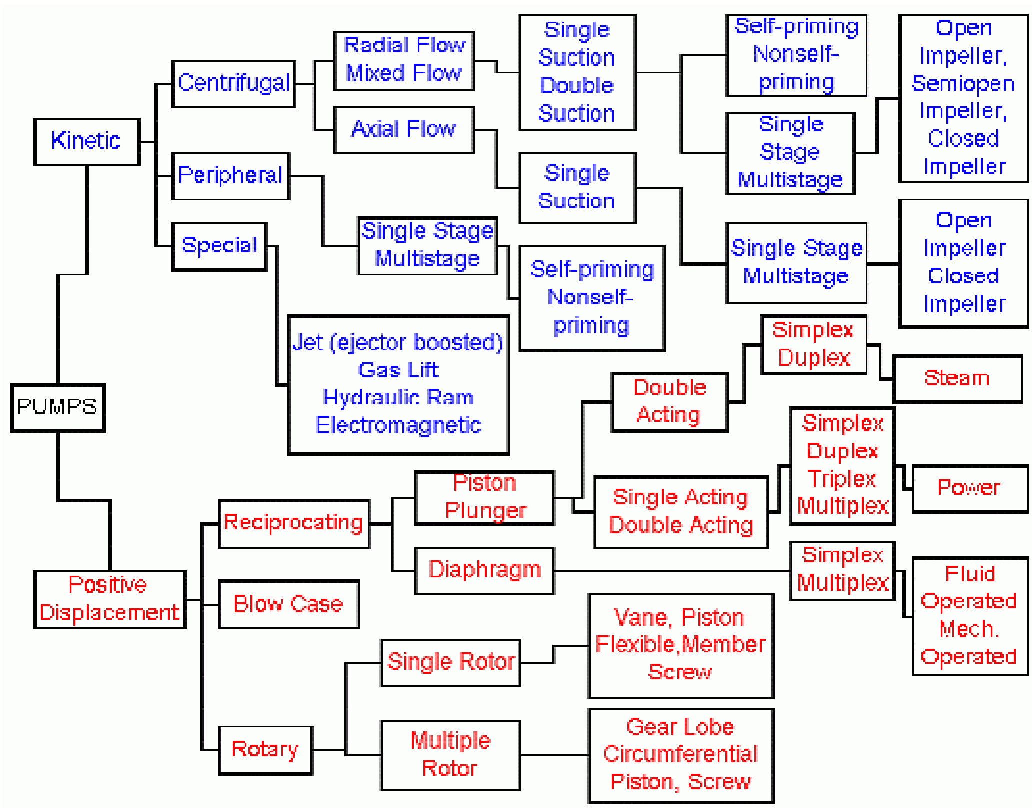



Fig 1.2 Classification of Pump

#### 1.2.1.1 RECIPROCATING PUMP

In reciprocating pumps the mechanical energy is converted into hydraulic energy by sucking the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards) which exerts the thrust on the liquid and increases its hydraulic energy(pressure energy), the pump is known as reciprocating pump. Reciprocating pumps are used where a precise amount of liquid is required to be delivered, also where the delivery pressure required is higher than that can be achieved with other types. Figure 1.2 shows line diagram of reciprocating pump.

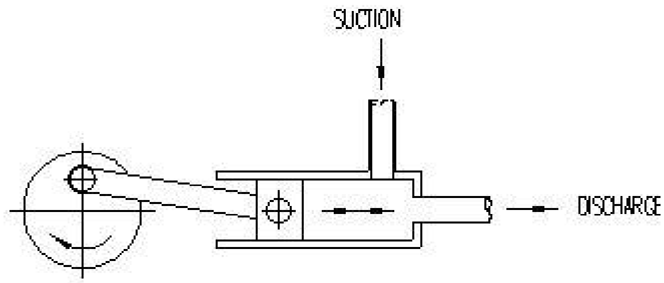



Figure 1.3 Reciprocating pump

#### 1.2.1.2 ROTARY PUMP

Rotary pump is used to move heavy or very viscous fluids. These employ mechanical means such as gear, cam and screw to move the liquid

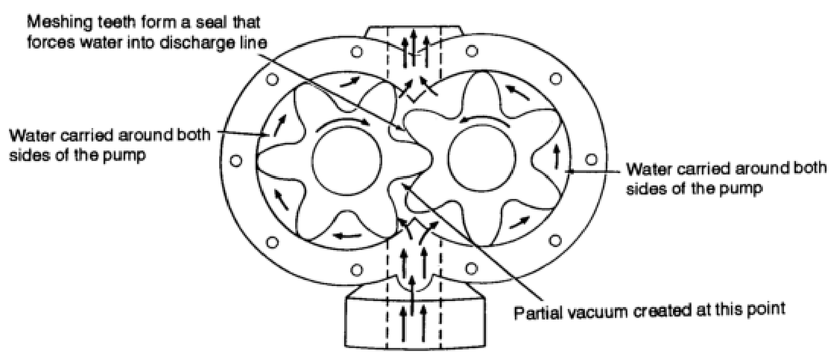



Figure 1.4 Sectional view of rotary pump

## 1.3 CENTRIFUGAL PUMPS

It is the Rotodynamic machine. By rotating action develop the pressure able to lifting of liquid lower level to higher level. Centrifugal pump is explained with the following headings:

### 1.3.1 WORKING PRINCIPLE OF CENTRIFUGAL PUMP

Centrifugal pumps works on the basis of 2$^{nd}$ law of Newton. Due to the rotation of the runner, called impeller the fluid at the inner radius moves to the outer radius & gain the Centrifugal head. Suction is created at the inlet to the pump which is called the eye. Continuous lifting of fluid thus takes place from sump to the pump while passing through the impeller the fluid take the energy from vane sin pressure & kinetic energy. A large amount of impeller outlet therefore made to convert the kinetic energy of fluid into pressure energy before the fluid enters the developing pipe.

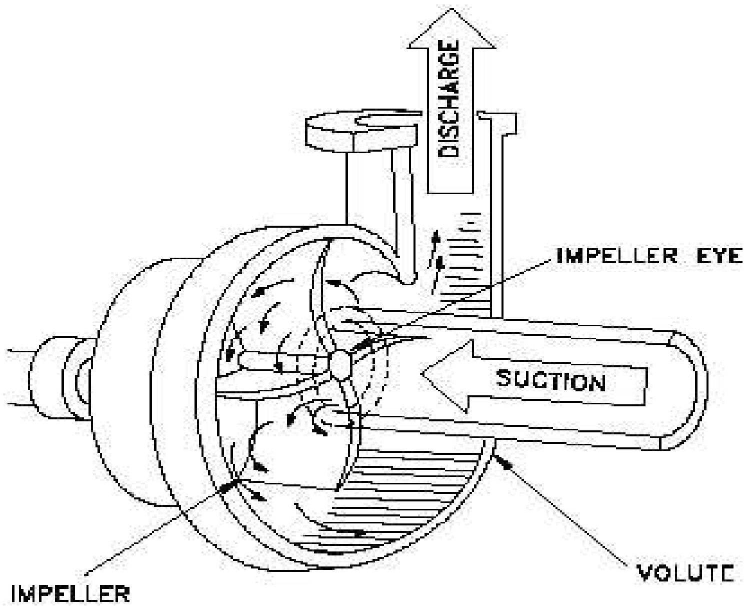



Fig. 1.5 working principle of Centrifugal pump

### 1.3.2 CLASSIFICATION OF CENTRIFUGAL PUMPS :

#### 1.3.2.1 WORKING HEAD:

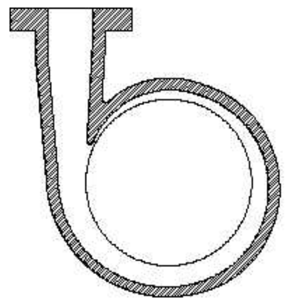
a) Low lift Centrifugal pumps: Impeller is surrounded by volute & there are no guide vanes.

b) Medium lift: They are generally provided with guide vanes.

c) High lift: They are generally multistage pumps because single stage can not easily build up such a high pressure.

#### 1.3.2.2 TYPE OF CASING:

a) Volute casing
b) Diffusion pump
c) Volute with vortex chamber



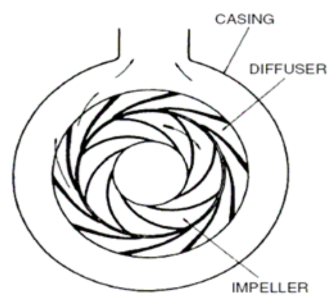



Fig 1.6(a) Volute casing Fig. 1.6(b) Diffuser casing

### 1.3.2.3 NUMBER OF STAGES:

a) Single stage pump
b) Multistage pump

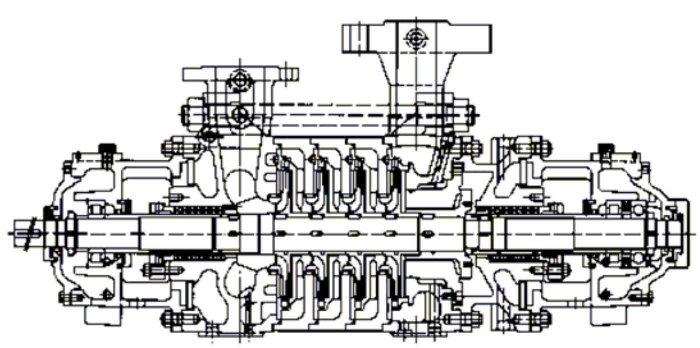


Fig 1.7Multistage Pump

### 1.3.2.4 RELATIVE DIRECTION OF FLOW THROUGH IMPELLER:

**a) Radial flow pump:**

Most centrifugal pumps are of radial flow. Radial flow impellers impart energy primarily by centrifugal force. Water enters the hub and flows radically to the periphery. Flow leaves the impeller at 90 degree angle from the direction it enters the pump.

**b) Mixed flow pump:**

Mixed flow impellers impart energy partially by centrifugal force and partially as an axial pump. This type of pump has a single inlet impeller with flow entering axially and discharging in an axial and radial direction. Mixed flow impellers are suitable for pumping untreated waste water. They operate at high speeds than the radial flow impeller pumps; are usually of lighter construction; and where applicable, cost less than other pumps. Impeller may be either open or enclosed, but enclosed is preferred.

**c) Axial flow pumps:**

Axial flow impeller imparts energy to the water by acting as axial flow pump. The axial flow pump has a single inlet impeller with flow entering and exiting along the axis of rotation (along the pump drive shaft). These pumps are used in low head, large capacity applications such as water supplies, irrigation, drainage etc.

### 1.3.2.5 NUMBER OF ENTRANCE TO THE IMPELLER:

a) The Single entry or single suction pump: Water is admitted from a suction pipe on the side of impeller.
b) Double suction pump: Admit water from both sides.

### 1.3.2.6 LIQUID HANDLED:

Depending on the tube & viscosity of the liquid to be pumped, it may have

a) Open impeller
b) Semi Open Impeller
c) Closed impeller

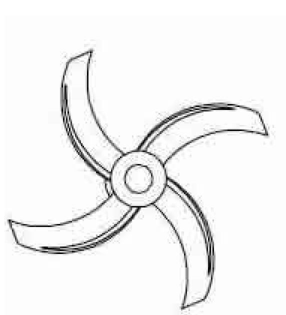


Fig 1.8(a) Open Impeller

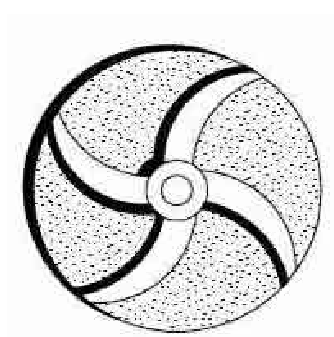


Fig 1.8(b) Semi Open Impeller

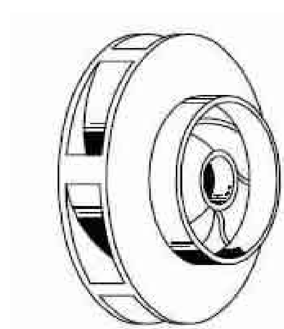


Fig 1-8(c) Closed Impeller

### 1.3.2.7 SPECIFIC SPEED:

a) Slow speed radial flow runner- 10 to 30

b) Normal speed radial flow impeller- 30 to 50

c) High speed radial flow impeller- 50 to 80

d) Mixed flow runner- 80 to 160

e) Axial flow runner- 110 to 150

### 1.3.2.8 DEPENDING ON THE POSITION OF IMPELLER:

a) Vertical impeller shaft pump

b) Horizontal Impeller shaft

Fig 1.9(a) Vertical impeller shaft pump

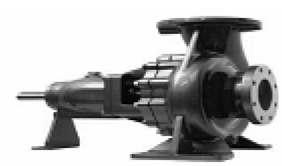


Fig 1.9(b) Horizontal Impeller shaft

# CHAPTER 2

# LITERATURE REVIEW

**V P Vasandhani et al[1] [1975]** conducted an experiment using a volute type radial pump with vertically split casing. They observed in the experiment that by changing the length or angle of the tongue of volute, best efficiency point can be shifted to different values of discharge and that the short tongue and a smaller tongue angle give broader efficiency curves without any change in the best efficiency of the pump. They determined the power-discharge, head discharge and efficiency discharge characteristics by setting tongue of volute at seven different positions.

**Jaroslaw mikielewicz et al[2] [1978]** have proposed a semi empirical method of ideal performance of a centrifugal pump to develop a head loss ratio by examining both single as well as two phase flow. They developed this ratio by dividing loss of head in two phases to loss of head in single phase flow using same values of flow rate and flow coefficient. The techniques used are first, second quadrant operations. In first quadrant, rotation is taken normal and in second quadrant reverse rotation is taken. They found that in both the cases results can be reproduced with acceptable accuracy.

**J.W Crisswell[3] [1982]** objective of this paper was to study the problems encountered in the pumping of slurries over short and long distances using centrifugal slurry pump. He discussed the effects of various parameters like friction loss, impeller speed, N.P.S.H, gland sealing in context of problems associated with selection and operation of slurry pump. He showed that wear is the most important factor related to slurry pump selection.

**J. Remisz et al[4] [1983]** have presented a method of transforming pump characteristics from clear water to slurry performance. This permits calculation of the main dimensions of a new slurry pump and also the prediction of the characteristic shape. It has been stated

that changes to the water characteristic depend basically on the type of solid forming the mixture its grain composition, grain geometry, mixture concentration and density.
**Koji Kikuyama et al[5][1985]** in this study the changes in the centrifugal pump head and the flow pattern were examined experimentally with use of pump impellers whose outlet edges are deformed stepwise by slicing off the blade on the suction or pressure side. Sharpening of blade caused a change in exit flow angle as well as decrease in velocity and an increase in the pump head was brought about. He presented a simple theory to predict the relationship between increase in pump head and the blade edge sharpening.

**W.Mez[6] [1984]** presented the influence of solids concentration, solid density and grain size distribution on the working behaviour of centrifugal pump. In tests 150 mm and 300 mm size pumps with channel type impellers were used. Clean coal, raw coal and gravel were used as slurry having maximum concentration of 40 % by volume and maximum grain size diameter of 125mm.He also compared the results with equations of several authors.

**C I Walker et al[7] [1984]** in this study the change in performance characteristics of centrifugal pumps when handling fine granular or homogeneous type non-Newtonian slurries were examined using two different slurry pumps. Coal/water and kaolin/water was used as slurry. Results showed that the pump performance is dependent on slurry's rheological properties with pump Reynolds number giving generally good correlation with the change in performance.

**K.K Sheth et al[8][1987]** carried out experiments to determine the effect of slip factor of slurry pump due to various parameters .Pumps were operated with three different slurries with different speeds. Euler's equation was used to find the equations of slip & friction factors of the flow. Results showed that slip factors deduced from head flow rate curves were more reliable than those deduced from power flow rate curves.

**Rayan and Shawky[9] [1989]** have evaluated erosion wear in the centrifugal slurry pump at different rotational speeds with different solid-liquid concentration by weighing method. They have reported that erosion wear rate increases with flow velocity as well as solid-liquid concentrations.

**Dong et al[10] [1992]** used PDV technique to visualize the flow inside the volute of a centrifugal pump. Neutrally buoyant particles of 30μm mean diameter were used as seed and it was observed that although most of the blade effects occur near the impeller tip, they are not limited to this region.

**T.Cader et al[11] [1992]** have investigated water and solid water mixture flow at the impeller outlet of a centrifugal slurry pump using LDV (Laser Doppler Velocity meter) system. Solid particle were taken as 0.8 mm diameter glass beads. They observed that solid particles have larger radial velocity than the carrier fluid at the impeller outlet, but they lag the water in the circumferential direction.

**V.K Gahlot et al[12] [1992]**. presented the effect of two different types of slurries namely zinc tailing & coal on the performance characteristics of centrifugal slurry pump. A correlation for predicting the reduction head in pump due to slurry flow was proposed. They observed that the head and the efficiency of the pump decrease with increase in solid concentration, particle size and specific gravity of solids where they are independent of the pump flow rate.

**Cader et al[13]. [1994]** have studied phase velocity distributions and overall performance of a centrifugal slurry pump by using LDA (Laser Doppler Anemometer). Experiment conducted with a dilute suspension of concentration of 1% micron size tracers and 0.8 mm glass beads at the impeller casing flow interface. Fluctuations in angular velocity up to 20%, radial velocity up to 90% and axial velocity up to 200% from their mean velocity components over various impeller angular positions were observed.

**W.Huang et al[14]. [1995]** Investigated that two phase flow structure at the impeller-volute interface by using laser doper velocitimetry (LDV). they observed that in the impeller casing slip velocity, solid liquid velocity fluctuations are the function of radial distance and impeller angle. In the impeller rotation region flow is approximately forced vortex type and in casing region free vortex type.

**S.Yedidiah[15] [1996]** discusses the present state of knowledge of the manner in which the impeller geometry affects the developed head. A comparison with test results shows a very impressive agreement between theory and practice

**S.Yedidiah[16] [1996]** discussed a novel approach for calculating the head developed by a centrifugal impeller. The approach was based on the fact that the head developed by an impeller depends on the shape of the total blade and not just upon the magnitude of its outlet angle. Presented approach was useful in solving many problems encountered with centrifugal pumps.

**Ni Fusheng et al[17] [1999]** have studied the effect of high delivered volumetric concentration on characteristics of a slurry pump. Experiments showed that pump efficiency in the coarse sand slurry service may develop almost 60% compared to that of water service, when delivered volumetric concentration of 42%.

**S. Gopalakrishnan[18] [1999]** have discussed the R & D efforts of past, present and future, in terms of core competencies, which are essential for today's pump manufacturer. These are hydraulics, vibrations and pump designs, which capitalize on improved understanding of the underlying technologies.

**Chung[19] [1999]** has developed optimum design code of the pump. They determined the geometric and fluid dynamic variables under the appropriate design constraints. Optimization problem has formulated with a non-linear objective function to minimize losses, net positive suction head required and product price of a pump stage depending on the weighting factor selected as the design compromise. Optimal solution obtained, efficiency $NPSH_R$ depends design variable of centrifugal pump. Selected in the range of weighting factor 0 to 1.designer can easily find the optimum value of design variable to meet their particular requirement of pump design.

**Sellgren et al[20]. [2000]** showed that the addition of clay to sand slurries has been found to reduce the pipeline friction losses, thus lowering the pumping head and power consumption. Pump water heads and efficiencies are decreased by the presence of solid particles. Experimentally results are presented for a centrifugal pump with an impeller diameter of 0.625 m for three narrowly graded sands with average particle sizes of 0.64, 1.27 and 2.2 mm. Reductions in head and efficiency are lowered by about one third for sand clay mixtures with sand to clay mass ratios between 4:1 and 6:1.

**Gandhi et al.[21] [2001]** have studied erosion wear at various locations inside the volute casing of a centrifugal slurry pump for the flow of solid-liquid mixtures. They reported that the wear increases all along the volute periphery with increase in the amount of solid suspended in the mixture and wear smaller when the pump operates near the (BHP).

**Gandhi et al[22]. [2001]** have studied the performance of two centrifugal slurry pumps for three solids materials having different particle size distribution (PSD) in terms of head, capacity and power characteristics. The results have shown that values of head and efficiency ratios are not only depended on solid concentration but are also affected by PSD of the solids and properties of slurry. They conclude that the head and efficiency of the pump decrease with increase in solid concentration, particle size and slurry viscosity, the decrease in the head being 2-10% higher than that of the efficiency. The presence of finer particles (<18 μm) in coarse slurries substantially attenuate the loss of the performance of the pump in terms of head and efficiency.

**M C Roco[23] [2001]** has identified qualitative aspects of the flow pattern as large scale periodical, two phase flow structures develop in the entire casing and are dominated by stationary works, particles generally lead the fluid in the radial direction and log in circumferential direction the averaged velocity distribution averaged over the casing width determines the flow rate.

**Oh and Kim[24] [2001]** developed a conceptual design optimization code for mixed flow pump to determine the geometric and fluid dynamic variables under appropriate design constraints. Optimization problem has been formulated with a nonlinear objective function to minimize the fluid dynamics losses.

**Chung M K et al[25] [2001]** developed a simple and accurate correlation for the slip factor of centrifugal impeller. Correlation provided was a function of number of vanes, vanes exit angle & the inlet-exit radius ratio. He investigated the radius of relative eddy inscribed by two adjacent vanes and the exit circle of a flow channel in the impeller to obtain the correlation.

**Engin and Gur[26] [2001]** have studied the effects of different solid-liquid mixture properties on the performance characteristics of a centrifugal enshrouded impeller pump, considering the variation of the tip clearance. The effect of the clearance between the impeller tip and the casing and of the solid concentration, density and mean diameter on the pump performance characteristics is investigated.

**Stephan Bross et al[27] [2002]** predicted the influence of different design parameters on the wear behaviour of centrifugal slurry pump's impeller suction sealing .For this purpose he developed a simple model and using this model he calculated the velocity field in the impeller suction side and also a comparison was done between analytical solution & numerical solution provided by a CFD package FLUENT.

**Gandhi et al[28] [2002]** have evaluated performance charactereristics of a centrifugal slurry pump at different rotational speeds with water as well as solid-liquid mixture. They found that the affinity relations applicable to conventional pumps for head and capacity can be applied to slurry pumps handling water and slurries at low concentrations (<20% by weight). For higher solids concentrations, these relationships needed to be corrected by taking into account the effect of solids.

**Goto Akira et al[29] [2002]** have proposed a computer aided design system for hydraulic parts of pumps including impellers, bowl diffusers, volutes and vaned return channels.

Technologies include 3D-CAD modeling, automatic grid generations, CFD analysis and a 3D inverse design method.

**Egin and Gur[30] [2003]** have evaluated some existing correlations to predict head degradation of centrifugal slurry pumps. A new correlation has been developed in order to predict head reductions of centrifugal pumps when handling slurries. The proposed correlation takes into account the individual effects of particle. The proposed correlation is therefore recommended for the prediction of performance factors of ‘‘small-sized’’ slurry pumps having impeller diameters lower than 850 mm size, particle size distribution, specific gravity and concentration of solids, and impeller exit diameter on the pump performance.

**Kadambi et al[31] [2004] )** have used Particle Image Velocimetry to investigate the velocities of the slurry in the impeller of a centrifugal slurry pump for sodium-iodide solution (NaI) and 500micron glass beads slurry. The experiments conducted at 725 rpm, 1000rpm speed, and1%, 2%, 3% volumetric concentration. They observed that the in clear fluid flow conditions for both the pump rpm, flow separation takes place on the suction side of the blade in the region below the blade tip. For the same flow conditions, the flow moves smoothly along the suction side of the blade depicting a recirculation zone. The intensity of this recirculation zone decreases at the higher concentration of 3% due to particle inertia effects. On the pressure side of the blade the particles are pushed along the blade surface and can result in the frictional wear.

**Graeme R. Addie et al[32] [2005]** have discussed numerical model of flow and particles. They have used the experiments which have been conducted to obtain the particle velocities inside an optically transparent acrylic pump using Particle Image Velocity (PIV). They have presented effect of different parameters on operating cost of pump. They concluded that wear parts cost of slurry pumps may be about 50% of the total operating cost of pumps.

**Addie et al.[33] [2007]** have developed ANSI/HI standard of centrifugal slurry pump. They studied the effect of slurry on pump performance; net positive suction head required and wear by using the ANSI/HI standard.

**Pullum et al.[34] [2007]** have calculated the performance reduction of the centrifugal slurry pump by using Hydraulic Institute method for handling non-Newtonian coarse particle suspensions. Suspensions up to 38% v/v of coarse particles with mean diameters in the range of $1.1 < d50 < 3.4$ mm suspended in carrier fluids with dynamic yield stresses of $0 < \tau\gamma < 17.2$ Pa and shear thinning indices in the range $0.35 < n < 0.79$ were examined. They found that the reduction in the head is the function of coarse solid concentration.

**Yang et al[35] [2007]** have for evaluated the internal flow in the impeller of the centrifugal chemical pump by CFD FLUENT software. Standard k -ε (two-equation) turbulence model was used.Simultaneousy the result of calculation is compared with PIV measurement. They found that the internal flow which is simulated in impeller is coincide with the general rule of flow in the impeller machinery, and validated with the result of PIV experiment

# CHAPTER 3

# DESIGN OFCENTRIFUGAL PUMP

## 3.1 CONVENTIONAL DESIGN OF PUMP:

Conventional design method of centrifugal pump are largely based on the application of empirical and semi-empirical rules along with the use of available information in the form of different types of charts and graphs in the existing literature. The program developed is best suitable for low specific speed centrifugal pump. Same program is also suitable for the design of high specific speed and multistage centrifugal pump with few modifications.

As the design of centrifugal pump involve a large number of interdependent variables, several other alternative designs are possible for same duty. Hence theoretical investigation supported by accurate experimental studies of the flow through the pump.
Impeller as it is the element which transfers energy to the fluid stream influences the performance of the pump. Different authors have suggested different design procedure, method of calculation.

The problem of calculation of the dimension of an impeller and hence of the whole pump for given total head may have several solutions but they are not likely to be of equal merit, when considered from the point of view of efficiency and production cost.

Designs suggested by Stenoff has been carefully studied. Each design parameter has been calculated using above procedures and an appropriate value adapt for present carefully analyzing the calculated values.

## 3.2 DESIGN PROBLEM

**INPUT DATA**

Head = 40m　　　Flow Rate = .080m$^3$/sec

Speed = 1450 rpm

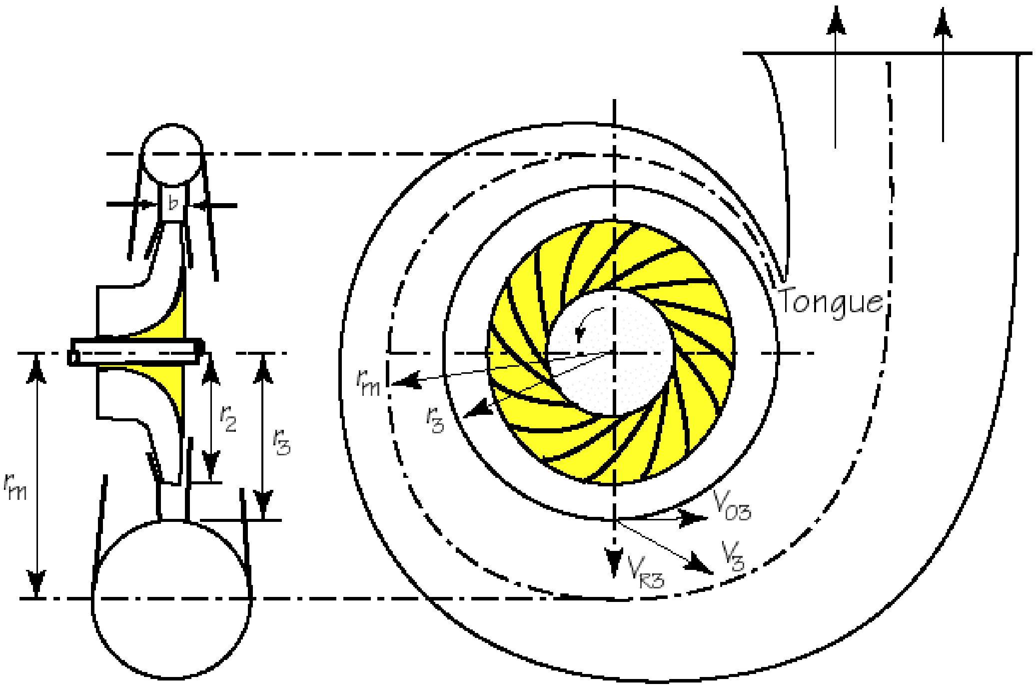


Fig. 3.1 Design parameters of centrifugal pump

## 3.3 DESIGN OF IMPELLER:

### SPECIFIC SPEED:

Specific speed of the pump is computed based on the power as well as discharge, different authors expressed the design parameter as function of specific speed.

$$\mathbf{Ns = N \sqrt{Q} / H^{3/4}} \qquad (3.1)$$

Where N = speed at pump shaft rotated.

Q = discharge in $m^3$ / sec

H = net head in m.

For given data $N_s = \dfrac{1450x\sqrt{.080}}{(40)^{3/4}} = 25.79rpm$

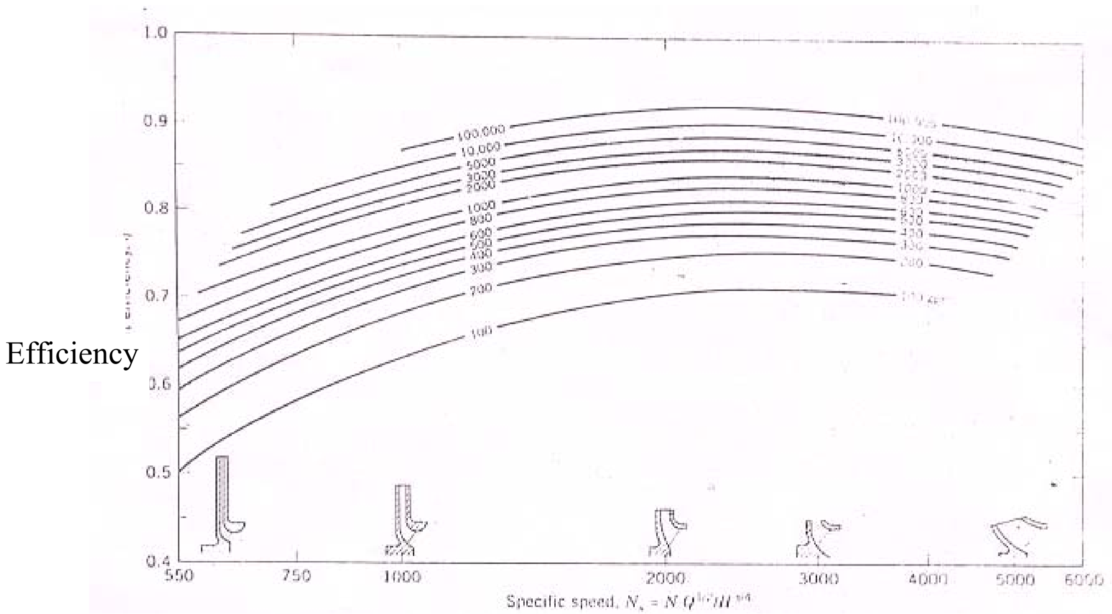



Fig 3.2 Efficiency Vs Specific Speed [37]

## POWER INPUT AND SHAFT INPUT POWER:

$$P_0 = \text{output} = \rho gQH = 1000 * 9.81 * Q * H / 745 \qquad (3.2)$$

Overall efficiency taking by the graph, input power required 15% more because of bearing and transmission loss consider.

For given data input power, Pin $= \dfrac{1000x9.81x.080x40}{745x.78}$

= 57.65 Hp

So,Input required power = 1.15 Pin

Psh = 1.15 x 57.65

= 65.72 Hp

## SHAFT DIAMETER:

Torque,

T = (P * 60 / 2ЛN) N m (3.3)

T = Л/16 Fs $d_{sh}^3$ (3.4)

Fs = stress depend the material constant

$d_{sh} = (16T/Л\ Fs)^{1/3}$

for given data $d_{sh} = \left(\frac{16x62.36x735x60}{2x3.14x1450x1.5}\right)$

$= 0.042486$ m

**HUB DIAMETER:**

(a) $D_{hb} = D_{sh} + 10$ mm for shaft 20mm diameter.[37]

$D_{hb} = D_{sh} + 20$ mm for shaft upto 100mm diameter.

(b) $D_h = (1.2 \text{ to } 1.3)\ D_{sh}$[37]

for given data $D_{hb} = 1.2 \times .042$

$= .0509$

**OUTLET BLADE VELOCITY ($U_2$):**

Head coefficient

$\varphi$ = pressure head generated / maximum Euler head

$\varphi = gh / \eta\ U_2^2$ (3.5)

Generally,

$\varphi = 0.5$ to $0.6$

$U_2^2 = (gh / \eta\ \varphi)$

$U_2 = \sqrt{(gh / \eta\ \varphi)}$

For given data $U_2^2 = \left(\frac{9.8 \times 40}{.78 \times .58}\right)$

$= 29.43$ m/sec

**OUTLET DIAMETER ($D_2$):**

$U_2 = Л\ D\ N / 60$ (3.6)

$D_2 = (60U_2 / ЛN)$

for given data $D_2 = \left(\frac{60x29.436}{3.14x1450}\right)$

$=0.388$ m

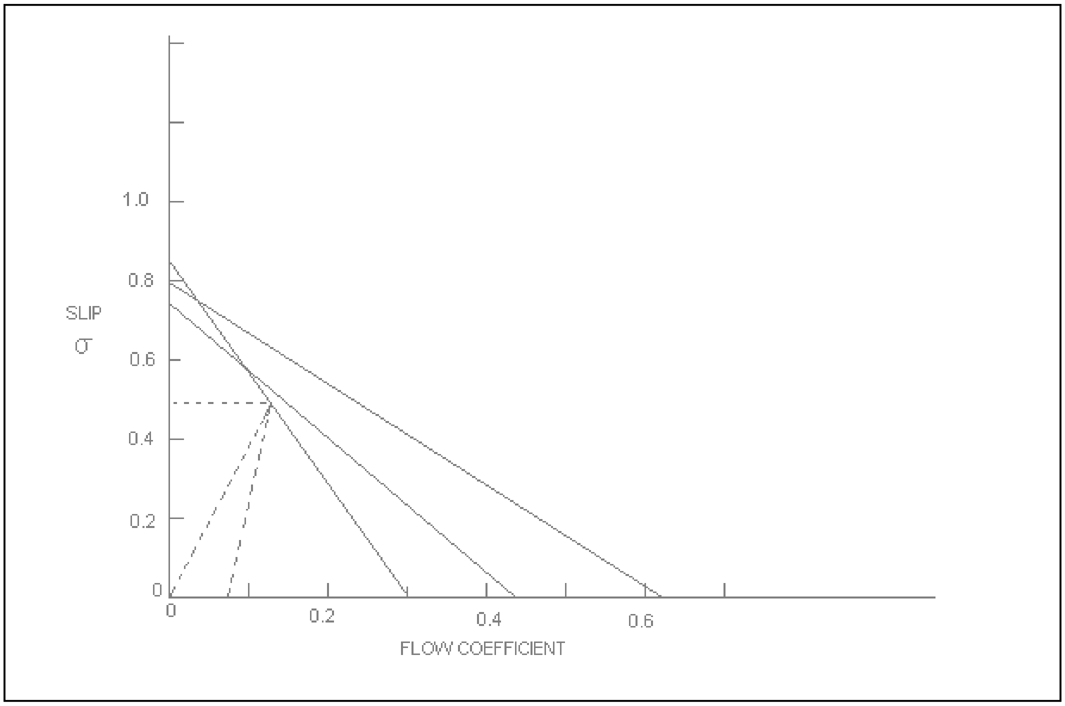



Fig 3.3 **Slip Vs Flow coefficient**[37]

**FLOW COFFICIENT:**

$$\Phi = V_{m2} / U_2 = \text{Flow velocity / Blade velocity} \qquad (3.7)$$

$\Phi$ taken 0.1 to 0.2[37]

$$V_{m2} = \Phi\, U_2$$

Mean meridional velocity of the steam just after entering the blade passage is denoted by $V_{m1}$ and $V_{m0}$ is the mean meridional velocity of steam just prior to blade inlet. $V_{m2}$ denotes the meridional velocity at the exit of the impeller. Ratio of √2gh is known as capacity constant.

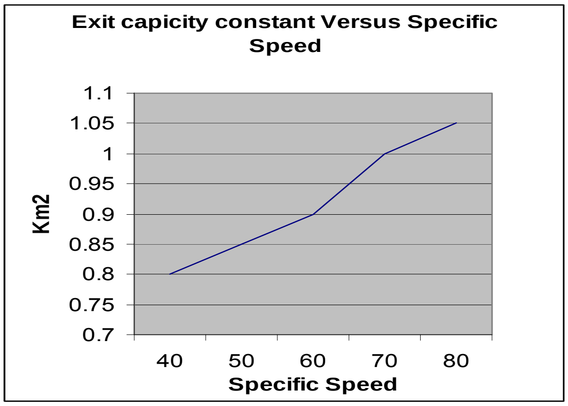


Fig 3.4(a)

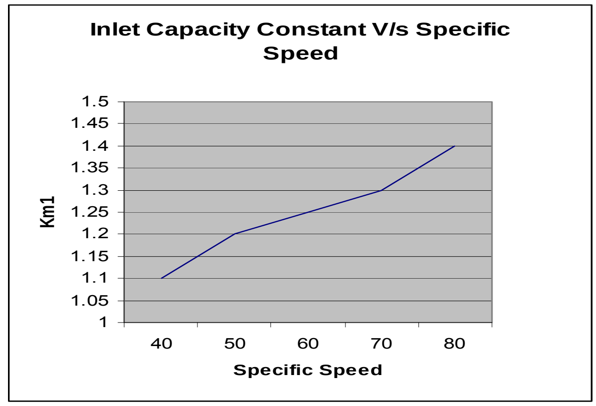


Fig.3.4(b)

$$K_{m1} = V_{m1} / \sqrt{2gH}$$

$$K_{m2} = V_{m2} / \sqrt{2gH}$$

To calculate $C_{m1.}$

$$V_{m0} = (0.8 \text{ to } 0.9)\ V_{m1}$$

Stepanoff recommended computation of $V_{m0}$ and $V_{m1}$

$$V_{m0} = (1.3 \text{ to } 1.5)V_{m2.}$$

For given data $V_{m1} = .175 \times 29.44$

$= 5.15\text{m/sec}$

$$V_{m2} = 1.15 \times 5.92$$

$= 5.924$ m/sec

**IMPELLER EYE DIAMETER ($D_E$):**

$$Л / 4( D_E^2 – D_H^2 ) V_{m1} = Q \text{ (Flow rate)}$$

$$D_e = [ 4Q / ЛV_{m1} + D_h^2 ]^{1/2}$$

$$D_e^{\ 2} = \left[.091^2 + .050^2\right]$$

$= 0.149$ m

**INLET DIAMETER ($D_1$):**

Stepanoff used the following relationship

$$D_1 = (D_e + 0.020) \text{ m}$$

For given data $D_1 = 0.149 + .020 = 0.1696$ m

**INLET BLADE VELOCITY ($U_1$):**

$$U_1 = (Л\, D_1\, N / 60)$$

For given data $U_1 = \left(\dfrac{3.14x.1696x1450}{60}\right)$

$= 12.87$ m/sec

**INLET BLADE ANGLE ($B_1$):**

Fluid at inlet assumed no pressure whirl

$$\beta_1 = \tan^{-1}\left(\frac{V_{m1}}{U_1}\right)$$

for given data $\beta_1 = \tan^{-1}\left(\frac{5.15}{12.87}\right)$

$= 21.82$

Thickness of the blade is mostly taken leading and trailing tips are 4 mm and 5 mm, respectively.

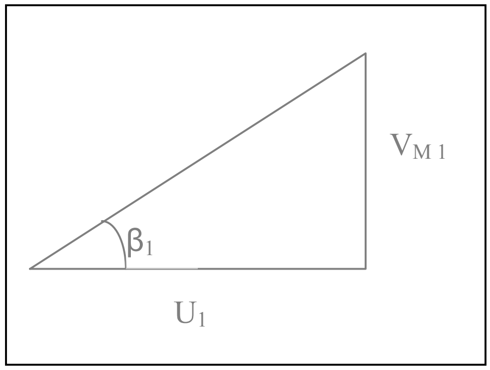



Fig. 3.5 Inlet velocity triangle

**WIDTH OF IMPELLER (B):**

Flow rate (Q) = $(\pi D_2 B_2) \cdot \phi \cdot u_2$

$$B_2 = \frac{Q}{(\pi \cdot D_2 \cdot u_2 \cdot \phi)}$$

$$B_2 = \frac{0.080}{(\pi x.388x29.43x.175)}$$

$$= 0.011 \text{ m}$$

**BLADE ANGLE AT OUTLET ($B_2$):**

Stepanoff recommends the equation

$$H = \frac{u_2^2}{g} - \frac{u_2 V_{m2}}{g \cdot \tan B_2^{'}}$$

$B_2^{'} = 1.4 \text{ x } B_2$

Assuming the fluid stream is entering the impeller without pre rotation and circulation is zero.

Stepanoff recommended

$$\Psi = \sigma - \Phi \tan \beta_2$$

for given data $40 = \dfrac{29.43^2}{g} - \dfrac{29.43x5.92}{g \cdot \tan B_2^{'}}$

$$B_2 = 24.58$$

**NUMBER OF BLADES (Z):**

Number of blades generally taken between 5 to 12.

According to Stepanoff Z = ($B_2$ / 3).

For given data Z = (24.58/3)

= 8.2

**OUTLET BLADE TRIANGLE:**

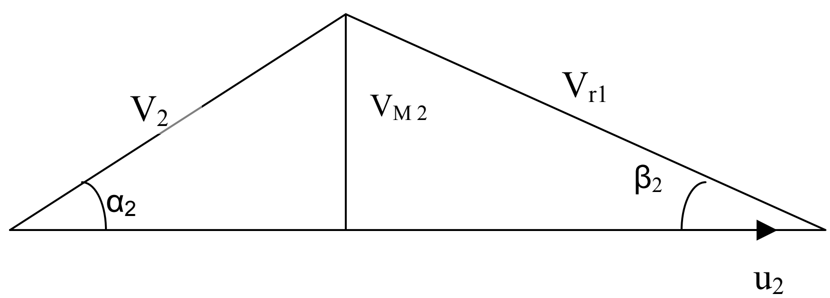



**Fig. 3.6**

From inlet velocity triangle

$V_1 = V_{m1} = 5.15$

$V_{r1}^2 = U_1^2 + V_{m1}^2$

$= 5.15^2 + 12.87^2$

= 13.86 m/sec

$V_{rui} = U_1$ = 12.87 m/sec

From outlet velocity triangle

$V_{r2} = V_{m2}/\sin \beta_2$

= 5.92/sin 24.58

=11.28 m/sec

## 3.4 DESIGN OF VOLUTE CASING:

### 3.4.1 INTRODUCTION:

The objective of the volute is to convert kinetic energy imparted to the liquid by impeller into pressure. Casing has no part in the dynamic generation of total head, it deals only with minimization of losses.

Following element are used to reduce the velocity and kinetic energy:

- Volute casing
- Vane less guide ring.
- Diffuser ring vanes.

Main advantage of volute casing as compared with a casing having diffuser vane is its mechanical simplicity, low cost and ease of manufacture. However the casing with diffuser vane is more efficient than volute casing. Volute casing are preferred for single stage pump and in case of multistage pumps diffuser ring is preferred.

### 3.4.2 DESIGN OF VOLUTE CASING:

Volute casing consists of a casing with gradually increasing area, tongue and a conical discharge nozzle. A circle centered on the axis of rotation and tangent to the volute called the base circle diameter. When rate of flow below the design value some flow return into the volute, passing impeller and tongue through the volute throat, a space between the impeller and tongue must be make small.

Water in the volute has very nearly the spiral flow so that

$$R.V_u = \text{constant.}$$

Let Φ represents the angular distance of any cross-section measured from volute tongue and R is inside radius, r is the radius of elementary strip. Within cross section considered, b is the axial width

Area of elemental ring

$$dA = bdR$$

Discharge through this ring

$$(dQ)_{\Phi} = dA.\ V_u = b.dR.V_u$$

$$(dQ)_{\Phi} = bdr / r\ .$$

Total discharge through the cross section is obtained by integrating the equation.

$$Q_{\Phi} = K \int_{r=r}^{r=R} (b/r)dr$$

Total Q discharge from the pump will be collected in the volute. When more angular distance of 360° from volute tongue.

Discharge at any volute section Φ

$$Q_{\Phi} = (\Phi/360).Q$$

$$\Phi(\text{degree}) = 360K/Q \int_{r=r}^{r=R} (b/r)dr$$

$$= 360\ R_2\, Cu_2 / Q\ K \int_{r=r1}^{r=R} (b/r)dr$$

Max total angle Qa, between the sides, about 60°. If more water is unable to flow the side hence turbulence and insufficiency result. Qa small and radius large give better result but casing diameter and weight of the pump are increased.

To avoid shock losses the tongue angle should be made the same as absolute outlet angle $\alpha_2$, water leaving the impeller, radius Rt is the 5 to 10% greater than outside radius of impeller to avoid turbulence and noisiness.

Zero point of the volute or point from angle is measured may be found by assuming that flow follows a logarithmic spiral

$$R = R_2\ e^{\tan\alpha 2.\theta}$$

$\Phi$ = Angle in radians

**Throat Angle**

Angle at which the throat of the volute start

$$\text{Throat angle } Qt = [\ \ln (Rt / R_2 ) / \tan \alpha_2$$

In order to avoid turbulence total divergence in this passage should not exceed 10°.

**Inlet width of volute**

$$B_3 / B_2 = 1.4 \text{ TO } 1.8 \text{ (for high Ns)}$$

$$= 2 \text{ (for low Ns)}$$

**Diameter of Inlet at Volute**

$D_3 = 1.1$ to $1.2\ D_2$

**Width of volute at any point**

$$B = B_3 + 2 X \tan (Q_A / 2)$$

$$Q_A = \text{Taken } 60°$$

X = Distance between any radius R and impeller outside radius

$$R2 = (Rav - R2)$$

**Velocity at throat $V_{th} = (D_2 \times V_{u2})/D_3$**

For given design problem

### 3.4.3 VOLUTE DESIGN CALCULATION

Inlet Width of Volute $B_3 = 1.8\ B_2$

$= 1.8 \times 0.11087$

$= 0.0199$ m

Dia. Of Inlet at Volute

$D_3 = 1.15D_2$

$= 1.15 \times 0.387$

$= 0.446$ m

$$R_v = \left(\frac{D_2 + D_3}{4}\right)$$

$$= \left(\frac{0.387 + 0.4461}{4}\right)$$

$= 0.2085$

Width at x distance $B_x = B_3 + \left(\frac{2R_v - D_2}{1.73}\right)$

$$= 0.0199 + \left(\frac{2\text{x}0.20854 - 0.388}{1.73}\right)$$

$= 0.0367$ m

Whirl component of velocity at volute

$V_{u3} = V_2 \cos \alpha_3$

$= 5.99 \cos 8.29$

$= 1.623$ m/sec

Throat angle Qt = [ ln (0.223 / 0.196 ) / tan 8.29

$= 38.56^o$

Velocity at throat $V_{th} = (0.387 \times 39.03)/0.448$

$= 33.94$ m/sec

# CHAPTER 4

# NPSH REQUIREMENT AND CAVITATION

When a pump impeller is designed to attain the required head at the design flow and maximum efficiency a stable flow characteristics and good cavitations performance, in a pump, if the pressure at any point drops below the vapor pressure corresponding to temperature of the liquid, liquid will vaporize and form cavities of vapor. Vapor bubbles are carried along with the stream until a region of higher pressure is reached where they collapse or implode with a tremendous shock on the adjacent wall. This phenomenon is called cavitations. Cavitation affects the pump performance and may damage pump parts in severe cases.

- Noise and vibrations
- Drop in head capacity and efficiency curve
- Impeller vane pitting and corrosion fatigue failure of metal.

A characteristic of the critical points where the head break down occurs can be found in terms of NPSE as a function of flow.

A characteristic where the flow is kept constant and absolute pressure (or NPSE) dropped until cavitation occurs in fig 4.2. Blade cavitation starts at a point A. Then gradually increase until at a point B, performance affected and finally at c. Cavitation is so extensive that the performance break-down occurs.

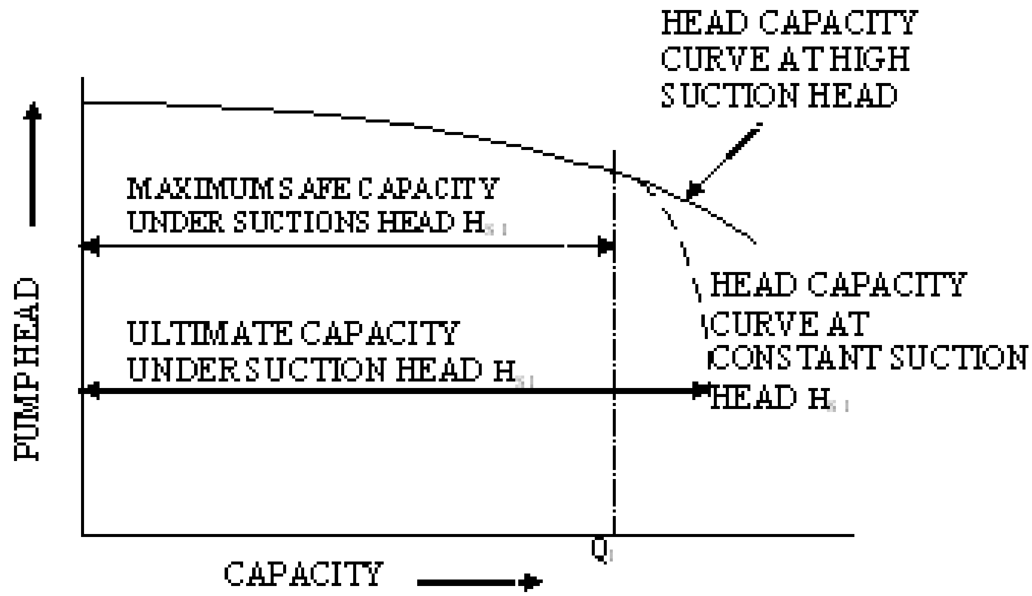



Fig. 4.1 Pump head Vs Capacity

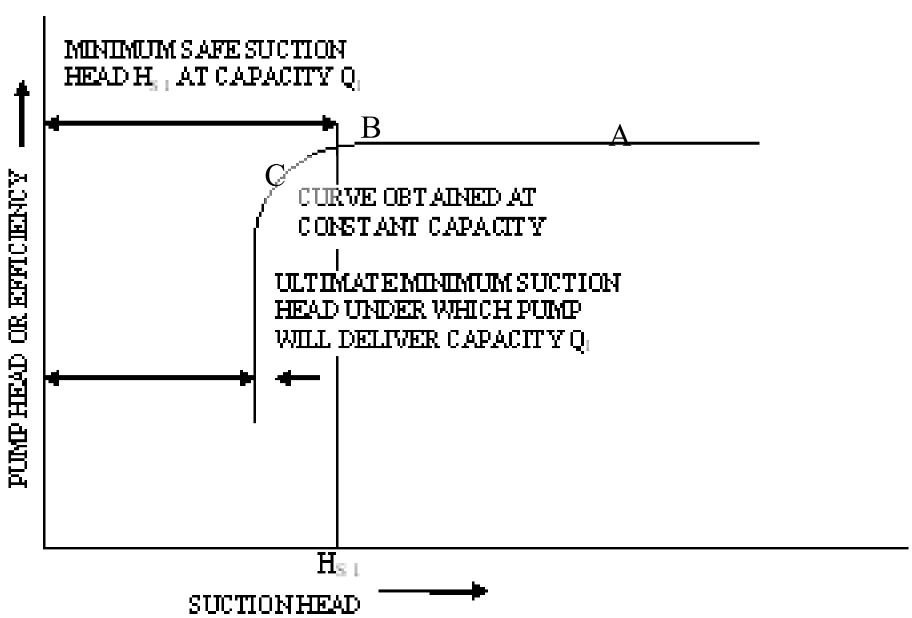


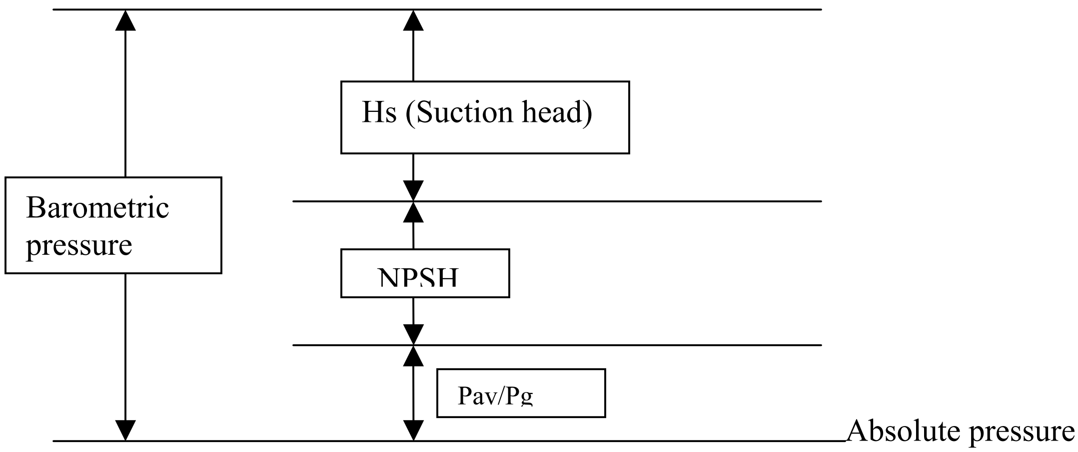

Fig. 4.2 Efficiency Vs Suction head

## 4.1 NPSH AVAILABLE:

Net positive suction available head is defined as the net head required to force the liquid into the pump through the suction pipe.

$NPSH_A = P_b$ - (Hs + Pvap / $\rho g$ + frictional head + kinetic head)

Gauge pr.




Cavitation occurs when $NPSH_A = 0$

$NPSH_A$ depends upon barometric pressure, location, up to sea level, suction height of machine , loss in suction head , all the factors depends on the layout independent of the pump performance.

THOMA CAVITATION COFFICIENT:

$$\sigma\, th = NPSH_A / H; \qquad (4.1)$$ [37]

## 4.2 NET POSITIVE SUCTION HEAD REQUIRED ($NPSH_r$):

Pressure of the liquid increase inlet to outlet. At any point of lowest pressure, is (H) below the suction side pressure of machine. This fall in pressure below the suction pressure is called net positive suction head required ($NPSH_r$). $NPSH_r$ depends on geometry of machine and performance parameter. In order to prevent the cavitation

$$NPSH_A > NPSH_r.$$

$$\text{And } \sigma_r = NPSH_r / H. \qquad (4.2)^{37}$$

To prevent cavitation

$$\sigma > \sigma_r$$

Cavitation inception will occur when

$$\sigma_r = \sigma$$

$$NPSH_r = \lambda_1 U_1^2 / 2g + \lambda_2 V_{m1}^2 / 2g \qquad (4.3)^{37}$$

The discharge and speed discharge depends on $V_{m1}$ and speed $U_1$.

BREAK DOWN POINT:

If the $\sigma$ is reduced further to a value $\sigma_{cl}$ at point A, efficiency falls rapidly. This point is called break down point.

## 4.3 CALCULATION OF NPSH:

**For given design problem**

According to stepanoff _this coefficient can be obtained as follows:

$P = K_s \times R$

Where P = Pfle i derer's coefficient 0.2447( for low specific speed)

R = Blade loading ratio

$K_s$ = constant for free vortex volute casing range 1.6 to 1.8

R = P/K = 0.2447/1.6 = 0.1529

Impeller mean Blade loading can be found as $dp/r = R \times H_t$

$H_t$ = theoretical head = 40 m

So $dp/\rho = 0.1529 \times 40 = 6.116$ m

Depression head

$H_d = K \times (dp/r)$

$$K = 0.20 + \left(\frac{12.95 x N_{sp}}{1000}\right)^3 x \frac{1}{68}$$

$$= 0.20 + \left(\frac{12.95 x 25.78}{1000}\right)^3 x \frac{1}{68}$$

$= 0.620$

$H_d = 0.62 \times 6.116$

$= 3.796$ m

Net Positive available suction head = $H_d + (V_{m1}^2/2g)$

$= 3.796 + (5.115^2/2g)$

$= 5.1294$ m

Thoma Cavitation Coefficient

$\sigma_{th} = NPSH/ H;$

$= 5.1294/40$

$= 0.12829$ m

Net positive suction required head

$$NPSH_r = \frac{(1+\sigma_b)V_{m1}^2}{2g} + \frac{\sigma_b U_1^2}{2g}$$

$$= \frac{(1+0.3)5.115^2}{2g} + \frac{0.30 x 12.87^2}{2g}$$

$= 4.29$ m

Thoma caviataion coefficient for required head

$\sigma_r = NPSHr/ H.$

$= 4.29/40$

$= 0.107$

And $5.129 > 4.2954$

Hence pump non cavitation condition

$NPSH_A > NPSH_r$

# CHAPTER 5

# SIMULATION OF LOSS MODELS

## 5.1 INTRODUCTION

In conventional design method of pump, efficiency is the function of specific speed, which is available in form of graph, empirical correlation in various text books and references .But in practical, efficiency has direct influence due to change of flow pattern, Renoldnumber, relative eddies in the impeller blade passage .

Present work is carried out under the following assumptions, the flow comes in through the inlet without any pre-swirl, the flow in the van less space is of a free-vortex type, and the volute casing is constructed of gradually increasing circular cross-sections with a constant average velocity. For calculation of optimum set of loss models for pump input data are design specifications and geometrical and hydraulic variables, given below. Empirical loss models from the open literature. Geometrical and hydraulic parameters are calculated with the help of conventional design method given in as mentioned in chapter 3.For simulation purpose computer program has been generated in C, which permits wide range of variables to be investigated in a short interval of time

## 5.2 DESIGN SPECIFICATION

Design of the Pump input data: Volume flow rate, Pump total pressure head, Specific speed, Density of liquid, Operating fluid viscosity.

## 5.3 GEOMETRIC PARAMETER

Vane angle, Number of vanes Impeller discharge width, Hub/Tip ratio, Inclination of the mean stream line to axial direction

## 5.4 HYDRAULIC PARAMETER

Flow coefficient, Head coefficient, Blade velocity, Relative velocity and other hydraulic parameter needed to describe the flow direction and magnitudes become direct function of geometry

## 5.5 LOSSES IN CENTRIFUGAL PUMP:

Losses divided mainly into two categories:

1. Internal losses
2. External losses

**INTERNAL LOSSES**

Losses which take place in the inner passages of the machine and are directly connected with the flow in the impeller. These are further divided as,

HYDRAULIC LOSSES:

Hydraulic losses are pressure losses due to friction, separation, contraction, diffusion, eddy formation, etc. of the flow passing through the machine from the inlet flange to the discharge flange.

LEAKAGE LOSSES:

The leakage flow loss is a loss on the flow quantity. A certain amount of flow will find its way from the pressure side of the impeller to the suction side without passing the impeller channels or will leak out of the machine through the sealing.

DISC FRICTION LOSSES:

The outside of the impeller is surrounded by the flow medium. While the impeller rotates, friction is generated between the outside of impeller and surrounding medium which result in friction loss. This loss is known as disc friction loss.

**EXTERNAL LOSS**

The external losses can be expressed as power loss due to friction in bearing, sealing and due to fluid friction. At the outer rotating parts of the machine like shaft End and coupling these losses are the mechanical losses.

BEARING LOSSES:

The loss due to the friction in the bearing is small compared to the other losses. Generally 3 to 5% of the power input loss in the bearing.

## 5.6 CALCULATION OF LOSSES:

**SUCTION LOSS**

Losses may arise from a change of direction in the impeller intake, liquid is directed through an angle of approx. zero degree before entry into blade cascade. These losses depend the velocity V1

$$\Delta h_{Suction} = f_1 \frac{V_1^2}{2g} \qquad (5.1)^{37}$$

Where f1 is the friction factor taken between 0.2 to 0.3

For given data

$$\Delta h_{Suction} = 0.2 \frac{5.15^2}{2g}$$

$$= 0.27077 \text{ m}$$

**INCIDENCE LOSS**

The friction losses in the impeller are due to the flow of fluid through the impeller and the retardation loss of the fluid.

Consider pure surface friction,

INCIDENCE LOSS:

$$\Delta h_{\text{Incidence}} = f_2 \frac{v_{rui}^2}{2g} \qquad (5.2)^{37}$$

Vrui - Tangential component of relative velocity f incidence= 0.5 to 0.7

For given data

$$\Delta h_{\text{Incidence}} = 0.5 \frac{12.87^{2}}{2g}$$

$$= 1.056 \text{ m}$$

**DISC FRICTION LOSS**

Disc friction loss is the most important single item among the pump losses.

Power required to rotate the disc in a fluid is known as disc friction. It will be treated as internal mechanical loss, a loss in power, hydraulic losses are losses of head or pressure. Thus if the outside of the impeller is polished, the efficiency will improve. But head will remain unchanged, whereas if the impeller passage and vanes are polished the efficiency and head also improve. Disc friction loss is grouped internal loss because heat due to disc friction loss is retained by the fluid.

This is due to viscous drag on the back surface of the impeller disc. It depends on density of the fluid, dia. of disc, peripheral speed and axial clearance between the housing and disc.

$H_3$ = disc friction loss

$$H_3 = \frac{f_{dR} D_2^2 u_2^3}{16Qg} \tag{5.3}$$ [36]

fd is the constant take into account effect of housing geometry and clearance,

Where

$$f_{dR} = \frac{2.67}{Re^{0.5}} \qquad Re<3x10^5$$

$$f_{dR} = \frac{0.0622}{Re^{0.2}} \qquad Re \geq 3x10^5$$

$$Re = \left(\frac{u_2 r_2}{\vartheta}\right)$$

$d_2$ = outlet dia of impeller

$u_2$ = peripheral blade velocity

Q = flow rate.

For given data

$$Re = \left(\frac{29.43x0.194}{0.32}\right)$$

$$= 1802.5$$

$$f_{dR} = \frac{2.67}{1802.5^{0.5}} \qquad Re<3x10^5$$

$$= 0.0628$$

$$H_3 = \frac{0.0628 \times 0.387^2 x29.43^3}{16x0.08g}$$

$$= 0.824 \text{ m}$$

**MIXING LOSS**

$$h_u = \frac{1}{(1+\tan^2\alpha_2)}\left(\frac{v_2^{\;2}}{2g}\right) \tag{5.4}$$ [37]

$V_2$ = absolute velocity

$\alpha_2$ = angle flow

$$h_u = \frac{1}{(1+\tan^2 32.16)}\left(\frac{5.99^{\;2}}{2g}\right)$$

$$= 0.895 \text{ m}$$

**RECIRCULATION LOSS**

$$(\Delta h)_{\mathrm{Re}} = \frac{8x10^{-5}\sinh(3.5\alpha_2^3)D_f^2u_2^2}{g} \tag{5.5}$$ [37]

$$(\Delta h)_{\mathrm{Re}} = \frac{8x10^{-5}\sinh(3.5x32.16^3)0.38^2 29.4^2}{g}$$

$$= 0.006 \text{ m}$$

**BLADE LOADING LOSS**

$$(\Delta h)_{Blade} = \frac{0.05\, D_f^2 u_2^2}{g} \qquad (5.6)^{37}$$

$$(\Delta h)_{Blade} = \frac{0.05x0.38^2 x29.4^2}{g}$$

$$= 0.665 \text{ m}$$

**SKIN FRICTION LOSS (SHROUDED IMPELLER)**

$$\Delta h = \frac{2CL_b V_r^2}{D_{hyd}\, g} \qquad (5.7)^{37}$$

$$\Delta h = \frac{20x0.29x11.27^2}{0.38g}$$

$$= 1.056 \text{ m}$$

**VOLUTE LOSSES:**

Kinetic energy of the liquid from the impeller is converted into head in the casing during the conversion friction losses occur.

$$\Delta h_{volute-loss} = h_{Expansion} + h_{inl\arg ement} + \Delta h_{skin-friction-volute}$$

**EXPANSION LOSS**

$$(\Delta h)_{Expansion} = \frac{0.75(V_{u2} - V_{th})^2 + V_m^2}{2g}$$

(4.10)[37]

$$(\Delta \mathrm{h})_{\mathrm{Expansion}} = \frac{0.75(39.03 - 33.9)^2 + 5.15^2}{2\mathrm{g}}$$

$= 1.44$ m

**ENLARGEMENT LOSS**

$$(\Delta h)_{Enlargement} = \frac{(V_{th} - V_d)^2}{2g}$$

(4.11)[37]

$$(\Delta h)_{Enl\arg ement} = \frac{(33.9 - 26.3\ )^2}{2g}$$

$= 1.98$ m

**VOLUTE SKIN FRICTION LOSS**

$$(\Delta h)_{volute-skin-friction} = \frac{0.5V_{th}^2}{2g}$$

(4.12)[37]

$$(\Delta \mathrm{h})_{\mathrm{volute-skin-friction}} = \frac{0.533.9^2}{2\mathrm{g}}$$

$$= 1.19 \text{ m}$$

Total Loss = $h_1 + h_2 + h_3 + h_4 + h_5 + h_6 + h_7 + h_8 + h_9 + h_{10}$

= 9.82862 m

$$\text{Efficiency} = \left(\frac{\mathrm{H} - \mathrm{H}_{\mathrm{loss}}}{\mathrm{H}}\right) \mathrm{x}100$$

$$= \left(\frac{40 - 9.8268}{40}\right) \mathrm{x}100$$

= 75.44 %

# CHAPTER 6
# RESULTS AND DISCUSSIONS

**Input data**

Head = 40 m, Flow rate = 0.8 $m^3$/sec, Speed = 1450 rpm

## 6.1 DESIGN OF IMPELLER

**Table 6.1 Output of impeller dimensions**

| | | |
|---|---|---|
| 1 | Specific speed of pump | 25.78 rpm |
| 2 | Input power of the pump | 57.65 HP |
| 3 | Shaft Power | 65.72 HP |
| 4 | Diameter of shaft | 0.04 m |
| 5 | Hub Diameter | 0.05 m |
| 6 | Axial velocity | 2.5487 m/sec |
| 7 | Eye diameter | 0.14 m |
| 8 | Inlet blade diameter | 0.166305 m |
| 9 | Hub diameter | 0.02362 m |
| 10 | Width at inlet | 0.011342 m |
| 11 | Inlet blade velocity | 12.87 m/sec |
| 12 | Inlet blade angle | $21.8218^\circ$ |
| 13 | Outlet blade velocity | 29.43 m/sec |
| 14 | Outlet blade diameter | 0.388 m |
| 15 | Number of blades | 8 |
| 16 | Width of blade at outlet | 0.007678 m |
| 17 | NPSH required | 1.64 |
| 28 | Outlet blade angle | $24.58^\circ$ |

## 6.2 DESIGN OF VOLUTE

**Table 6.2 Output of volute dimensions**

| | | |
|---|---|---|
| 1 | $B_3$ | 0.0199 m |
| 2 | $D_3$ | 0.446 m |
| 3 | $R_v$ | 0.2085 |
| 4 | $B_x$ | 0.0367 m |
| 5 | $V_{u3}$ | 1.623 m/sec |
| 6 | $\theta_t$ | $38.56^o$ |
| 7 | $V_{th}$ | 33.94 m/sec |

## 6.3 OUTPUT OF NPSHR

**Table 6.3 Output of NPSHr**

| | | |
|---|---|---|
| 1 | R | 0.1529 |
| 2 | $dp/\rho$ | 6.116 m |
| 3 | $H_d$ | 3.796 m |
| 4 | NPSH | 5.1294 m |
| 5 | $\sigma_{th}$ | 0.12829 m |
| 6 | $NPSH_R$ | 4.29 m |
| 7 | $\sigma_r$ | 0.107 |

## 6.4 OUTPUT OF LOSSES

**Table 6.4 Results of losses (in m)**

| | | |
|---|---|---|
| 1 | $H_1$ | 0.27 |
| 2 | $H_2$ | 1.05 |
| 3 | $H_3$ | 0.82 |
| 4 | $H_4$ | 0.89 |
| 6 | $H_6$ | 0.66 |
| 7 | $H_7$ | 1.05 |
| 8 | $H_8$ | 1.44 |
| 9 | $H_9$ | 1.98 |
| 10 | $H_{10}$ | 1.19 |
| 11 | $H_L$ | 9.82 |
| 12 | H | 75.44 % |

Fig. 6.1 shows variation of Efficiency with density at different pump speeds. This figure shows that Efficiency decreases as density increases and when speed increases efficiency increases.

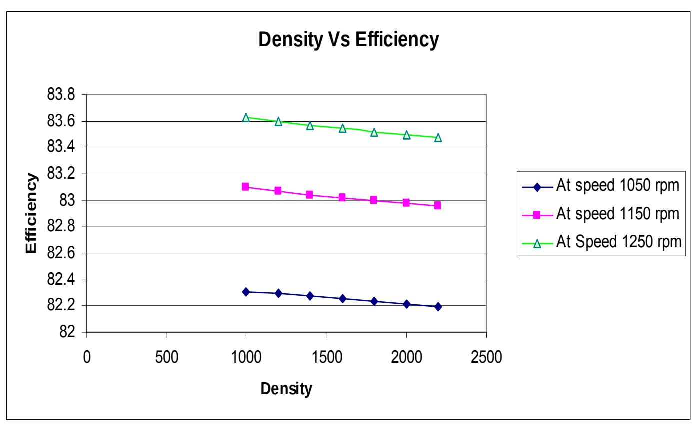
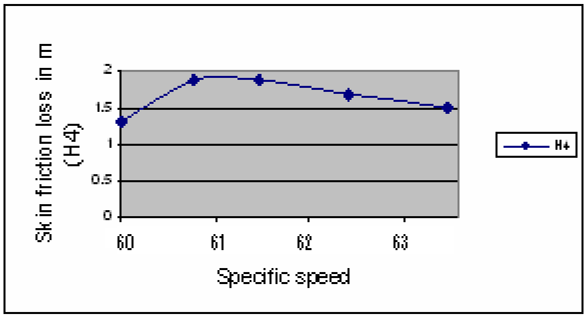



Fig. 6.1 Density-Efficiency



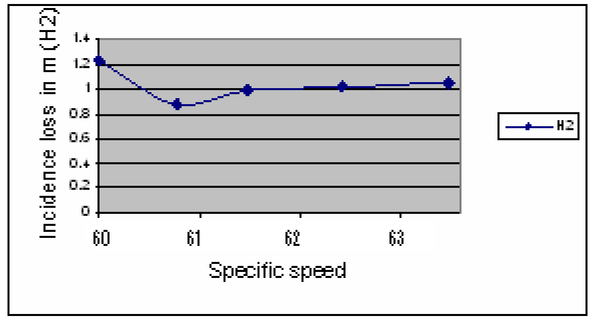
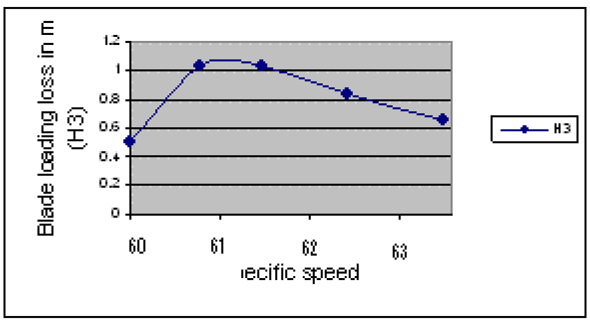




Fig 6.2 specific speed -suction head loss

Fig 6.3 Specific speed - incidence loss



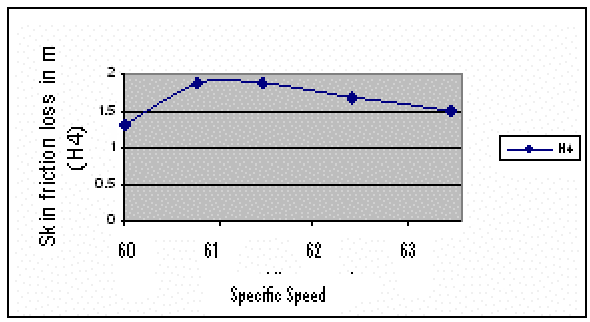
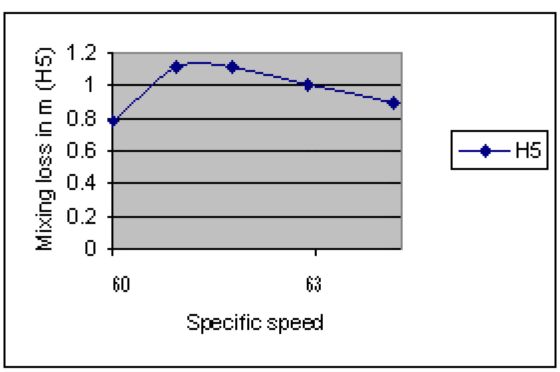




Fig 6.4 Specific speed - blade loading loss
loss

Fig 6.5 Specific speed -skin friction



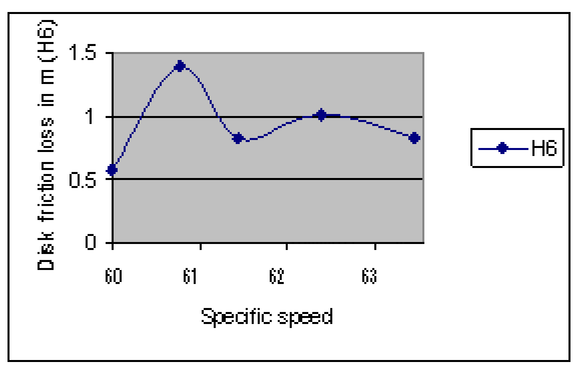




Fig6.6 Specific speed - mixing loss
loss

Fig6.7 Specific speed - disk friction

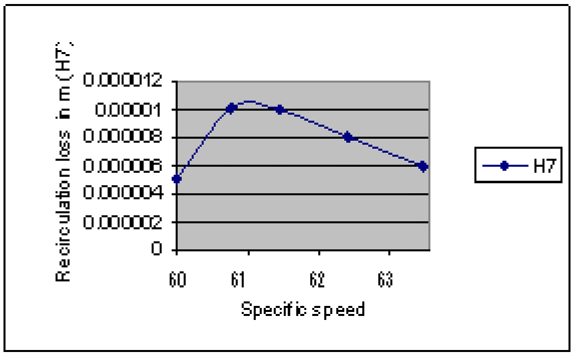



Fig6.8 Specific speed- recirculation loss

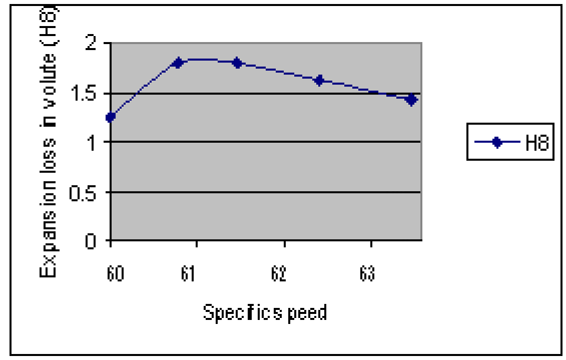



Fig 6.9 Sp speed – volute expansion loss

Figure 6.2 to 6.9 show variation of different internal hydraulic losses with variation of specific speed. These figures show that the various hydraulic losses first increase with specific speed and after some value start decreasing except the incidence loss which decreases first and after some value becomes almost stable. Thus the total internal hydraulic efficiency first decreases and then increases after some value.

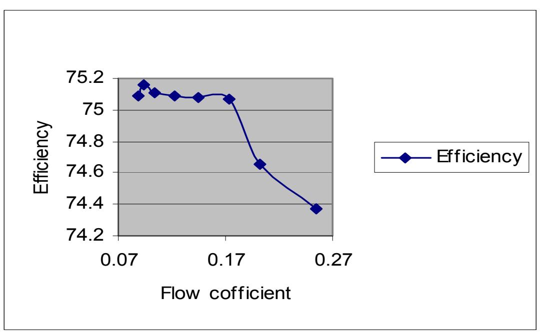



Fig 6.10 Flow coefficient - efficiency

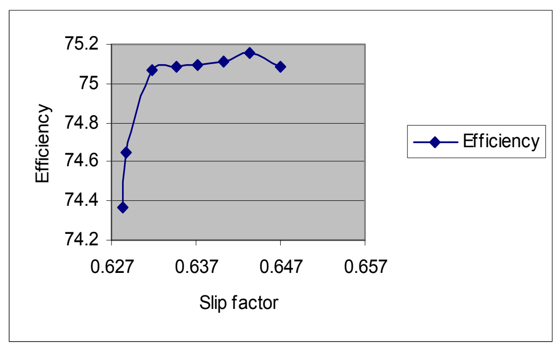



Fig 6.11 Slip factor - efficiency

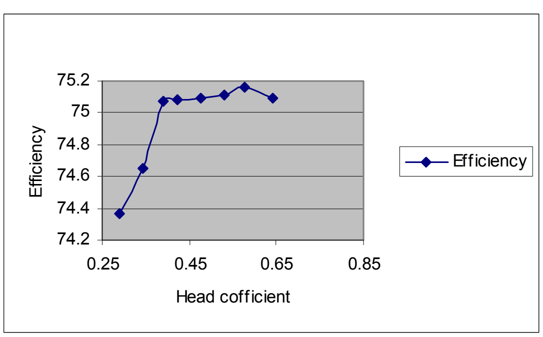



Fig 6.12 Head coefficient - efficiency

Figure 6.10-6.12 show the variation of efficiency with different non dimensional fluid parameters. With increase in slip factor and head coefficient, efficiency increases but with increase in flow coefficient, efficiency decreases. Thus efficiency is stable for the given range of non dimensional parameters.

# CHAPTER 7

# CONCLUSION AND SCOPE FOR FUTURE WORK

## 7.1CONCLUSIONS

Design program has been developed a configuration of centrifugal pump order the consideration of pump efficiency, NPSHR, to avoid cavitaton. Advantage of computer permit the wide range of design variable to investigated in a very short time. Accuracy of the result depends on the accuracy of the information fed into the computer and choice of suitable variable. Design variable range selected in the present study has been described in reference research paper, book. The present work can easily calculate the actual hydraulic losses of pump at given operating condition. As these loss models directly depend on geometrical and hydraulic parameters of pump.

## 7.2 SCOPE OF FUTURE WORK:-

1) Using finite element analysis pressure distribution, stress variation can be analyzed.

2) Performance of theoretical analysis can be compared with experimentally.

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# ANNEXURE I

# PROGRAM FOR DESIGN OF CENTRIFUGAL PUMP

/* Design of impeller*/

```
#include<stdio.h>
#include<conio.h>
#include<math.h>
double  sai= 0.58 ;
double  fi = 0.175;
double  eta= 0.78;
double  g  = 9.8;
double  ro = 1000;
double  pi = 3.14;
double sigma =0.66;
void main()
   {  double H,z,b2,N,Q,P,Ns,D2,U2,Psh,Dhb,Dsh,De,Vm1,D1,U1,B1,B2,Z,B,Ku;
        double V1,Vr1,Vm2,W1,Vrui,Vr2,Alpha2,V2,Vu2;
        double Re,Fd,Vth,Vd,S,Ath,HL,sigb,NPSHR;
        char ch;
        clrscr();

        printf("\n New session of calculating :: \n Following variables ::\n1-Ns\n2-U2\n3-
D2\n");
        printf("Values of the constants taken are :::\n ");
        printf("sai =%lf\nfi=%lf\n eta=%lf\ng=%lf\nro=%lf\n",sai,fi,eta,g,ro);
        printf("\nThe value  of Sigma=%lf",sigma);


        printf("\nPress any key to go ahead");
        getch();
        printf("For Calculation pls Enter following value");

        printf("\nH====");
        scanf("%lf",&H);

        printf("\nN====");
        scanf("%lf",&N);

        printf("\nQ====");
        scanf("%lf",&Q);


        Ns= (N * sqrt(Q))/pow(H,0.75);


        U2=sqrt((g*H)/(eta*sai));
```

```
        D2 = (60 * U2)/ ( pi * N);
        printf("\npress any key to get results");
        getch();

        printf("\nValues obtained are =>>\n1-Ns=%lf\n2-U2=%lf\n3-
D2=%lf\n",Ns,U2,D2);


   //   printf("\nDo you want more processing Y/N\n");

   //   scanf("%c",&ch);

   //   if(ch!='Y' && ch!='y')
   //   return;

        P=(ro*g*Q*H)/(eta*735);

        Psh=1.14*P;

        Dsh =0.12*pow((Psh/N),0.33);


        Dhb= Dsh*1.2;
    //  if(0.020<Dsh<=0.100)
   //   Dhb= Dsh +.016;
        Vm1 = fi*U2;

        De = (Q*4)/(Vm1*pi) + pow(Dhb,2);
        De = sqrt(De);

        D1 = De + 0.020;
        Vm2 = 1.15*Vm1;
        U1 = (pi*D1*N)/60;
        W1=Q/(pi*D2*Vm2);
        B1 = atan(Vm1/U1);
        B1 = (B1 *180)/pi;
        printf("\nValues calculated are =\n P=%lf\n Psh=%lf\n Dsh=%lf\nDhb=%lf\n
Vm1=%lf\n De=%lf\n D1=%lf\n U1=%lf\n
B1=%lf\n",P,Psh,Dsh,Dhb,Vm1,De,D1,U1,B1);
   //    printf("Do you want more processing Y/N\n");
   //   scanf("%c",&ch);

  //    if(ch!='Y' && ch!='y')
   //   return;
```

```
b2 = atan((sigma - sai)/fi);

b2= (b2*180)/pi;
z= b2/3;
printf("\nValues calculated are =\n B2=%lf\n z=%lf\n",b2,z);
/*************************************************************/

V1=Vm1;
Vr1=(U1*U1 + Vm1*Vm1);
Vr1=sqrt(Vr1);
Vrui=U1;
Vr2=Vm2/(sin(b2));
Alpha2=atan(Vm2/(U2-Vr2*(cos(b2))));
V2=(Vm2/(cos(Alpha2)));
Vu2=(U2-Vr2*cos(b2));
printf("Values calculated
are\nW1=%lf\nVm2=%lf\nV1=%lf\nVr1=%lf\nVrui=%lf\nVr2=%lf\nAlpha2=%lf\nV2=
%lf\nVu2=%lf\n",W1,Vm2,V1,Vr1,Vrui,Vr2,Alpha2,V2,Vu2);

/*********************************************************/
getch();
return;
}
```

/* Design of valute casing*/

```
#include<stdio.h>
```

```
#include<conio.h>
#include<math.h>
double ALPHA3 =8.13;
double  sai= 0.58 ;
double  fi = 0.175;
double  eta= 0.78;
double  g  = 9.8;
double  ro = 1000;
double  pi = 3.14;
double sigma =0.66;
void main()
    {  double H,z,b2,N,Q,P,Ns,D2,U2,Psh,Dhb,Dsh,De,Vm1,D1,U1,B1,B2,Z,B,Ku;
        double V1,Vr1,Vm2,W1,Vrui,Vr2,Alpha2,V2,Vu2;
        double Re,Fd,Vth,Vd,S,Ath,HL,sigb,NPSHR,D3,B3,Rv,Bx,thita,Vthx,Vu3;
        char ch;
        clrscr();

        printf("\n New session of calculating :: \n Following variables ::\n1-Ns\n2-U2\n3-
D2\n");
        printf("Values of the constants taken are :::\n ");
        printf("sai =%lf\nfi=%lf\n eta=%lf\ng=%lf\nro=%lf\n",sai,fi,eta,g,ro);
        printf("\nThe value  of Sigma=%lf",sigma);


        printf("\nPress any key to go ahead");
        getch();
        printf("For Calculation pls Enter following value");

        printf("\nH====");
        scanf("%lf",&H);

        printf("\nN====");
        scanf("%lf",&N);

        printf("\nQ====");
        scanf("%lf",&Q);


        Ns= (N * sqrt(Q))/pow(H,0.75);


        U2=sqrt((g*H)/(eta*sai));

        D2 = (60 * U2)/ ( pi * N);
        printf("\npress any key to get results");
        getch();
```

```
        printf("\nValues obtained are =>>\n1-Ns=%lf\n2-U2=%lf\n3-
D2=%lf\n",Ns,U2,D2);


   //   printf("\nDo you want more processing Y/N\n");

   //   scanf("%c",&ch);

   //   if(ch!='Y' && ch!='y')
   //   return;

        P=(ro*g*Q*H)/(eta*735);

        Psh=1.14*P;

        Dsh =0.12*pow((Psh/N),0.33);


        Dhb= Dsh*1.2;
     // if(0.020<Dsh<=0.100)
    //  Dhb= Dsh +.016;
        Vm1 = fi*U2;

        De = (Q*4)/(Vm1*pi) + pow(Dhb,2);
        De = sqrt(De);

        D1 = De + 0.020;
        Vm2 = 1.15*Vm1;
        U1 = (pi*D1*N)/60;
        W1=Q/(pi*D2*Vm2);
        B1 = atan(Vm1/U1);
        B1 = (B1 *180)/pi;
        printf("\nValues calculated are =\n P=%lf\n Psh=%lf\n Dsh=%lf\nDhb=%lf\n
Vm1=%lf\n De=%lf\n D1=%lf\n U1=%lf\n
B1=%lf\n",P,Psh,Dsh,Dhb,Vm1,De,D1,U1,B1);

    //   printf("Do you want more processing Y/N\n");
   //   scanf("%c",&ch);

  //    if(ch!='Y' && ch!='y')
   //   return;


         b2 = atan((sigma - sai)/fi);
```

```
        b2= (b2*180)/pi;
        z= b2/3;
        printf("\nValues calculated are =\n B2=%lf\n z=%lf\n",b2,z);
  /************************************************************/

  V1=Vm1;
   Vr1=(U1*U1 + Vm1*Vm1);
   Vr1=sqrt(Vr1);
   Vrui=U1;
   Vr2=Vm2/(sin(b2));
   Alpha2=atan(Vm2/(U2-Vr2*(cos(b2))));
   V2=(Vm2/(cos(Alpha2)));
   Vu2=(U2-Vr2*cos(b2));
   printf("Values calculated
are\nW1=%lf\nVm2=%lf\nV1=%lf\nVr1=%lf\nVrui=%lf\nVr2=%lf\nAlpha2=%lf\nV2=
%lf\nVu2=%lf\n",W1,Vm2,V1,Vr1,Vrui,Vr2,Alpha2,V2,Vu2);


  /*******************************************************/
    getch();

        B3= 1.8 * W1;
        D3=1.15 * D2;
        Rv=(D2+D3)/4;
        Bx=B3+(2*(Rv-0.5*D2)/1.73);
        Vu3=V2*cos(ALPHA3);
        thita =log(D3/D2)/tan(Alpha2);
        Vthx=(D2*Vu2/D3);
        getch();
        printf("Values calculated are
\nB3=%lf\nD3=%lf\nRv=%lf\nBx=%lf\nVu3=%lf\nthita=%lf\nVthx=%lf\n",B3,D3,Rv,
Bx,Vu3,thita,Vthx);
        getch();
    return;
  }
```

```
                    /*Calculation of Lossess and NPSHR*/

#include<stdio.h>
```

```
#include<conio.h>
#include<math.h>
double  sai= 0.58 ;
double  fi = 0.175;
double  eta= 0.74211;
double  g  = 9.8;
double  ro = 1000;
double  pi = 3.14;
double sigma =0.66;
void main()
        {  double H,z,b2,N,Q,P,Ns,D2,U2,Psh,Dhb,Dsh,De,Vm1,D1,U1,B1,B2,Z,B,Ku;
        double V1,Vr1,Vm2,W1,Vrui,Vr2,Alpha2,V2,Vu2,C;
        double h1,h2,h3,h4,h5,h6,h7,h8,h9,h10,Re,Fd,Vth,Vd,S,Ath,HL,sigb,NPSHR;
        char ch;
        clrscr();

        printf("\n New session of calculating :: \n Following variables ::\n1-Ns\n2-U2\n3-
D2\n");
        printf("Values of the constants taken are :::\n ");
        printf("sai =%lf\nfi=%lf\n eta=%lf\ng=%lf\nro=%lf\n",sai,fi,eta,g,ro);
        printf("\nThe value  of Sigma=%lf",sigma);

        printf("\nPress any key to go ahead");
        getch();
        printf("For Calculation pls Enter following value");

        printf("\nH====");
        scanf("%lf",&H);

        printf("\nN====");
        scanf("%lf",&N);

        printf("\nQ====");
        scanf("%lf",&Q);


        Ns= (N * sqrt(Q))/pow(H,0.75);


        U2=sqrt((g*H)/(eta*sai));

        D2 = (60 * U2)/ ( pi * N);
        printf("\npress any key to get results");
        getch();
```

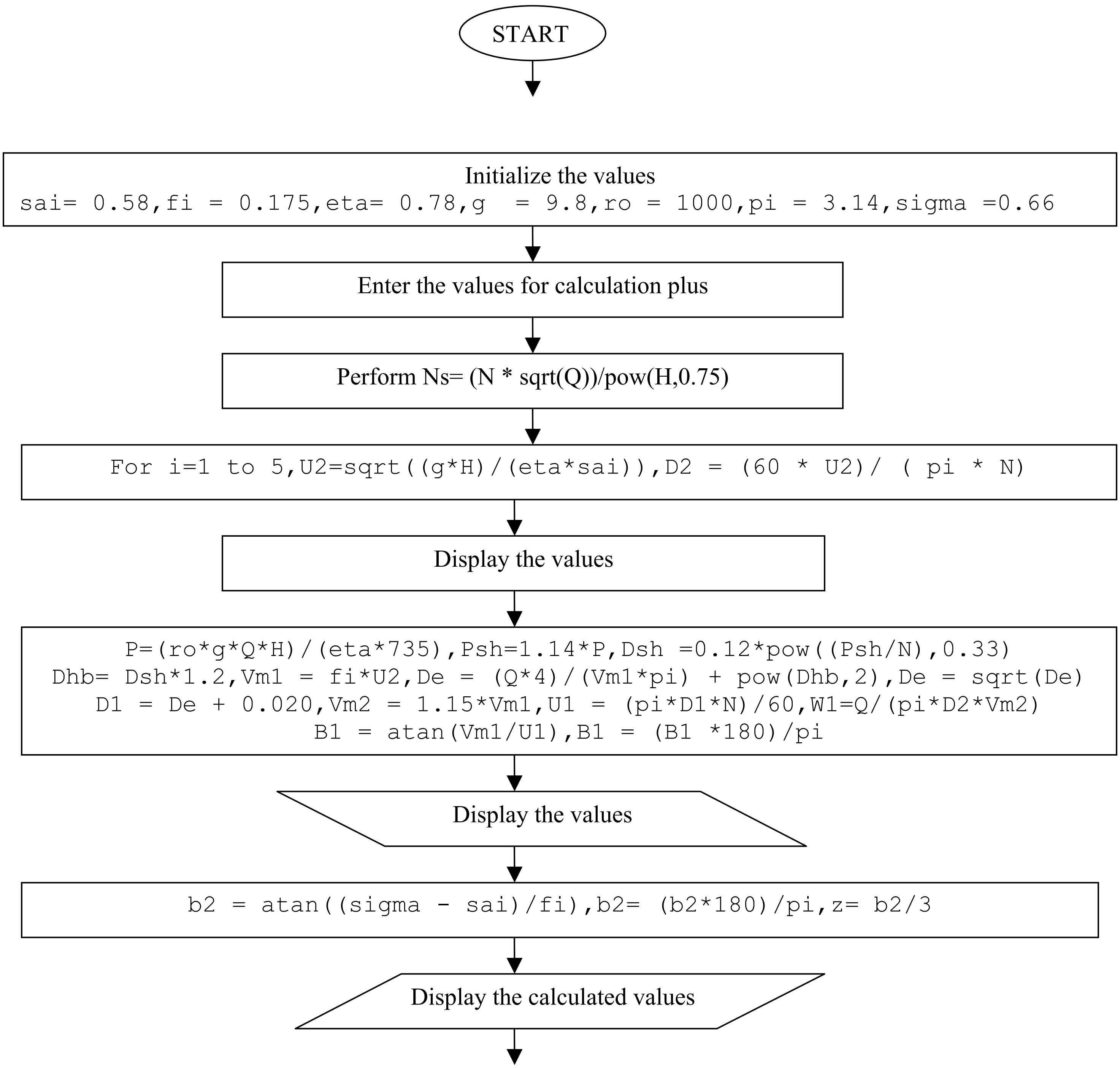
```
	printf("\nValues obtained are =>>\n1-Ns=%lf\n2-U2=%lf\n3-
D2=%lf\n",Ns,U2,D2);


  //	printf("\nDo you want more processing Y/N\n");

  //	scanf("%c",&ch);

	//	if(ch!='Y' && ch!='y')
  //	return;

	P=(ro*g*Q*H)/(eta*735);

	Psh=1.14*P;

	Dsh =0.12*pow((Psh/N),0.33);


	Dhb= Dsh*1.2;
   //	if(0.020<Dsh<=0.100)
   //	Dhb= Dsh +.016;
	Vm1 = fi*U2;

	De = (Q*4)/(Vm1*pi) + pow(Dhb,2);
	De = sqrt(De);

	D1 = De + 0.020;
	Vm2 = 1.15*Vm1;
	U1 = (pi*D1*N)/60;
	W1=Q/(pi*D2*Vm2);
	B1 = atan(Vm1/U1);
	B1 = (B1 *180)/pi;
	printf("\nValues calculated are =\n P=%lf\n Psh=%lf\n Dsh=%lf\nDhb=%lf\n
Vm1=%lf\n De=%lf\n D1=%lf\n U1=%lf\n
B1=%lf\n",P,Psh,Dsh,Dhb,Vm1,De,D1,U1,B1);
   //	 printf("Do you want more processing Y/N\n");
	//	scanf("%c",&ch);

 //	if(ch!='Y' && ch!='y')
  //	return;


	 b2 = atan((sigma - sai)/fi);

	 b2= (b2*180)/pi;
	 z= b2/3;
```

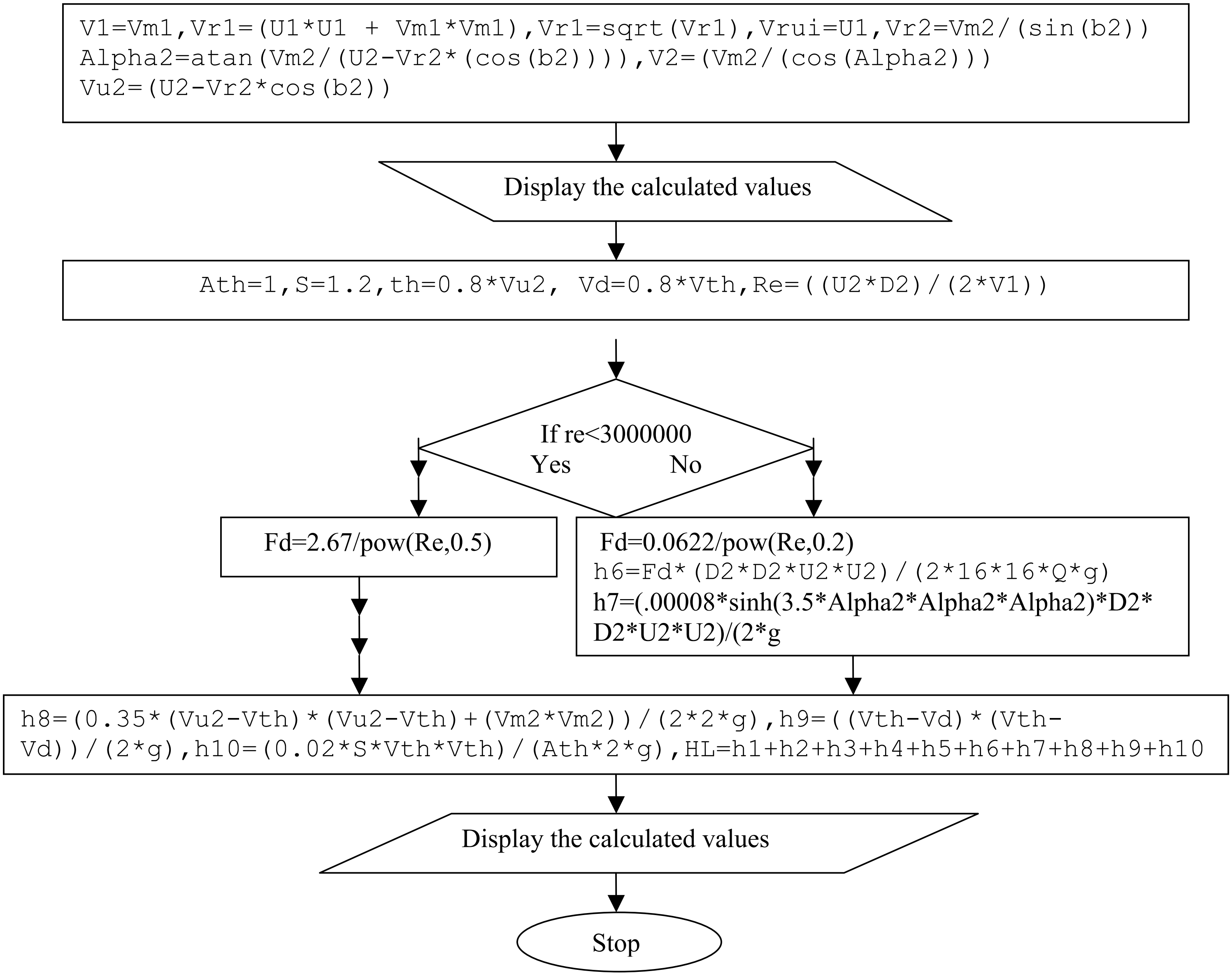
```
        printf("\nValues calculated are =\n B2=%lf\n z=%lf\n",b2,z);
    /****************************************************************/

    V1=Vm1;
        Vr1=(U1*U1 + Vm1*Vm1);
     Vr1=sqrt(Vr1);
     Vrui=U1;
     Vr2=Vm2/(sin(b2));
     Alpha2=atan(Vm2/(U2-Vr2*(cos(b2))));
     V2=(Vm2/(cos(Alpha2)));
        Vu2=(U2-Vr2*cos(b2));
     printf("Values calculated
are\nW1=%lf\nVm2=%lf\nV1=%lf\nVr1=%lf\nVrui=%lf\nVr2=%lf\nAlpha2=%lf\nV2=
%lf\nVu2=%lf\n",W1,Vm2,V1,Vr1,Vrui,Vr2,Alpha2,V2,Vu2);

    /**********************calculation of Losses***************/
    h1=0.2*((V1*V1/(2*g)));
    h2=0.5*((Vrui*Vrui)/(4*2*g));
    h3=0.05*((D2*D2*U2*U2)/g);
    h4=0.23*((Vr2*Vr2)/(2*g));
    h5=((V2*V2)/((2*2*g)*(1+(tan(Alpha2)*tan(Alpha2)))));
   /**************************************************************/

   Ath=1;
   S=1.2;
   Vth=0.8*Vu2;
   Vd=0.8*Vth;
   Re=((U2*D2)/(2*V1));
   if(Re<300000)
   Fd=2.67/pow(Re,0.5);
   else
   Fd=0.0622/pow(Re,0.2);
   h6=Fd*(D2*D2*U2*U2)/(2*16*16*Q*g);
   h7=(.00008*sinh(3.5*Alpha2*Alpha2*Alpha2)*D2*D2*U2*U2)/(2*g);
   h8=(0.35*(Vu2-Vth)*(Vu2-Vth)+(Vm2*Vm2))/(2*2*g);
   h9=((Vth-Vd)*(Vth-Vd))/(2*g);
   h10=(0.02*S*Vth*Vth)/(Ath*2*g);
   HL=h1+h2+h3+h4+h5+h6+h7+h8+h9+h10;
   C=(H-HL)*100/H;
   getch();
   printf("\ncalculated values
are=\n\nH1=%lf\nH2=%lf\nH3=%lf\nH4=%lf\nH5=%lf\nH6=%lf\nH7=%lf\nH8=%lf\n
H9=%lf\nH10=%lf\nHL=%lf\nEfficiency
=%lf\n",h1,h2,h3,h4,h5,h6,h7,h8,h9,h10,HL,C);
   /*******calculation of NPSHR********************/
   getch();
```

```
sigb=0.30;
NPSHR=(((1+sigb)*Vm1*Vm1)/(2*g))+((sigb*U1*U1)/(2*g));
printf(" \n Value of NPSHR =%lf",NPSHR);

getch();


        return;
 }
```

# ANNEXURE 2

# FLOW CHART OF LOSS MODELS





Flow chart of loss models in centrifugal pump