NUMERICAL STUDY OF PERFORMANCE OF CENTRIFUGAL PUMP FOR HANDLING ASH SLURRY

A dissertation
Submitted for the partial fulfillment of the requirement for the award of degree
of

MASTER OF ENGINEERING

In

CAD/CAM AND ROBOTICS

By

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CERTIFICATE

This is to certify that the work which is being presented in this dissertation entitled as **"NUMRICAL STUDY OF PERFORMANCE OF CENTRIFUGAL PUMP FOR HANDLING ASH SLURRY"** submitted by **Mr. Ashutosh Kumar** in partial fulfillment of the requirement for the award of degree of **MASTER OF ENGINEERING** in **CAD/CAM AND ROBOTICS** in the Mechanical Department, **THAPAR UNIVERSITY, PATIALA** is an authentic record of candidate's own work carried by him from January-2008 to June-2008 under the supervision and guidance of **Mr. Satish Kumar,** Lecturer, Mechanical Engineering Department, Thapar University, Patiala. The matter embodied in this dissertation has not been submitted anywhere else for the award of any other degree.

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ABSTRACT

Centrifugal pumps are used extensively for hydraulic transportation of solids over short to medium distance through pipelines where the requirements of head and discharge are moderate. A centrifugal pump designed to handle solid-liquid mixtures is normally single stage, end suction type having radial and mixed flow configuration to facilitate the motion of solid particles. Some of the special features of the centrifugal slurry pumps are larger flow passages to accommodate solid particles, robust impeller with lesser number of vanes, special seals and proper material of construction to ensure longer life. These have to be operated with relatively wide clearance at impeller-casing contacts to minimize choking and localized wear. These modifications increase the hydraulic losses in the pump and deteriorate the pump performance. During the design of pump handling abrasive slurries, the requirement of longer life and reliability have to be balanced by the constraint of high initial cost and efficiency.

The present works is concerned with the evaluation of the performance and wear characteristics of a centrifugal slurry pump. To achieve the above objectives the performance of the centrifugal slurry pump has been evaluated at different speed with clear water and with ash slurry at different solid concentration (by weight) with bed ash as solid material whose specific gravity is 2.65. At all the speed, the head developed and efficiency of the pump is found to reduce with increase in solid concentrations. The power input to the pump is also found to increase almost linearly with increase in specific gravity of slurry.

The wear characteristics of centrifugal slurry pumps are much influenced by various particle parameters like particle size, concentration and flow velocity of slurry etc. For this purpose design code has been written in C++. Thereafter, output of the program has been used for modeling of impeller and volute casing of pump and then numerical analysis is done on it to find out the wear rate for slurries of different concentration and solid particle size on the volute casing of the centrifugal pump for handling ash slurry.

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NOMENCLATURE

SYMBOLS	STANDS FOR
N_s	Specific speed
N	Impeller RPM
D_i	Impeller diameter
H	Total head developed
V	Velocity at periphery of impeller
g	Acceleration due to gravity
h_S	Static suction head
h_d	Static discharge head
h_f	Friction Head
h_{vp}	Vapor pressure head
h_p	Pressure head
h_v	Velocity head
H_S	Total suction head
H_d	Total discharge head
H_T	Total Differential head
N_s	Specific speed
$NPSH_a$	Net positive suction head available
$NPSH_r$	Net positive suction head required
Q	Flow rate
η_0	Efficiency
P_0	Output power
Pin	Input power
T	Torque
D_{hb}	Hub diameter
D_{sh}	Shaft diameter
U_2	Outlet blade velocity
φ	Pressure head generated / maximum Euler head
D_2	Outlet diameter

V_{m1}	Mean meridional velocity of steam just prior to blade inlet
V_{m2}	Mean meridional velocity at the exit of the impeller
D_e	Impeller eye diameter
U_2	Inlet blade velocity
D_2	Inlet diameter
β_1	Inlet blade angle
β_2	Outlet blade angle
B	Width of impeller
Z	No. of blades
Qt	Throat angle
B_3	Inlet Width of Volute
D_3	Dia. Of Inlet at Volute
V_{u3}	Whirl component of velocity at volute
V_{th}	Velocity at throat

CHAPTER 1

INTRODUCTION

1.1 Slurry

Slurry is essentially a mixture of solids and liquids. Its physical characteristics are dependent on many factors such as size and distribution of particles, concentration of solids in the liquid phase, level of turbulence, temperature, and absolute (or dynamic) viscosity of the carrier fluids. Nature offers examples of slurry flows such as seasonal floods that carry silt and gravel.

a) Non-settling slurry

A slurry in which the solids do not settle to the bottom, but remain in suspension for a long time. A non-settling slurry acts in a homogeneous, viscous manner, but the characteristics are non-Newtonian. A non-settling slurry can be defined as a homogeneous mixture.

Particle size: less than 60-100 µm.

Homogeneous mixture-

Homogeneous mixture is defined as the mixture of solids and liquid in which the solids are uniformly distributed.

b) **Settling slurry**

This type of slurry settles fast during the time relevant to the process, but can be kept in suspension by turbulence. Settling slurry can be defined as a pseudo-homogeneous or heterogeneous mixture and can be completely or partly stratified.

Particle size: greater than 100 µm.

Pseudo-homogeneous mixture-

Pseudo-homogeneous mixture is defined as the mixture in which all the particles are in suspension but where the concentration is greater towards the bottom.

Heterogeneous mixture-

Heterogeneous mixture is defined as the mixture of solids and liquid in which the solids are not uniformly distributed and tend to be more concentrated in the bottom of the pipe or containment vessel (compare to settling slurry).

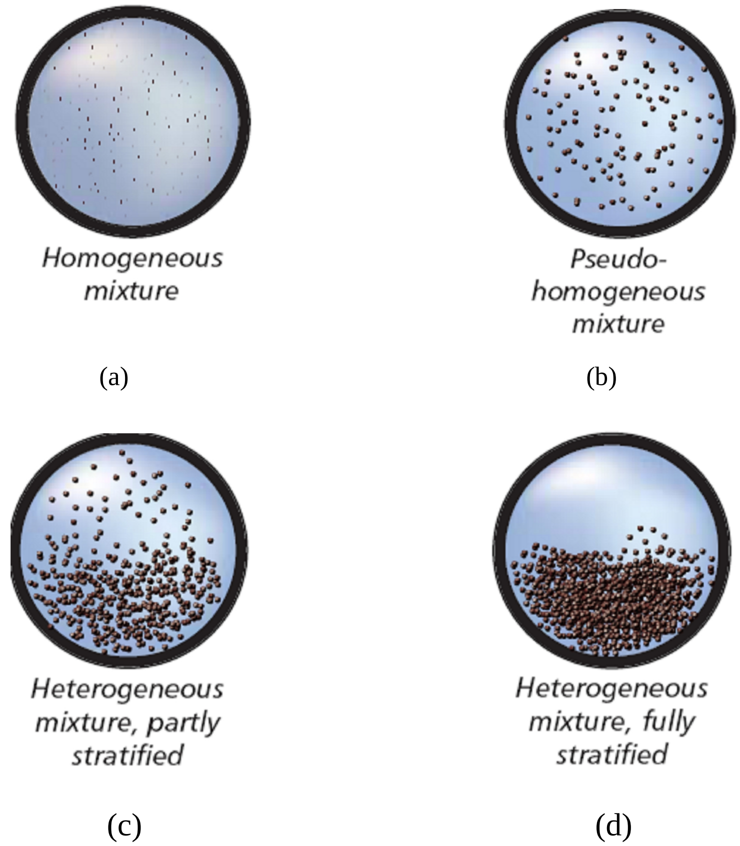


Fig. 1.1: Types of slurry

1.1.1 Ash

Ash is the residual by-products of coal burning to produce electricity. Power generation in India is primarily coal based in the present scenario. The coal used in our thermal power plants produce a large amount of ash, which is around 100 millions tons per year, at present. Out of this, around 80 millions tons will be fly ash and the remainder will be bed ash. It is expected that at the best 30% of the total ash produced will be utilized which will still leave the remaining 70% for safe disposal. Currently, both the type of ash are being mixed together and transported hydraulically from thermal power plant to ash ponds through pipelines at the solid concentration of around 10-20% by weight. The operation of ash disposal pipelines is highly uneconomical due to requirement of large quantity of water and increase in power consumption for pumping 80-90% of clear water. Further, bed ash is relatively coarse as compared to fly ash which is extremely fine. The transportation of the mixture of both the ashes at low concentration also require higher flow velocities and result in existence of highly skewed concentration profile across the pipe cross-section. This causes excessive erosion wear of the components of ash disposal system and therefore reduces their working life. Thus the design of existing ash disposal systems is highly conservative as it is based on the criteria of reliability with little or no emphasis on the optimization of various parameters like water and energy consumption, economic considerations etc. The present design procedures results in excessive wastage of water, energy and construction material of transportation equipment. It has become imperative to optimize the design methodology of these systems, as this procedure of ash handling will be going to remain the primary method for many more years to come. In order to optimize the design a detailed knowledge of the parameters affecting its operation is essential.

1.1.2 Types of ash

Ash slurry is mainly of two types:
Bed ash and fly ash, both are ash but having different properties like specific gravity, representative particle size, particle size distribution, static settling characteristics, pH value and rheological parameters. It remains homogenous and behaves Newtonian for certain maximum particle size and after that becomes heterogeneous and behaviour changes.

1.2 Centrifugal pumps used for transportation slurries

The choice of pumps or pumping systems for slurry transport will depend not only on the flow and head required, suction conditions, type of installation and location, as for any other pump application, but also on the slurry flow regime and properties, i.e. mixture concentration, apparent viscosity, particle size. Hence, the range may be quite wide or relatively limited, depending on the particular application involved.

Rotodynamic pumps, of which the centrifugal or radial-flow type is the most common in slurry service, are usually considered for the higher flow, lower head duties, whereas conversely, positive-displacement reciprocating types tend to be used for the lower flow, high pressure applications, e.g. long-distance pipelines. However, relatively high pressures may also be achieved with centrifugal pumps, depending on casing pressure limitations, by arranging them in series. For a given duty, centrifugal pumps are usually cheaper, occupy less space and have lower maintenance costs than positive displacement types, and can handle much larger solids.

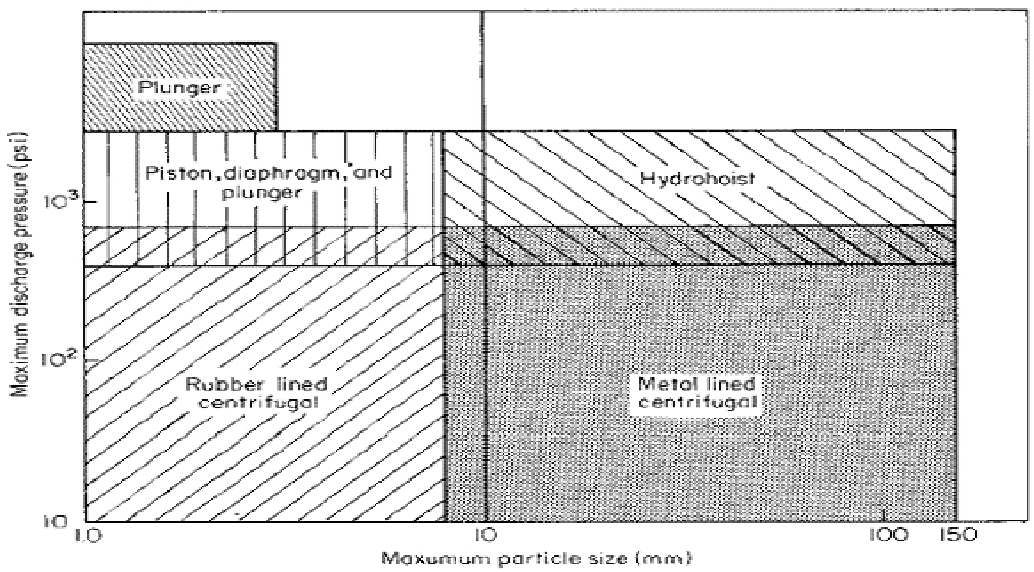


Fig. 1.2: Variation shown for max. discharge pressure vs max. particle size

Centrifugal pumps are used in a variety of applications, such as, water supply and irrigation, power –generating utilities, flood control, sewage handling and treatment, process industries, transporting liquid-solid mixtures. The list can go on and on. Because of their wide spectrum of application, centrifugal pump are manufactured in a variety of shapes and with a variety of performance characteristics.

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 as proposed by successful designers. When pump is designed, due concentration is focused on important factors like required head, max efficiency, a stable flow characteristics and non-cavitation performance of the pump. This is best suitable for a range of specific speed centrifugal pump. Same design is also suitable for the design of multistage centrifugal pump with few modifications. As the design of centrifugal pump involves a large number of interdependent variables, several other alternative design are possible for same duty. Hence, theoretical investigation supported by accurate experimental studies of the flow through the pump is required to investigate. Different authors have suggested different design procedure, method of calculation. The problem of calculation of the dimension of an impeller and thereby 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.

Generally; the pump is used to handle the clear liquids. The efficiency of a well-designed pump can be as high as 92%. Same pump, we are using for slurry transportation, but the performance characteristics of slurry pumps are littlie poor as compared to the conventional pump, because slurry pump efficiency has influence on various factors like size and concentration of the solid particles, abrasive property of the slurry, pumping pressures, pipe diameter, reactivity between the solids and the liquid and the surfaces, viscosity of the liquid, critical velocity and slurry properties.

Slurry pumps are similar to water pumps, so a lot of the available water pump technology can still be used. When it comes to determining pipeline velocities and heads, and the effect of slurries on performance, however, different studies including numerical modeling are found to be necessary.

Pump system evaluations require a completely different approach. Cost of ownership calculations for slurry pumps must consider wear, parts life, as they can be 50% of the total operating cost. When the whole system is considered, it needs to be evaluated in terms of the cost of transporting a given weight of dry solids over a given distance, not in water transported or in the efficiency of the pumps alone.

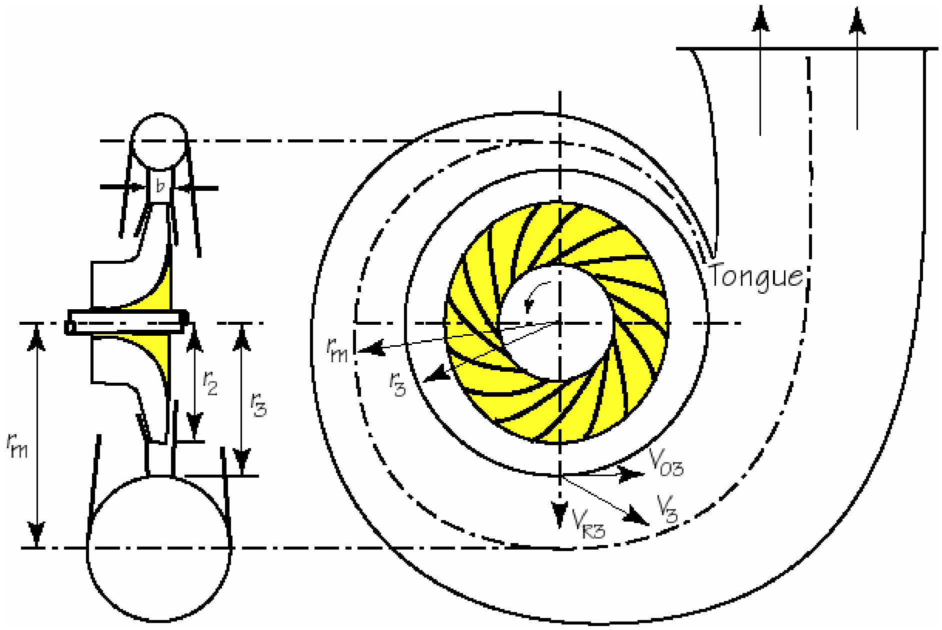


Fig. 1.3: Transportation of slurry

A designer needs to know the pump performance for handling the solid liquid mixture for design and operation of a slurry transportation system. It is difficult and time consuming to evaluate the performance characteristics of these pumps with all type slurries. Many investigators have proposed different correlation for the estimation of these correction factors which are dependent on solid concentration, pump design, flow rate and solid properties. These correction factors, in general, are function of change in flow field inside the pump due to suspended solids and additional losses due to solids.

1.3 Classification of centrifugal pump

Based on their utility, design and Constructional features, centrifugal pumps can be classified with respect to the following characteristics.

1.3.1 Working head

a) Low lift Centrifugal pumps: Low lift centrifugal pumps are meant to work against heads up to 15m, Impeller is surrounded by a volute and there are no guide vanes.

b) Medium lift: Medium lift centrifugal pumps are used to build up heads as high as 4m. They are generally provided with guide vanes.

c) High lift: High lift centrifugal pumps are employed to deliver liquids at heads above 40m, High pumps are generally multistage pumps because single impeller can not build up such a high pressure.

1.3.2 Type of casing

Pump casing should be so designed as to minimize the loss of kinetic head through eddy formation etc. Efficiency of the pumps largely depends on the type of casing.

a) Volute casing:

A casing operates on the principle of increasing the pressure energy in a free-vortex or spiral flow. In free-vortex, Angular momentum is constant

So, $mvr = cons\tan t$ and $\therefore r \propto 1/v$.

Volute casing cross-section of the moving stream gradually increases from torque towards the discharging pipe. This increase in area results in a gradually decrease in velocity (kinetic energy) with corresponding increases in pressure. Most of the single stage pumps are built with volute casing.

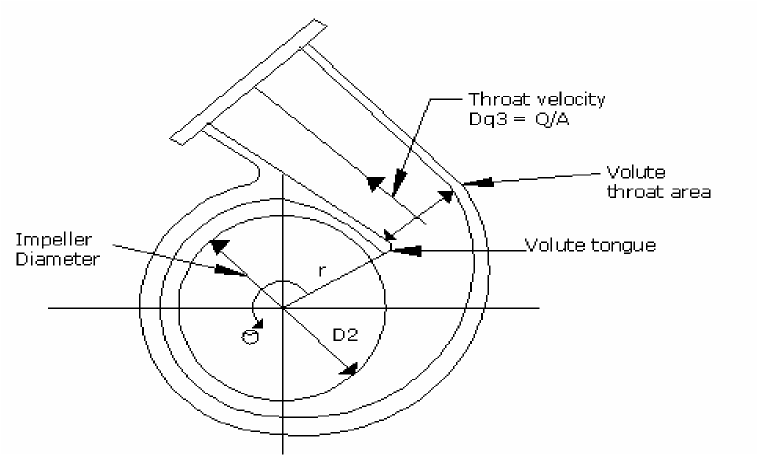


Fig 1.4: Volute casing

b)Volute with vortex chamber: Vortex or whirl pool chamber: Annular space is provided between the volute and impeller. This arrangement arrests the formation of eddies and gives an improved performance.

c) Diffusion pump: Impeller surrounded by a guide wheel consisting of a number of stationary vanes or diffuser providing outlets with cross-section gradually enlarging towards the periphery. Water emerging from impeller flow past the guide vanes and as the section across flow increases, velocity falls and pressure is build up. Angle of guide vanes at the entrance should coincide with the direction of absolute velocity of water at impeller outlet. This arrangement is employed in all multistage pumps.

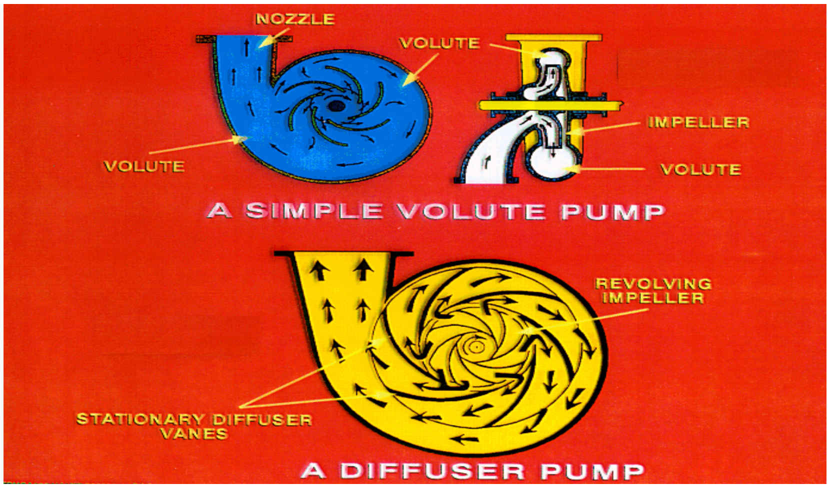


Fig 1.5: Diffuser pump

1.3.3 Number of stages

a) Single stage pump: It has one impeller keyed to the shaft. This is generally horizontal but can be vertical also. It is usually low lift pump.

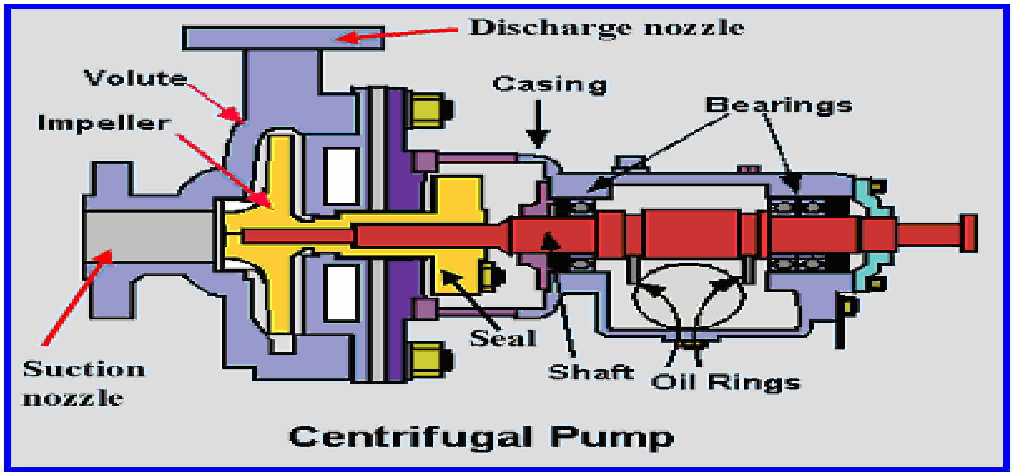


Fig. 1.6: Single stage pump

b) Multistage pump: Multi Stage Centrifugal Pump:-It has two or more impellers keyed to a single shaft enclosed in the same casing. Pressure is built up in steps. The impeller is surrounded by guide vanes and the water is led through a by-pass channel from the outlet of one stage to the entrance of the next until it is finally discharged into a wide chamber from where it is pushed on to the delivery pipe . These pumps are used essentially for high working heads and the number of stages depends on the head required

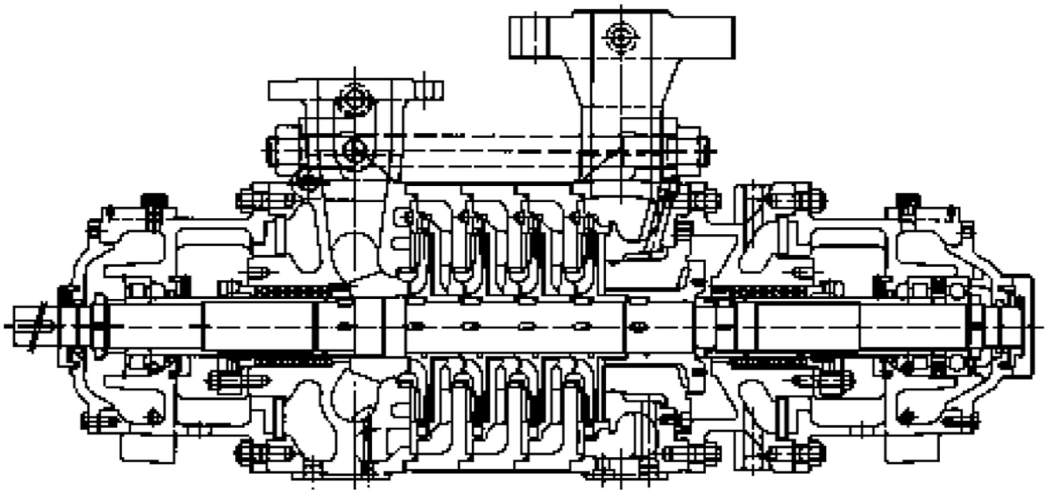


Fig. 1.7: Multistage pump

1.3.4 Relative direction of flow through impeller

a) Radial flow pump: It is that pump in which the liquid flows through the impeller in the radial direction only. Ordinarily all the Centrifugal pumps manufacture with radial flow impeller.

b)Mixed flow pump: : In mixed flow pumps the liquids flows through the impeller axially as well as radially i.e. there is a combination of radial and axial flows. A mixed flow pump is just a modification of radial flow type in this respect that the former is capable of discharging a large quantity of liquid.

c) Axial flow pumps: In axial flow pumps the impeller is in the axial direction only. Axial flow pumps are usually designed to deliver very large quantities of liquid at relatively low heads. However, it is not justified to call axial flow pumps as centrifugal pumps because there is hardly any centrifugal action in their operation.

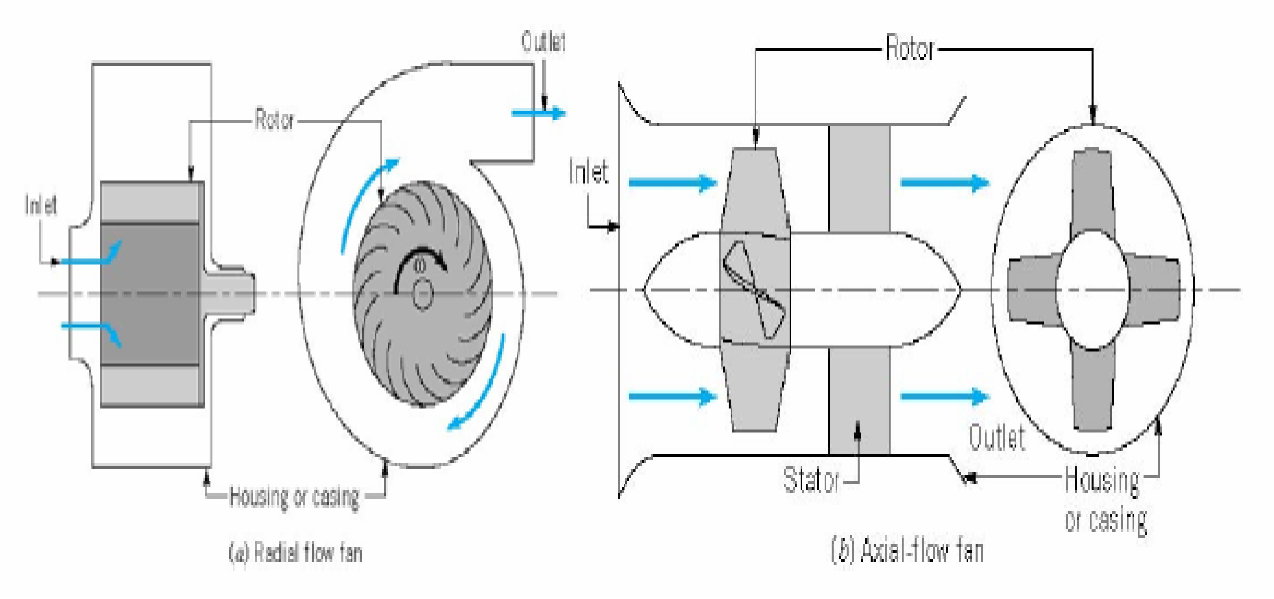


Fig. 1.8: Relative direction of flow through impeller

1.3.5 Number of entrance to impeller

a) Single entry or single suction pump: In a single suction pump liquid is admitted from a suction pipe on one side of impeller.

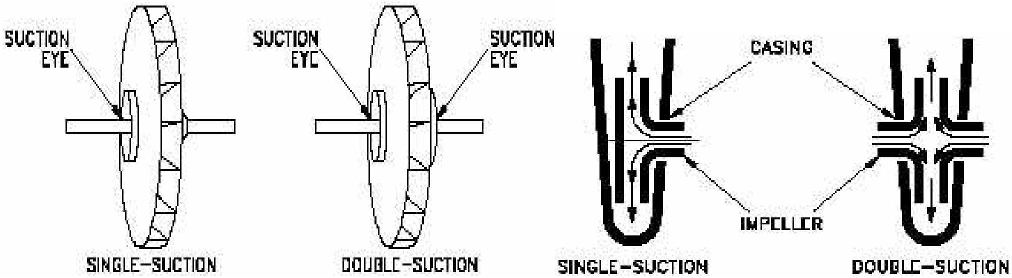


Fig. 1.9: No. of entrance to impeller

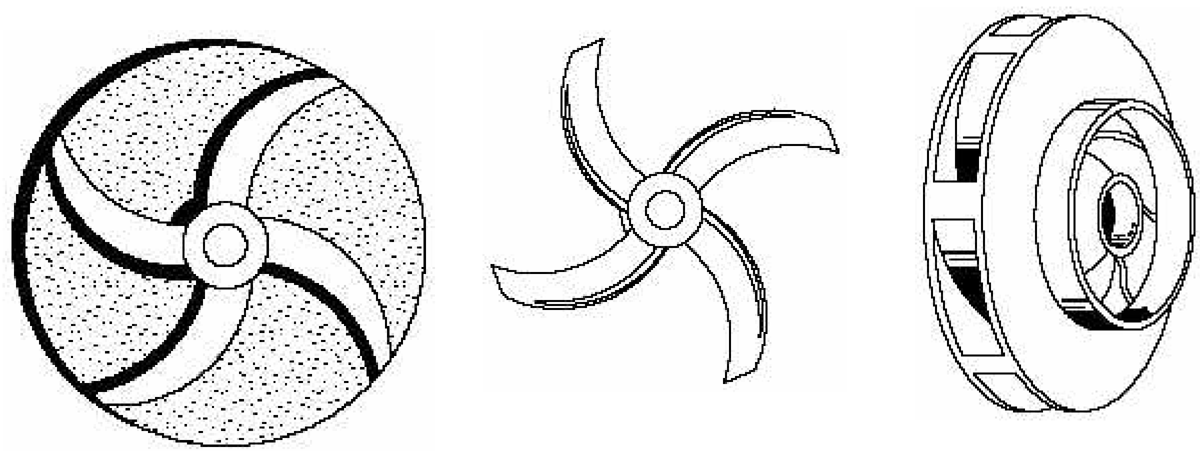
b) Double suction pump: In double suction pumps liquid enters from both sides of impeller. A double suction pump has an advantage that by this arrangement the axial thrust on the impeller is neutralized.

1.3.6 Liquid handled

Depending on the type and viscosity of liquid to be pumped, the pump may have a closed or open impeller.

a) Closed impeller:

An ordinary centrifugal pump is equipped with a closed impeller in which the vanes are covered with shrouds on both sides. This type is meant to handle non-viscous liquid such as ordinary water, hot water, hot oils and chemicals like acids etc, material of the impeller should be selected according to the chemical properties of liquid used. For hot water at temperature exceeding 150 degree celcius cast steel impeller is recommended.



(a) Semi-open impeller (b) Open impeller (c) Closed impeller

Fig. 1.10: Types of impeller

b) Semi open impeller: The impeller is provided with shroud on one side only. This pump is used for viscous liquid such as sewage; paper pulp etc, choice of material for manufacturer of impeller is influenced by chemical nature of liquid to be handled.

c) Open Impeller Pump: The impeller is not provided with any shroud. Such pumps are used in dredgers, and else where for handling mixture of water, sand, clay etc, it is generally made of forged steel.

1.3.7 Specific speed (N)

a) Slow speed radial flow runner- 10 to 30 rpm
b) Normal speed radial flow impeller- 30 to 50 rpm
c) High speed radial flow impeller- 50 to 80 rpm
d) Mixed flow runner- 80 to 160 rpm
e) Axial flow runner- 110 to 150 rpm

1.3.8 Outlet blade angle

a) Backward Blade: Outlet blade curves in a direction opposite to that of motion , & the angle between the blade tip & the tangent to rotor at exit is below 90 (β_2<90)

b) Radial Blade: Liquid leaves the blade with relative velocity in a radial direction & angle β_2=90

c) Forward Blade: Outlet tip of blade curves in the direction of motion & the angle between blade tip & the tangent to rotor at exit is obtuse (β_2>90)

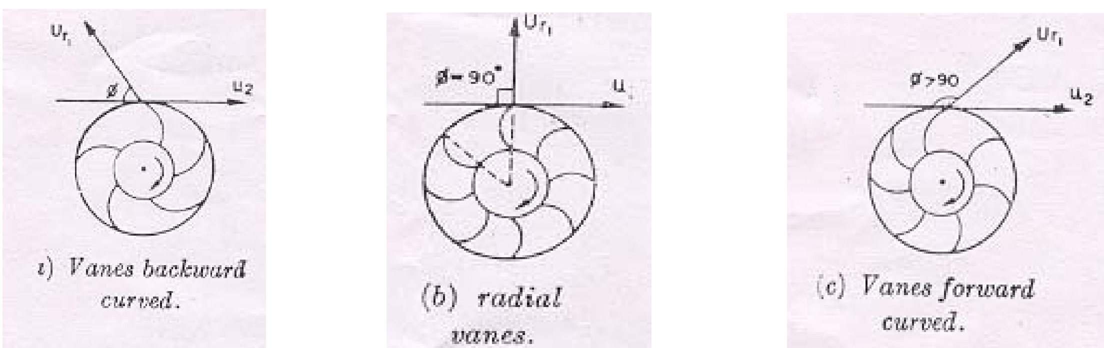
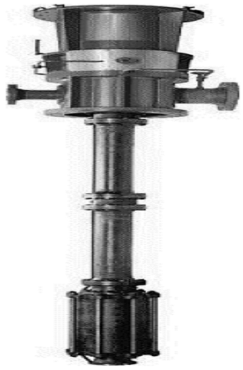


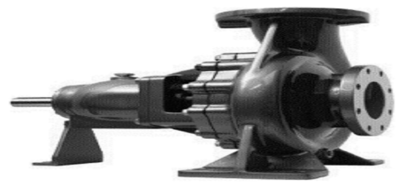
Fig. 1.11: Outlet blade angle

1.3.9 Depending on the position of impeller

a) Horizontal impeller shaft pump
b) Vertical impeller shaft pump

The centrifugal pumps may be designed with either horizontal or vertical position of shaft, generally the pumps are provided with horizontal shaft. For deep wells and mines the pumps with vertical shaft are more suitable because the pumps with vertically disposed shaft occupy less space.





(a) Horizontal shaft pump (b) Vertical shaft pump

Fig. 1.12: Position of impeller

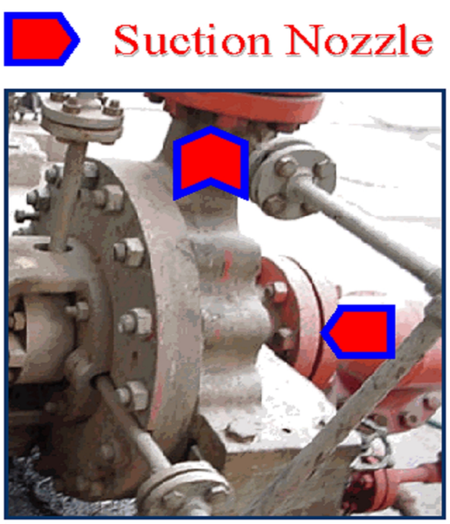
1.3.10 Suction and Discharge Nozzle

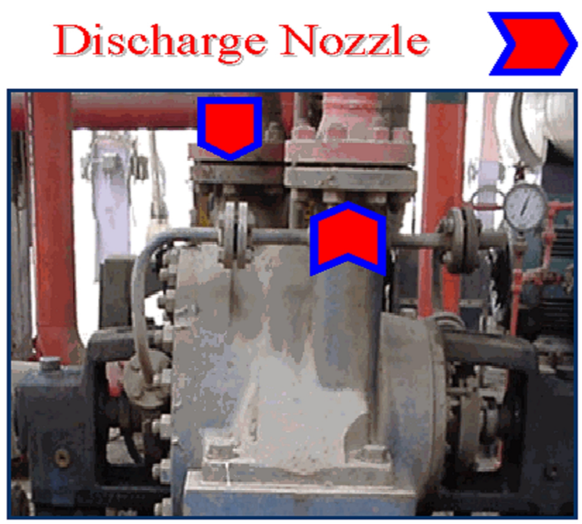
a) End suction/Top discharge –

This pump is always of an overhung type and typically has lower NPSHr because the liquid feeds directly into the impeller eye.

b) Top suction/Top discharge -

This pump can either be an overhung type or between-bearing type but is always a radially split case pump.





a) End suction/Top discharge b) Top suction/Top discharge

Fig. 1.13: Position of suction & discharge

1.4 Construction Details

1) Accelerating nozzle and inlet guide vanes

The flow enter a three dimensional impeller through an accelerating nozzle and a row of inlet guide vanes .the inlet nozzle accelerates the flow from its initial condition to the entry of inlet guide vanes. The inlet guide vanes direct the flow in the desired direction at the entry of the impeller.

2) Impeller

The impeller through its blades transfers the shaft work to the fluid and increases its energy level. It can be made in one piece consisting of both the inducer section and a largely radial portion the inducer receives the flow between the hub and the tip diameters of impeller eye and passes it on to the radial portion of the impeller blades

Fig.1.14: Impeller

The impeller is mounted on a shaft with bearings so that it can be rotated inside the casing. In majority of centrifugal compressors, the impeller has straight radial blades after the inducer section. Note that where the rotating shaft enters the casing, it must be sealed against leakage of liquid.

3) Diffuser

The impeller discharges the flow in to the diffuser through a vane less space. Here the static pressure of the liquid rises further on the account of the deceleration of the flow. The diffuser may be merely a vane less space or may consist of blade ring. For high performance, the design of diffuser is as important as that of the impeller.

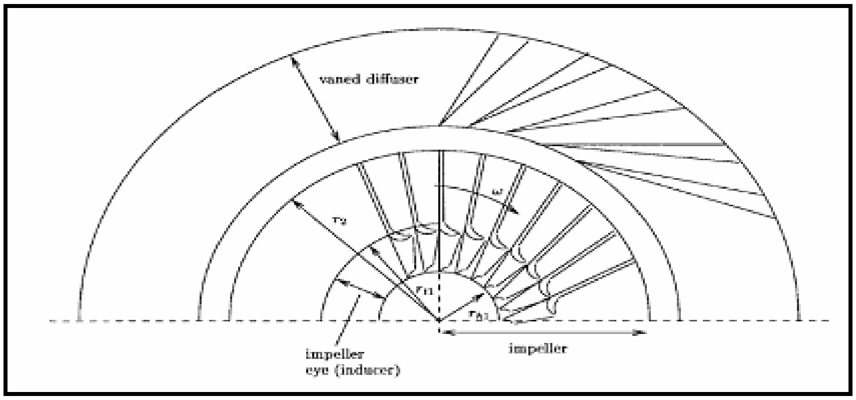


Fig.1.15: Diffuser

4) Shafts and bearings

Compressor shafts are designed to support the impeller on one end, overhung from the bearings or between the bearings. The overhung design allows straight liquid flow into the impeller, but results in greater radial load on the bearings. The overhung design

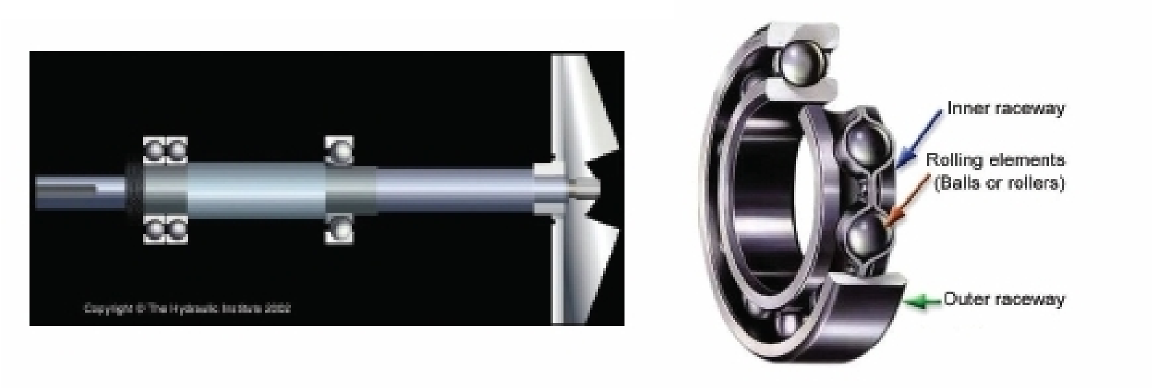


Fig. 1.16: Shaft and bearing

eliminates one seal around the shaft and simplifies the construction of the casing. Some seal less compressor designs use a stationary shaft or spindle on which the impeller and drive element rotate. Rolling element bearings are most commonly used to support centrifugal compressor shafts. Single row or double row ball bearings are good for carrying radial and axial loads. Roller bearings are good for carrying high radial.

5) Seals and packing

In order to seal rotating shafts against leakage of the pumped liquid, soft packing is often used. Such packing is usually made of braided fibers impregnated with graphite or other lubricating material. The packing is retained by a gland which can be tightened to squeeze the packing close to the shaft. During operation, some leakage of liquid is necessary to keep packing from overheating.

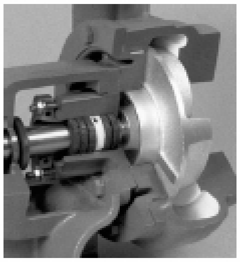


Fig. 1.17: Seal

6) Volute Casings

Compressor casings collect the liquid from the impeller, convert the velocity energy to pressure energy, and guide the liquid to the compressor discharge nozzle. This casing shows a dual (double) volute which helps balance the hydraulic forces.

CHAPTER 2

LITERATURE REVIEW

Pullum et al.[1] **[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.

Min-Guan, Y. et al.[2] **[2007]** have observed the phenomena of two-phase flow with salt crystallizing in the chemical pump, the 3-D turbulent flow in the impeller of chemical pump was simulated at the condition of rinsing. The internal flow between the impellers of chemical pump was investigated. Based on the Reynolds-averaging N-S equations and the standard k-ε two equations turbulent model, the simulations of turbulent flow between the impellers were performed using the flow computing software Fluent under different operating conditions. Based on the analysis of the calculated results of velocity and pressure profiles in the chemical pump and experimentally observed phenomenon of flow impact, secondary flow and recirculation, some design improvements were proposed, which give suggestions on the optimal design and internal two-phase flow study of the chemical pump.

Addie et al [3] **[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.

Addie et al.[4] **[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.

Kadambi et al.[5] [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.

Engin et al.[6] [2003] presented a new correlation to predict the head reduction of centrifugal pumps when handling slurries. The proposed correlation takes into account the individual effect of particle size, particle size distribution, specific gravity and concentration of solids, and impeller exit diameter on the pump performance. The correlation developed provides remarkably closer approximation with the experimental data compared to all others for most of experimental data used. Overall it produces only a mean deviation of 8.378% and an average deviation of 0.620%.

Goto Akira et al.[7] [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.

Stephan Bross et al.[8] [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.[9] **[2002]** through experimental investigation of pump characteristics at different rotational speeds concluded 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.

M C Roco[10] **[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[11] **[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.[12] **[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.

Gandhi et al.[13] **[2001]** also reported 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

(1) 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.

(2) At low solid concentration (<30% by weight), the increase in the pump input power directly proportional to the specific gravity of slurry whereas the same relationship is not applicable at higher concentrations.

Gandhi et al.[14] [2000] presented a methodology based on a loss analysis procedure to predict the performance of centrifugal slurry pumps handling solid-liquid mixtures. Various energy-head losses in the pump have been estimated using analytical models based on pump geometry and properties of suspended solids. The predicted values of head over the operating range of flow rates of the pump have been found to be in reasonable agreement with the measurements made at various rotational speeds and solid concentrations.

Sellgren et al [15] [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.

Ni et al.[16] [2000] have experimentally evaluated the effect of high delivered volumetric concentration (C_{vd}) on characteristics of a slurry pump. They performed extensive experiments by using three sorts of narrowly graded sands tested in the laboratory of DN150 mm pump loop of the Delft University of technology, for the observation of pump and pipeline characteristics. They conclude that high solid concentration has a strong influence on the pump head, efficiency and power consumption and this influence behaves differently with different sand size. The pump efficiency in coarse sand slurry service may drop almost 60% compared to that of water service, when c_{vd} = 42%. Within the measured range of concentrations in each passages may experience similar stratification process occurred in pipelines. The mechanical friction regime in the impeller passages could be similar to that in pipelines. Therefore the delivered volumetric concentration and the size affect the head loss in the same way both in pumps and pipelines.

S. Gopalakrishnan[17] [1999] has 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[18] [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.

S.Yedidiah[19] [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[20] [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.

Walker et al.[21] **[1993]** have investigated the effect of impeller geometry on the performance of centrifugal slurry pump with sand slurries of two sizes. They found that except for the vane shape, other vane parameters do not have any significant effect on either the head or the efficiency.

Gahlot et al.[22] **[1992]** investigated the performance of two 50 mm centrifugal slurry pumps made of two different materials namely Ni-hard(metal) and rubber-lined having closed and open impellers respectively with coal and zinc tailings slurries at 1450 rpm in the concentration range of 0-57% by weight. 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 as they are fairly independent on the pump flow rate. They observed higher reduction in the head compared to the efficiency and have shown linear decrease in the head and the efficiency ratios with increase in solid concentration upto 50% by weight. For higher solid concentrations, relatively lager drop in both the head and efficiency were observed.

Sellgren et al. [23] **[1990]** have reported that the weighted mean diameter is a better representative particle size for multi-sized slurries as compared to the median particle diameter for analyzing the pump performance. He has also attempted to correlate the drop in the head for different pumps to normalized impeller diameter and vane width.

Sellgren and Vappling[24] **[1986]** have investigated the effect of solid concentration for two tailings materials on the performance of a 100 mm centrifugal slurry pump having closed impeller. They observed fall in the head and the efficiency with increase in solid concentrations, the effect being more for efficiency at higher concentrations as compared to head. They attributed this phenomenon to the increased disc friction losses caused by fine particulate slurries and to the break-down of boundary layers in the flow through the impeller due to coarse sized particles. Based on existing data on the pump performance in the literature and their own data.

Eschells[25] [1986] has suggested that thick slurries may behave as thin liquids due to highly turbulent flow conditions existing inside the pump. He, however, has not given any method to determine the effective slurry viscosity for estimating the pump performance.

Roco et al.[26] [1986] have experimentally evaluated the head-capacity characteristics of few centrifugal slurry pumps of different geometrical configurations handling silica sand slurries. They reported that different head losses in the pump vary differently with the solid properties and flow velocities, and hence the effect of suspended solids on individual losses has to be evaluated separately to obtain the performance of the pump in solid-liquid mixture flows.

Chand et al.[27] [1985] have investigated the effect of solid particles and drag reducing polymers on the centrifugal slurry pump performance. They performed extensive experiments using various materials such as fly ash, sand, iron ore and coal dust at concentrations up to 26% by volume, by running the pump at three different speeds namely 1000, 1200 and 1400 rpm. Unlike the observations of other investigators, they have reported improvements in head and efficiency of the pump with increase in concentrations of solid as well as polymer additives.

Sellgren[28] [1979] has suggested that the head loss also depends on the basic mineral structure and chemical composition of the solid particle in addition to their size, shape and distribution.

Burges and Riezes[29] [1976] have experimentally evaluated the pump performance for slurries at various pump speeds and supported the applicability of affinity laws. They found that the head loss is a function of particle size, solid concentration and specific gravity of solids. They attributed the measurements of lower loss values for the efficiency as compared to the head to the uncertainties in the power measuring instruments. Based on his experiments on a rubber-lined slurry pump

Vocaldo et al.[30] [1974] have experimentally investigated the performance of rubber-lined and metal centrifugal slurry pumps handling sand material of different particle sizes and clay at two speed of 1180 and 1780 rpm. The head loss for rubber-lined pump was found to be higher than that for the metal pump which they attributed to the higher absorption of particle kinetic energy by the lining. The loss in the head and efficiency of the pump was found to be equal at various solid concentrations. However in the case of clay suspension of 8% by volume. Affinity laws were also found to be applicable for these pumps handling slurries.

Pagalthivarthi et al.[31] [1998] presented a 2-dimensional computer program based on Galerkin finite element method to solve two dimensional turbulent flow in a centrifugal slurry pump. A mixing length model was used for turbulent stresses. Through this program they found 2.5% difference in inflow and outflow and predicted the recirculation in casing when flow rate exceeds the design flow.

Sun and Tsukamoto[32] [1990] have studied pump off-design performance using the commercial software Fluent. They also predicted reverse flow in the impeller shroud region at small flow rates. They validated the predicted results of the head-flow curves, diffuser inlet pressure distribution and impeller radial forces by comparing with the experimental data over the entire flow range. They observed back flow at small flow rates, however no back flow was observed at higher flow rate.

Walker and Goulas[33] [1984] developed a computer program for selection of centrifugal slurry pumps by assuming that the head developed, the input power and NPSH of the pump could be represented in the form of polynomial of the flow rate. They have also suggested separate procedure for correcting the pump performance for settling and non-settling slurries.

Roco and Reinhart[34] [1980] presented a numerical method for calculating the distribution of solid particles concentration in centrifugal slurry pump impeller. They used finite element technique to solve the convection-diffusion differential equation between blades. By used of this method they were predicted best optimum impeller design. They found that this method is valid for any form and number of impeller blades

CHAPTER 3

DESIGN OF CENTRIFUGAL PUMP

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 Stepanoff have been carefully studied. Each design parameter has been calculated using above procedures and an appropriate value adapt for present carefully analyzing the calculated values.

3.1 Design problem

<u>Input data</u>

Head = 36m   Flow Rate = .090m^3/sec

Speed = 1450 rpm

3.3 Design of impeller

1. 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.

$$N_s = \frac{N\sqrt{Q}}{H^{0.75}}$$

Where N = speed at pump shaft rotated.

Q = discharge in m^3 / sec

H = net head in m.

For given data,

$$N_s = \frac{1450x\sqrt{.090}}{(36)^{3/4}} = 29.598rpm$$

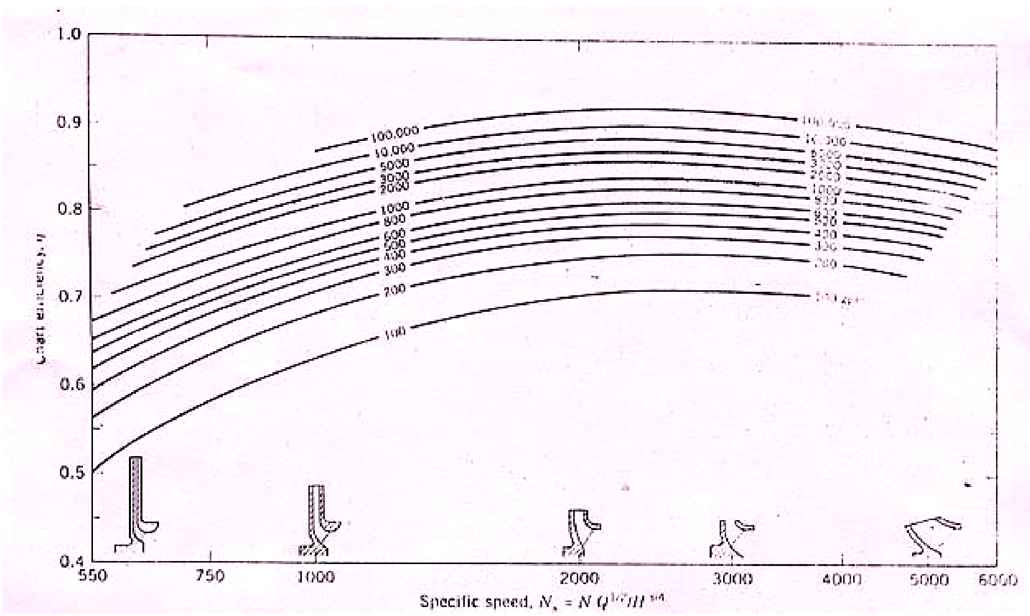


Fig 3.1: Efficiency Vs Specific Speed

Overall efficiency taken from the graph is taken as 80%, for specific speed = 29.598.

2. Power input and shaft input power

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

$$\text{Input power, } P_{in} = \left(\frac{\rho gQH}{745\ \eta_0}\right) \tag{3.4}$$

Where,

g= Acceleration due to gravity

ρ=Density of the fluid

$$\text{For given data input power} = \frac{1000x9.81x.090x36}{745x.80}$$

$= 53.32$ Hp

$P_{sh} = 1.15 \text{ x } 89.54$

$= 61.329$ Hp

3. Shaft diameter:

According to Stepanoff,

Torque

T = (P * 60 / 2ЛN) N m

$T = Л/16\ Fs\ d_{sh}^3$

Fs = stress depend the material constant

$d_{sh} = (16T/Л\ Fs)^{1/3}$

$$\text{for given data } d_{sh} = \left(\frac{16x60.572x60}{2x3.14x1450x1.5}\right)$$

$= 0.0362$ m

4. Hub diameter

The diameter of the hub is given as:

$D_h = 1.2 \text{ x } D_{sh}$

for given data $D_{hb} = 1.2 \text{ x } .0361 = 0.0434$ m

5. Outlet blade velocity (U_2)

Head coefficient

φ = pressure head generated / maximum Euler head

Generally,

φ = 0.5 to 0.6

so, φ has been taken as 0.58 for design calculation

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

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

For given data $U_2^{\,2} = \left(\dfrac{9.8 \times 36}{.80 \times .58} \right)$

$= 27.58$ m/sec

6. Outlet diameter (D_2)

$U_2 = Л\, D\, N / 60$

$D_2 = (60U_2 / ЛN)$

for given data $D_2 = \left(\dfrac{60x27.58}{3.14x1450} \right)$

$= 0.363$ m

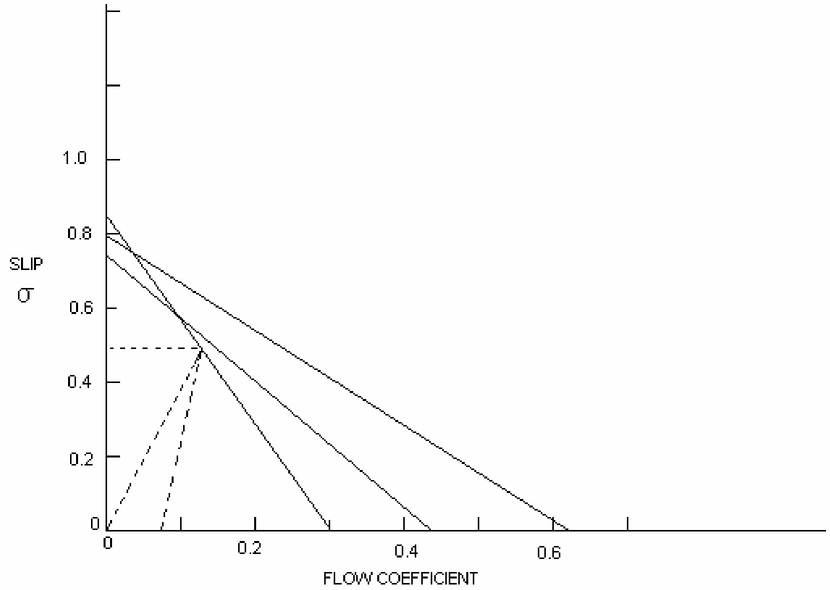


Fig 3.2: Slip Vs Flow coefficient

7. Flow coefficient

$\Phi = V_{m2} / U_2$ = Flow velocity / Blade velocity

Φ taken 0.1 to 0.2

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

Mean meridional velocity of the steam just after entering the blade passage is denoted by V_{m1} 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 $\sqrt{2gh}$ is known as capacity constant.

For given data $V_{m1} = .175 \times 27.58$

$= 4.827$ m/sec

$V_{m2} = 1.15 \times 4.827$

$= 5.521$ m/sec

8. Impeller eye diameter (D_E)

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

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

$$D_e^2 = [.02388 + .0499^2]$$

$= 0.160$ m

9. Inlet diameter (D_1)

Nyiri used the following relationship for the inlet diameter,

$D_1 = (D_e + 0.020)$ m

For given data,

$D_1 = 0.160 + .020$

$= 0.180$ m

10. Inlet blade velocity (U_1)

U_1 = (Л D_1 N / 60)

For given data U_1

$$= \left(\frac{3.142x.180x1450}{60}\right)$$

= 13.67 m/sec

11. Inlet blade angle (β_1)

Fluid at inlet assumed no pressure whirl

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

for given data $\beta_1 = \tan^{-1}\left(\frac{4.797}{13.67}\right)$

= 19.44

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

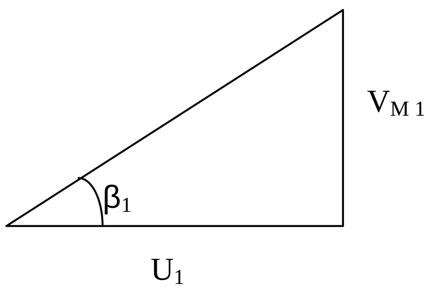


Fig. 3.3: Inlet velocity triangle

12. Width of impeller (B)

According to Stepanoff,

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

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

$$B_2 = \frac{0.090}{(\pi x.361x27.58x.175)} = 0.0163 \text{ m}$$

13. Blade angle at outlet (β_2):

Kovats recommends the equation

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

$B_2^{'} = 1.2 \text{ x } B_2$

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

Peck recommended

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

For given data $36 = \dfrac{27.58^2}{g} - \dfrac{27.58x5.521}{g \cdot \tan B_2^{'}}$

$B_2 = 24.69$

14. Number of blades (Z)

Number of blades generally taken between 5 to 12. According to Pleider

$Z = (B_2 / 3)$.

For given data Z = (24.69/3)

= 8.23

from inlet velocity triangle

$V_1 = V_{m1} = 5.15$

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

$= 13.67^2 + 4.827^2$

$= 14.50$ m/sec

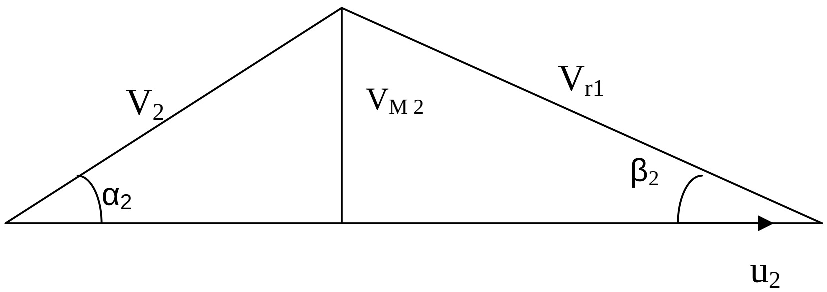


Fig. 3.4: Outlet velocity triangle

$V_{rui} = U_1 = 13.8423$ m/sec

From outlet velocity triangle

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

$= 5.521/\sin 24.69$

$=13.21$ m/sec

3.3 Design of volute casing

3.3.1 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 elements 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.

3.3.2 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 α_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

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

$$\Phi = \text{Angle in radians}$$

1. Throat Angle

Angle at which the throat of the volute start

Throat angle Qt = $\ln (R_t / R_2) / \tan \alpha_2$

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

2. Inlet width of volute

B_3 / B_2 = 1.4 to 1.8 (for high Ns)

= 2 (for low Ns)

3. Diameter of Inlet at Volute

$D_3 = 1.1$ to $1.2\ D_2$

4. Width of volute at any point

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

Q_A = Taken 60°

X = Distance between any radius R and impeller outside radius

$R_2 = (R_{av} - R_2)$

5. Velocity at throat

$V_{th} = (D_2 \times V_{u2})/D_3$

For given design problem

3.3.3 Volute design calculation

Inlet Width of Volute $B_3 = 1.8\ B_2$

$= 1.8 \times 0.0165$

$= 0.0293$ m

Dia. Of Inlet at Volute

$D_3 = 1.15 D_2$

$= 1.15 \times 0.361$

$= 0.415$ m

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

$$= \left(\frac{0.361 + 0.415}{4}\right)$$

$= 0.195$ m

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

$$= 0.0297 + \left(\frac{2x0.194 - 0.361}{1.73} \right)$$

$$= 0.0451 \text{ m}$$

Whirl component of velocity at volute

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

$$= 5.521 \cos 8.3$$

$$= 5.494 \text{ m/sec}$$

Throat angle Qt = ln (0.2075 / 0.18) / tan 8.3

= 34.25°

Velocity at throat V_{th} = (0.361 x 39.03)/0.415

= 33.93 m/sec.

CHAPTER-4

MODELING OF CENTRIFUGAL PUMP

4.1 Introduction of CFD

Computational Fluid Dynamics (CFD) is the analysis of systems involving fluid flow, heat transfer and associated phenomena such as chemical reactions by means of computer-based simulation. The heat and mass transfer, fluid flow, chemical reaction, and other related processes that occur in engineering equipment, in the natural environment, and in living organisms play a vital role in a great variety of practical situations. Nearly all methods of power production involve fluid flow and heat transfer as essential processes. The same processes govern the heating and air conditioning of buildings. Major segments of the chemical and metallurgical industries use components such as furnaces, heat exchangers, condensers, and reactors, where thermo-fluid processes are at work. Aircraft and rockets owe their functioning to fluid flow, heat transfer, and chemical reaction. In the design of electrical machinery and electronic circuits, heat transfer is often the limiting factor. The pollution of the natural environment is largely caused by heat and mass transfer, and so are storms, floods, and fires. In the face of changing weather conditions, the human body resorts to heat and mass transfer for its temperature control. The processes of heat transfer and fluid flow seem to pervade all aspects of our life. Since the above processes have such an overwhelming impact on human life, we should be able to deal with them effectively. This ability can result from an understanding of the nature of the processes and from methodology with which to predict them quantitatively. The designer of an engineering device can ensure the desired performance, if the power of prediction enables him. Predictions of the relevant processes help us in forecasting, and even controlling, potential dangers such as floods, tides, and fires. In all these cases, predictions offer economic benefits and contribute to human well-being. The prediction of behavior in a given physical situation consists of the values of the relevant variables governing the processes of interest. Let us consider a particular example. In a combustion chamber of a certain description, a complete prediction should give us the values of velocity, pressure, temperature, concentrations of the relevant cherI1ical species, etc., throughout the domain of interest; it should also provide the shear stresses, heat fluxes, and mass flow rates at the confining walls of the combustion chamber. The prediction should state how any of these quantities would change in response to proposed changes in geometry, flow rates, fluid properties, etc

4.2 Mathematical model

The starting point of any numerical method is the mathematical model, i.e. the set of partial differential or integro-differential equations and boundary conditions. One chooses an appropriate model for the target application (incompressible, inviscid, turbulent; two- or three dimensional, etc.). The numerical model may include simplifications of the exact conservation laws. A solution method is usually designed for a particular set of equations. Trying to produce a general purpose solution method, i.e. one which is applicable to all flows, is impractical, if not impossible and, as with most general purpose tools, they are usually not optimum for anyone application.

The physical aspects of any fluid flow are governed by the following three fundamental Principles:

1. Conservation of mass;
2. Newton's second law; and
3. Energy conservation

These fundamental principles can be expressed in terms of mathematical equations which in their most general form are usually partial differential equations. CFD a art of replacing the governing partial differential equations of fluid flow with numbers and advancing these numbers in space and or time domain to obtain a final description of complete flow field of interest. With the advent of high-speed digital computers, CFD has become a powerful tool to predict flow characteristics in varied problem, in an economical way.

4.3 History of CFD

Computers have been used to solve fluid flow problems for many years. Numerous programs have been written to solve either specific problems, or specific classes of problems. From the mid-1970's, the complex mathematics required to generalize the algorithms began to be understood, and general purpose CFD solvers were developed. These began to appear in the early 1980's and required what were then very powerful computers, as well as an in-depth knowledge of fluid dynamics, and large amounts of time to set up simulations. Consequently, CFD is a tool used almost exclusively in research.

Recent advances in computing power, together with powerful graphics and interactive 3D manipulation of models have made the process of creating a CFD model and analyzing results much less labour intensive, reducing time and, hence, cost. Advanced solvers contain algorithms which enable robust solutions of the flow field in a reasonable time. As a result of these factors, Computational Fluid Dynamics is now an established industrial design tool, helping to reduce design time scales and improve processes throughout the engineering world. CFD provides a cost-effective and accurate alternative to scale model testing with variations on the simulation being performed quickly offering obvious advantages.

4.4 CFD METHODLOGY

4.4.1 Initial design

In order to obtain better design in CFD, following procedure is applied so that fluid flow can easily be modeled. Initial design of the model is a planning decision and the geometry is generated depending on these initial design considerations, using either CFD modeling tools or other Design tools.

4.4.2 Geometry generation

The first task to accomplish in a numerical flow simulation is the definition of the geometry, followed by the grid generation. This step is the most important step for the study of an isolated impeller assuming an axis symmetric flow simplifies the domain to a single blade passage.

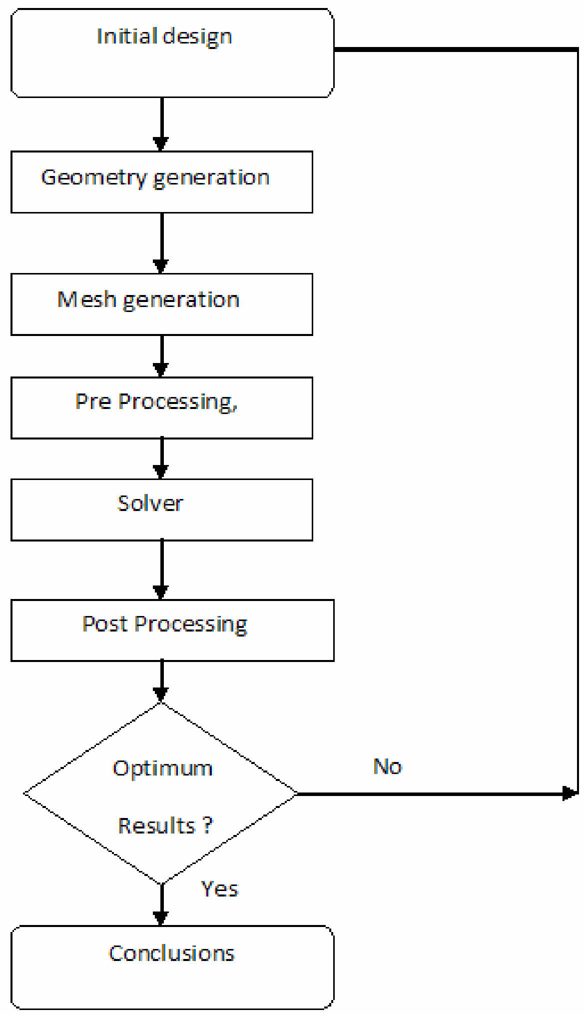


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Fig. 4.1: Flow chart showing CFD solving methodology

The geometry of design needs to be created from the initial design. Any modelling software can be used for modelling and then shifted to other simulation software for analysis purposes.

4.4.3 Mesh generation

Mesh generation (Girding) is the process of subdividing a region to be modelled into a set of small control volumes. Associated with each control volume there will be one or more values of the dependent flow variables (e.g., velocity, pressure, temperature, etc.) Usually these represent some type of locally averaged values. Numerical algorithms representing approximations to the conservation laws of mass, momentum and energy are then used to compute these variables in each control volume.

Meshing is a method to define and break up the model into small elements. In general, a finite element model is defined by a mesh network, which is made up of the geometric arrangement of elements and nodes. Nodes represent points at which features such as displacements are calculated. Elements are bounded by sets of nodes, and define localized mass and stiffness properties of the model. Elements are also defined by mesh numbers, which allow references to be made to corresponding deflections, stresses, pressures, temperatures at specific model locations. The traditional method of mesh generation is block-structure (multi-block) mesh generation. The block-structure approach is simple and efficient technique of mesh generation.

4.4.3.1 Topology

The topology is a structure of blocks that acts as a framework for positioning mesh elements. Topology blocks represent sections of the mesh that contain a regular pattern of hexahedral (hex) elements. They are laid out adjacent to each other without overlap or gaps, with shared edges and corners between adjacent blocks, such that the entire domain is filled. By using topology blocks to control the placement of hex elements, a valid hexmesh can be generated to fill a domain of arbitrary shape. The topology is invariant from hub to shroud and is viewed/edited on 2-D layers which are located at various spanwise stations. The topology blocks can be arranged in a regular (structured) pattern, an irregular (unstructured) pattern, or in a pattern consisting of structured patches and unstructured patches. The choice of which approach should be followed should be based on whichever method minimizes the maximum skew of the topology blocks, since the skew in the hex elements of the mesh is directly related. The topology should then be investigated at various layers (especially the hub and shroud layers) to check its quality since the mesh quality is directly dependent on topology. Topology blocks generally contain the same number of mesh elements along each side. The mesh elements vary in size across topology blocks in a way that produces a smooth transition within and between blocks. This is accomplished by shifting the nodes toward, or away from, certain block edges.

There are basically 3 types of topologies which is generally used

1) H-Grid
2) J-Grid
3) O-Grid

In actual conditions, a combination of these topologies is used. An O-Grid is a topology that forms a continuous loop around the blade profile. Using an O-Grid around the blade yields excellent boundary layer resolution and near-orthogonal elements on the blade.

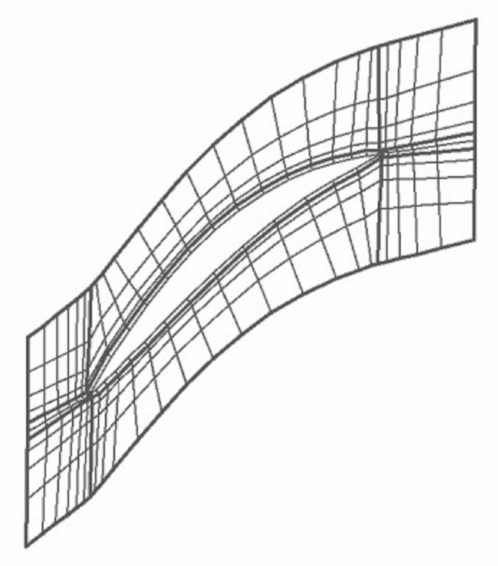

Fig.4.2: Topology of Type H

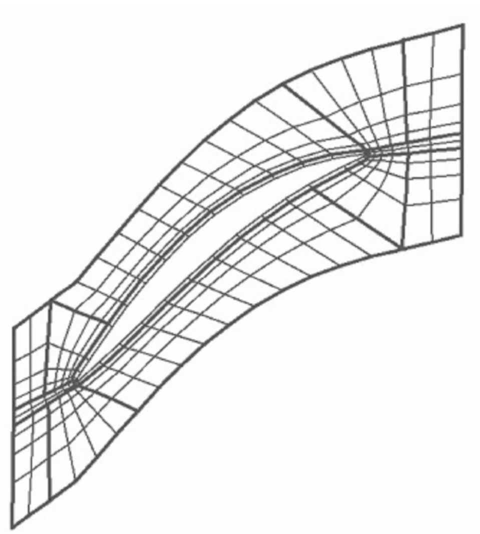

Fig. 4.3: Topology of Type J

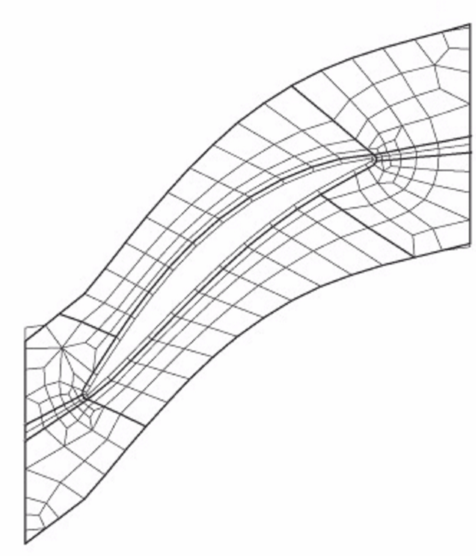

Fig. 4.4: Topology of type H-Grid

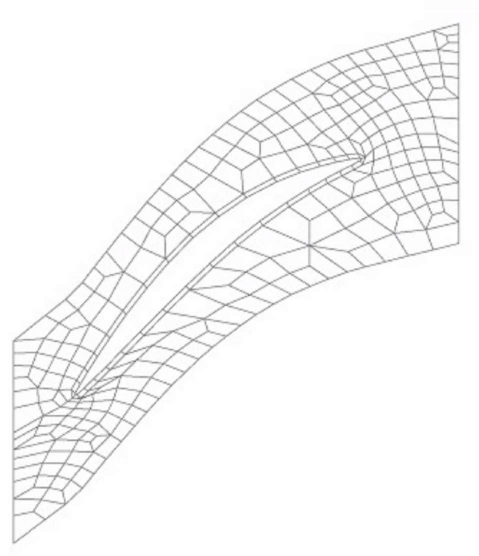

Fig. 4.5: Topology of Type Automatic dominan

H-Grid Dominant adds some topology blocks in a structured manner around an optional embedded O-Grid that surrounds the blade (as for a topology of type H-Grid), then completes the mesh by adding topology blocks in an unstructured manner. The structured blocks extend upstream of the leading edge, downstream of the trailing edge, and from the blade to the periodic surfaces. Automatic adds topology blocks in an unstructured isotropic manner around an optional O-Grid that surrounds the blade.

4.4.4 Preprocessing

4.4.4.1 Boundary condition

The first step in Pre-processing is setting up the boundary conditions. Boundary condition will be different for each type of problem. In Cartesian and cylindrical-polar coordinates, the location of boundary features (inlets, outlets, blockages, etc) can be linked to named 'objects' defined during the grid-generation procedure. This obviates the need to enter the coordinates twice: once when defining the grid, and again when specifying boundary conditions.

If an 'object' is subsequently repositioned or re-sized, then the boundary condition is also changed automatically. If an object is deleted, any associated boundary-conditions will also be deleted without further instructions from the user. If a new 'object' is created by copying an existing one, the boundary conditions are not automatically copied, but a new boundary condition may be linked to the new object.

In low speed and incompressible flows, disturbances introduced at an outflow boundary can have an effect on the entire computational region. As a general rule, a physically meaningful boundary condition, such as a specified pressure condition, should be used at out flow boundaries whenever possible. Generally, a pressure condition cannot be used at a boundary where velocities are also specified, because velocities are influenced by pressure gradients. The only exception is when pressures are necessary to specify the fluid properties, e.g., density crossing a boundary through an equation of state.

The inlet condition for velocity and temperature can be specified using profile of grid. The turbulent kinetic energy (k) and its dissipations rate can be calculated from the value of turbulence intensity specified in the inlet.

4.5 Modeling of impeller and casing

Before the modeling of blade, a generalized program is written for the design of the blade. The program is based on the design methodology discussed in the previous chapter. The parameters which were considered initially are Head, Flow Rate, Pump Speed, Volumetric efficiency and Overall Efficiency. The output of the program is given in the table 6.1. Some of the output parameters of this design were used as input of the Blade Modeler.

Pump part	**Dimensions**
Impeller details	
Type	Open
No. of vanes	5
Inlet diameter	50 mm
Outlet diameter	100 mm
Inlet width	31 mm
Outlet width	20.4 mm
Inlet vane angle	20 degree
Outlet vane angle	25 degree
Casing details	
Type	Semi-volute
Tongue angle	11 degree
Cut water radius	154 mm
Cut water area	4900 mm^2

Table 4.1: centrifugal pump design parameters

The numerical model as shown in fig 4.6 was created using the Blade Modeler tool of ANSYS-CFX package. From the solid model of the impeller, a negative mold was created that describes the flow passages. This solid model was the base geometry for creating the finite volume mesh.

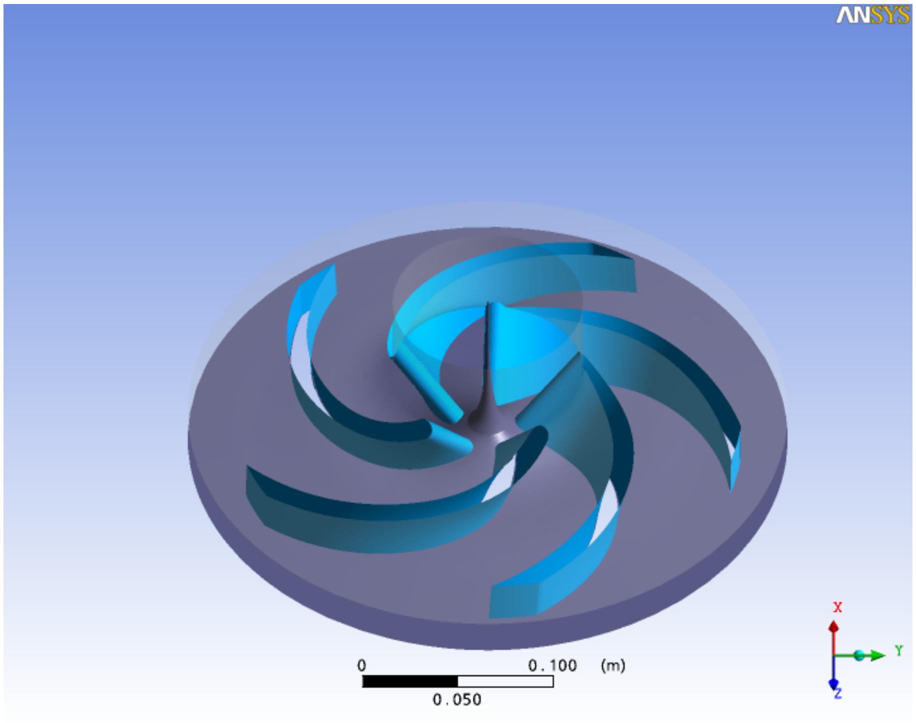

Fig. 4.6: Impeller model in ANSYS

The figure 4.7 shows the model of the volute casing in default orientation in PRO-E window.

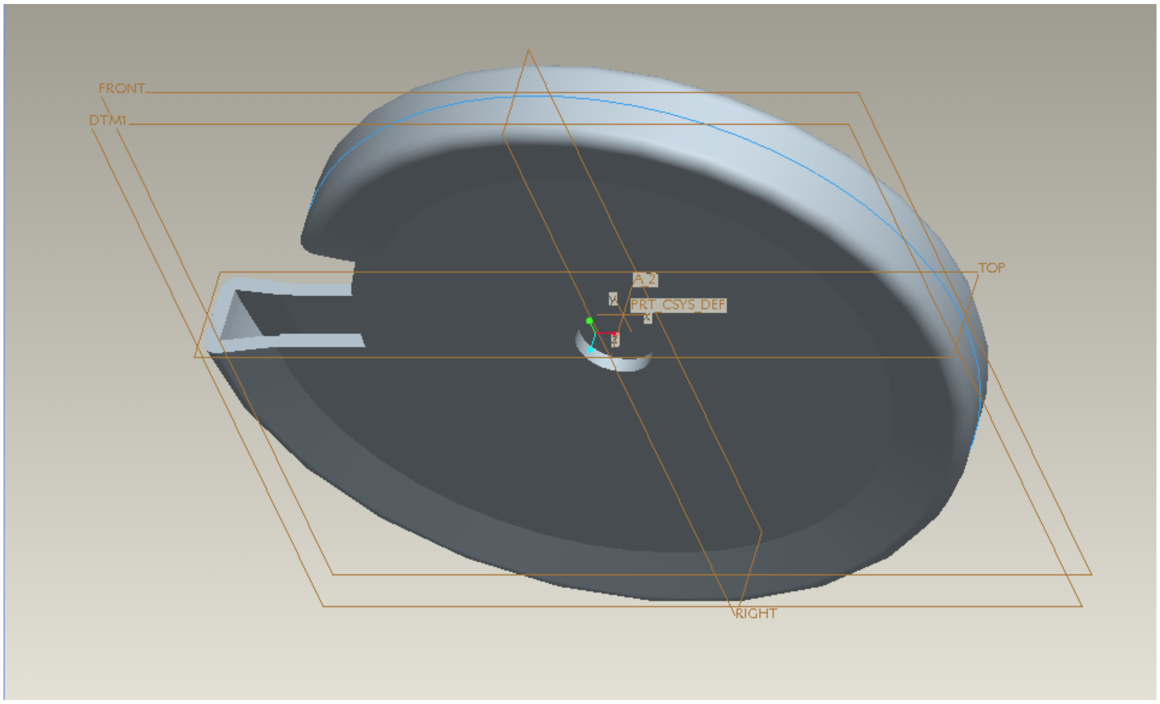

Fig 4.7: Model of volute casing

CHAPTER 5

PERFORMANCE AND WEAR CHARACTERISTICS

5.1 Performance of pumps-characteristics

A pump is usually designed for a particular speed, flow rate and head, but in actual practice the operation may be at some other condition of head, and for the changed condition the behavior of the pump is less efficient than the quantity the value of velocity of flow of liquid through impeller will be changed. As a result, the value of v, r and v_r will be changed, and at the same time the loss will be increased so the efficiency of pump will lowered, therefore, in order to predict the behavior and performance of a pump under varying condition, tests are performed, and the result of test are plotted, the curve thus obtained are known as characteristic curve of a pump. Characteristic curve are usually prepared for the centrifugal pump

1) Main and operating characteristic curve
2) Constant efficiency curves
3) Constant head & constant discharge curve

5.1.1 Main characteristics

These are obtained by fixing the speed at some arbitrary value of and plotting separately H, HP, & η against Q. The rate of flow Q is varied by means of the quantity H, HP, η are calculated. A number of different values of N are chosen and one such set of curves is drawn for each speed.

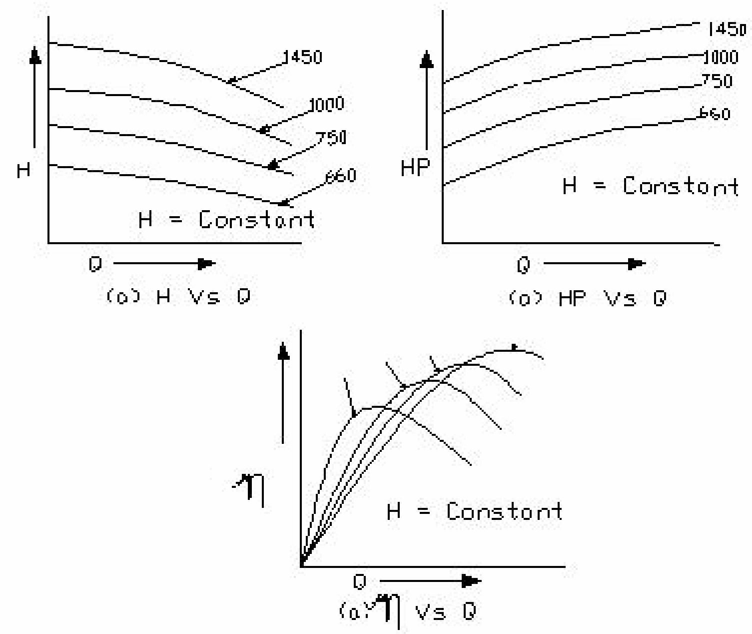

Fig. 5.1: Main characteristic curves of centrifugal pump[35]

5.1.2 Operating characteristics

During operation the pump must run at a constant speed. Normally, this is the designed speed. The particular set of main characteristics which correspond to the designed speed is mostly used in operation and is therefore known as operating characteristics.

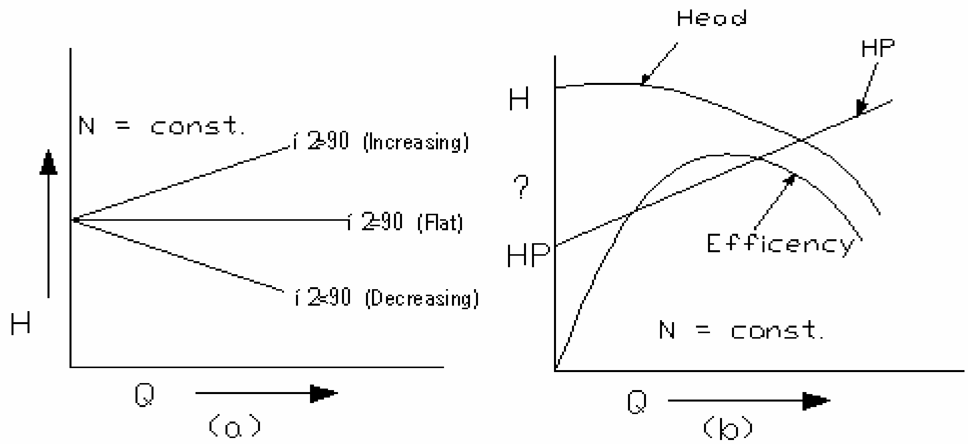

Fig. 5.2: Operation characteristic curve for centrifugal pump[35]

5.1.3 Muschal Curve or Constant Efficiency Curve

With the help of data obtained from the above curve, a series of const. Efficiency curve can be obtained. They facilitate the job of the salesman and enable the prospective customer to see directly the range of operation with a particular efficiency,

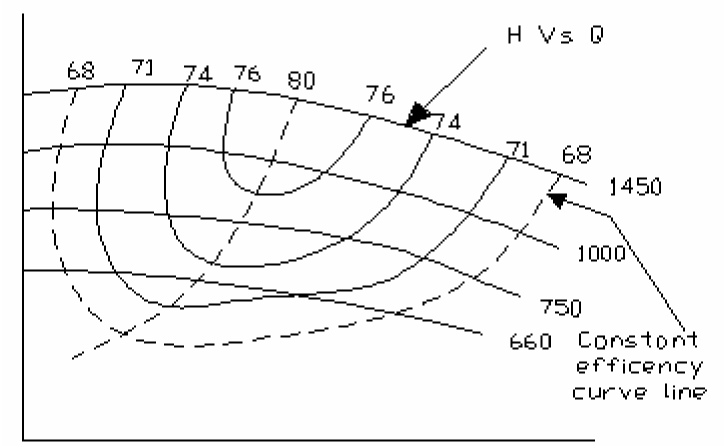

Fig. 5.3: Constant efficiency curve for centrifugal pump[35]

5.1.4 Constant Head and Constant Discharge Curve

It is quite possible that a pump may be required to deliver water at a certain height, in which case it is fixed. If for some reason the speed varies, discharge will also be affected.

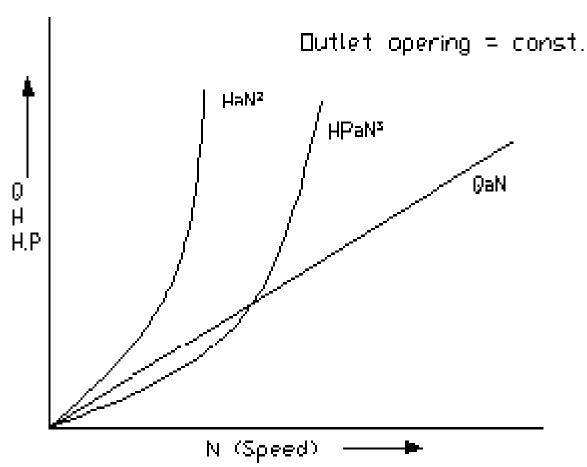

Fig. 5.4: Constant head & constant discharge curve for centrifugal pump[35]

The performance of the pump under such condition, it is necessary to drawn a constant H curve by plotting Q vs. N. Similarly to determine the speed required to discharge a certain quantity at diff. pressure it is convenient to draw constant Q curves showing H against N.

5.2 Erosion wear

Wear is one of the most common problems encountered in industrial applications. It is defined as the volume loss of material from a solid surface by the action of another solid or liquid particles. In order to establish the life of pump and period of replacement of its parts, wear studies in the impeller and casing have to be carried out. The concentration of the bed ash taken for the analysis purpose varies from 10 to 20% and particle size varies from 0.2235 to 0.890 mm.

Then Erosion wear, E_w, is calculated by using the formulae given below:

$$E_w = 2.568 \times V^{2.56} \times C_w^{0.825} \times d^{0.847}$$

Where,

V = velocity of flow, m/s,

C_w = solid concentration by weight,

d = representative particle size, mm.

5.2.1 Parameters affecting erosion wear:

The prominent parameters and their effect on erosion wear are as under:

1) **Impact angle**

Impact angle is defined as the angle between the target surface and the direction striking velocity of the solid particle. The variation of erosion wear with the impact angle depends on the characteristics of the target surface material namely brittle or ductile type.

2) Velocity of solid particles

Velocity of solid particle strongly affects the erosion wear. As particle velocity increases there is significant increase in erosion rate. The erosion rate is generally related to the particle velocity using power law relationship in which the power index for velocity varies in the range of 2-4.

3) Hardness

Hardness is the characteristic of a solid material expressing its resistance to permanent deformation.Surface hardness as well as hardness of solid particles has profound effect on the erosion wear mechanism. Hardness ratio has been defined as the ratio of hardness of target material to the hardness of solid particles.

4) Particle size and shape

Particle size and shape is also one of the prominent parameter, which affect erosion wear. Many investigators have considered solid particle size important to erosion. The erosion wear increases with increase in particle size according to power law relationship. The effect of particle shape on the erosion is not very well established due to difficulties in defining the different shape features. Generally roundness factor is taken into consideration. If roundness factor is one then the particles are perfectly spheres and a lower values show the particle angularity.

5) Solid concentration

Concentration is amount of solid particles by weight or by volume in the fluid. As concentration of particle increases more particles strike the surface of impeller which increase the erosion rate, the concentration of slurries can vary from 2% to 50% depending upon the type of slurry. However, at very high concentrations particle-particle interaction increases and this decreases the striking velocity of particle on the surface.

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CHAPTER-6

RESULT AND CONCLUSION

6.1 Input data

Head = 36 m, Flow rate = 0.09 m^3/sec, Speed = 1450 rpm

Design of centrifugal pump program generated in C++ and output for the given data given below:

6.2 Centrifugal pump dimensions

Design parameters	**Values**
Specific speed	29.598 rpm
Input required power	102.972 Hp
Shaft diameter	0.0430098 m
Hub diameter	0.0516118 m
Outlet blade velocity	27.4176 m/s
Outlet diameter	0.361082 m
Mean meridional velocity of steam just prior to blade inlet	4.79808 m
Mean meridional velocity at the exit of the impeller	5.51779 m
Impeller eye diameter	0.162922 m
Inlet diameter	0.182922 m
Inlet blade velocity	13.8896 m/s
Inlet blade angle	19.0548°
Width of the impeller	0.0165334 m
Outlet blade angle	25.0959°
No. of blades	8
Relative velocity at inlet	14.6949 m/s

Relative velocity at outlet	11.3458 m/s
Inlet width of volute	0.0297602 m
Diameter of inlet at volute	0.415245 m
Width at x distance, Bx	0.045414 m
Whirl component of velocity at volute	5.45998 m/s
Velocity at throat	33.9391 m/s

Table 6.1 Output of impeller dimensions

6.3 Simulation results of the performance characteristics:

Figure 6.1 shows that total head of the pump decreases with increase in discharge rate for water and bed ash both. Figure 6.2 shows that pump input power increases with increase in discharge rate for water and bed ash both. Figure 6.3 shows that pump efficiency decreases with increase in discharge rate for water and bed ash both.

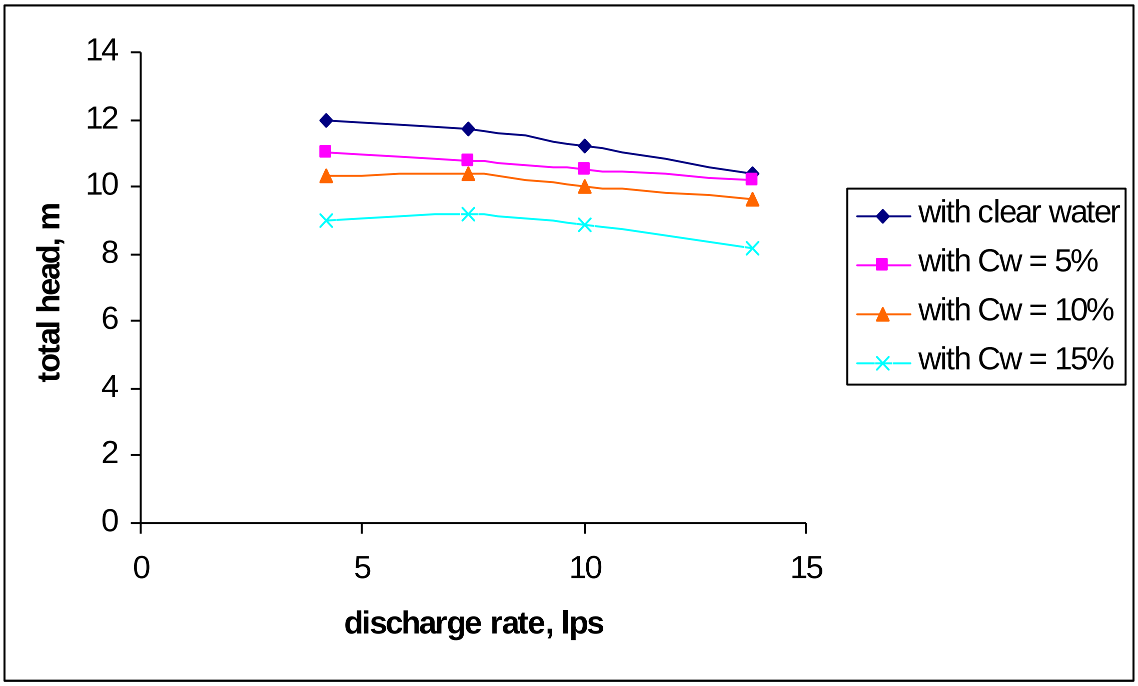

Fig. 6.1: Total head vs. discharge rate

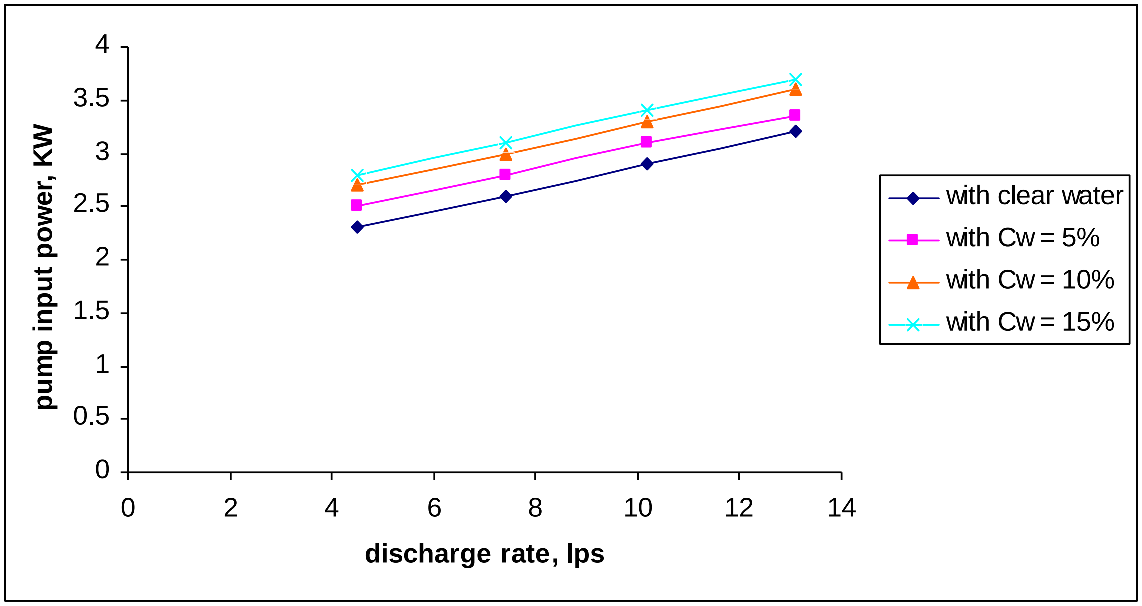

Fig. 6.2: Pump input power vs. discharge rate

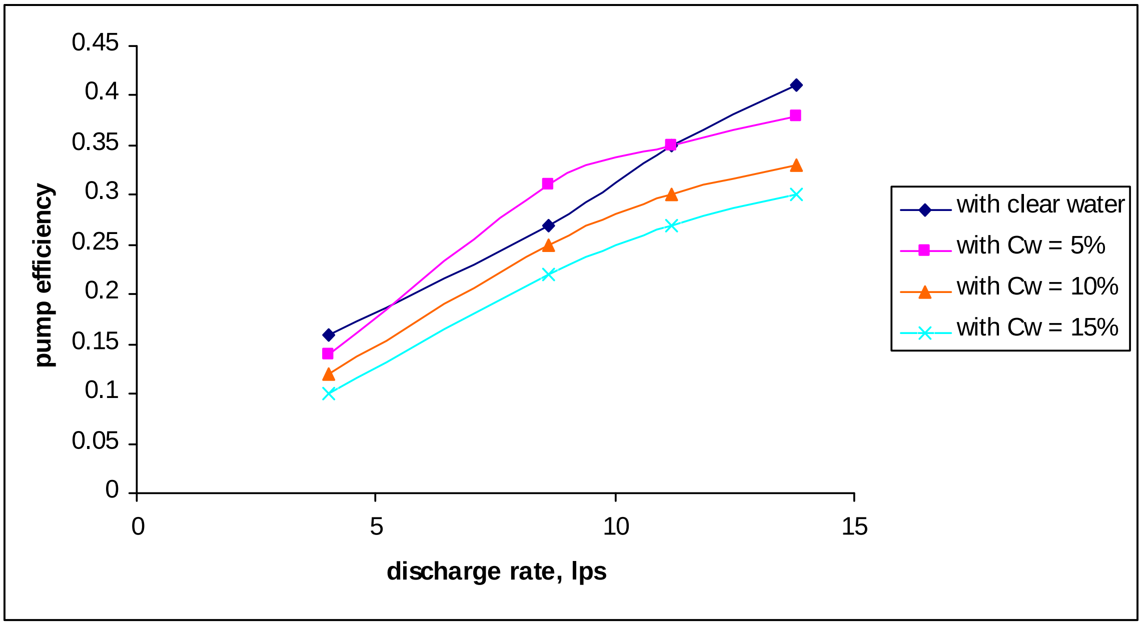

Fig. 6.3: Pump efficiency vs. discharge rate

Figure 6.4 shows that specific head decreases with increase in specific discharge for water and bed ash both.

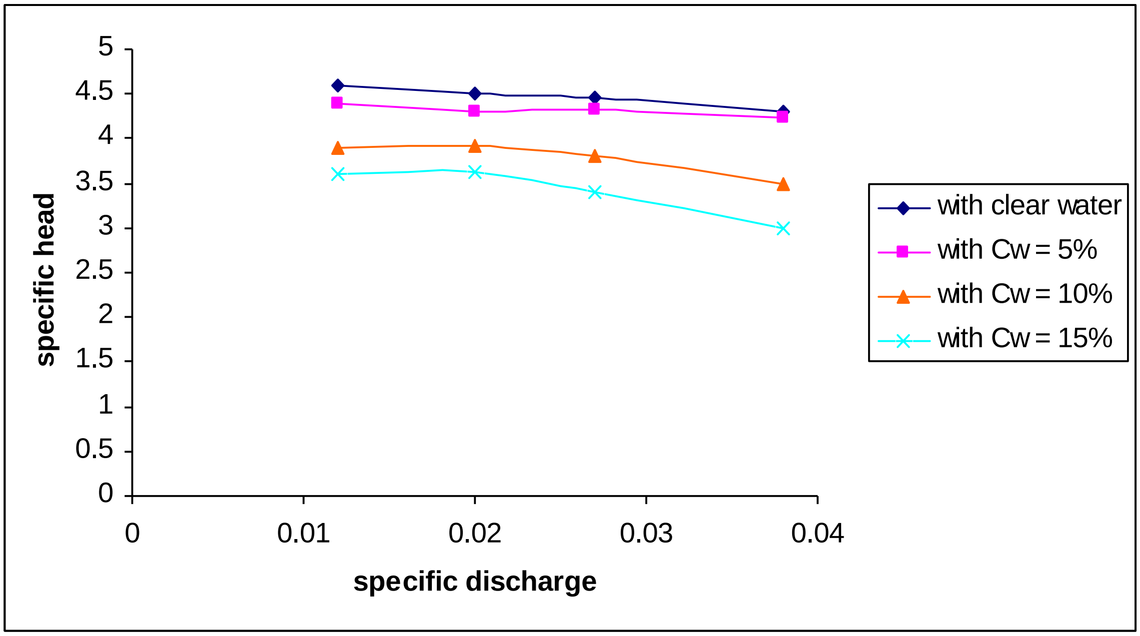

Fig. 6.4: Specific head vs. specific discharge

Figure 6.5 shows that specific power of the pump increases with increase in specific discharge rate for water and bed ash both. Figure 6.6 shows that specific head decreases with increase in specific discharge for water and bed ash both.

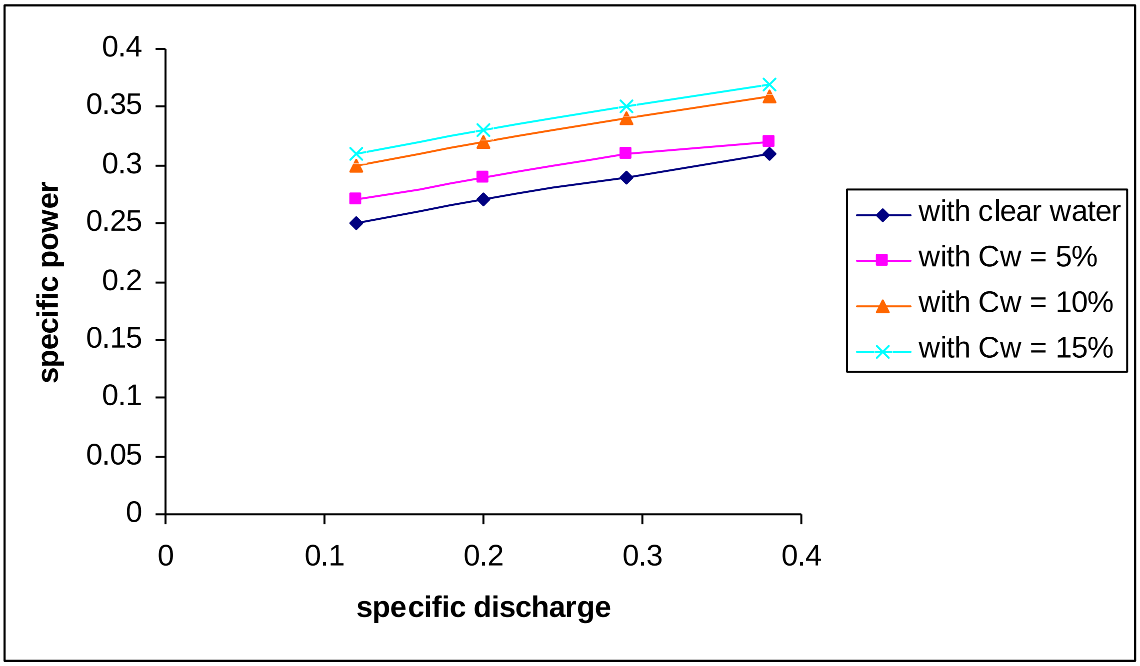

Fig. 6.5: Specific power vs. specific discharge

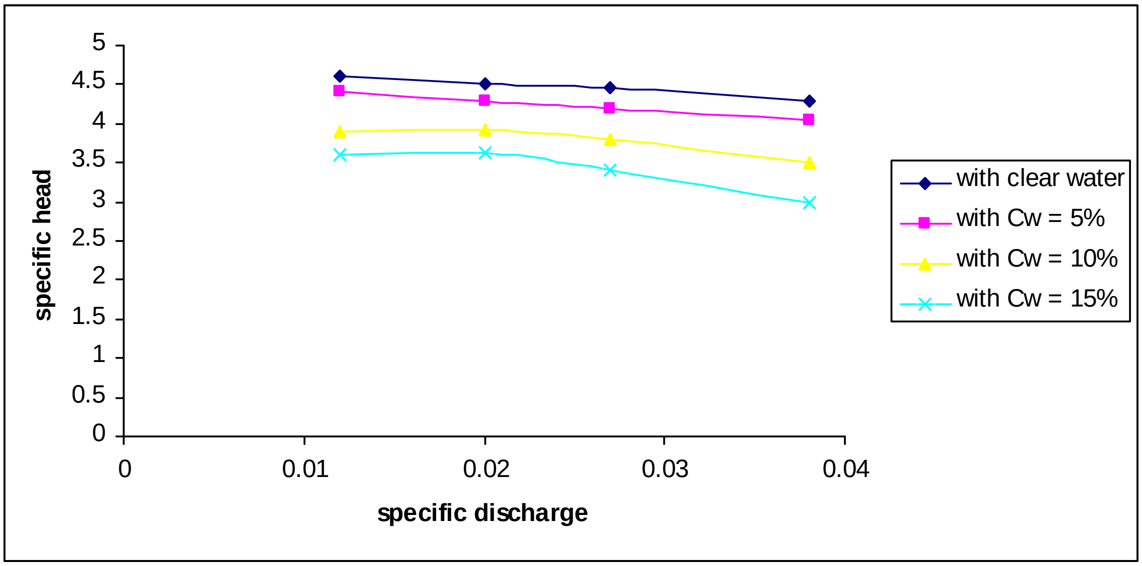

Fig. 6.6: Specific head vs. specific discharge

6.4 Results of the wear characteristics

Figure 6.7, 6.8, 6.9 and 6.10 show that wear rate increase with the increase in solid concentration with varying particle size, particle concentration at varying speed of

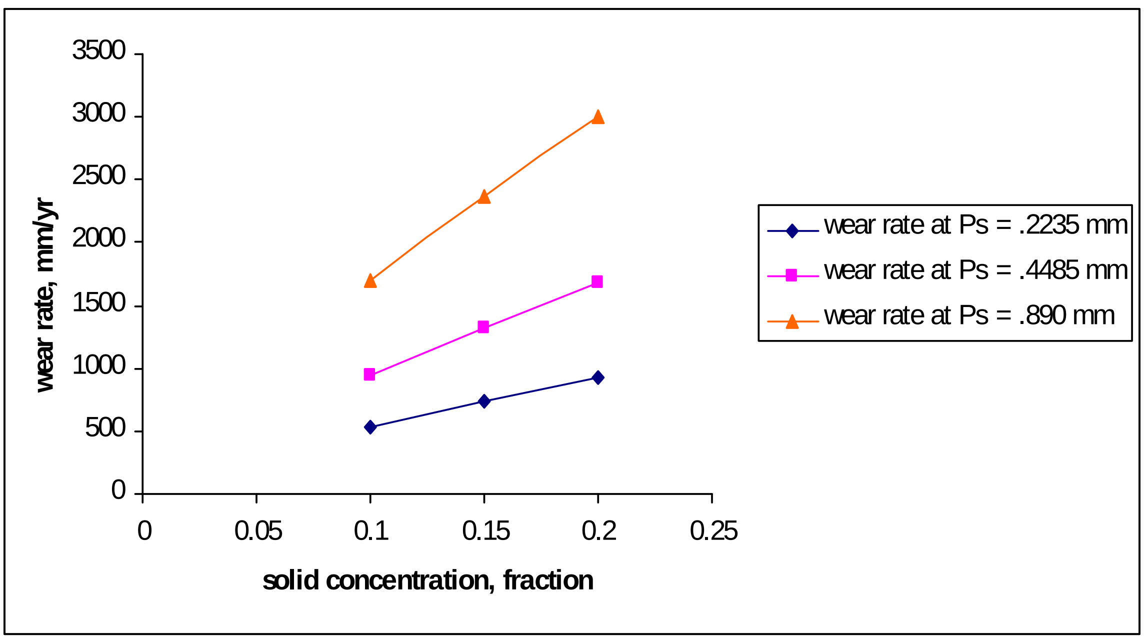

Fig. 6.7: Wear rate vs. particle concentration at varying particle size

impeller, particle concentration at varying speed of impeller and particle size at varying speed of impeller respectively.

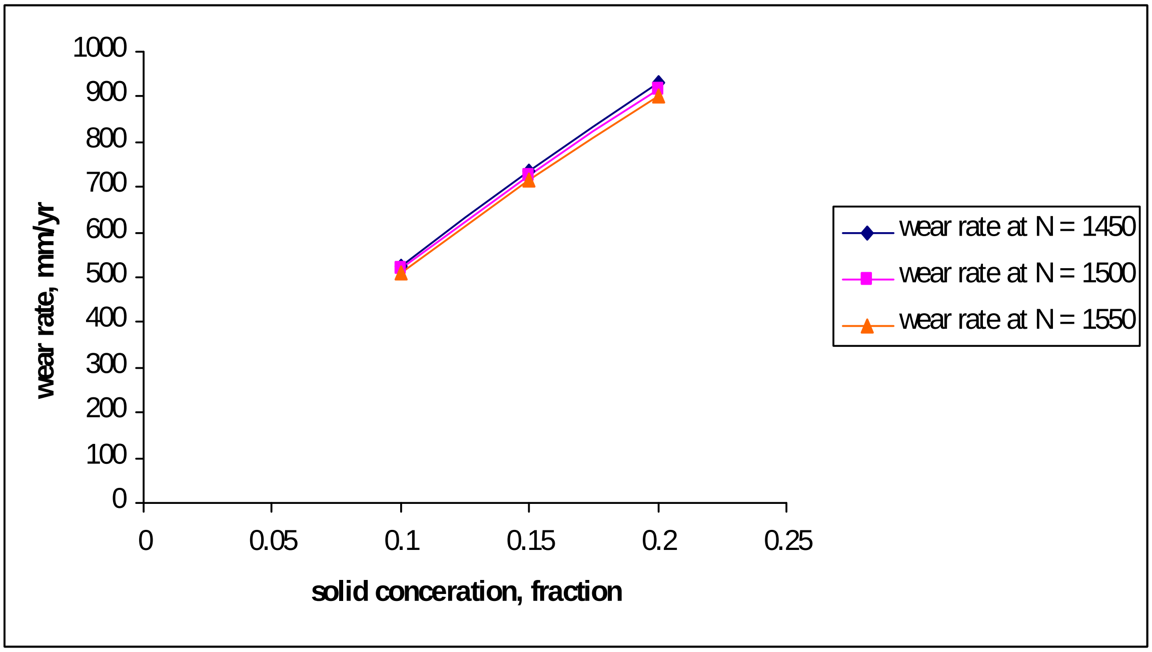

Fig. 6.8: Wear rate vs. particle concentration at varying speed of impeller

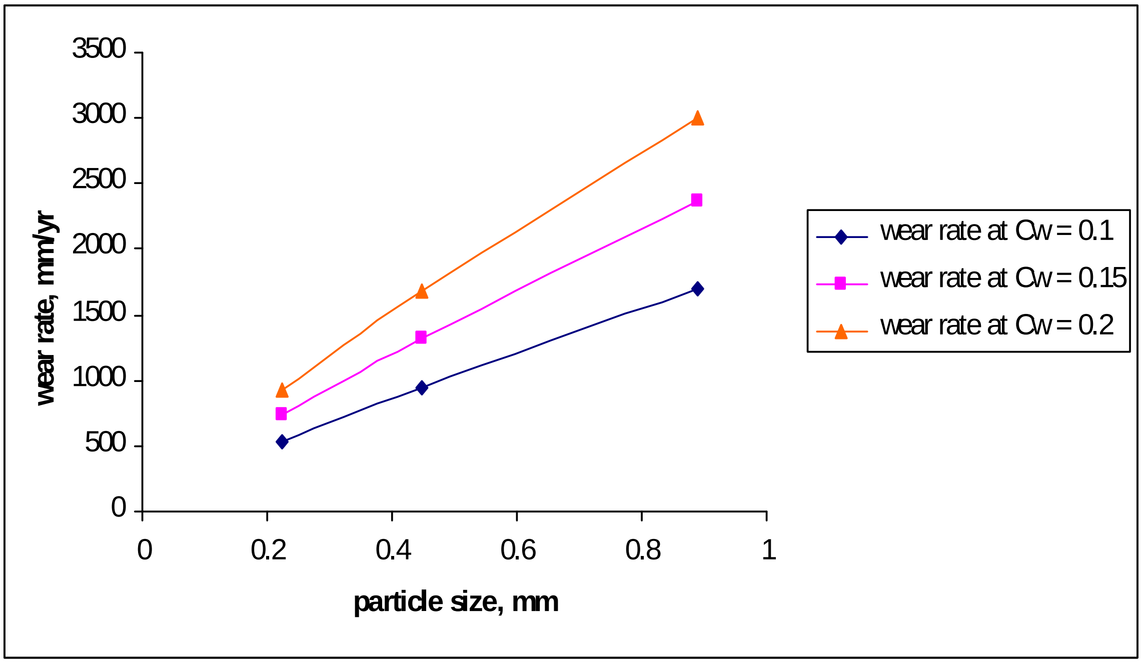

Fig. 6.9: Wear rate vs. particle size at varying particle concentration

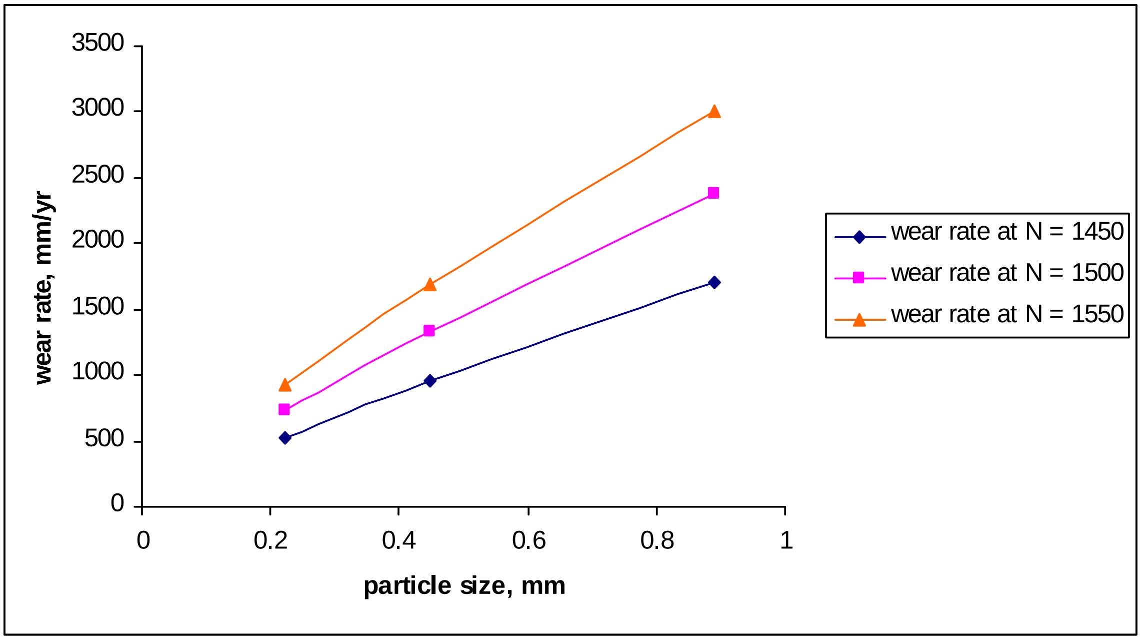

Fig. 6.10: Wear rate vs. particle size at varying speed of impeller

Figure 6.11 and 6.12 shows the variation in wear rate with flow rate at different particle concentration and size and wear rate decreases with increase in particle size with increasing particle concentration and size respectively..

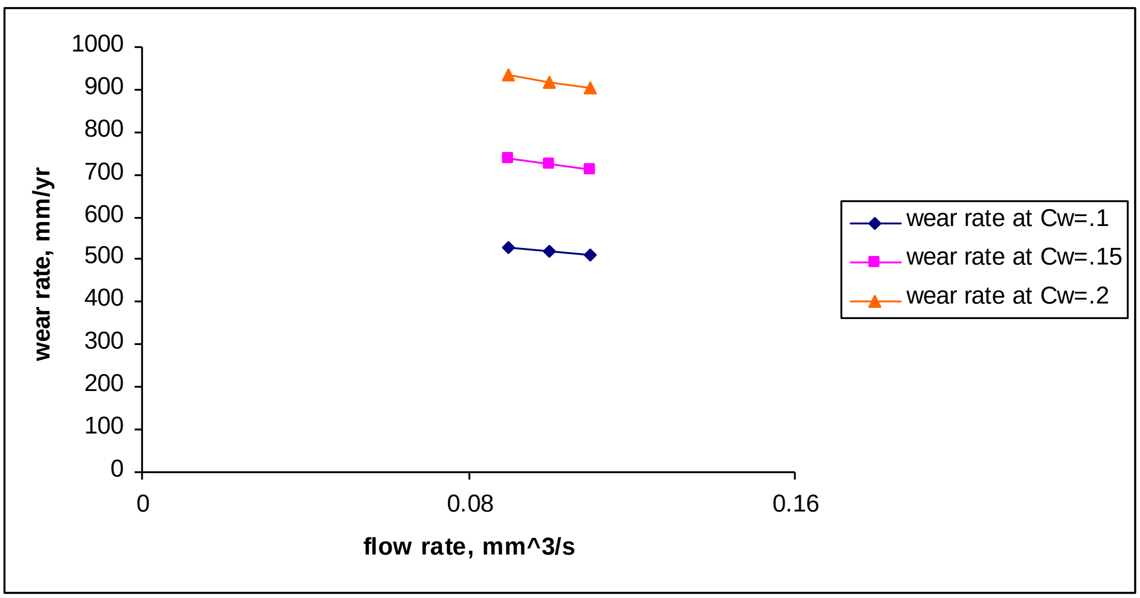

Fig. 6.11: Wear rate vs. flow rate at varying particle concentration

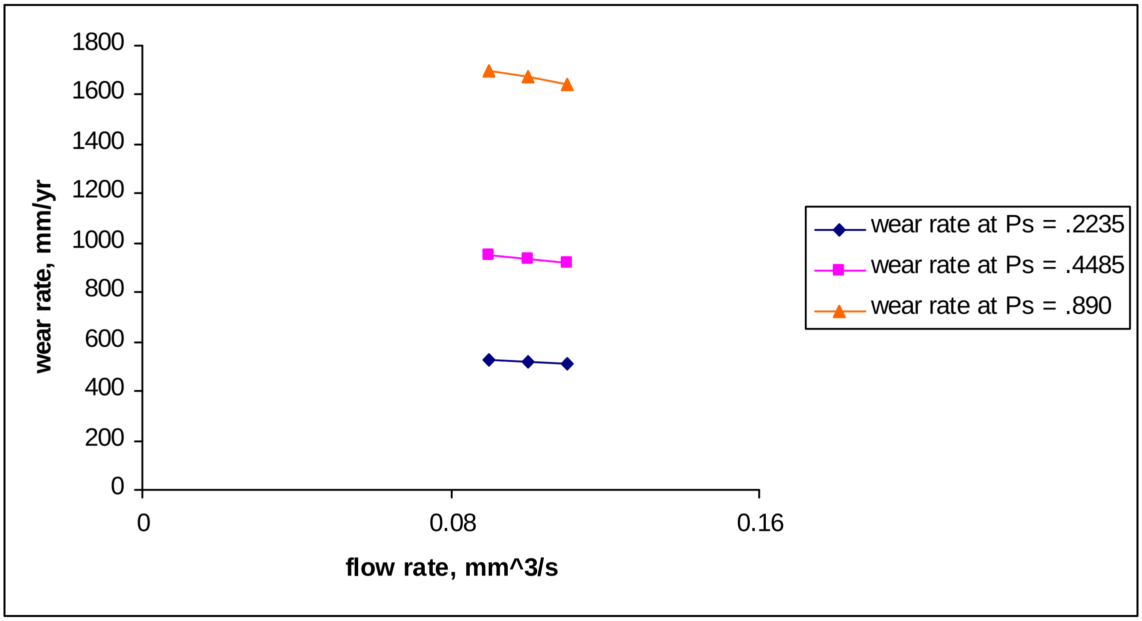

Fig. 6.12: Wear rate vs. flow rate at varying particle size

Figure 6.13 and 6.14 shows the variation in wear rate with impeller speed at different particle concentration and size and wear rate increases with increase in impeller speed with increasing particle concentration and size.

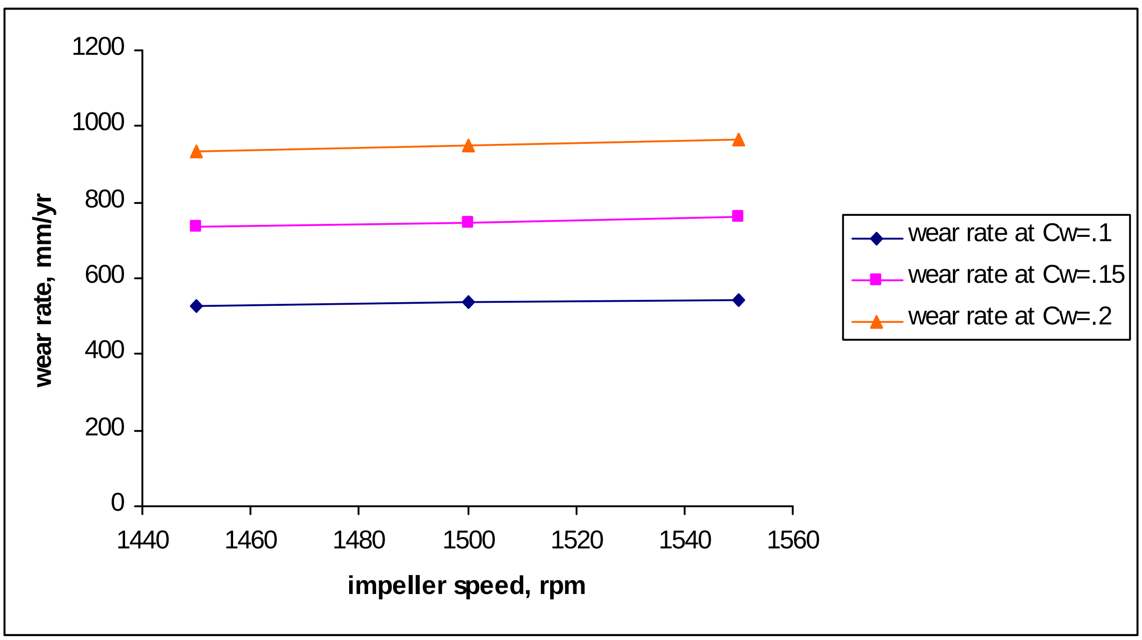

Fig. 6.13: Wear rate vs. impeller speed at varying particle conc.

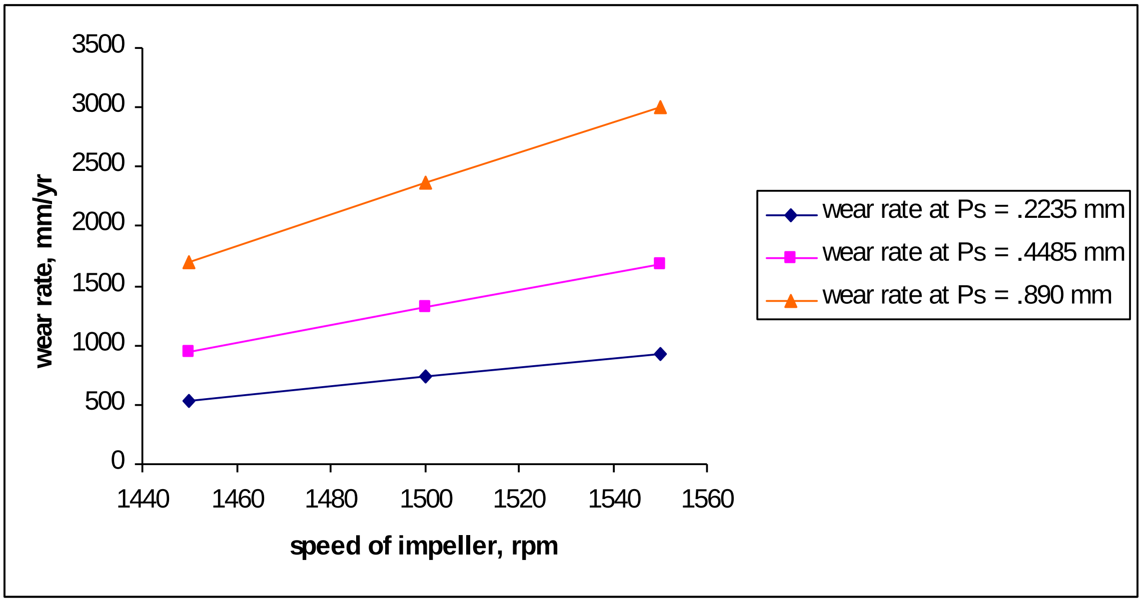

Fig. 6.14: Wear rate vs. impeller speed at varying particle size

Conclusion

Design program has been generated for the impeller and volute casing of the pump Advantage of computer simulation permit the wide range of design variable to investigated in a very short time.

A numerical modeling of an impeller and volute casing has been also successfully generated by using the ANSYS simulation code and PRO-E part modeling. Performance characteristics of the centrifugal slurry pump evaluated using ANSYS-CFX and C++, simulation results show the good agreement with the open literature.

Wear characteristics of the volute casing also evaluated using empirical correlation for handling ash slurry and found that wear rate increases with increasing particle size, particle concentration, impeller speed and decreases with increasing flow rate.

Scope for future work

- Pressure and velocity distribution in the casing of pump can be analyzed.
- Same simulation can be used for calculating performance and wear characteristics of other turbo machinery.
- Wear analysis of the pump by experiments and by numerical simulation using local values of velocity and concentration.

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Generalized C++ program of centrifugal pump design

```
#include <iostream.h>
#include<math.h>
#include <conio.h>
#include<stdio.h>
#include<process.h>

void main()
{
        float H=36;
        float N;
        float Q;
        float g=9.81;
        float d;
        float Cw;
        float t=0.175;
        float s=0.58;

        cout<<" Given head = "<<H<<" m "<<endl;
        cout<<" Enter particle size = ";
        cin>>d;
        cout<<" Enter solid concentration = ";
        cin>>Cw;
        cout<<" Enter speed of impeller = ";
        cin>>N;
        cout<<" Enter flow rate = ";
        cin>>Q;

        float a=sqrt(Q);
        float b=pow(H,0.75);
        float Ns=N*a/b;
        cout<<"\n Specific speed = "<<Ns<<" rpm "<<endl;

        float efficiency=.8.15;
        float Po=(1700*g*Q*H)/(745*efficiency);
        float Psh=1.15*Po;
         float c=pow(10,-5);
        float Dsh=pow((16*Psh*60*c)/(2*3.142*1.5*N),0.33);
        float Dh=1.2*Dsh;

        float U2=sqrt((g*H)/(s*efficiency));
```

```
cout<<"\n Outlet blade velocity = "<<U2<<" m/s "<<endl;

float D2=(60*U2)/(3.142*N);
float Vm1=t*U2;
float Vm2=1.15*Vm1;
float De=pow((4*Q)/(3.142*Vm1) + pow(Dh,2),0.5);
float D1=(De+0.020);
float U1=(3.142*D1*N)/60;
float m=Vm1/U1;
float Beta1=atan(m)*(180/3.142);
float B2=(Q/(3.142*D2*U2*t));
float Beta2=atan(Vm2*U2/(U2*U2-H*g))*1.4*180/3.142;
int Z=Beta2/3;
float Vr1=pow((pow(U1,2) + pow(Vm1,2)),0.5);
float n=sin(Beta2*(3.142/180));
float Vr2=Vm2/n;

float B3=1.8*B2;
cout<<"\n Inlet width of volute = "<<B3<<" m "<<endl;

float D3=1.15*D2;
cout<<"\n Diameter of inlet at volute = "<<D3<<" m "<<endl;

float Rv=((D2+D3)/4);
cout<<"\n Rv = "<<Rv<<" m "<<endl;

float Bx=B3+((2*Rv-D2)/1.73);
cout<<"\n Width at x distance Bx = "<<Bx<<" m "<<endl;

float Vu3=Vm2*cos(8.3*(3.142/180));
cout<<"\n Whirl component of velocity at volute = "<<Vu3<<" m/s "<<endl;

float Vth=(D2*39.03)/D3;

cout<<"\n velocity of flow, cutoff = "<<U2<<" m/s "<<endl;

float Ew1=2.568*pow(d,0.847)*pow(Cw,0.825)*pow(U2,2.56);
cout<<"\n Erosion wear rate, cutoff = "<<Ew1<<" mm/yr "<<endl;


}
```

APPENDIX B

Flowchart of the program

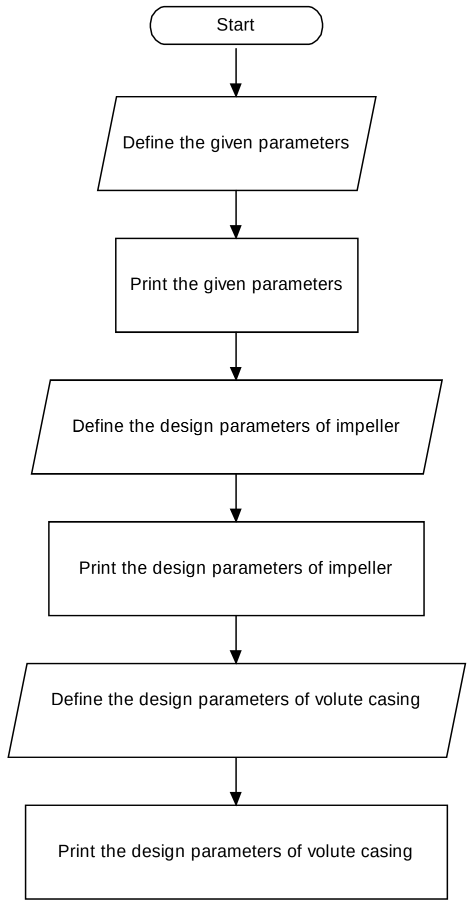

Fig.: Flow chart of pump design

The flow chart is made according to the design steps needed to design the volute casing, impeller and calculating the NPSH, cavitation of the centrifugal slurry pump. For making the flowchart, UML software is used.

APPENDIX C

Output of the program

Specific speed = 29.598 rpm

Outlet blade velocity = 27.4176 m/s

Inlet width of volute = 0.0297602 m

Diameter of inlet at volute = 0.415245 m

Rv = 0.194082 m

Width at x distance Bx = 0.045414 m

Whirl component of velocity at volute = 5.45998 m/s

Velocity of flow, cutoff = 27.4176 m/s

Erosion wear rate, cutoff = 526.858 mm/yr