DESIGN OF DOUBLE CURVATURE BLADE

A PROJECT REPORT

Submitted by

Keerthikanth.M - 71205114018
Manoj.S - 71205114021
Vinoth Kumar.C - 71205114059

in partial fulfillment for the award of the degree

of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY, COIMBATORE

ANNA UNIVERSITY : CHENNAI 600 025

APRIL 2009

ANNA UNIVERSITY : CHENNAI 600 025

BONAFIDE CERTIFICATE

Certified that this project report "...**DESIGN OF DOUBLE CURVATURE BLADE**..." is the bonafide work of

Keerthikanth.M	-	Register No. 71205114018
Manoj.S	-	Register No. 71205114021
Vinoth Kumar.C	-	Register No. 71205114059

Who carried out the project work under my supervision.

24/4/2009

Signature of the Head of the Department

Dr.T.Kannan Ph.D.,

HEAD OF THE DEPARTMENT

24 April 2009

Signature of the Supervisor

Prof.S.Bhaskar M.E.,

SUPERVISOR

Internal Examiner 27/4/2009

27/4/09

External Examiner

DEPARTMENT OF MECHANICAL ENGINEERING

KUMARAGURU COLLEGE OF TECHNOLOGY

COIMBATORE 641 006

ABSTRACT

In many incidents of pump development, the inlet edge of vane cannot always start radially from the eye diameter as in the case of cylindrical vane development. When the specific speed of the pump 'n_s' is increased, the diameter ratio D_2/D_1 reduces. When $(D_2/D_1) < 1.5$, the radial inlet of cylindrical vanes cannot provide enough area to develop the required head. In other words the blade loading is higher, over and above the permitted blade loading calculated for the cylindrical vane development.

The blade angle has to be increased to bring down the blade loading to the normal level. This can be obtained only by extending the inlet edge from pure radial entry to diagonal entry. The outlet edge is located at higher diameter, which is the eye diameter, whereas the inner diameter will be on the hub. The inlet edges, from hub to periphery, lie at different radii. Since meridional velocity C_m is constant at all radii and since the vane velocity are different, blade angle decreases from hub to periphery. The blade becomes 3 dimensional at inlet. When the specific speed 'n_s' increases further i.e, n_s>250 to 450, the outlet edge of the impeller also changes from radial form to diagonal form. In other words vane becomes a purely 3 dimensional. Hence the vane will be in twisted form, that is, double curvature.

The design of a double curvature blade for the specifications 7.5m head, 16 lps capacity, 2850rpm and 2.4hp input power, is completed successfully and the efficiency is found to be higher than the normal blade.

KEYWORDS: Vanes, blade loading, impeller, 3 dimensional, double curvature

ACKNOWLEDGEMENT

I wish to express my sincere gratitude to my project guide **Prof.S.Bhaskar**, Assistant professor, Mechanical Engineering Department, Kumaraguru College of Technology, Coimbatore, for his encouragement, guidance and moral support through out the work, for providing valuable suggestions, fruitful discussions for improvement and the successful completion of the project.

I also wish to express my sincere thanks to **Dr.T.Kannan** Professor and The Head of the Department, Department of Mechanical Engineering KCT Coimbatore for the encouragement and moral support throughout the B.E. degree and especially during the project work.

I thank our beloved **Prof.R.Annamalai**, Vice-Principal of Kumaraguru College of Technology, Coimbatore, for providing such an opportunity and the necessary facilities to carry out this project work successfully.

We specially thank **Dr.K.M.Srinivasan,** Dean (R&D) Mechanical Sciences, Kumaraguru College of Technology, Coimbatore, who guided us in the project and rendered all the help needed to us to complete this project successfully.

We have great pleasure in extending our gratitude to all the teaching staff and non teaching staff, Department of Mechanical Engineering for their support in this project.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS, ABBREVIATIONS AND NOMENCLATURE

$\mathbf{A_i}$ - areas of the impeller sections

$\mathbf{b_1}$, $\mathbf{b_2}$ - Width of the blade

$\mathbf{C_{m3}}$, $\mathbf{C_{m2}}$, $\mathbf{C_{m1}}$, $\mathbf{C_{m0}}$, $\mathbf{C_{mi}}$ - Meridional velocities

$\mathbf{c_0}$ - Flow velocity

$\mathbf{D_0}$ - Eye diameter

$\mathbf{D_{1nom}}$ - Nominal diameter

$\mathbf{d_{a\text{-}out}}$, $\mathbf{d_{a\text{-}in}}$, $\mathbf{d_a}$,$\mathbf{d_{b\text{-}out}}$, $\mathbf{d_{b\text{-}in}}$, $\mathbf{d_b}$, $\mathbf{d_c}$ - Diameters at the points of the stream line

$\mathbf{d_h}$ - Hub diameter

$\mathbf{d_s}$ - Shaft diameter

FS- Factor of safety

$\mathbf{f_m}$ - Ultimate stress

$\mathbf{f_s}$ - Working stress

H- Head

$\mathbf{H_m}$- Manometric Head

$\mathbf{k_1}$, $\mathbf{k_2}$- Flow coefficients

m- Number of streamlines

N, $\mathbf{n_s}$ - Speed and specific speed of the pump

$\mathbf{N_i}$ - Input power

$\mathbf{N_0}$ - Output power

n- Number of passages

p - Slip factor

$\mathbf{p_{atm}}$, $\mathbf{p_{vp}}$ - Atmospheric pressure and vapour pressure

Q- Discharge

$\mathbf{Q_{th}}$ - Theoretical discharge

$\mathbf{R_{Ds}}$ -Radius for the front shroud

$\mathbf{r_i}$ - Radius at the points on the stream line

T - Torque

t - Circumferential distance between two blades

$\mathbf{u_1}$, $\mathbf{u_2}$, **u.** - Blade velocity

$\mathbf{w_1}$, $\mathbf{w_2}$- Relative velocity

$\mathbf{z_E}$ - Axial extension

z - Number of blades

$\Delta\varepsilon$, Δg, Δz, ΔL, Δm, $L_{a,m}$, $L_{b,m}$, $L_{c,m}$- Various constructional lengths at the vane development

α - Angle of attack

β_{10} ,β_1, β_2 - Blade angles

ϕ, θ - Angles of the points on the streamlines

η_h - Hydraulic efficiency

η_v - Volumetric efficiency

η - Overall efficiency

Ψ - Coefficient depending upon blade configuration.

γ - Specific weight of liquid

δ - Thickness of the blade

ε_{Ds}, ε_{Ts}, ε_{EK} - Various angles at the meridional section diagram

CHAPTER 1

INTRODUCTION

CHAPTER 1
INTRODUCTION

1.1 PUMPS:

One of the first pieces of powered machinery to be invented at the dawn of the industrial age was a crude form of pump. The pump has since evolved into an endless variety of types, sizes, and applications. This module will give an overview of the general types of pumps that are in common use in buildings and industrial plants.

Operators should become familiar with the diversity of pumps that are in existence as they may be required to safely operate pumps in the normal course of their daily routines. A functional understanding of pumps, their use, and application, is essential to understanding how most processes are handled in plants today.

1.2 DEFINITION:

A pump is a mechanism that is used to transfer a liquid from one place to another by imparting energy to the liquid being transferred.

1.3 APPLICATIONS:

Pumps are used in numerous locations for many purposes. A car may contain several different types of pumps: one pumping fuel, one pumping lubricating oil, one pumping engine coolant, possibly another pumping high pressure hydraulic fluid for power steering, and a hydraulic pump attached to a foot-pedal, which activates the brakes.

Pumps are employed to move materials ranging from molten metals at very high temperatures, to cryogenic materials at extremely low temperatures. They are used to generate pressures so small as to be barely perceptible or pressures so high that the liquid being pumped is capable of cutting through material as though it were a saw. Also, they are designed to supply quantities from as small as one drop per day to four billion litres per day. They have power requirements from a few watts to nearly 75 megawatts.

Some of the more common types of pumps required in industrial plants are:

1. Boiler feed water pump - supplies the boiler with feed water as required. It must be capable of forcing this water into the boiler against the pressure existing in the boiler.

2. Fuel oil pump - used in oil-fired boilers to pump fuel oil to the burners.

3. Lubricating oil pump - used to circulate oil to the bearings of a machine such as a turbine, engine, pump, or compressor.

4. Circulating water pump - also called a cooling water pump. It is used to pump water through a heat exchanger such as a condenser or oil cooler.

5. Chemical feed pump - small capacity units are used to pump chemicals into boilers; larger units are used as process pumps.

6. Fire pump - used to supply water to plant fire lines.

7. Domestic water pump - used to supply water to plant washrooms, etc.

1.4 CENTRIFUGAL PUMPS:

A centrifugal pump may be defined as a pump which uses centrifugal force to develop velocity in the liquid being handled. The velocity is then converted to pressure when the liquid velocity decreases. As kinetic energy is decreased, pressure is increased. Centrifugal pumps can be subdivided into the following types: volute, diffuser, axial flow, mixed flow, and regenerative. Although the regenerative pump is not truly a centrifugal pump, it will be considered in this classification.

The general construction of the volute centrifugal pump is shown in Fig 1.1. The liquid being pumped is drawn into the centre or eye of the impeller and is discharged from the impeller periphery into the volute casing. The volute casing has an increasing cross-sectional area as it approaches the pump discharge. In this area, the velocity of the liquid discharged from the impeller is lowered and converted to pressure.

To make the conversion from velocity to pressure more effective, stationary diffuser vanes can be installed around the rim of the impeller. This construction gives rise to the term diffuser centrifugal pump as shown in Fig. 1.2.

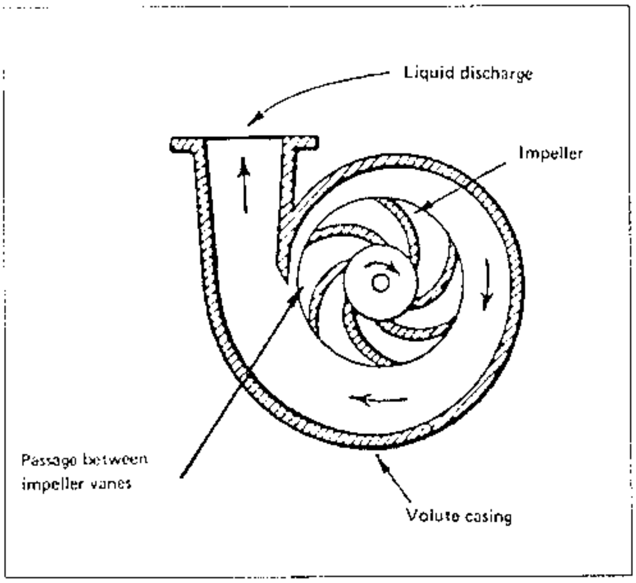

Fig1.1 Volute Centrifugal Pump

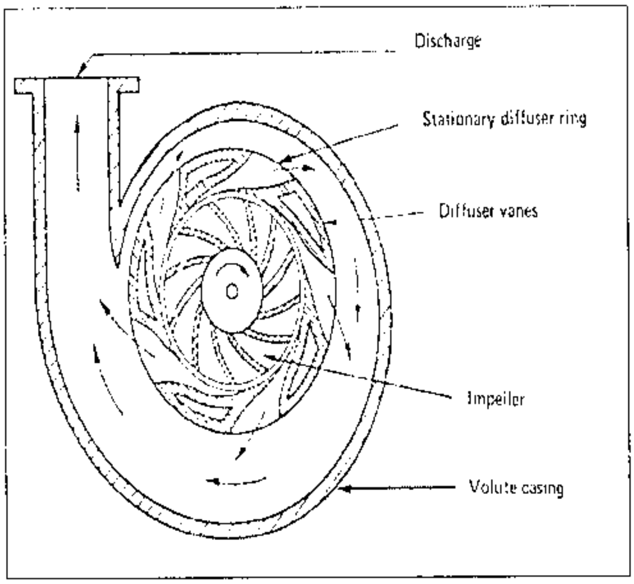

Fig1.2 Diffuser Centrifugal Pump

The combination of centrifugal force and pressure created by velocity decrease accounts for the total pressure developed by the volute or diffuser pumps.

A very popular pump used to provide forced circulation in hot water heating and chilled water cooling systems of small and medium capacity is illustrated in Fig.1.3. It is known in the trade as a circulator. The pump is driven by an electric motor attached by means of a flexible coupling to the pump frame. The pump is directly installed in the piping and, therefore, is also known as an in-line pump. A circulator must be capable of operating quietly and reliably for long periods of time without shut down.

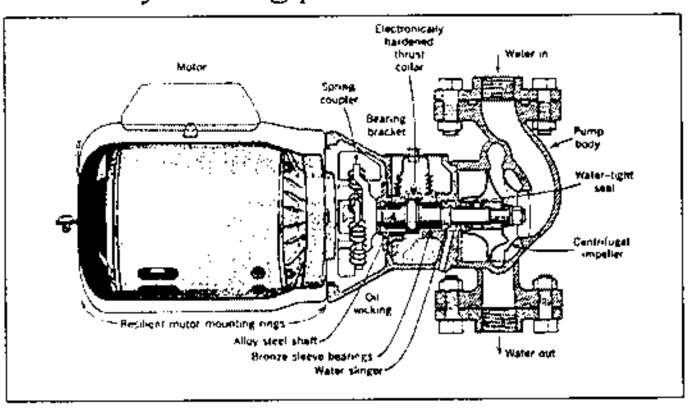

Fig1. 3 Centrifugal Circulator

1.5 CENTRIFUGAL PUMP CHARACTERISTICS:

Reciprocating and rotary pumps are classified as positive displacement pumps, which mean that at a constant speed they move a specific amount of liquid regardless of pump head. The capacity of a centrifugal pump, however, changes with a change in head, thus this pump is not a positive displacement pump.

When the head is increased, the capacity of the centrifugal pump decreases and when the head is lowered, the capacity increases. When the head is increased so much that it exceeds the design head for the pump, the output drops to zero.

It is, therefore, vitally important that the designer of a pumping system carefully calculates the total head of the system in order to be able to select a centrifugal pump that can deliver the required amount of liquid against this head.

The flow of a centrifugal pump can be regulated by adjusting the discharge valve. Throttling the discharge valve increases the flow resistance, thus enlarges the friction head, and the flow will be reduced. The discharge pressure gage on the pump will show an increase in pressure, however this increase is moderate and there will be no danger to the pump as with positive displacement pumps. Even with the discharge valve completely closed the pressure build-up will be well within safe limits.

When the flow is throttled down, the power requirement of the pump is also reduced, notwithstanding the resulting pressure increase. We take advantage of this fact by starting large centrifugal pumps with a closed discharge valve. Since the no-flow power requirement is relatively small, excessive power surging during start-up of the pump can be avoided. This is very important in buildings with a large electric light load, such as large office buildings, where severe dimming of lights due to a power surge can be disturbing.

Centrifugal pumps with double curvature type of impeller gives higher efficiency than normal blades.

CHAPTER 2

PROBLEM IDENTIFICATION

CHAPTER 2
PROBLEM IDENTIFICATION

PROBLEM:

In pump development, the inlet edge of vane cannot always start radially from the eye diameter as in the case of radial vanes. When the specific speed of the pump 'n_s' is increased, the diameter ratio reduces. As a result, the radial inlet of cylindrical vanes cannot provide enough area to develop the required head. In that case the vane extends like an axial vane and becomes a mixed flow type vane. Even then the blade loading is higher, over and above the permitted blade loading calculated for the normal blade.

PROPOSED SOLUTION FOR THE PROBLEM:

To overcome this problem the blade angle has to be increased to bring down the blade loading to the normal level. This can be obtained only by extending the inlet edge from pure radial entry to diagonal entry. The outlet edge is located at higher diameter, which is the eye diameter, whereas the inner diameter will be on the hub. Since meridional velocity C_m is constant at all radii and since the vane velocity are different, blade angle decreases from hub to periphery. The blade becomes 3 dimensional at inlet. When the specific speed 'n_s' increases further i.e., n_s>250 to 450, the outlet edge of the impeller also changes from radial form to diagonal form. In other words vane becomes a purely 3 dimensional or twisted form. Thus it becomes a double curvature blade which can overcome the difficulties of the blade loading.

CHAPTER 3

LITERATURE SURVEY (VANE DEVELOPMENT)

CHAPTER 3

LITERATURE SURVEY (VANE DEVELOPMENT)

3.1 VANE DEVELOPMENT IN THREE DIMENSIONAL COORDINATES:

In many incidents of pump development, the inlet edge of vane cannot always start radially from the eye diameter as in the case of cylindrical vane development. When the specific speed of the pump 'n_s' is increased, the diameter ratio D_2/D_1 reduces. When $(D_2/D_1) < 1.5$, the radial inlet of cylindrical vanes cannot provide enough area to develop the required head. In other words the blade loading is higher, over and above the permitted blade calculated for the cylindrical vane development. The blade angle has to be increased to bring down the blade loading to the normal level. This can be obtained only by extending the inlet edge from pure radial entry diagonal entry. Due to extension of inlet edge from radial to diagonal, the inlet edge now lies at different radii. The outlet edge is located at higher diameter, which is the eye diameter, whereas

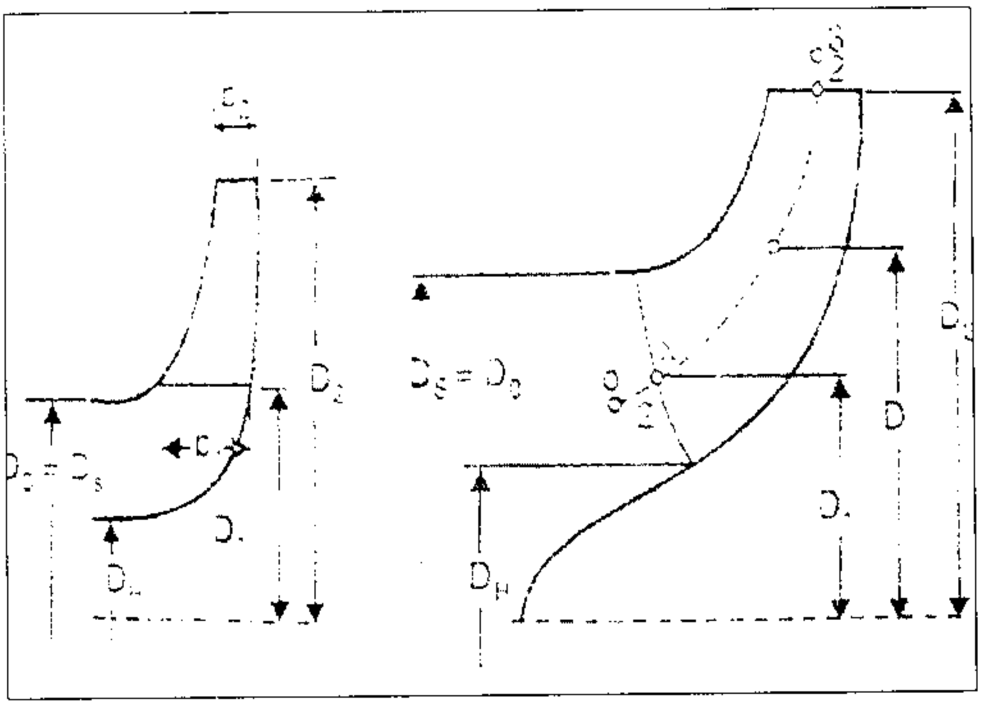

Fig 3.1 Position of inlet edge

a)Single curvature blade b)Double curvature blade

the inner diameter will be on the hub. Other inlet edges are hub diameter and eye diameter. The inlet edges from hub to periphery lie at different radii, since meridional velocity C_m is constant at all radii and since the vane velocity are different, blade angle decreases from hub to periphery. The blade becomes 3 dimensional at inlet.

However, when the specific speed 'n_s' increases further i.e., n_s>250 to 450, the outlet edge of the impeller also changes from radial form to diagonal form. In other words vane becomes a purely 3 dimensional. The outer radius of outlet edge of the outlet larger than the inner radius of the outer edge. When such changes occur i.e., from pure radial to inclined vane pattern. The velocity and pressure distribution change from inner stream line to outer stream line to outer stream line of the blade for same head across the same potential points on all stream lines. So also when blade changes from pure axial. In axial flow pumps too, the velocity distribution from hub to periphery changes along the potential line.

Axisymmetric flow is assumed for such diagonal flow as well as for axial flow conditions. Different flow conditions prevail between streamlines at the periphery. The blade becomes 3 dimensional instead of two dimensional. The blade angle at inlet and at outlet changes to hub to periphery, between the streamlines, drawn in between hub and periphery.

3.2 CONSTRUCTION OF DIFFERENT STREAMLINES:

Construction of different streamlines, called as flow lines are essential because each streamline, due to its location from hub to periphery will have a different vane angle at inlet and at outlet with respect to the neighbouring streamlines. Each streamlines forms one cylindrical circular surface. In order to get the correct change over all parameters from hub to periphery, it is necessary to have additional streamlines between hub and periphery. Similar to the cylindrical surface of revolution, here also additional surfaces of revolution are formed. The number of such surfaces of revolution depend upto 'n' the number of intermediate streamlines, created for the design of vane or blade passage. Each streamline or each surface of revolution is considered as streamline of one radial pump. Each streamline is designed as single curvature radial pump. It is assumed that an equal amount (portion) of fluid passes between from each surface of revolution from inlet to outlet. Accuracy of calculation will be higher if more number of streamlines is selected between periphery and hub. Determination of flow passages i.e., stream surfaces of revolution are determined without vane in the passage. The vane will be added afterwards on the streamline and will deviate or deform in the same way as that of stream surface. However the accuracy of the vane angle determination on the stream surface for each streamline separately will be higher if the blade or passage is near radial and will be inaccurate more and more when the passage deviates from radial to axial pattern.

3.3 PURE AXIAL TYPE (PROPELLER TYPE):

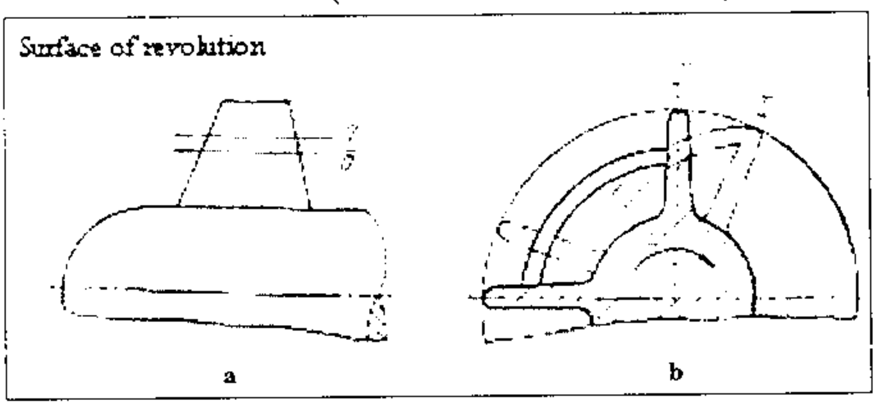

Fig 3.2 Flow in propeller pump

Kaplan type or propeller type blades are axial. Streamlines are cylindrical surfaces. A view of axial blades illustrated in fig is the two elementary streamlines. These two lines are forming as concentric arc of circles in the front view. When impeller rotates at an angular velocity 'ω' the pressure at the leading edge is higher at lower at the trailing edge. Correspondingly as per Bernoulli's equation the velocity at the leading edge is lower than that at the trailing edge. Owing to that, stream cylindrical surfaces 'a' and 'b' rotate in anti-clockwise direction. From inlet to outlet the deviation continuous. The rotation is maximum when the pump is axial and zero in the radial of impellers. In radial blades, stream line shifts to a higher radius and stays there itself. So the hypothesis of keeping stream surfaces as surfaces of revolution is maximum in axial flow and zero in radial flow. In spite of this hypothesis it is followed in all pumps. The stream surfaces are important and studied in meridional section. The inner and outer shrouds are used as steam surfaces. Additional stream surfaces about 1 to 3 members as required for the design the intermediate streamlines are formed. Based on the equal velocity 'C_m' construction. The flow rate is divided equally in all passages. The equal area construction is formed by the law.

$$\frac{\pi\left(d_2^2 - d_1^2\right)}{4} = n\frac{\pi\left(d_0 - d_h\right)^2}{4}$$

Where d_2 and d_1 are the two adjacent streamlines

d_1 is the diameter of the inner stream surface

d_2 is the adjacent streamline diameter

d_h is the hub diameter

d_0 is the diameter of the outer shroud surface

n is the number of flow passages

or

$$(d_2^2 - d_1^2) = n(d_0^2 - d_h^2)$$

These lines form different stream lines in meridional view.

The equal area construction is adopted at inlet to the impeller eye (0-0) and at the outlet end (2-2). It should be remembered that distance between the different streamlines gradually reduce from 0-0 i.e., hub to 2-2 periphery.

In the turning area the pressure at the hub diameter will be more due to centrifugal force and pressure at the peripheral stream line will be less. Correspondingly velocity at the hub streamline will be less and at the outer periphery it will be more. While constructing the intermediate streamlines in between the inlet and outlet especially at the bend an assumption is made as 'C_m' is constant along the normal. This assumption coincides with the practical flow conditions. In practise the pump impellers are designed under equal velocity distribution operates at high efficiency. Normally streamlines are drawn under equal area construction. Referring to fig 3.3 the area is $2\pi r_x \Delta\sigma$ where $\Delta\sigma$ is the distance between the two adjacent streamlines and r_x is the radius of the centre of the circle drawn to these two adjacent streamlines. The diameter of the circle will be $\Delta\sigma$. Normals are drawn by 1st approximation such that normals and streamlines intersect exactly perpendicular to each other since the perpendicularity between the normal and the streamlines is not possible in 1st approximation, correction are made by calculation at all places where the flow turns from radial to axial i.e., at the bend..

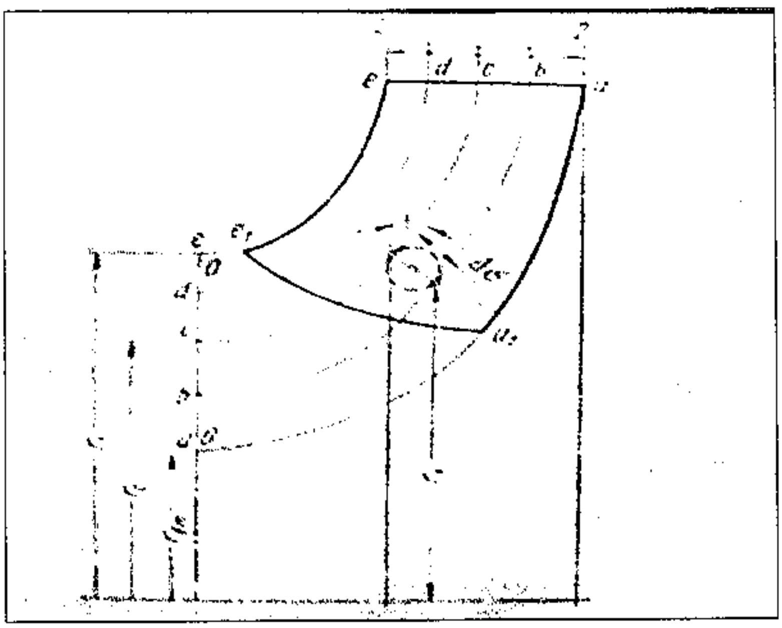

Fig.3.3 Streamline constructing a, b, c, d, e and normal $\Delta\sigma$ for a double curvature

blade system in meridional plane.

It was experimentally verified and found that there is no change in impellers made with corrections and another without correction. Ultimately it is concluded that highly accurate corrections are not necessary.

3.4 CORRECTION OF ENTRY AND EXIT ANGLES:

Having determined the streamlines, the profiling can be developed, for which the location of the inlet and outlet edges are to be determined. However locations of the inlet and outlet edges are to be determined. However locations of the inlet and outlet edges are to be determined later after the profile formation.

Having known the values of u_i , C_{mi} and C_{ui} (between streamline normally(=0) D_i, the diameter of streamlines, and the speed of the pump 'N'. the inlet velocity triangle is constructed for each streamline and the values C_{m1} ,β_1, and w_1 are determined. Inlet edge a-e, (Figure 3.4) is drawn such that it is perpendicular to each streamline. It is desirable to have the point a_1, a little away from the axis. The length of the streamline should be sufficient enough to take the blade loadings and each streamline develops same total head. Angle β is provided graphically to find the shockless entry and then the coefficient of the vane thickness is calculated.

$$\psi = 1 - \frac{\delta z}{\pi D \sin \beta_1}$$

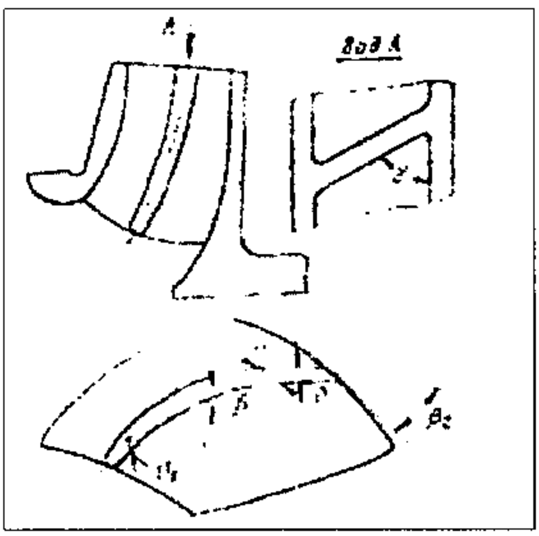

Fig3.4 The pattern of entrance and exit edges in a double curvature blade system.

Procedure to determine β_1 is as follows.

- Determine $\beta_1 = \arctan \dfrac{C'_{m1}}{u_1 - C'_{u1}}$
- Calculate Ψ_1 and β_1 with correction for vane thickness. An angle of attack $\Delta\beta$ is now given
- Find $\beta_1 = \beta_1' + \Delta\beta$

Calculations are carried out for each streamline separately. The inlet edge is finalised after a few position of inlet blade angle. The location of inlet edge depends upon the property laid out at plan. For pumps of small n_s ($n_s \leq 120$) it is better to have small angle of attack at each streamline at the inlet edge. For pumps of higher specific speed ($n_s \leq 200$) where the number of blades are smaller, the condition of suction and hydraulic losses in the vane passage purely depend upon the unequalness of w_1 on the inlet edge. That is why it is recommended to have at inlet edge ($\tan\beta$ / $\tan\beta_1'$) is constant and $w_{1i} = 0.9$ to $1.15\ w_{1av}$

Calculations are carried out for each streamline for constant total head. Corrections for the outlet vane angle is done by the formulae,

$$H_{th} = \frac{\omega}{g}(1-k)\left[\omega r_a^2 - \frac{Q}{2\pi b_2 \psi_2 \tan\beta_2} - (C_u r)_{av}\right]$$

Usually taken values are,

k=0 (for more number of vanes)

$\beta_{2n} = \beta_2$ (blade angle at outlet edge)

$$r_a^2 = r_2^2 y_n$$

$$\text{Where, } y_n = 1 - \frac{\pi \sin \beta_2}{Z}$$

The area reductions coefficient due to vane thickness at outlet will be

$$\Psi_2 = 1 - \frac{\delta_2 Z}{\pi D_2 \sin \beta_2} \qquad \text{usually (0.95 to 0.9)}$$

3.5 VANE CONSTRUCTION FOR III DIMENSIONAL FLOW:

From the outlet blade angle β_2 (β_2 goes in inclined direction (plane y_n and Ψ_2)

$$y_n = 1 - \frac{\pi \sin \beta_2}{Z}$$

$$\Psi_2 = 1 - \frac{\delta_2 Z}{\pi D_2 \sin \beta_2}$$

And to find the value of r_2^2,

These calculations are repeated a few times, especially for this dimensional inlet and for the outlet edges of the impeller vane. Calculations are done for the each streamline separately when the inlet and the outlet edges of the vane are inclined. After determining β_2, r_2 all parameters are known to determine the meridional sections of the impeller. The profile form are now determined with the vane angles β_1 and β_2.

The profile formation for the double curvature blade system is done by conformal mapping method. The vane surface developed in conformal mapping is transformed into a plane which could be spread out. While transforming, the blade angles are kept without any change between any surface and the vaned surface. Conformal mapping gives also similarity of the figures in both the surfaces. Coefficient of similarity remains constant at all the time, in order to have the facility of spreading out, the conical cylindrical surface is developed from the surface, where the vane is located.

The process of conversion is as follows.

1) The selection of the diameter of the cylinder for conformal mapping and the construction of the vane on the cylindrical surface that could be spread out are done with corresponding network.

2) Construction of co-ordinates on flow surface of the vane of the impeller.

3) Transfer of vane profile on the spread out cylinder let us must the original system of co-ordinate for both surfaces. For this lines on the meridional section at every interval of 10 are known ($\Delta\phi = 10$), which when crosses the conical and cylindrical gives a system of meridional plane I,II,III etc.,

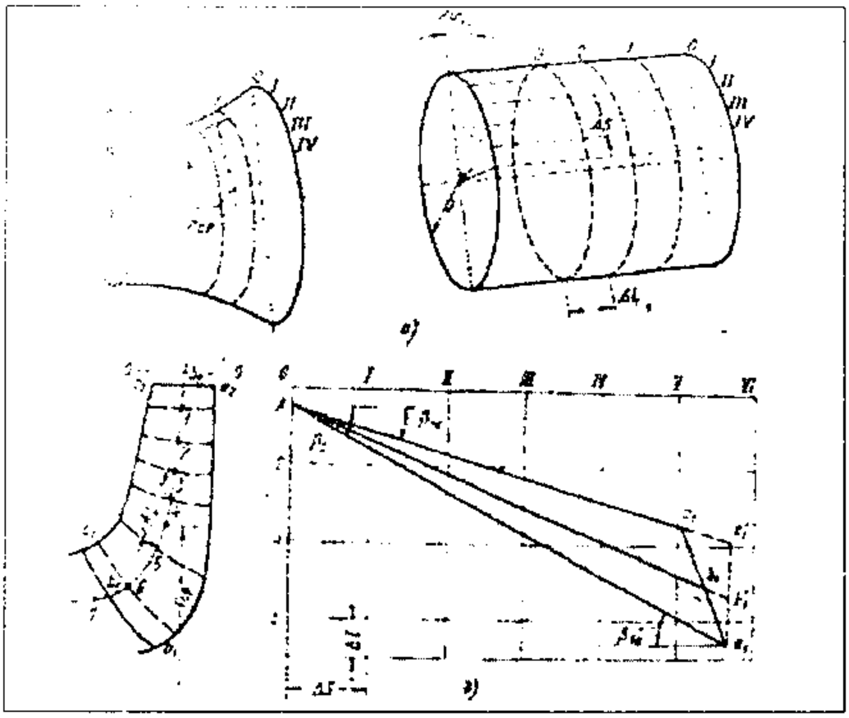

Fig 3.5 Scheme of conformal transformation of vane system (a) streamline and conformal transformation cylinder (b) spread out view.

While developing the second system in parallel it is essential to provide the law of conformal mapping, taking into effect $\Delta\phi_k = \Delta\phi_u$.

Now an equal spaced line parallel on cylinder 1, 2, 3 etc., with a distance of ΔL and on the cone with the distance of ΔL_l (variable pitch) (fig 3.5).

Conformness is affected, if the similarity of the elementary lines perpendicular to each other set in parallel and the meridian of conical and cylindrical as per (fig 3.5)

i.e.,

$$\frac{\Delta L}{\Delta S} = \frac{\Delta l}{\Delta s} \qquad , \Delta S = R\Delta\phi_u$$

$$\Delta s = r_{av}\Delta\phi k$$

$$\Delta\phi_k = \Delta\phi_u$$

$$\frac{\Delta l}{r_{av}} = \frac{\Delta L}{R}$$

$$\Delta l = \frac{\Delta L}{R} r_{av}$$

Since r_{av} changes when moving along the meridional plane (fig 3.5) the value ΔL also changes. This concludes that in equal distributed net parallel on cylindrical surface corresponds to unequal net parallel on the flow surface (conical). The above equations in the differential form will be,

$$\frac{dl}{dL} = \frac{r}{R}$$

On integration we get the relation $l = f(L)$ for which $\Delta l = f(r)$ must be known for the formation of flow surface (conical).

By approximation, for the change in the conical surface, the differential equation will be,

$$\Delta l = \frac{r_0}{\sin\frac{\alpha}{2}}\left(1 - e^{-\frac{\Delta L \sin\frac{\alpha}{2}}{R}}\right)$$

Where r_0 is the radius from which the calculation starts α is the cone angle which changes the cone of curved shape. That's why the above equation holds only good for one angle α and ΔL. For another section α and ΔL are different. In practise, the net is meridional and in parallel and are drawn very close, to get accurate results.

3.6 METHODS TO GET SECTIONS AS PER CONFORMAL MAPPING :

1) The radius of the transformation cylinder R is selected suitably so that the distance ΔS is in whole number in the meridional cylinder (fig 3.5a).

   when $\Delta\varphi = 10^0$

$$\Delta S = \frac{\pi}{18}R \qquad R = \frac{18\Delta S}{\pi} = 5.43\Delta S$$

2) The distance between the parallel lines in the cylinder ΔL does not depend upon ΔS, hence can be selected suitably.

3) It is recommended and not advisable that the starting point of the blade construction should not be at the inlet edge.

4) While making ΔL_i on the streamline (fig 3.5a), the following rule must be followed.

$$\Delta l = \frac{\Delta L}{R} r_{av}$$

However at the beginning it is better to give r_a and find ΔL. After wards rav is checked and corresponding ΔL is calculated correctly by the formulae.

5) To mark the length ΔL_i on streamline a, b, c (fig 3.5b), which can be shows as streamline on conformal spreading of cylinder, the inlet edge is marked approximately.

6) The conformal mapping lines on transfer (average lines of vanes with inlet and outlet vane angle β_1 and β_2 respectively, must be smooth without any abrupt changes. The angles β_1 and β_2 are marked from the point 'a' only (fig 3.5b). the line smoothness does not come the abrupt change comes only if the points a, b, c of inlet edge lie in the section between the directions with angles marked from the point A.

While transforming the streamline by conformal transformation on cylinder the profiling must be done in such a way that the inlet edge lies along one meridional plane (a', b', c') the location of inlet edge of streamline can be located only with one and the same valuation of $w = f(s)$ along the stream. It is necessary to get nearly same angle of coverage of the blade in plans 'Θ' for all streamlines i.e., a_i, b_i, c_i, etc., the location of inlet edge in one single meridional plane, simplifies the production process and also get control. In some cases it provides a good anti cavitating quality in impeller.

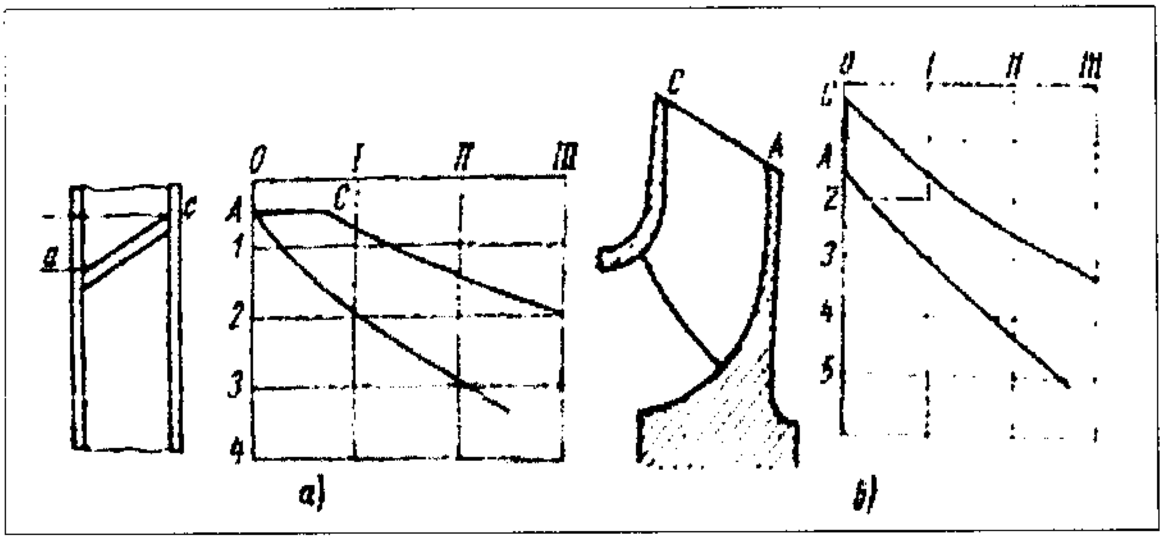

Fig.3.6. Correction in exit edge of the blades on conformal transformation diagram.

Profiling of vanes on conformal cylinder is done, on the meridional sections.

The points of intersection in meridional sections with the vane on conformal net of the cylinders are marked and then the points of transformation are transferred as per the general rule on the meridional plane of the impeller. Marking I, II, III etc., the skeleton of the vane sections (a_i, b_i, c_i) and then the profile thickness (fig 3.5b) we get the leading and trailing edge of the blade in meridional plane of the impeller.

While transferring the meridional part on the meridional projection of impeller. It is essential to know that the meridional part should not lie in the plane of evaluation, but located are and above.

As per the drawing the meridional parts can be considered to be a quality for profiling of vane. The distance between two meridional parts along the stream must change if not, smoothness of the profile cannot be obtained.

3.7 METHOD OF CONSTRUCTION OF VANE ON CONFORMAL:

The difficulty lies when the impeller drawings are prepared for the pump having n_s = 300 to 350. Especially the outer streamlines 'C' because during the profiling the vane becomes very short. For improving the drawback,

1) By increasing the vane angle in places
2) Straightening the outlet end (fig 3.6a)
3) By bending the edges in the meridional plane (3.6b)

After construction of meridional parts of vanes, it is essential to check the profile thickness and its effect on flow.

The coefficient of flow reduction is $\Psi = 1 - \frac{\sigma}{t}$

Where 't' is the pitch = $\frac{2\pi r}{2}$

The vane thickness along the circumferential direction (Fig 3.6a) of radius 'r'.

For cylindrical blades $\sigma = \frac{\delta}{\sin \beta_v}$

For three dimensional blades $\sigma = \frac{\delta}{\tan \beta_v \sin \lambda}$

If when λ is the angle between the parts and the streams (fig 3.6). The angle should not be $\lambda = 60$, because of the difficulty in manufacturing and poor flow pattern when flow is with angle, λ must be checked at the walls, where the conditions are difficult.

3.8 CALCULATION OF VELOCITIES AND MOMENTS OF VELOCITY IN IMPELLERS:

The calculation of moments and velocities are done for each streamline, from inlet to outlet. This is done graphically in the form of (C_u, r); β_v, w_∞, $t/(t-\sigma)$, $1/r \tan\beta_v$, $1/\Psi$ and β_v, vs streamline distance from inlet to outlet. (Streamline a_1, a_2 (r_1 to r_2), $\beta_1 - \beta_2$). From the graph (C_u, r) and w_∞ are calculated. Corrections are made if necessary so as to get smooth change from inlet to outlet of all parameters, which provides the change in thickness uniformly. The change in w_∞ gives the diffuser effect of the impeller channel. The flow pattern in blades is known from the change of (C_u, r). (C_u, r) from inlet to outlet must increase for pumps, where as it should reduce for turbines. It remains constant at non-working regions.

First C_m the meridional velocity along the channel is determined from the elevations or meridional plane of the impeller.

$$C'_m = \frac{C_m}{\psi}$$

The coefficient Ψ is determined from the conformal net of β_v. The relative velocity $w_\infty = C'_m / \sin \beta_v$ is determined from the velocity triangle. For infinitesimal blade, the value Cu is determined as,

$$C_{u r} = u - \frac{C'_m}{\tan\beta_v} + (C_u r)_v$$

Next is to construct the values of w, Θ, C_m for each streamline (a_1,a_2). Similarly for other streamlines b1-b2 etc., At inlet $C_{u1}r_1$ = 0,However ($C_{u1}r_1$ is not equal to 0at inlet due to the provisions of angle of attack, $(C_{u1}r_1)_r$ - $(C_{u1}r_1)$ is the angle of attack.

At outlet $(C_{u2}r_2) = \frac{gH_m}{w} + (C_{u1}r_1)_\infty$

This value lies below the value of $(C_u r)_r$ due to the effect of finite number of impeller blades.

Corrections are made in (C_ur) i.e.,to increase uniformly from inlet to outlet referring to (fig 3.3b), curve 1 indicates abrupt change of (C_ur) at inlet and at outlet as well as area with dead moment of velocity i.e., no change in (C_ur). Due to drastic change in Cur at inlet flow separation and cavitation can occurred. Due to abrupt change in at outlet flow separation will take place. The dead area is at the centre of the length wherein there is no change in (C_ur).

Curve 2 gives amore streamlined variation of C_ur. $C_{u2}r_2$ here is above $C_{u2}r_2$ of the curve drawn, due to the effect of finite number of blades. The change of curve 1 from low level to high level is due to the angle of attack and finite number of vanes. Practically large part of channel should be with a diffuser effect. The value w_∞ does not change near outlet and at inlet. After corrections, graph is drawn for 1/ Ψ and β to determine the smoothness of the blade passage. The value of C_ur at inlet is more than $C_u r_r$ due to the effect that the blade cannot immediately act on the flow and conversion is not yet done. Due to relative vortex there will not be a change in C_ur at the middle i.e., dead zone. The relative vortex changes the eppur of the velocity but there is no change in average moment. The increase in C_ur at inlet and at outlet due to the above facts changes the relative velocity w_r, which can be seen from the velocity triangle as inlet and at outlet.

3.9 DEVELOPMENT OF DRAWINGS OF IMPELLER BLADE :

Vane development is completed with the drawing of impeller in meridional as well as in plan. At meridional section a number of equal distant lines perpendicular to axis is drawn covering the impeller complete, i.e., from end to end, sections 1 to 9 (fig 3.7). The radial distance of these lines are taken by selecting a number of points at the

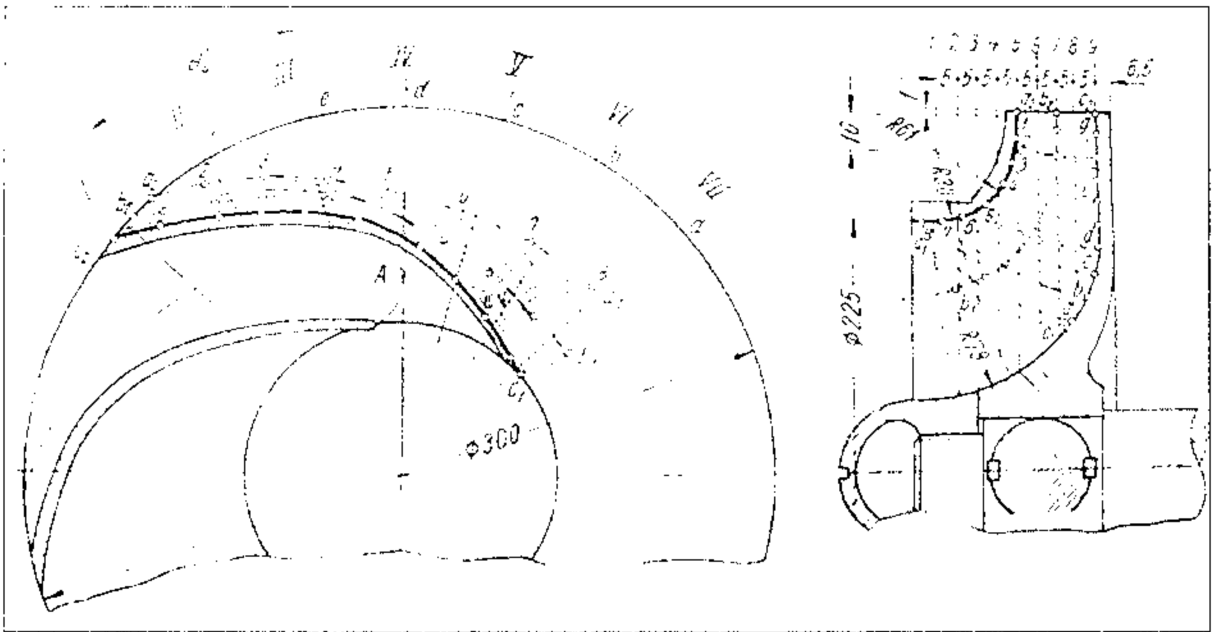

Fig.3.7 Model impeller - developed three dimensional blade.

meridional section and then is transferred to the plan. Then points on this line are drawn in plan to get a complete line. The vane thickness is included on this line to get the other end of the blade i.e., leading edge. The original edge is the trailing edge.

CHAPTER 4

IMPELLER DESIGN

CHAPTER 4

IMPELLER DESIGN

4.1 IMPELLER DESIGN:

Three fundamental parameters namely (1) total head "H" (2) quantity of flow "Q" And (3) either speed of the pump "n" or suction head "H_s" are necessary for impeller design.

The speed of rotation "n" however is related to the size of the pump, and cavitational characteristics of the pump.

If suction head "H_s" is known ,the speed can be determined from suction specific speed (C).from the known value of H, Q, n, specific speed n_s for the pump is calculated by which the type of pump can be determined.

When speed "n" is increased for the given value of Q and H. Specific speed n_s increases. The type of pump changes such as radial, or diagonal, or mixed or axial flow. Also the overall size of the pump is reduced.

It is found that maximum hydraulic as well as overall efficiencies are attained between n_s =150 to 200 for radial type centrifugal pumps.

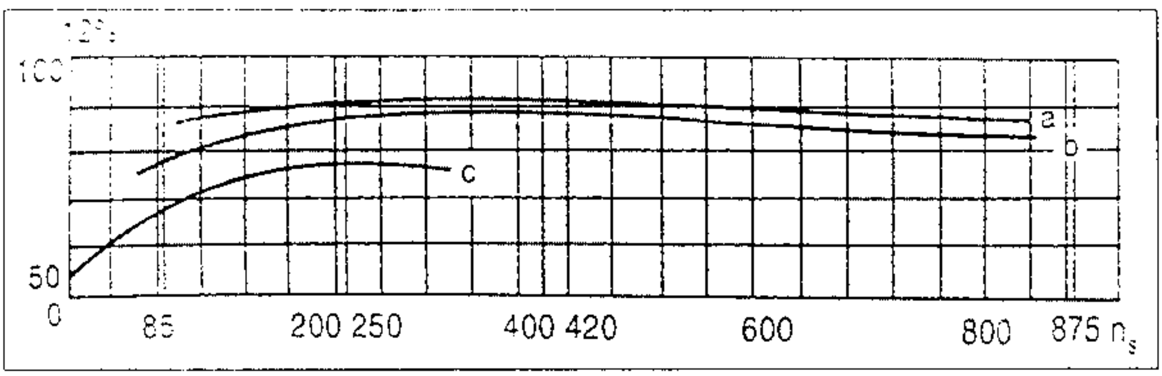

Fig 4.1 Graph h= f(ns, Do)

Suction head H_s is reduced, when speed is increased. Cavitational specific speed "C" can be taken as C=800 to 1000 from which suction head H_s can be determined under first approximation using the formula.

$$\frac{p_{atm}}{\gamma} - \frac{p_{vp}}{\gamma} - \left[\frac{n\sqrt{Q_p}}{c}\right]^{4/3}$$

p_{atm} = atmospheric pressure

p_{vp} = vapour pressure

γ =specific weight of liquid

Correct suction head H_s or specific speed "n" can be established by applying cavitation conditions.

In pump industries, pump is selected mostly from the among the available models Manufactured in the industry. For the available data of H, Q, pump model, so selected, must be capable of meeting the hydraulic and constructional requirement of the field condition. For example: impellers of multistage pump having hub extended into the impeller eye should not be selected for a single stage end suction pump, since the entry in multistage pump impeller is different from entry of liquid in single stage end suction impeller. In single stage end suction pumps entry is radial, whereas in multistage pump entry at suction need not be radial. By applying model analysis, the available models are selected to suit the new requirement. If pumps are not available from the existing model, new designs are made using systematic design procedure. Total head of single stage pumps with standard speed of rotation 1440 rpm will be $H<=30$m.in order to keep the impeller size and weight of the pump within limit. If head for the single pump is more than 30m, then the impeller size and corresponding the total weight of the pump considerably increases. Hence, head and quality for a single stage pump should be selected up to maximum of 30m for n=1440rpm .If pumps are in series, then head per stage will be $H= H_T /i$ where h - head of single stage pump, H_T - total head of the multistage pump, i is the number of stage .If the pump are in parallel then quantity of flow per pump will be $Q_p = Q_T/i$ where Q_p - quantity of flow for one pump, Q_T - total quality required and I - number of pumps to be kept in parallel. If a double suction pump is used then $Q=Q_T/2$.In case of single stage pump, excess quantity is required to take care of axial thrust, Leakage through wearing rings, stuffing box cooling, etc. actual quantity must be increased by an extra of 3 to10 %.i.e., $Q_p = Q_{act} = Q$ when high suction characteristics are essential such as condensate or for gas-liquid pumping speed of rotation must be selected a little lower than normal. A double suction pump is preferred. If a multistage pump is used for such condition, the first stage impeller must be specially designed. The suction head H_s is determined as per the equation.

For the calculated specific, approximation overall efficiency (η) can be obtained by referring the graph ,

The power of the prime mover will be $N_i = \frac{\gamma Q H}{\eta}$.

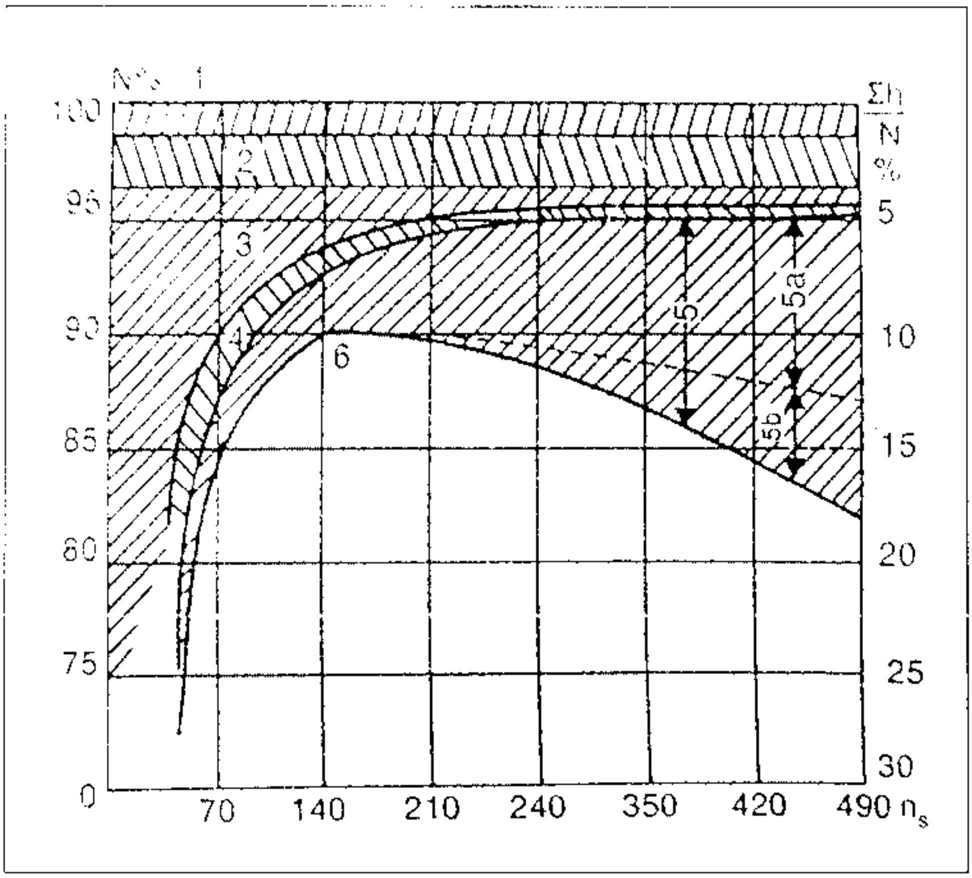

Fig 4.2 Energy balance for pumps of different n_s (1)Mechanical loss (2)Impeller loss (3)Discfriction loss (3)Volumetric loss (4)Hydraulic loss

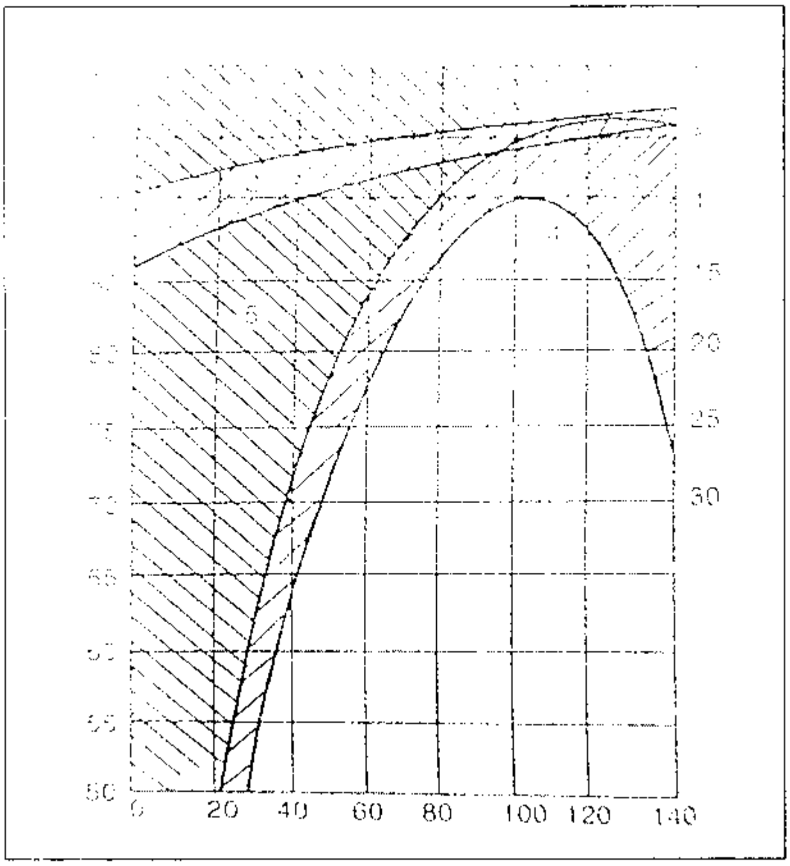

Fig4.3 Q/Q_{nor} % Energy balance for pump (1)Mechanical loss (2)Volumetric loss (3)Hydraulic loss (4) Useful power (5)Recirculation loss

Overall efficiency $\eta = \eta_v \, \eta_m \, \eta_h$ where the $\eta_v \, \eta_m \, \eta_h$ are the volumetric mechanical and hydraulic efficiencies. it is necessary to reduce the volumetric efficiency to 1 to 2 percentage depending upon the condition , in case excess volume is used axial thrust balancing and stuffing box cooling. Prof. A.A.Lomakin has suggested that volumetric and hydraulic efficiencies can be determined as per the equation given below. Mechanical efficiency can be assumed as 1% for larger pumps and 1.5% for smaller pumps. volumetric efficiency is given below

$$\frac{1}{\eta} = 1 + 0.68(n_s)^{-2/3}$$

And $Q_{th} = Q_{act} / \eta_v$ where Q_{act} is the quantity of flow for one pump. η_v the volumetric efficiency lies between85% to 95% for pumps. Hydraulic efficiency is given by ,

$$\eta_h = 1 - \frac{0.42}{(\log d_o - 0.172)^2}$$

$$D_{0(nom)} = (4.5 \text{ to } 4)\sqrt[3]{\frac{Q}{n}} \text{ meters}$$

And $H_{th} = H_{act}/\eta_h$ where h_{act} - total head for one pump. η_h the hydraulic efficiency lies between 75% to 95% and depends upon the shape of the vane passages, surface roughness of the passages and size of the impeller.

Mechanical efficiency lies 1% for larger pumps and 1.5% for smaller pumps.

Based on the head, quality of flow, the power required to drive the pump $N_i = \gamma QH/\eta$ where γ is the specific weight of the pumping liquid. The above equation can be written in different forms such as $N_i = WH/\eta$ since $\gamma Q = W$, weight of the pumping liquid flowing per unit time. Also $N_i = PQ/\eta$, since $\gamma H = P$

The total pressure required for the pumping liquid. N_i is the power input to the pump at coupling and is equal to the power output from the prime mover. If the efficiency of the prime mover is known, the power input to the prime mover

$$N_{ipr} = \frac{N_{op} - N_t}{\eta_{pr}}$$

4.2 DETERMINATION OF SHAFT DIAMETER AND HUB DIAMETER:

Having known the total head, quantity of flow and power, the shaft diameter "ds" can be determined based on the material selected for shaft, its yield strength for bending and torque to be transmitted .A factor of safety of 2to 6 is used depending upon the type of the pump.

In order to take care of the operation of pumps under overloading, a 10% to 15% extra power, over and above normal rated power is taken for shaft diameter design

Power required is $(1.1 \text{ to } 1.5)N_i = T\omega$ where T is the torque transmitted in N.m and ω is the angular velocity of the shaft $\omega = 2\Pi n/60$, where n is the speed rpm. If f_u is the ultimate strength of the shaft material selected, the yield strength f_s for boring, fatigue and shear operating condition, $f_s = f_u/FS$ where FS is the factor of safety (2 to 6) shaft diameter d_s is determined from the formula

$$\frac{\pi d_s^3}{16} f_s = T$$

Hub diameter d_h will be d_h=1.2 to1.3 d_s

Depending upon the pump capacity. It is necessary to select the hub diameter to accommodate impeller key with sufficient space especially for smaller pumps.

4.3 DETERMINATION OF INLET DIMENSIONS FOR IMPELLER:

Normally the eye velocity C_0 will be 3 to 5 mps. however, it can be determined as,

$$C_0=0.06 \text{ to} 0.08\ 3\sqrt{(Qn^2)}$$

Eye diameter D_0 is determined as

$$Q_{th}=c_0.(\Pi/4)(D_o^2-d_h^2) \text{ or}=c_0\ (\Pi/4)D_0^2$$

Depending upon type of construction of the pump such as multistage pumps or double suction or single stage end suction pumps. Eye diameter d_0 is rounded off nearest standard pipe size and then correct value of c_0 is again determined from the continuity equation

The position of the inlet edge of the blade in impeller must be based on the required cavitational characteristics. Radial type , low specific speed centrifugal pump will have the inlet edge of the impeller parallel to the axis. At higher ranges of specific speeds the inlet edge of the impeller is extended into impeller eye in order to provide batter cavitational characteristics. The inlet edge of the impeller blade will be inclined instead of purely parallel to shaft axis. In other words, the inlet edge of the impeller blade gradually extended from purely axial to diagonal when specific speed of radial type centrifugal pumps increases, in order to improve the cavitation characteristics of the pump.

Diameter D_1 is selected as D_1=0.70 to 1.1 D_0, when specific speed ranges from 300 to 70.

Taking c_{m0} the meridional velocity before the blade inlet as $C_{m0}=C_0$ or 1.05 to 1.1C0 the breath B_1 at inlet is calculated from the continuity equation.

$$Q_{th}=\Pi D_1 B_1 C_{m0}$$

$$B_1=\frac{Q_{th}}{\pi D_1 C_{mo}}$$

The inlet blade angle is determined as

$$\text{Tan } \beta_{10}=\frac{C_{m1}}{u_1}=\frac{K_1 C_{mo}}{u_1}$$

$$C_{m1}=k_1 C_{m0}=\frac{\pi D_1 C_{mo}}{\pi D-\frac{Z\delta}{\sin\beta_{10}}}=\frac{1.C_{mo}}{1-\frac{Z\delta}{\sin\beta_{10}}}=\frac{t}{t-\frac{\delta}{\sin\beta_{10}}}$$

$$t=\frac{\pi D_1}{Z}$$

Selection of number of blades may be carried out .normally number of vanes is selected as Z_i=6 to 8 depending upon the specific speed. The pitch or blade spacing (t) can be calculated as $t_1=\Pi D_1/Z$. vane thickness can be selected for strength and at the same time as minimum thickness as possible to get more flow passage area between any two blades and also to get proper vane shape while casting in foundry.

As first approximation β_{10} is determined from the above equation. This value is substituted in equation and the coefficient K_1 is calculated.

This value is now substituted in equation this value is substituted in equation to get new value of K_1. this value of K_1 is now substituted in equation to get the second value of β_{10}. this process is repeated until two successive values of β_{10} and K_1 are same. The blade angle β_1 is determined by adding the angle of attack δ ex: $\beta_1 = \beta_{10}+\delta$ as mentioned earlier .Final value of C_{m1} is determined from $C_{m1}=K_1C_{m0}$. thus, all parameters for impeller blade inlet D_1,B_1,C_{m1},K_1,u_1, β_{10},Z are available for further the calculation to determine the parameters at impeller blade outlet

4.4 DETERMINATION OF OUTLET DIMENSIONS OF IMPELLER:

The relative velocity "w_1" at inlet will be

$$W_1 = \frac{C_{m1}(=K_1 C_{m0})}{\sin\beta_1}$$

The meridional velocity at outlet Cm2 is selected as C_{m2}=0.8 to 0.9 C_{m1} and the relative velocity at outlet W_2 is determined asw_1/w_2=1.1 to 1.15, since the blade passage is a divergent passage. It is also necessary to take uniform change of w and C_m between inlet and outlet of impeller passage, in order determined the blade angle β at different radii between inlet and outlet of the impeller blade passage. Also losses will be less and hydraulic efficiency will be higher.

Outlet parameters are determined by approximation method, and then corrected, since the coefficient ψ and p to determine the total head reduction due to finite number of blades, determination of number of blade, are all function of outlet blade angle and outlet diameter.

As first approximation $\hat{C}_{u2}$ is selected as $\hat{C}_{u2}$=0.8 to 0.5 for specific speeds 75 to 250.

Manometric head $H_m = \frac{H}{\eta_h} = \frac{u_2 C_{u2}}{g} = \frac{\overleftrightarrow{c_{u2}} u_2^2}{g}$ for normal entry at inlet

$$u_2 = \sqrt{\frac{gH_m}{C_{u2}}} \quad \text{and} \quad D_2 = \frac{60u_2}{\pi n}$$

D_2 determined from first approximation, is used to determined outlet blade angle β_2, number of blades Z and the head correction coefficient ψ and p. from velocity triangles at inlet and at outlet,

$$W_1 = \frac{C_{m1}}{\sin\beta_1} \text{ and } w_2 = \frac{C_{m2}}{\sin\beta_2} \text{ from which}$$

$$\frac{W_1}{W_2} = \frac{C_{m2}}{\sin\beta_2}.\frac{\sin\beta_1}{C_{m1}} = \frac{C_{m2}}{C_{m1}}.\frac{\sin\beta_1}{\sin\beta_2}.\frac{K_2 C_{m3}}{K_1 C_{m0}}.\frac{\sin\beta_1}{\sin\beta_2}$$

$$\text{Sin}\beta_2 = \frac{C_{m2}}{C_{m1}}.\frac{W_2}{W1}\sin\beta_1 = \frac{K_2 C_{m3}}{K_1 C_{m0}}.\frac{W_2}{W_1}\sin\beta_1$$

Since β_1 is known C_{m2}/C_{m1}, W_2/W_1, value β_2 can be determined. Values ψ, Z, p are determined from equation the value $H_{th}=((1+p)H_m$ is determined. The outlet vane velocity u_2 is determined from equation and then $d_2 60u_2/\pi n$. Outlet breadth. B_2 is determined as

$$k_2 = \frac{t_2}{t_2 - \dfrac{\delta_2}{\sin\beta_2}} \text{ and } C_{m3} = \frac{C_{m2}}{K_2} \quad Q_{th}=\pi D_2 B_2 C_{m3}$$

Relative velocity at outlet $W_2 = \dfrac{C_{m2}}{\sin\beta_2}$

If D_2 value determined by I and II approximation vary too much, then D_2 determined from IInd approximation should be substituted in all equation to determined the outlet dimension and the process should be repeated until successive values of D_2 are same.

4.5 DELVOPMENT OF FLOW PASSAGE IN MERIDIONAL PLANE:

After determining inlet and outlet parameters of impeller blade, the development of flow passage in meridional plane (elevation) should be determined before developing the blade shape in plan

Selection and formation of flow passage depend upon the specific speed of the pump. The radius of curvature at the bend portion must be large as possible in order to provide a smooth change over from axial to radial direction. The criteria for construction of such flow passage is to provide an uniform change in area from eye to outlet of impeller and at the same time providing velocity C_0 at eye, C_{m0} at inlet C_{m3} at outlet.

From the established dimension at inlet and outlet for the impeller, a graph indicating the variation of C_m, w, β, δ, B from inlet to outlet as a function of diameter D should be prepared. The uniform change in C_m and W is suitably assumed between the inlet and outlet and the graph is drawn. The blade angle β will be $\beta = \sin^{-1} C_m/w$. similarly, the blade thickness δ can be assumed. Blade thickness is always determined based on the blade loading and facility available at foundry to cast as minimum thickness as possible which provides more flow passage area. Normally blade thickness is gradually increased from inlet to some distance approximation up to 1/3 to 2/5 of the blade length and then decreases up to outlet. Usually 4mm to 6mm for smaller pumps and 10mm to 12mm for larger pumps are selected. A graph δ=f(D) is drawn. The breath of the blade at any diameter can be determined from the equation

$$Q_{th} = \left(\pi D - \frac{Z\delta}{\sin\beta} \right) B C_m$$

The value β, δ,C_m are taken from the graph for the selected diameter "D" a graph B=f(D) is drawn in the same graph.

From impeller eye to blade inlet edge, the graph can be extended to get complete the flow passage.

The continuity equation at impeller eye portion will be $Qth = \pi D_0^2/4.C_0$ for end suction pumps.

This can be changed as $Q_{th} = \pi D_0^2/4.c_0 = \pi D_m B C_0$ where D_m is mean diameter=$D_0/2$ and B is the equivalent breadth $B=D_0/2$

Similarly $Q_{th}=\pi/4(D_0^2-d_h^2).C_0$ for double suction and multistage pumps. This can be modified as

$$Q_{th}=\frac{\pi}{4}\left(D_o^2-d_h^2\right)C_o=\pi D_m B C_0$$

$$D_m=\frac{D_o+d_h}{2} \text{ and } B=\frac{D_0-d_h}{2}$$

Depending upon the specific speed, the shape of the middle stream line (D_m from eye to inlet and D from inlet to outlet)is drawn. it should be remembered, that the radius of curvature at the bend, where the flow direction changes from axial to radial must be as larger as possible at inner and outer should for better performance. On this streamline, a number of circles are drawn, at frequent intervals, selecting to the breadth "B" for selected diameter (D) this value of B can be obtained from the graph. Lines are drawn at both ends of the circle such that line drawn must be tangent to all circles. These two lines from inner and outer shrouds of the impeller.

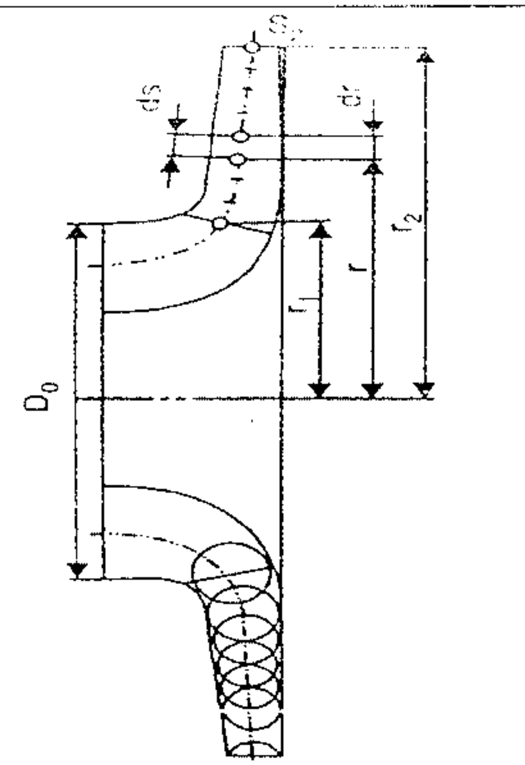

Fig 7.10 Three Dimensional front view of double curvature blade

If an arc is drawn connecting the meeting tangent points on shrouds and the centre of circle the angle between the arc and tangent should be 90^0. A graph can be drawn between areas "A", A=f((S). which must have the shape in the fig.

For better cavitational characteristics the rate of area increase at the bend portion, where the flow changes from axial to radial direction must be at a larger rate then the area increase at the radial portion. By providing considerable increase in area at the inlet suction, the rate of increase in area at the radial direction will be at a lower rate. moreover, significant increase in area at inlet compensates the area reduction due to vane thickness at inlet.

Radius of curvature at the bend portion of the meridional passage at the outer side be as large as possible since smaller radius of curvature at this point yields high velocity of flow as well as flow separation after the bend, which drastically reduce cavitational property and hydraulic efficiency flow separation at this point will create very poor flow in the following radial portion as well as the inlet of the impeller, all will reduce the hydraulic efficiency.

In general, meridional flow passage development must possess,

1. Smooth, streamlined and uniform area change from eye to the outlet must be ensured.
2. Then radius of curvature at the outer side of the bend portion must be large as possible.
3. Contour of flow passage must be in the same pattern as that recommended for that specific speed.

The diameter D_2/D_1 reduced, when specific speed ns increase when $D_2.D_1<1.6$, the surface area of the vane significantly reduces if the inlet edge of the blade lies in the radial portion of the passage. The blade loading will be higher, which in turn, reduces the cavitational. Hydraulic losses are increased. To overcome this, the blade inlet is extended into the bend portion. The inlet edge of the blade, instead of parallel to axis, will be inclined. The blade passage changes from diagonal at inlet to radial at outlet.

This inurn reduces the blade velocity and relative at inlet. This reduces hydraulic losses and improves cavitational characteristics and reduces blade loading. due to the inclined location of inlet edge, the radius from hub to outer changes. Since medicinal velocity Cm is constant throughout the inlet cross-section, blade angle β_1 reduces from hub to periphery. Blade curvature changes, from single curvature to double curvature. The inlet edge will be diagonal and outlet edge will be parallel to axis for specific n_s=200 to 300. when specific speed increases still further ex: for n_s=300 to 500 the outlet edge also become inclined and the pumps will be mixed or diagonal type in stead of radial.

4.6 DEVELOPMENT OF SINGLE CURVATURE BLADE—RADIAL BLADES :

Single curvature blade or plane development is adopted for pure radial blades, where the inlet and outlet edges lie parallel to axis. The specific speed "n_s" of such pump will be less then 100, n_s<100 and normally the diameter D_2<70 to 100 mm.

Vane development, either by single or by double arc method or by step by step method called point by point method, most provide uniform variation in relative velocity "w" meridional velocity C_m and angle of divergence from inlet to outlet alone the flow passage.

Blade thickness "δ" is selected either constant or changing from inlet to outlet, smaller thickness at inlet and at outlet end and higher thickness at the middle. However,

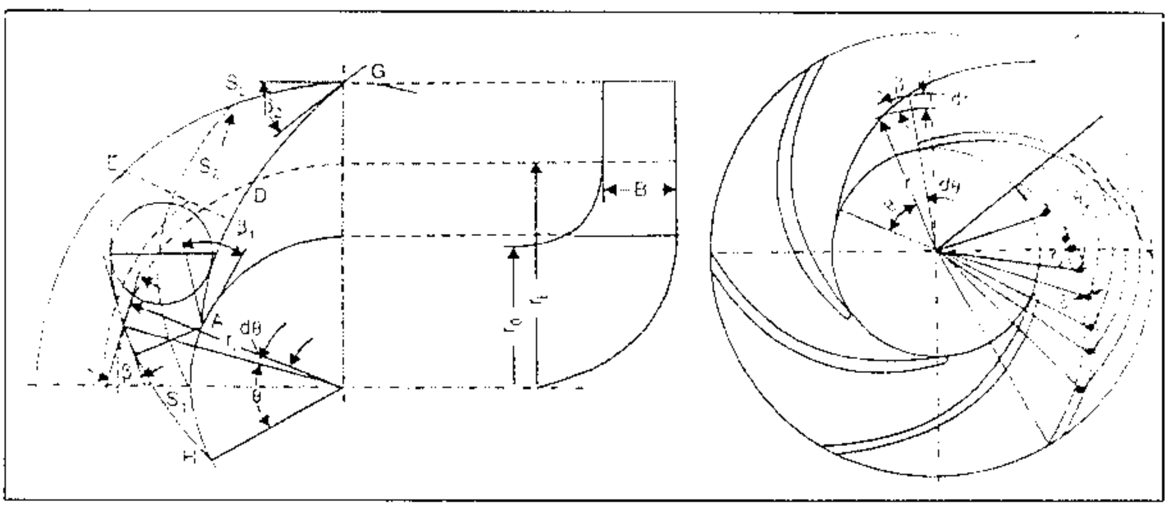

Fig 4.5 Vane development by point by point method

blade thickness is determined based in the blade loading and the type of casting adopted in foundry for casting the impeller. The vane thickness will be a little higher at inlet then that at outlet and will be rounded off at inlet for shock ness entry. For smaller pumps the blade thickness will be 3mm at inlet,5mm to 6mm at the middle and 1mm to 2mm at the outlet. For larger pumps the blade thickness is increased up to 10 to 12 mm. selection of minimum thickness provide a larger flow passage between blades. The velocities Cm and w in the flow passage is reduced, which yields to higher hydraulic efficiency. Flow is also without separation for a wide range of flow rate. Now-a-days airfoils are used, for maximum economy and for better anticavitating property. These profiles are positioned on the stream line "S_1 to S_2" determined by point by point method.

The differential equation at any point between "S_1 toS_2" for the central stream line in plan can be written as

$$\text{Tan}\beta = \frac{dr}{rdr} \text{ or } d\,\theta = \frac{dr}{r\tan\beta}$$

Taking $\theta = 0$ when r=r1 and β from the graph β=f(D)

$$\theta = \int_0^\theta d\theta = \int_{r_1}^{r_2} \frac{dr}{r\tan\beta}$$

Integration is carried out step by step summation of d θ

$$\frac{1}{r\tan\beta} = B(r)$$

$$\Delta\,\theta = \frac{B_i + B_i + 1}{2}\Delta r_i$$

Where $\Delta\,\theta$ and Δr are the increment in central angle and radius Bi and Bi+1 are the integrals at the beginning and at the end of the selected radius. Total value of θ will be

$$\theta_i = \sum_{i=1}^{i=i} \frac{B_i + B_i + 1}{2}\Delta r_i$$

All calculation are carried out in tabular form .

The values of S or r can be arbitrarily selected for which streamline is constructed from the table where θ and the corresponding r are known. Blade thickness is added on the streamline, to get the blade in complete shape.

4.7 DEVELOPMENT OF DOUBLE CURVATURE BLADE:

Increase the speed of the impeller reduces the overall dimensions, total Wight and the cost of the pump. The specific speed of the pump ns increases. Diameter ratio D_2/D_1 reduces.

If radial vanes are provide when $D_2/D_1<1.6$ and specific speed is $150 \leq n_s \leq 250$, the specific, load on the vane increases, due to the reduction in the effective vane area. Cavitational property of the pump also reduces. In order to overcome this, the vane is extended into impeller eye i.e.; vane will be diagonal at the inlet of instead of radial.

If the increase in specific speed is still further, $300 \leq n_s \leq 600$ the outlet edge of the vane also becomes diagonal. Each stream line of the vane will have its own configuration. i.e.; the vane angle β_1 and β_2 are different from hub to periphery. The vane will be twisted form i.e.; double curvature.

Due to the change in direction of flow for axial, to diagonal, uniform steady flow no longer exist. The velocity field considerably changes at the inlet and at outlet. This complicates the pattern of flow. Existing element theory of pumps with average velocity assumption along the circumferential and along the radial direction cannot be assumed. A simple but considerably accurate scheme has to be developed. Axisymmetric flow. i.e.; flow with infinite number of vane is commonly adapted for this type of flow.

Theoretical investigation under axisymmetric flow with infinite number of vanes in medicinal section of flow will be equal velocity construction. This has been suggested by so many authors.

One of the methods of construction of diagonal type of impellor is the assumption of constant head along all surface of revaluation where the flow line lies. By applying Kelvin theorem the vertex free flow. Ex- potential flow $\omega_u = 0$ suggested by Bowersfield is attained in the vane system as a result of which the circulation along any contour is constant.

CHAPTER 5

SINGLE AND DOUBLE CURVATURE BLADES

CHAPTER 5

SINGLE AND DOUBLE CURVATURE BLADES

5.1 GENERAL REMARKS ON DESIGN OF RUNNERS AND IMPELLERS:

The shape of a flow passage for either a runner or an impeller depends on the H, Q, and N—represented by the specific speed. These in turn are functions of:

1. The outlet tip speed, u_2, and the outlet meridional velocity, C_{m2}
2. The outlet blade angle, b_2
3. The number of blades, z
4. The ratio. C_{u2}/C_{u3}
5. The diameter ratio, d_1/d_2

5.2 SINGLE-CURVATURE DESIGN:

As an example of the design problems and techniques used for single curvature blades, the problem of centrifugal pump design will be used. The methods will be equally applicable to turbine runners.

For a centrifugal pump, head may be maintained at the same value with a smaller u_2 and a smaller d_2 with the same rotational speed N by increasing b_2 and z. Thus, the problems of design and associated calculations may result in several solutions, but they will not be of equal value in terms of efficiency and possibly production costs.

5.2.1 MERIDIONAL VELOCITIES, INLET DIAMETER, AND INLET ANGLE:

Meridional velocities are calculated from:

$$C_{m1} = K_{Cm1} \times (2gH)^{0.5}$$

$$C_{m2} = Kc_{m2} \times (2gH)^{0.5}$$

K_{cm1}; K_{cm2} are velocity coefficients.

c_{m1}; c_{m2} are meridional velocities

Values of K_{cm1} and K_{cm2} given in a plot by Stepanoff (1957) have been modified for units and plotted on an arithmetic basis rather than a logarithmic basis by Stepanoff. An empirical Russian equation that is sometimes used for the initial calculation of an inlet diameter is:

$$d_0 = (4.0 - 4.5)\,(Q/N)^{1/3}$$

b1, the inlet angle, is calculated from:

$$\tan\,\beta_1 = c_{m1}/u_1$$

Where,

$$u1 = (N/60)\ \pi\, d_1$$

Experiments have shown that Equation does not give the volumetric flow rate at the best efficiency point. b_1 must be increased by the value of the incidence angle d_1. This is = 2–6°.

Thus,

$$\beta_1^{1} = \beta_1 + \delta_1$$

5.2.2 TIP IMPELLER VELOCITY (U_2) AND OUTLET DIAMETER (D_2):

The theoretical head for a centrifugal pump with an infinite number of blades was given in

$$H_{th}(\infty) = (1/g)(u_2c_2\cos a_2 - u_1c_1\cos a_1)$$

From the outlet velocity triangle, it follows that:

$$c_{u2} = u_2 - c_{m2}/\tan\,\beta_2$$

Substitution of Equation in one another yields:

$$gH_{th}(\infty) = u_2(u_2 - c_{m2}/\tan\,\beta_2) - u_1c_{u1}$$

or

$$u_2^2 - u_2\,(c_{m2}/\tan\,\beta_2) = gH_{th}(4) + u_1c_{u1}$$

Hence,

$$u_2 = c_{m2}/2 \tan \beta_2 + [(c_{m2}/2 \tan \beta_2)^2 + gH_{th}(4) + u_1c_{u1}]^{0.5}$$

Usually, $c_{u1} = 0$ because of axial entry of the fluid. Equation becomes:

$$u_2 = c_{m2}/2 \tan \beta_2 + \{(c_{m2}/2 \tan \beta_2)^2 + (gH/\eta_h)(1 + C_p)\}^{0.5}$$

where,

$(1 + C_p)$ is evaluated from the Pfleiderer correction for a finite number of blades.

$$(1 + C_p) = 2(y/z)[1/(1 - (d_1/d_2))^2]$$

Where,

$$y = k(1 + \sin \beta_2)(d_1/d_2)$$

The value of k = 1 or 1.2, depending on whether or not the pump has guide vanes. For pump

without guide vanes k = 1.2.

The outlet diameter is given by:

$$d_2 = [(60)(u_2)]/[(\pi)(N)]$$

5.2.3 INLET AREAS AND IMPELLER WIDTHS:

Inlet areas are given by:

$$A_1 = yQ_1/c_{m1}$$

y = a coefficient of constriction. This allows for reduction of flow blade area because of the presence of the blades.

$$Q_1 = Q/0.96$$

The value 0.96 is a commonly used value for volumetric efficiency.

Impeller widths at inlet and outlet are calculated from:

$$b_1 = A_1/\pi d_1$$

and

$$b_2 = A_2/\pi d_2$$

5.2.4 DIMENSION CALCULATIONS, CONTINUITY ADJUSTMENTS:

There are three principal methods for designing blades:

1. Circular arc method
2. Point-by-point method
3. Conformal representation

The first of these methods may be carried out by a single-arc or double-arc method. Both methods are less accurate than the second—the point-by-point method. The point-by-point method, given was originally introduced by C. Pfleiderer (1957) and is illustrated in Figure 5.1.

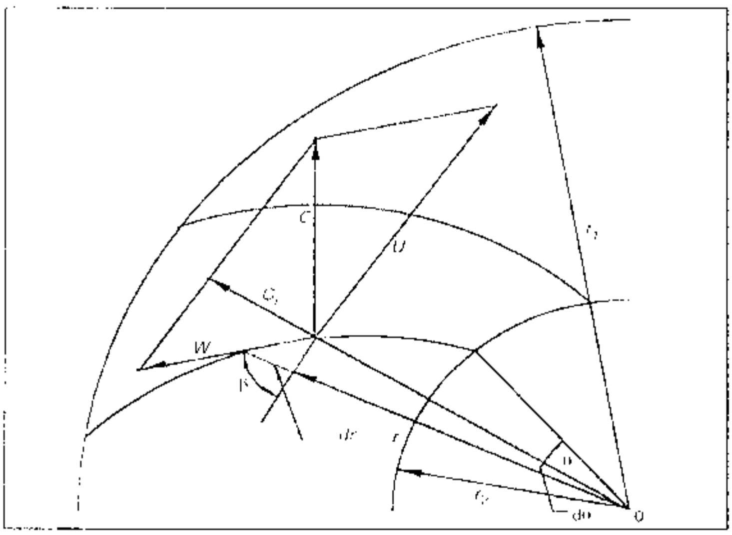

Fig 5.1 Point by point method of determination of blade profile

Referring to the figure, the angle increment dq is given by:

$$d\theta = dr/(r \tan \beta)$$

Integrating between r2 and r gives q, expressed in degrees:

$$\theta = (180/\Pi) \int_{r2}^{r} (r \tan \beta)^{-1} \, dr$$

When the initial blade design is done, a final check on the dimensions of the passages between the blades must be made. In effect, this is a check for flow continuity. The normals to the blade surfaces are drawn by means of inscribed circles, as illustrated by Figures 5.2 and 5.3. A given cross section has the shape of a trapezium of height Δe and breadth b. Figure illustrates the type of deviation from the trapezoidal shape. The blade shape is then corrected for each cross section.

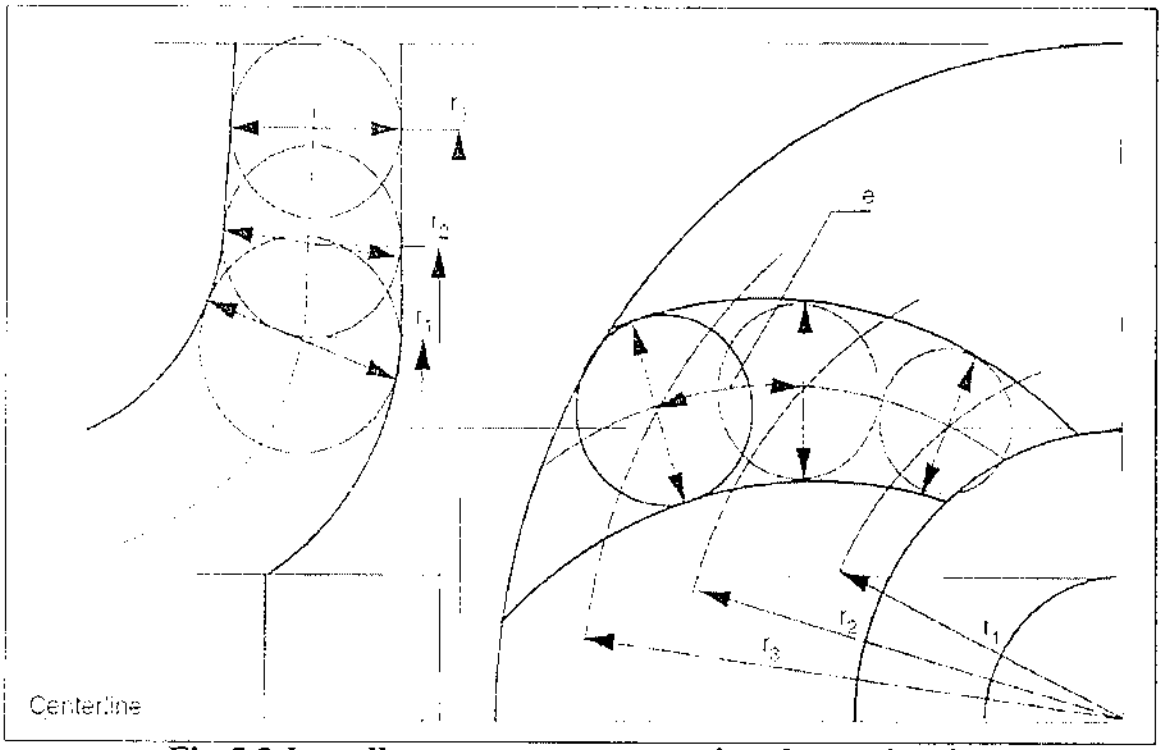

Fig 5.2 Impeller passage cross section determination

5.3 DESIGN OF DOUBLE-CURVATURE BLADES BY CONFORMAL MAPPING:

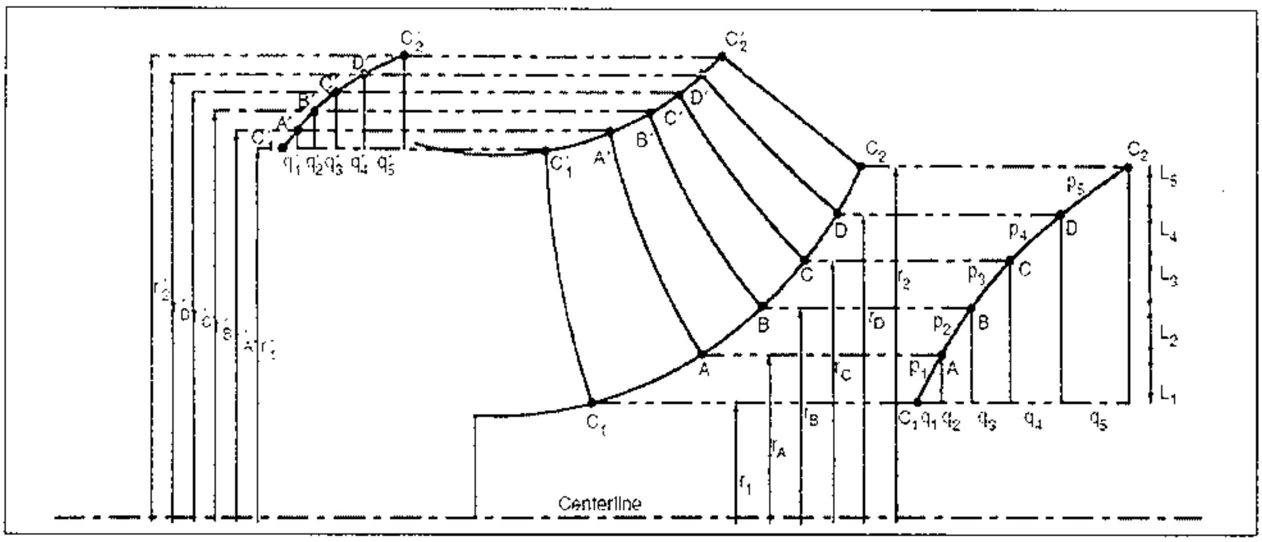

Fig 5.3 Determination of blade surface by conformal mapping

V. Kaplan, the turbine designer, presented a variation of conformal mapping that gives rapid results. Figures best illustrate the method.

The basic construction is as follows:

1. The back and front shroud streamlines are divided into several segments. In Figure 5.3, five segments have been arbitrarily chosen: p_1, p_2 . . . etc. For greater accuracy, 10 to 15segments would be preferable.

2. Planes are drawn through the division points C1 : J : K : L : C2 perpendicular to the impeller axis. The traces of the intersection with the surface of the back shroud are concentric circles - $q_1 : q_2 : q_3$ .....

3. The segments $e_1 : e_2 : e_3 : e_4$ and $f_1 : f_2 : f_3 : f_4$ and $g_1 : g_2 : g_3 : g_4$ form curvilinear triangles.

4. The radii of the points C_1: J : K : L : C_2 are determined as in Figure .

5. From the number of blades and the angle of overlap (usually 35°–50°) the central angle j may be determined.

6. If the outside edge is oblique, then the streamlines will be separated accordingly. From the hydraulic point of view it is preferable that the angles between the impeller and shrouds be as close to 90° as possible. If the shrouds are highly curved, this is almost impossible to manage.

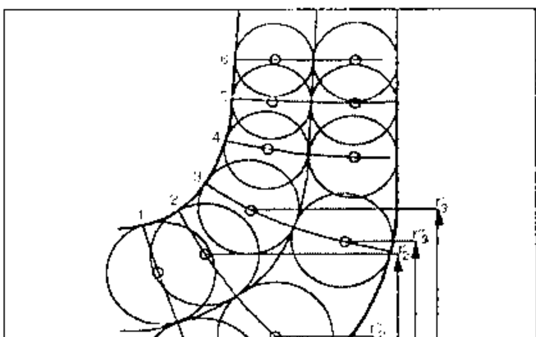

Fig 5.4 Construction of Stream -lines in an impeller channel

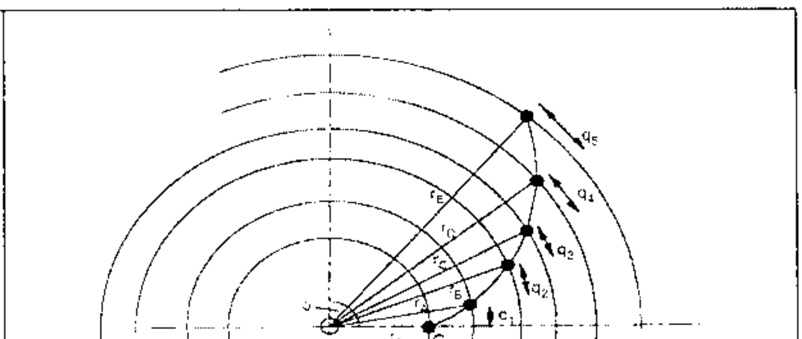

Fig 5.5 End elevation projection of the back shroud showing curvature

CHAPTER 6

KAPLAN METHOD

CHAPTER 6
KAPLAN METHOD

6.1 DESIGN OF THE MERIDIONAL SECTION:

A section through the impeller axis is called "meridional section". In this presentation the blade leading and trailing edges are projected into the drawing plane through "circular projection".

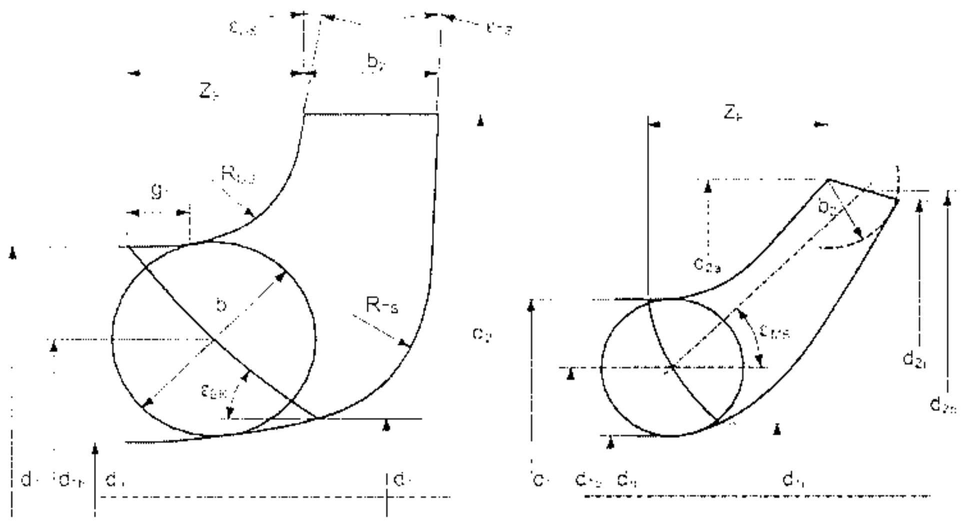

Fig. 6.1. Design parameters for the meridional section of the impeller. Left: radial impeller, right: semi-axial impeller

To be able to design the meridional section, details for the position of the leading edge are required next to the dimensions (d_2, b_2, d_1, d_{1i}, d_n) determined above. Recommendations for the axial extension z_E and the radius of curvature R_{Ds} of the front shroud are given by Equation .

$$z_E = (d_{2a} - d_1)\left(\frac{n_q}{n_{q,Ref}}\right)^{1.07}$$

$R_{Ds} = (0.6 \text{ to } 0.8)\ b_1$
$b_1 = ½\ (d_1 - d_n)$
$n_{q,Ref} = 74$

Favorable flow conditions are achieved by moving the leading edge forward into the impeller eye. In this way low blade loadings and correspondingly moderate low-pressure peaks are obtained and cavitation is reduced. The leading edge at the outer streamline should not be located in the region of high curvature but preferably before the start of the front shroud curvature. Therefore, the radius R_{Ds} should not be tangent to the point defined by z_E, but a short section g_1 = (0.2 to 0.3)×$b_1$should be introduced with only a minor increase in radius in order to achieve flatter pressure distributions. For small pumps, or when a short axial extension of the impeller is a priority, smaller values are selected for z_E and R_{Ds} than calculated from Equation.

The angle ε_{Ds} which defines the shape of the front shroud near the impeller outlet is determined according to the specific speed. It can be used to influence the velocity profile at the impeller outlet. For $n_q < 20$, $\varepsilon_{Ds} = 0$ is often selected. For radial impellers with higher specific speeds ε_{Ds} increases to approximately 15 to 20°. With d_2, b_2, z_E, d_1, g_1, ε_{Ds} and R_{Ds} it is now possible to draw the outer streamline which can be defined by a free curve or assembled from straight lines and circular arcs. When using a design program, Bezier functions may be used. Unfortunately, general rules are not known which would ensure favorable hydraulic features. Meridional section and blade development must rather be optimized together. The angle ε_{Ts} which defines the shape of the rear shroud near the impeller outlet can be chosen positive or negative. At n_q < 30, $\varepsilon_{Ts} = 0$ is often designed (some manufacturers also use negative values). At high specific speeds ε_{Ts} is always positive with $\varepsilon_{Ts} < \varepsilon_{Ds}$.

The inner streamline must be shaped in a way that the cross sections A = 2×π×r×b vary continuously along the mean streamline. To this end, a mathematical rule may be selected which describes the cross section A or the channel width b in the meridional section from b_1 to b_2 as a function of the developed length of the outer streamline, Fig. 6.2. In addition to a linear or parabolic function, a cubic law could also be considered to obtain smaller changes in cross section near the inlet and outlet of the impeller. After the desired rule b = f(L) has been defined, the width is marked off with dividers at a number of checkpoints on the outer streamline. Subsequently, the inner streamline is drawn as an envelope curve. The same construction can also be performed in sub-operations with several streamlines. Finally, the streamlines can be constructed from the theory of potential flows.

The blade leading edge is initially obtained from the chosen parameters d_1, z_E, d_{1i} and the designed inner streamline. The position of the leading edge has a major effect on the part load behavior accordingly. The following criteria serve for shaping the leading edge: (1) In multistage pumps with $n_q < 25$ the angle ε_{EK} must be designed as large as possible to prevent the Q-H-curve from drooping towards Q = 0. (2) For impellers with an axial inlet ε_{EK} = 30 to 40° is about the optimum according to equation. (3) If ε_{EK} is selected too small, problems with hub cavitation and instabilities of the Q-H- curve may be encountered. ε_{EK} too large can result in too low values of d_{1i} and too

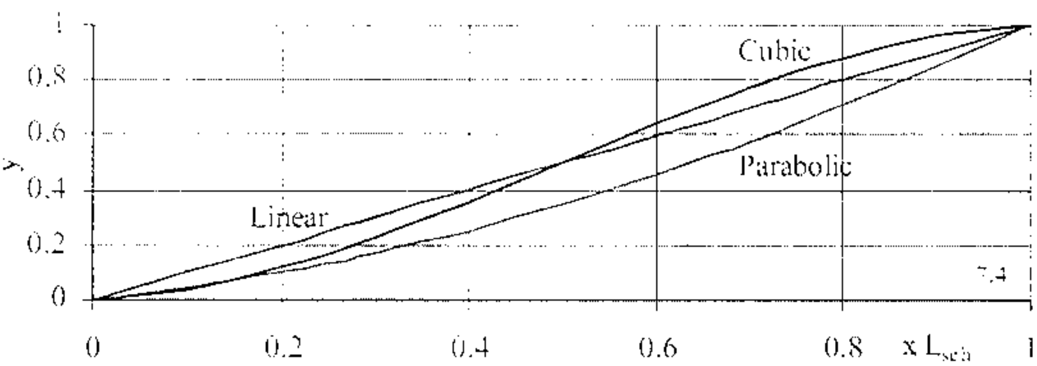

Fig. 6.2. Options for the development of the flow area in the impeller; L_{sch} = blade length

large blade angles $\beta_{1B,i}$ on the hub. The meridional section largely determines the flow distribution over the blade width and, consequently, the turbulent dissipation losses, the part load behavior and the stability of the Q-H-curve.

Criteria for the meridional section to be checked:

- The leading edge should join the outer streamline with the largest possible angle to reduce the local blockage in the case of twisted blades.
- The curvatures of the outer and inner streamlines and the cross sections A =

$2 \times \pi \times r \times b$ should vary continuously along the mean streamline.

6.2 BLADE DESIGN:

The blade design serves to define the shape of the blades along the outer, inner and mean streamlines so that the inlet and outlet angles established are obtained. The blade describes a three-dimensional curve along each surface of revolution; for instance, along the outer streamline defined by the meridional section (or front shroud). This curve can be described through its projections into the meridional section and the plan view. Every point of this curve is defined in space by the coordinates r, z and ε.

With reference to Fig. 6.4a consider an element of the outer streamline which is positioned between the points 5 and 6 on the radii r5 and r6. In the meridional section it has the (approximate) length:

$$\Delta m = \sqrt{\Delta r^2 + \Delta z^2}$$

In the plan view, Fig. 6.4b and 6.4c, the position of the points 5 and 6 is defined by these radii and the angle $\Delta\varepsilon$. The distance between points 5 and 6 is:

$$\Delta g = \sqrt{\Delta r^2 + \Delta u^2}$$

The length of the streamline element in space is obtained from the length in the plan view Δg and the extension in axial direction Δz(normal to the plan view):

$$\Delta L = \sqrt{\Delta g^2 + \Delta z^2}$$

By means of Equations the relationship between the true length ΔL of the streamline element and its projection Δm in the meridional section and the projection Δu in the plan view is obtained:

$$\Delta L = \sqrt{\Delta g^2 + \Delta z^2} = \sqrt{\Delta r^2 + \Delta u^2 + \Delta z^2} = \sqrt{\Delta m^2 + \Delta u^2}$$

The distances ΔL , Δu and Δm according to Fig. 6.4c constitute a right-angled triangle which includes the true angle β between the streamline element ΔL and the tangential direction Δu.

If a streamline is developed step by step into the drawing plane according to this procedure, its true length and the true angles relative to the circumferential direction are obtained. This procedure is used to design the blade coordinates in the plan view from a defined meridional section and a blade development to be specified. In the above method, the curvilinear triangles have been replaced by rectilinear ones. The smaller the elements are selected, the better this approximation becomes. Of the different methods for designing the blades, the Kaplan method is described; it comprises the following steps:

1. Depending on the width of the impeller (i.e. n_q) one to five additional surfaces of revolution are drawn into the meridional section in addition to the outer and inner streamlines. These surfaces can be designed as streamlines of identical part flows. To this end, (estimated) normal to the streamlines are drawn. In doing so, the width between two streamlines is determined in a way that the same flow $\Delta Q = 2\times\pi\times r\times\Delta b\times c_m$ travels through each partial channel (frequently with the assumption cm = constant).

2. All surfaces of revolution or streamlines are sub-divided into n elements of identical length Δm (e.g. Δm = 8 mm) according to Fig. 6.4a so that the points a_1 to a_n, b_1 to b_n etc. are obtained.

3. In the blade development according to Fig. 6.4d, n parallel straight lines are drawn at the distance Δm. Lengths $L_{a,m}$, $L_{b,m}$, $L_{c,m}$ etc. of the developed meridional section are obtained (which constitute the projection of the streamlines into the meridional section).

4. The length $L_{a,u}$ of the streamlines in circumferential direction is still unknown; within certain limits it can be freely selected. A possible approach consists in defining a development law according to Fig. 6.5. Starting with $\beta(j=0) = \beta_{2B}$ at point A in Fig.

6.4d, the circumferential length Δu of the element Δm is calculated from Eq for each step j to obtain the position of the next point.

$$\Delta u_j = \frac{\Delta m_j}{\tan \beta_j}$$

The angles for each step are obtained from the selected law according to which the angle is to develop as a function of the blade length from β_{2B} to β_{1B}. In general it is possible to write:

$$\beta_j = \beta_{2B} - y(x)\,(\beta_{2B} - \beta_{1B})$$

Any random function which satisfies the conditions $y(0) = 0$ and $y(1) = 1$ can be substituted for $y(x)$. If a linear angle development from β_{2B} to β_{1B} is desired, the function $y(x)$ has to be formulated as:

$$y(x) = \frac{1}{L_{a,m}} \sum_0^j \Delta m_j$$

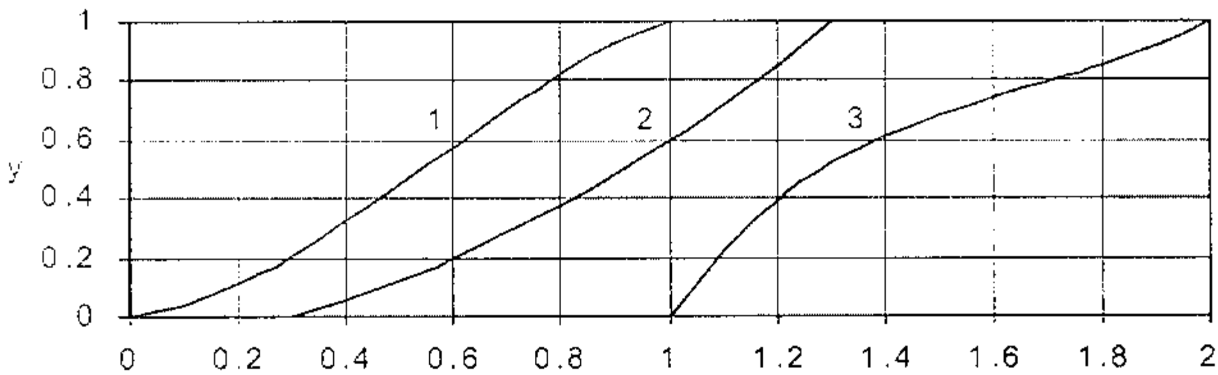

Fig. 6.3. Blade development options (the figures on the abscissa have no meaning).

This method can also be used in a way that the outlet angle is kept constant over n_a sections ($\beta_j = \beta_{2B}$ for $j < n_a$) and/or that the inlet angle is kept constant over n_e sections ($\beta_j = \beta_{1B}$ for $j > n - n_e$). Accordingly, equations are only formulated for the middle part of the blade in which the angle changes from β_{2B} to β_{1B}.

5. An alternative way of determining the blade development is as follows: a line S1 is drawn from point A with an angle β_{2B}, Fig. 6.6d. A point E at distance $L_{a.m}$ is then selected such that a line S_2 started with the angle β_{1B} intersects the line S_1 in a point P. The ratio $L_p/L_{a,m}$ has an effect on the blade length and the channel cross sections. It must be selected so that the intended blade distances a_1 and a_2 as well as acceptable blade loadings according to the criteria in equations are obtained. A smooth curve is then nestled into the geometry defined by the points A and E and the lines S_1 and S_2. This curve forms the chosen blade development.

6. The blade geometry must be represented in the plan view so as to allow the manufacturing of the impeller or pattern. The blade points 1 to 13 in the development (Fig. 6.4d) define triangles with the sides ΔL, Δu and Δm. These points are situated on the radii r_1 to r_{13} (see the meridional section in Fig. 6.4a). Starting on the outer radius the points are transferred to the radii r_1, r_2 etc. with Δu and Δr according to Fig. 6.6e. This yields the points 1, 2, 3, etc. in the plan view. Joining these points produces the streamline projected into the plan view.

7. The wrap angle ε_{sch}, formed by the blade in the plan view depends on the blade angles β_{1B} and β_{2B}, the radii ratio d_1/d_2 (hence n_q) and the streamline length in the meridional section. Wrap angles of radial impellers usually are within the following ranges: with $z_{La} = 5$: $\varepsilon_{sch} = 130$ to $160°$; with $z_{La} = 6$: $\varepsilon_{sch} = 120$ to $140°$ and with $z_{La} = 7$: $\varepsilon_{sch} = 100$ to $130°$.

8. Steps 4 to 6 are performed for all surfaces of revolution (streamlines).

9. To generate sufficient checkpoints for the manufacturing process, radial sections (A to Q in Fig. 6.5) are drawn in the plan view and their intersections with the surfaces of revolution or streamlines transferred into the meridional section (shown in Fig. 6.5 by way of Section K). The radial sections should yield smooth curves in the meridional section in order to avoid waviness in the blade surface. These radial sections need not always be curves; they can also be straight lines.

10. For manufacturing it is also possible to use contour lines ("board lines"). These are equidistant sections in the meridional section normal to the axis, No. 0 to 12 in Fig. 6.5. Their intersections with the streamlines and radial sections are transferred to the plan view and joined to form the board lines. These must present smooth curves in the plan view in order to prevent wavy blade surfaces.

11. The blade surface designed in this way constitutes the camber, the suction or the pressure surface -- depending on the angles (camber, suction or pressure surface) that were used for the development. The selected blade profile can be drawn into the development as section normal to the blade surface in order to better appreciate its effect on the approach flow.

12. The blade profile is represented as developed onto the camber line where it is dimensioned as thickness = f(L) (dimensioning not shown in Fig. 6.5).

6.3 CRITERIA FOR SHAPING THE BLADES:

When constructing the blade development, the designer enjoys a certain freedom which he may utilize to obtain specific hydraulic features. Impellers designed for high NPSHA, i.e. when cavitation is not an issue, will be primarily optimized for efficiency and the stability of the Q-H-curve. In order to minimize the losses, a smooth variation of geometric parameters should be targeted (for example by increasing the blade angles to a linear function from inlet to outlet, i.e. $\partial\beta/\partial L$ = constant). On the outer streamline the outlet angle β_{2B} is usually several degrees larger than the inlet angle $\beta_{1B,a}$ and a development similar to curve 2 in Fig. 6.5 is obtained. On the inner streamline of an impeller with axial inlet (end suction pump) $\beta_{1B,i}$ is generally much larger than β_{2B} so that a development similar to curve 3 in Fig. 6.5 is obtained. It is also possible to specify a law for $u \times c_u$ as a function of the blade length. For example, the design could be done with low blade loading at the inlet because of cavitation, with the strongest blade load in the middle part where the blades overlap and moderate blade loading towards the outlet with the object to achieve as uniform a flow as possible. It is thus attempted to minimize turbulent dissipation losses and pressure pulsations. The developments of all streamlines must be matched to each other so that the desired leading edge is obtained. Since the inflow angles are usually small (between 12 and

18°), the approach flow of the blade leading edge can be assessed qualitatively by way of its position in the plan view. If the leading edge is radial (No. 1 in Fig. 6.6), it is hit by an almost perpendicular approach flow. If the leading edge is inclined by the angle ε against the radial direction, it meets the approach flow at an angle which tends to cause lower velocity peaks and losses (compare to a cylinder exposed to an inclined versus a perpendicular approach flow). A leading edge shape according to Nos. 2, 3 or 4 in Fig. 6.6 corresponds to a "sweep back" which is used in axial pumps, compressors and inducers. It tends to displace fluid away from the hub. Special shapes such as No. 5 in Fig. 6.6 ("sweep forward") are sometimes used to extend and unload the outer streamline. The leading edge also influences the part load behavior – especially shapes such as 4 and 5 in Fig. 6.6.

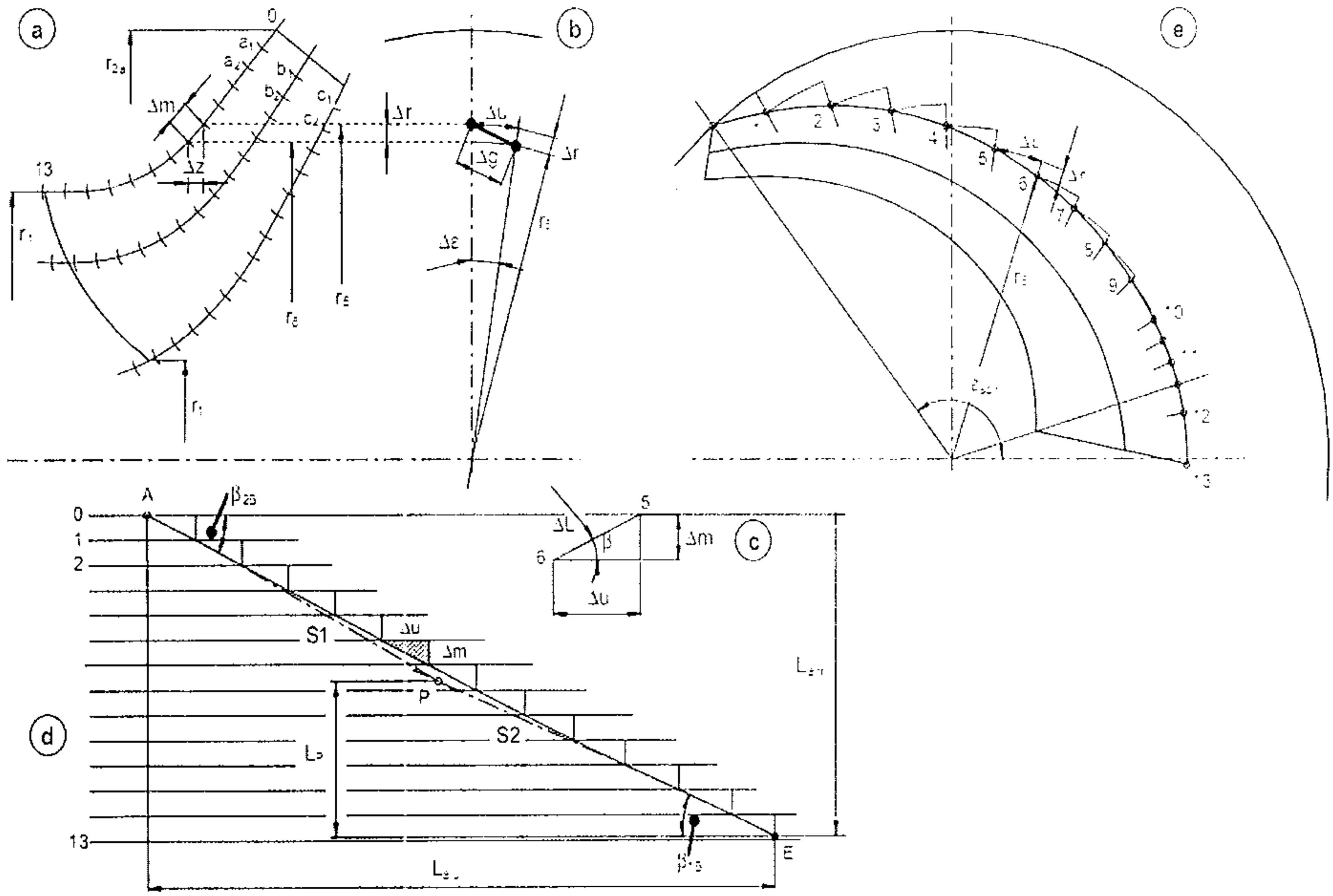

Fig. 6. 4 Blade design according to Kaplan's method

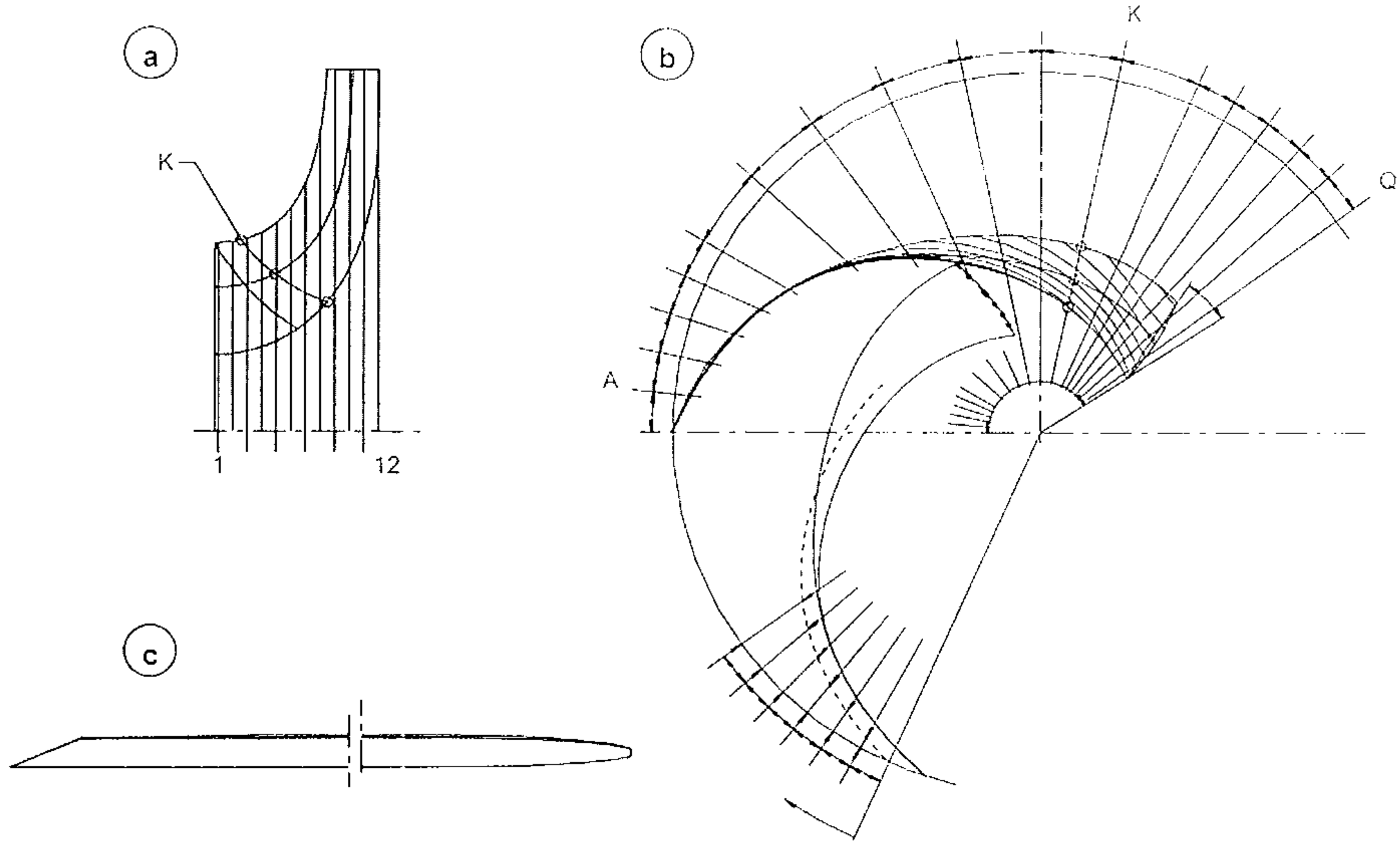

Fig. 6.5. Representation of the impeller coordinates by board lines 0 to 12 and radial sections A to Q.
a) meridional section, b) plan view, c) blade profile (developed)

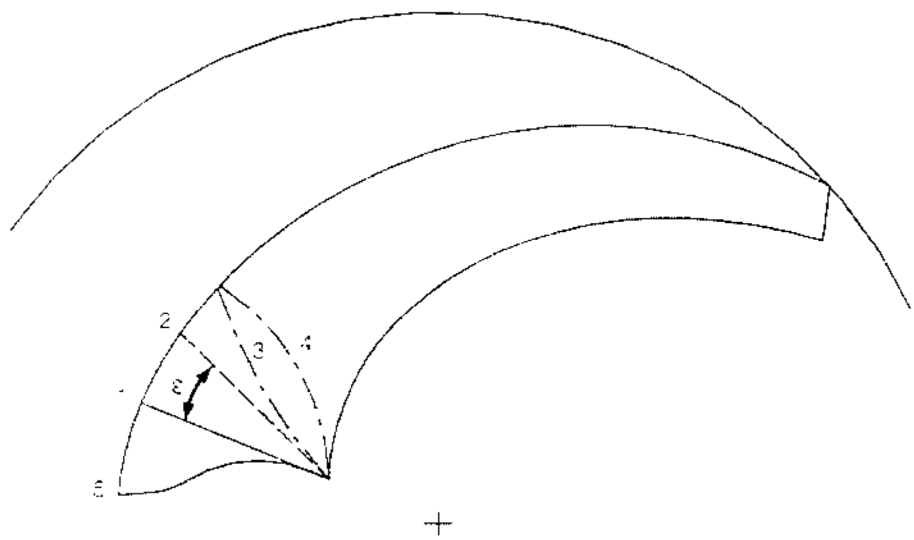

Fig. 6.6. Different shapes of the impeller blade leading edge

The blade twist at the inlet is obtained from equation from the calculated flow angles and the selected incidences. This involves major angle variations over the blade height if $d_{1i} << d_{1a}$ and $\beta_{1B,i} >> \beta_{1B,a}$. In contrast, the blade twist at the impeller outlet can be selected freely. The blades of radial impellers of low specific speeds are frequently not twisted at the outlet. The higher the specific speed, the more an outlet twist will be required to optimally match the streamline lengths. In order to reduce pressure pulsations, the blades can be twisted at the impeller outlet. The effectiveness of this measure increases with the ratio of the outlet width b_2 to the pitch of the blades t_2 and with the specific speed. The blades can be arched deliberately in order to increase the throat area A_{1q}. The radial sections will then appear as pronounced curves in the meridional section. This design is used especially with radial impellers of high specific speeds (compare to the impellers of Francis turbines). In general, all streamlines, blade developments, radial sections, board lines and channel cross sections should vary continuously along the flow path in order to prevent unnecessary accelerations, decelerations and velocity differences over the channel. Turbulent dissipation losses caused by uneven velocity distributions can thus be minimized.

CHAPTER 7

DESIGN DETAILS

CHAPTER 7
DESIGN DETAILS

7.1 IMPELLER BLADES WITH DOUBLE CURVATURE:

The techniques used in the design of impellers and runners of single curvature may also be used for double-curvature blade design. The methods are applicable to radial flow and diagonal impellers. For pumps care must be taken with regard to the choice of blade length. Too short a length has an unfavorable influence on suction capacity and efficiency. If the velocity of the liquid before the blade and the constriction coefficient are constant, c_{m1} will be the same for all points along the blade. Therefore, the blade inlet angle b_1 must be variable along the edge; that is, the blade must be twisted or have double curvature. In certain cases it may be necessary to design a purely radial blade with double curvature because the inlet is wide; these are usually low-specific speed impellers. Wide inlets can mean large variation of C_{m1} along the inlet edge of the blade. Boiler feed pumps, which require a stable H-Q characteristic, also have extended edges; these also should be designed with double curvature.

7.2 PROCEDURE:

1. A value of Ns for given values of Q and H is calculated, u_2 is calculated, impeller diameter d_2 for the central streamline.

2. c_{m2} is found, b_2 is assumed, and impeller width b_2 is calculated.

3. The impeller profile is provisionally assumed together with the position of inlet edge, making sure that the shape is smooth and continuous and that the change from c_{m1} to c_{m2} is gradual.

4. Corrections are made to the profile and position of inlet edge if necessary.

7.3 DESIGN SOLUTION:

7.3.1 Given specification:

Head , H=7.5m

Discharge, Q=15.8 to16 lit/sec

Speed, N=2850 rpm

Size= 8 inch (suitable for 9' bore)

Power= 2.4 hp

Liquid= water

7.3.2 Assumption:

Single stage centrifugal pump

No of blades, Z= 6

Flow coefficients, k_1=1.3, k_2=1.1,

Relative velocity, w_1/w_2=1.15

Ratio, c_{m3}/c_{m0}=0.8

Angle of attack, $\alpha = 4.5^0$

7.3.3 Design solution:

Specific speed, $n_s=3.65n\sqrt{Q}/H^{0.75}$

$=3.65*2850*\sqrt{.016}/7.5^{0.75}$

$=\mathbf{290.34}$

n_s>250, blade will be diagonal at the inlet , to prevent blade loading over and above the permitted value calculated. The vane becomes twisted (i.e. double curvature).

7.3.4 General dimensions:

Nominal diameter, $D_{1nom} = 4.5*10^3*\sqrt[3]{\frac{Q}{n}}$

$$= 4.5*10^3*\sqrt{\frac{0.016}{2850}}$$

$= \mathbf{79.98mm}$ **D_{1nom}= 79.98mm**

Hydraulic efficiency, $\eta_h = \frac{1-0.42}{(\log D_{1nom} - 0.172)^2}$

$= 0.8598$

$= \mathbf{85.98\%}$ **η_h= 85.98%**

Volumetric efficiency $\eta_v = \frac{1}{(1+0.68n_s^{-2/3})}$

$= 0.9847$

$= 98.47\%$ **η_v= 98.47%**

Assuming mechanical efficiency, η_m=0.96 **η_m=0.96%**

Overall efficiency $\eta = \eta_h\ \eta_v\ \eta_m$

$= 0.8598*0.9847*.96$

$= 0.8128$

$= 81.28\%$ **η= 81.28%**

Output power $N_0 = \frac{\gamma QH}{const}$

Input power $N_i = \frac{N_0}{\eta}$

Assuming an over load of 15% Input power N_i

But input power is given as 2.4 hp.

i.e. N_i = 2.4 hp

= 2.4*0.7457 = **1.79 kw**

Torque $T = N_i/\omega = \frac{1.79}{2*\pi*2850/60}$

$= 5.99*10^{-3} = 0.006$ kN.m

Taking the shaft material as **30Ni4cr1**

Ultimate stress, f_m = 1100 N/mm^2 (from design data book)

Taking factor of safety (Fs) as 4

Working stress f_s=fm/Fs =1100/4 = 275 N/mm^2 = 275000 kN/m^2

Shaft diameter $d_s = \left(\frac{16T}{\pi f_s}\right)^{1/3} = \left(\frac{16*0.006}{\pi*275*10^3}\right)^{1/3}$

$= 4.81*10^{-3}$ m ~ **5 mm**.

Taking fatigue stress (bending and shear) into account, minimum shaft diameter d_s is taken as

d_s= 10mm

Hub diameter, **d_h= 16 mm**

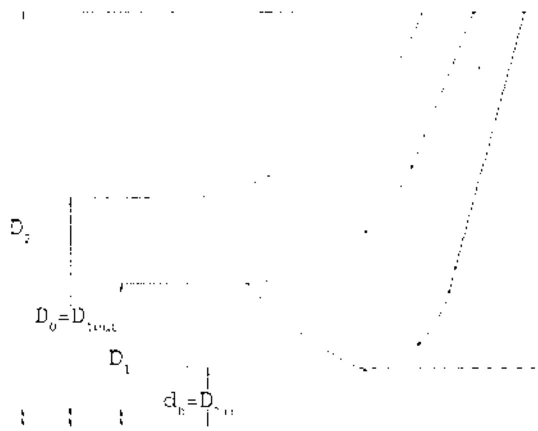

Fig 7.1 Different Diameters in Meridional view

7.3.5 Inlet dimensions:

Theoretical discharge, $Q_{th} = \frac{Q}{\eta_v} = \frac{0.016}{0.9847} = 0.01625$ m^3/sec

The flow velocity before the inlet edge of the impeller blade

$$(C_0)=0.06\sqrt[3]{Q_{th}n^2}$$

$$= 0.06*\sqrt[3]{0.01625*2850^2}\ \ (0.01625*2850^2)^{1/3}$$

$$= \mathbf{3.0549\ m/s} \qquad \mathbf{C_0 = 3.0549\ m/s}$$

Taking, $C_{m0} = C_0$ =3.0549 m/s

Taking, K_1=1.3, $C_{m1} = K_1 C_{m0}$ = 1.3*3.0549 = 3.9714 m/s

Taking C_{m1}= **3.9714 m/s** **C_{m1}= 3.9714 m/s**

The axial velocity at the impeller eye, $C_{m0} = \frac{4Q_{th}}{[\pi(D_0^2 - d_h^2)]}$

$$3.0549\text{ m/s} = \frac{4*0.01625}{[\pi(D_0^2 - d_h^2)]} = \frac{4*0.01625}{[\pi(D_0^2 - 0.016^2)]}$$

D_0= 0.08384 m = 83.84 mm~ **84 mm**

Eye diameter (D_0) = 84 mm **D_0 =84 mm**

Taking, D_0 (Eye diameter) = D_{1out} (Diameter of outer edge of the blade inlet)=84 mm

D_{1out}=84 mm

Diameter at the inner edge of the blade inlet, D_{1in} =Hub diameter = 16 mm

D_{1in}= 16 mm

Taking, $D_1 = 0.7\,D_0 = 0.7*84 = 58.8$ mm

D_1 =58.8 mm

$u_1 = \frac{\pi D_1 n}{60} = \frac{\pi * 58.8 * 2850}{60} = 8774.47$ mm/s = **8.77 m/s** **u_1= 8.77 m/s**

Assuming normal entry, (C_{u1}=0),

Inlet blade angle 'β_1' will be,

$\beta_{10} = \tan^{-1}\left(\frac{C_{m1}}{u_1}\right) = \tan^{-1}\left(\frac{3.9714}{8.77}\right) = 24.36^0$

Allowing an angle of attack, $\alpha=4.5^0$, $\beta_1 = \beta_{10}+\alpha = 24.36^0 + 4.5^0 = 28.86^0 \sim$ **29^0**

β_1=23^0

7.3.6 Outlet dimensions:

Manometric head, $H_m = \left(\frac{H}{\eta_h}\right) = \left(\frac{7.5}{0.8598}\right) =$ **8.723 m** **H_m= 8.723 m**

Taking, $\tilde{c}_{u2}=c_{u2}/\tilde{u_2}=0.5$

$H_m = H/\eta_h = c_{u2}u_2/g = \tilde{c}_{u2}u^2/g$

7.3.7 First approximation:

$u_2 = \left(\frac{g\,H_m}{\tilde{c}u_2}\right)^{1/2} = \left(\frac{9.81 * 8.723}{0.5}\right)^{1/2}$ = **13.0822 m/s** **u_2=13.0822 m/s**

Outer diameter, $D_2 = \frac{60u_2}{\pi n} = \frac{60 * 13.0822}{\pi * 2850} = 0.08767$ m

= **87.67 mm** **D_2 = 87.67 mm**

Taking, $C_{m3}=0.8C_{m0}= 0.8*3.0549=$ **2.4439 m/s**

C_{m3}= 2.4439 m/s

Taking, k_2=1.1 and $\frac{w_1}{w_2}$ =1.15

$\sin\beta_2 = \sin\beta_1 \cdot \frac{k_2}{k_1} \cdot \frac{w_1}{w_2} \cdot \frac{C_{m2}}{C_{m1}} = \sin 29^0 * \frac{1.1}{1.3} * 1.15 * 0.8 = 0.3774$

Outer blade angle, β_2= 22.17~**23^0** **$\beta_2 = 23^0$**

$C_{m2}/C_{m1} = \frac{k_2 cm_2}{k_1 cm_1} = \frac{1.1}{1.3} * 0.8 = 0.6769$

Outlet flow velocity, c_{m2}= 0.6769*3.9714 = **2.6882 m/s**

cm2= 2.6882 m/s

No of blades, $Z = 6.5 \frac{(D_2 + D_1)}{(D_2 - D_1)} \sin\left(\frac{(\beta_2 + \beta_1)}{2}\right)$

$= 6.5 * \frac{(87.67 + 58.8)}{(87.67 - 58.8)} * \sin\left(\frac{(29 + 23)}{2}\right)$ Z = 14.45~15 (not applicable)

Therefore assuming **Z=6** **Z=6**

$\Psi = 0.6(1 + \sin\beta_2) = 0.6 * (1 + \sin 23^0) = 0.8344$

Ψ= 0.8344

$p = \frac{2\psi}{Z} \cdot \left(\frac{1}{1 - (r_1/r_2)^2}\right) = 2 * 0.7854/6 * \left(\frac{1}{1 - (58.8/87.67)^2}\right) =$ **0.4758** **p= 0.4758**

$H_\infty = (1+p)\, H_m = (1+0.4758) * 8.723 =$ **12.87 m** **H_∞= 36.58 m**

7.3.8 Second approximation:

Outlet blade velocity, $u_2 = \left(\frac{c_{m2}}{2\tan\beta_2}\right) + \left[\left(\frac{c_{m2}}{2\tan\beta_2}\right)^2 + gH_\infty\right]^{1/2}$

$$= \left(\frac{2.6882}{2\tan 23}\right) + \left[\left(\frac{2.6882}{2\tan 23}\right)^2 + 9.81 * 12.87\right]^{1/2}$$

= **14.84 m/s** **u_2= 14.84 m/s**

Outer diameter, $D_2 = \frac{60 u_2}{\pi * n} = \frac{60 * 14.84}{\pi * 2850}$ = 0.09945 m = 99.45 ~ 100 mm

D_2 I[st] approximation (D_{2I}=87.67) and D_2 II[nd] approximation (D_{2II}=100).

Final value of outer diameter D_2 is taken as D_2= **100 mm**

D_2= 100 mm

C_{m3}=cm_2/k_2= 2.6882/1.1=**2.4438 m/s**

C_{m3}=2.4438 m/s

7.3.9 Verification for flow coefficients:

δ_1 δ_2 thickness of blade = 5mm

$$K_1=\frac{1}{\left[1-\frac{Z\delta_1}{\pi D_1 \sin\beta_1}\right]}=\frac{1}{\left[1-\frac{6*0.005}{\pi 0.0588\sin 29}\right]}=1.5$$

$$K_2=\frac{1}{\left[1-\frac{Z\delta_2}{\pi D_2 \sin\beta_2}\right]}=\frac{1}{\left[1-\frac{6*0.005}{\pi 0.1\sin 23}\right]}=1.32$$

$$w_1=\frac{c_{m1}}{\sin\beta_1}=\frac{3.9714}{\sin 29}=9.397=\mathbf{8.19\ m/s}$$

w_1= 8.19 m/s

$$w_2=\frac{c_{m2}}{\sin\beta_2}=\frac{2.6882}{\sin 23}=7.8597=\mathbf{6.88\ m/s}$$

w_2= 6.88 m/s

NOTE:

Taking 3 streamlines , m=3

No of passages, (m-1) = n = 2

$\pi(D_0^2 - d_h^2) = 2 * \pi(D_{known}^2 - D_?^2)$

$84^2 - 16^2 = 2 * (D_{known}^2 - \mathbf{D_?}^2)$

$6800=2 * (D_{known}^2 - \mathbf{D_2}^2)$

$\mathbf{D_?}=\sqrt{(D_{known}^2 - 3400)}$

Breath of the passage B = $(D_{known} - D_?)/2$

7.4 GRAPHS AND TABULATIONS

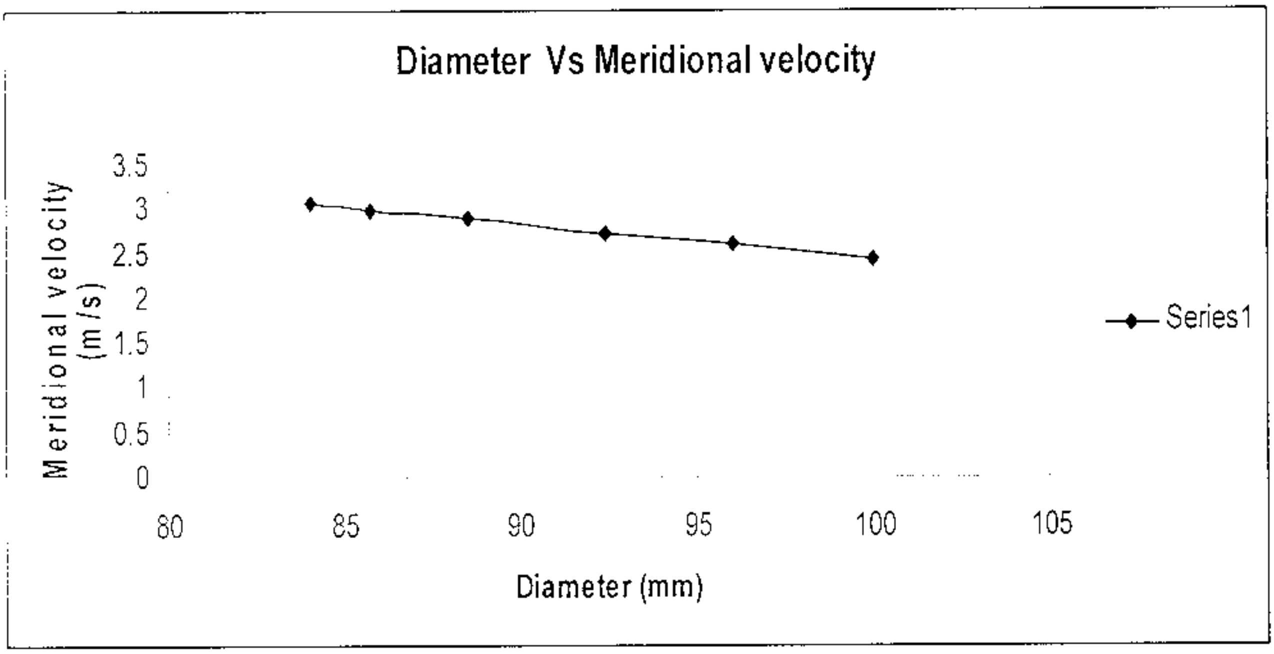

Fig 7.2 Diameter Vs Meridional velocity graph

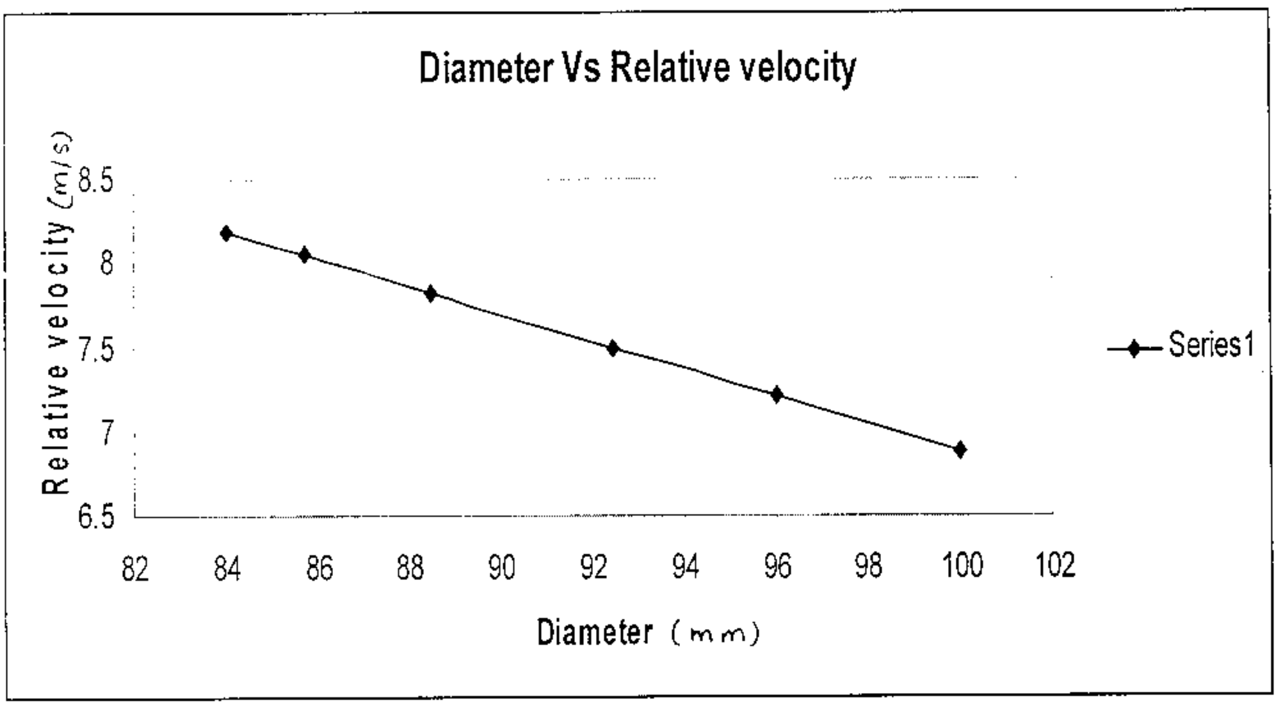

Fig 7.3 Diameter Vs Relative velocity graph

Table 7.1 Vane development for the stream line 'a':

S.No	r (mm)	$d_{a\text{-out}}$ (mm)	C (m/s)	$d_{a\text{-in}}$ (mm)	b (mm)	d_a (mm)	w (m/s)	C_m/w
1	42	84	3.0549	60.46487	11.77	36.27	8.19	0.3730
2	42.85	85.7	2.989971	62.80517	11.45	37.50	8.05	0.3714
3	44.24	88.48	2.883792	66.54856	10.97	39.74	7.82	0.3686
4	46.21	92.42	2.733309	71.70395	10.36	42.24	7.50	0.3644
5	48.01	96.02	2.595811	76.28788	9.87	44.62	7.21	0.3602
6	50	100	2.4438	81.24038	9.38	47.15	6.88	0.3552

δ (mm)	$t = 2\pi r/Z$ (mm)	δ/t	Sinβ $=[(C_m/w)+(\delta/t)]$	B	Tanβ	B= $1/(r\tan\beta)$ (rad/m)	▲r $=r_{i+1}-r_i$ (m)
5	44.0000	0.1136	0.4866	29.1083	0.5570	42.7422	0.0009
5	44.8905	0.1114	0.4828	28.8548	0.5513	42.3339	0.0014
5	46.3467	0.1079	0.4765	28.4458	0.5420	41.7055	0.0020
5	48.4105	0.1033	0.4677	27.8735	0.5291	40.8978	0.0018
5	50.2962	0.0994	0.4596	27.3533	0.5176	40.2445	0.0020
5	52.3810	0.0955	0.4507	26.7751	0.5048	39.6175	

x $=\frac{B_i + B_{i+1}}{2}$	▲Ø=x* ▲r	Ø=$\sum \Delta Ø$ (rad)	(rad)	r (mm)	d (mm	pi* d_a *b
		0.3284	18.81	42	84	1341.402
42.5380	0.0362					
		0.2923	16.74	42.85	85.7	1349.159
42.0197	0.0584					
		0.2339	13.39	44.24	88.48	1369.587
41.3016	0.0814					
		0.1525	8.73	46.21	92.42	1375.072
40.5711	0.0730					
		0.0795	4.55	48.01	96.02	1383.56
39.9310	0.0795					
		0.0000	0.00	50	100	1389.954

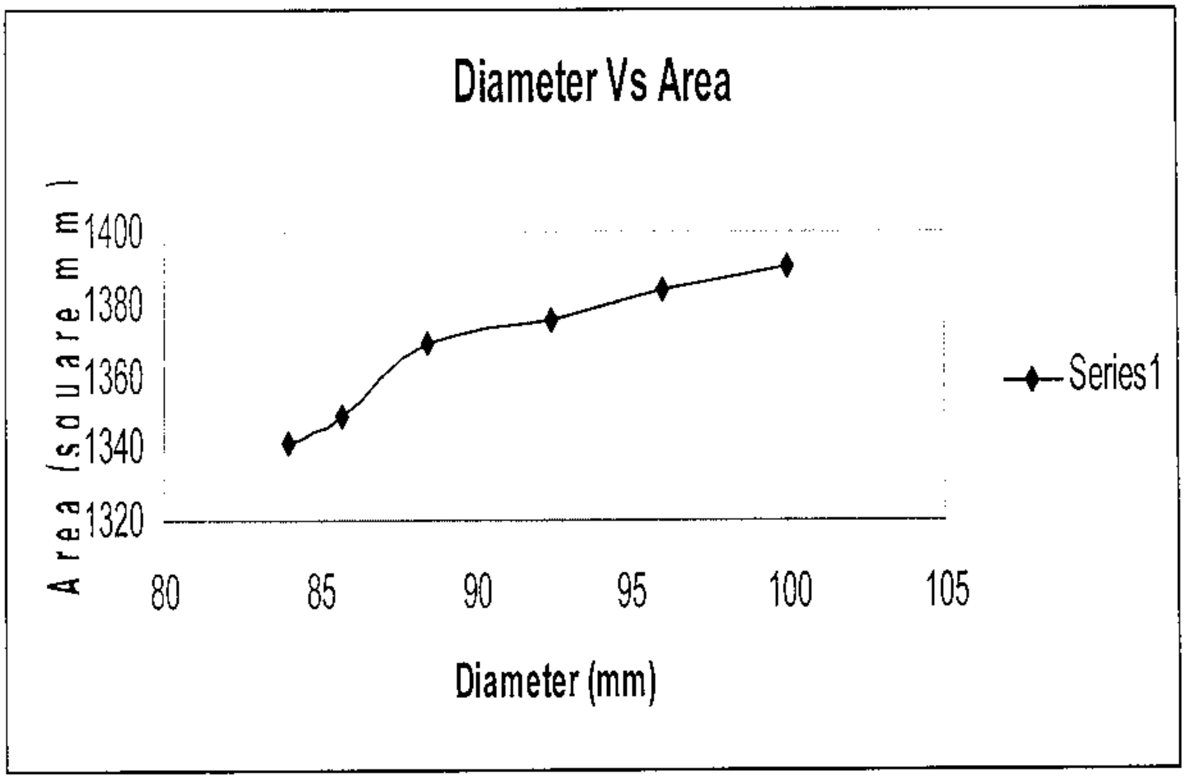

Fig 7.4 Diameter Vs Passage area between Streamline 'a' & 'b' graph

Table 7.2 Vane development for the stream line 'b':

S.No	r (mm)	d_b (mm)	C (m/s)	$d_{b\text{-}in}$ (mm)	b (mm)	d_b (mm)	w (m/s)	C_m/w
1	30.54	61.08	3.0549	18.18698	21.45	20.07	8.19	0.3730
2	32.15	64.3	2.9385	27.10148	18.60	23.74	8.16	0.3601
3	35.24	70.48	2.8221	39.59079	15.44	29.16	7.98	0.3536
4	38.27	76.54	2.7057	49.58197	13.48	33.49	7.69	0.3518
5	41.23	82.46	2.5893	58.30653	12.08	37.52	7.33	0.3532
6	44.3	88.6	2.4438	66.70802	10.95	41.66	6.88	0.3552

δ (mm)	t =2πr/Z (mm)	δ/t	Sinβ =[(C_m/w)+(δ/t)]	β	tanβ	B= 1/(rtanβ) (rad/m)	▲r $=r_{i+1}-r_i$ (m)
5	31.9943	0.1563	0.5293	31.9441	0.6238	52.4890	0.0016
5	33.6810	0.1485	0.5086	30.5558	0.5906	52.6612	0.0031
5	36.9181	0.1354	0.4891	29.2685	0.5607	50.6078	0.0030
5	40.0924	0.1247	0.4766	28.4494	0.5421	48.2042	0.0030
5	43.1933	0.1158	0.4690	27.9585	0.5310	45.6736	0.0031
5	46.4095	0.1077	0.4629	27.5659	0.5223	43.2211	

$x = \frac{B_i + B_{i+1}}{2}$	▲Ø=x*▲r	Ø=∑ΔØ (rad)	Ø=∑ΔØ (deg)	r (mm)	d (mm)	pi* d_b *b
		0.0000	0.0000	30.54	61.08	1352.784
52.5751	0.0846					
		0.0846	4.8479	32.15	64.3	1387.718
51.6345	0.1596					
		0.2442	13.9858	35.24	70.48	1415.432
49.4060	0.1497					
		0.3939	22.5595	38.27	76.54	1418.724
46.9389	0.1389					
		0.5328	30.5170	41.23	82.46	1424.089
44.4474	0.1365					
		0.6693	38.3320	44.3	88.6	1433.174

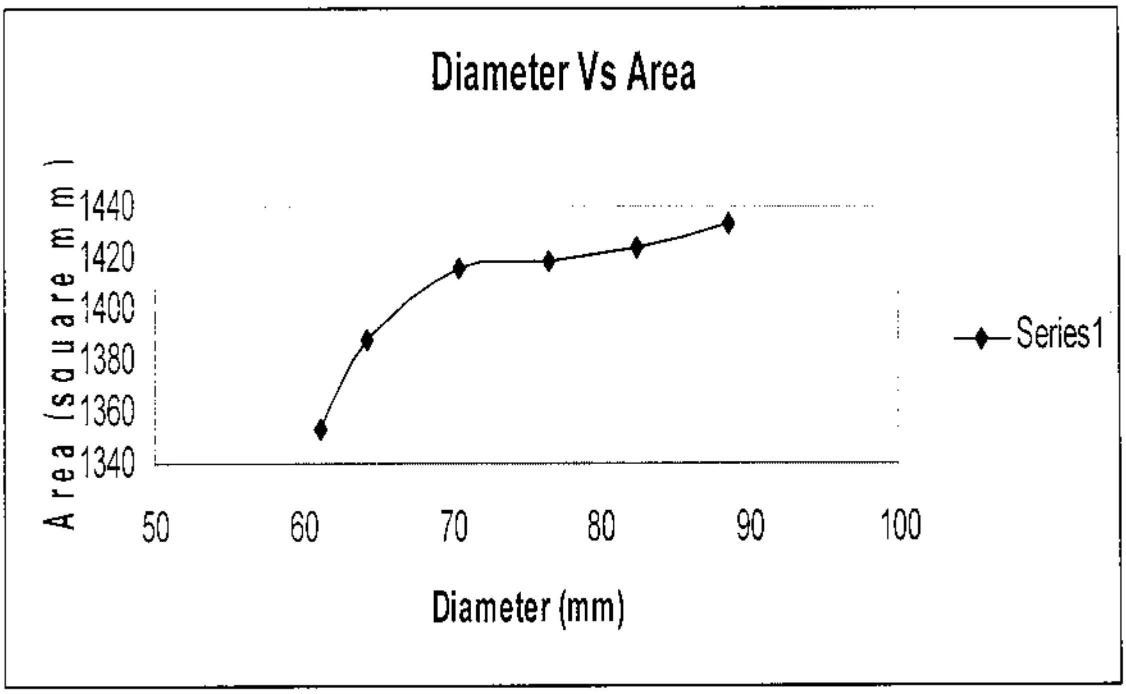

Fig 7.5 Diameter Vs Passage area between Streamline 'b' & 'c' graph

Table 7.3 Vane development for the stream line 'c':

S.No	r (mm)	d_c (mm)	C (m/s)	w (m/s)	C_m/w	δ (mm)	t =2πr/Z (mm)
1	11.97	23.94	3.0549	8.19	0.3730	5	12.5400
2	17.96	35.92	2.9385	8.16	0.3601	5	18.8152
3	25.39	50.78	2.8221	7.98	0.3536	5	26.5990
4	30.8	61.6	2.7057	7.69	0.3518	5	32.2667
5	35.11	70.22	2.5893	7.33	0.3532	5	36.7819
6	39.89	79.78	2.4438	6.88	0.3552	5	41.7895

δ/t	Sinβ =[(C_m/w)+(δ/t)]	β	Tanβ	B= 1/(rtanβ) (rad/m)	▲r $=r_{i+1}-r_i$ (m)	x $=\frac{B_i+B_{i+1}}{2}$	▲Ø =x*▲r
0.3987	0.7717	50.4890	1.2135	68.8441	0.0060	69.0910	0.4139
0.2657	0.6259	38.7292	0.8024	69.3879	0.0074	65.3774	0.4858
0.1880	0.5416	32.7810	0.6443	61.1280	0.0055	58.2385	0.3209
0.1550	0.5068	30.4390	0.5879	55.2263	0.0043	53.0038	0.2284
0.1359	0.4892	29.2751	0.5609	50.7813	0.0048	48.6215	0.2324
0.1196	0.4749	28.3382	0.5396	46.4616			

$Ø=\sum\Delta\Theta$ (rad)	$Ø=\sum\Delta\Theta$ (deg)	d (mm)
0.0000	0.0000	23.94
0.4139	23.7026	35.92
0.8996	51.5231	50.78
1.2205	69.9015	61.6
1.4489	82.9853	70.22
1.6814	96.2961	79.78

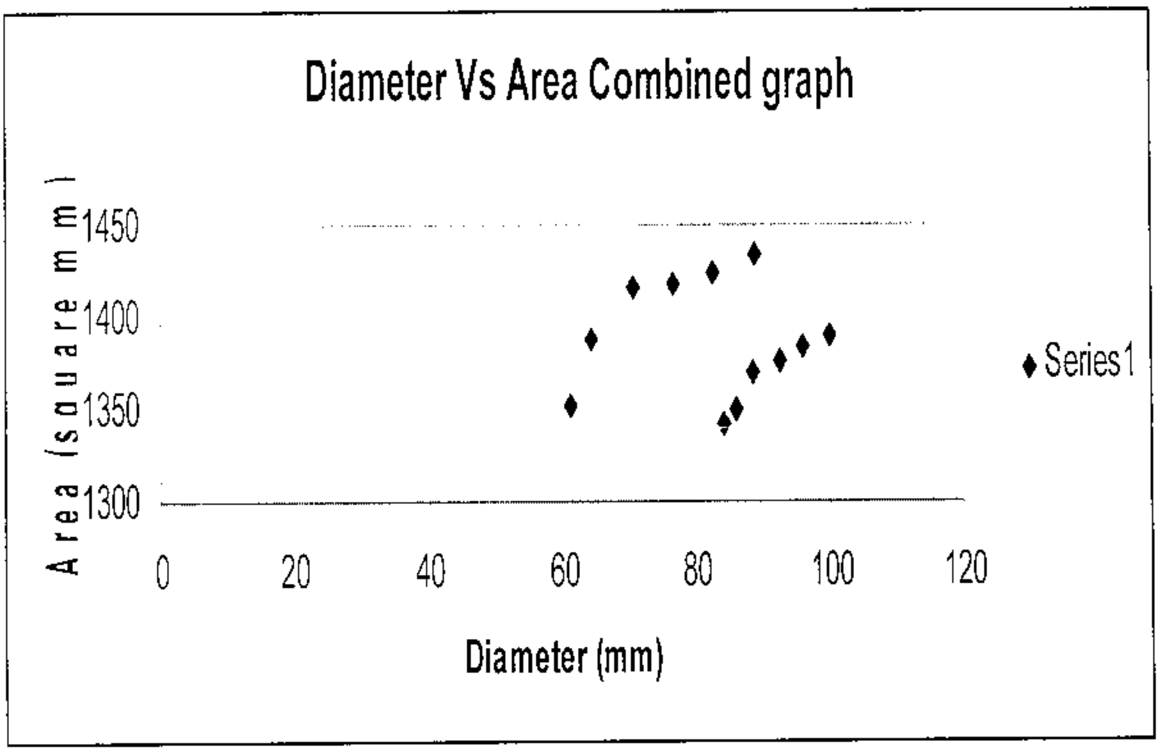

Fig 7.6 Diameter Vs Passage areas between Streamline 'a' ,'b' & 'c' combined graph

7.5 VANE DEVELOPEMENT

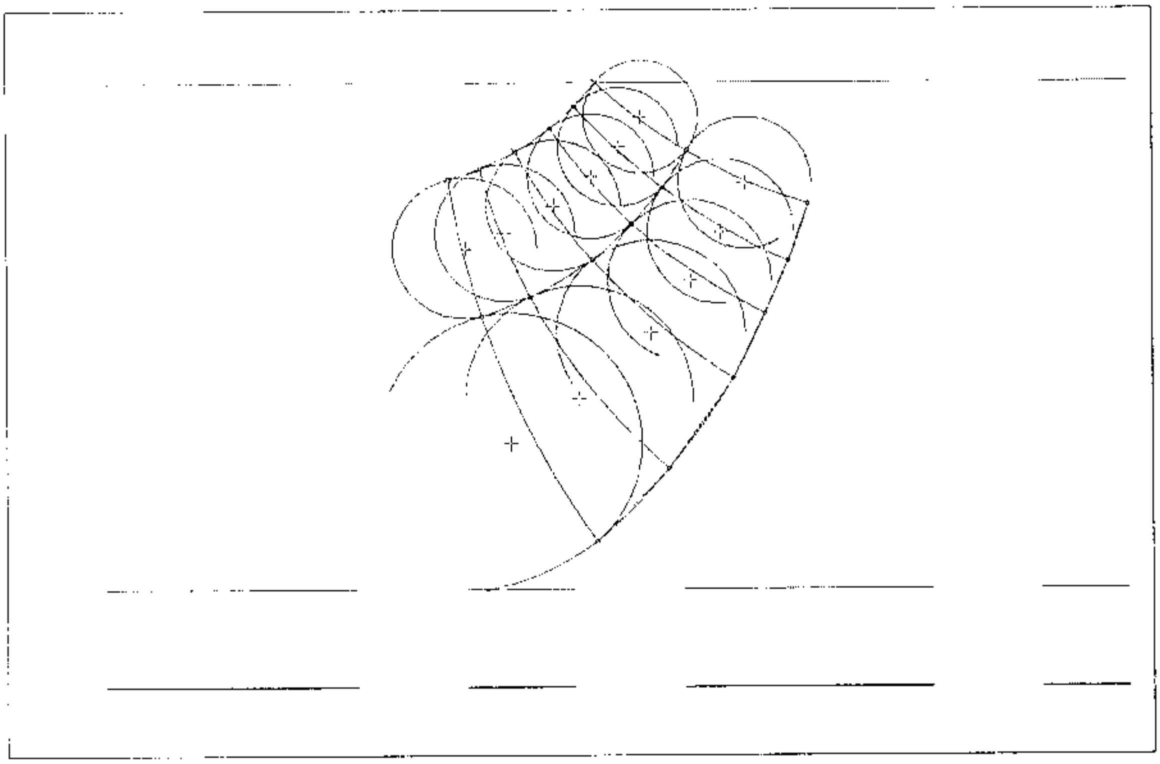

Fig 7.7 Meridional view of double curvature blade

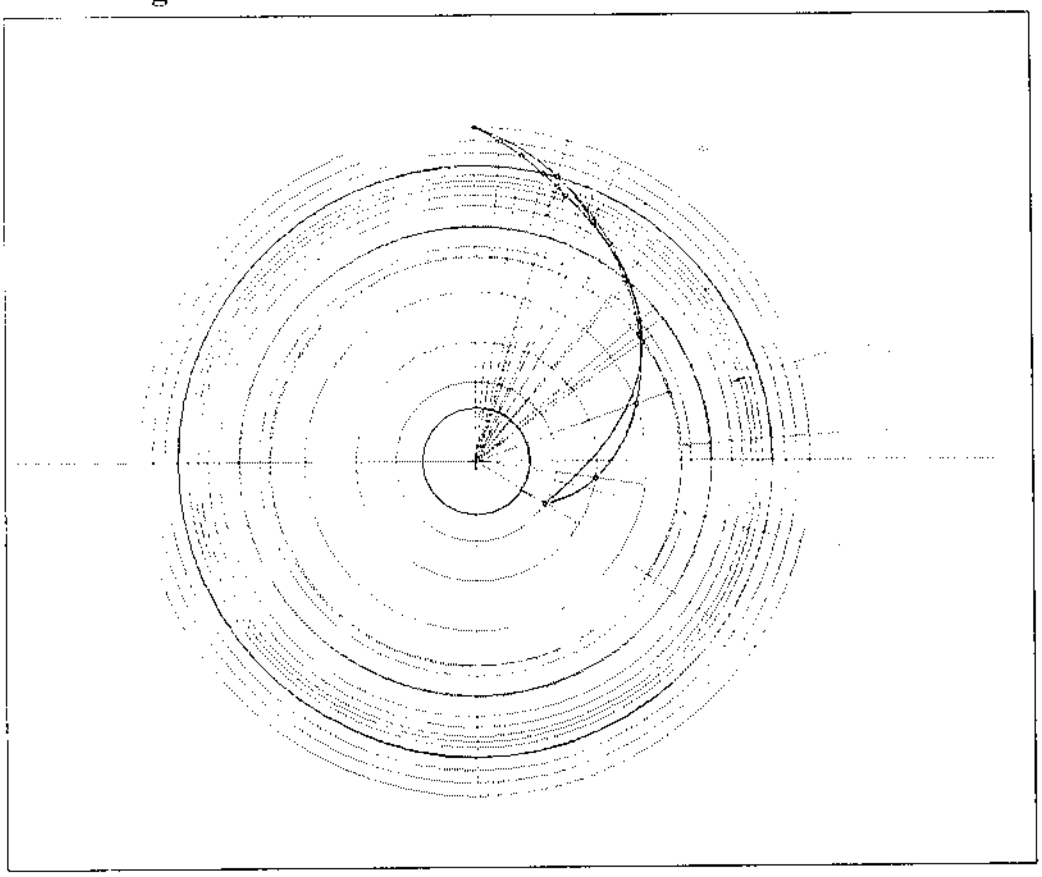

Fig 7.8 End view of double curvature blade

7.6 THREE DIMENSIONAL VIEWS

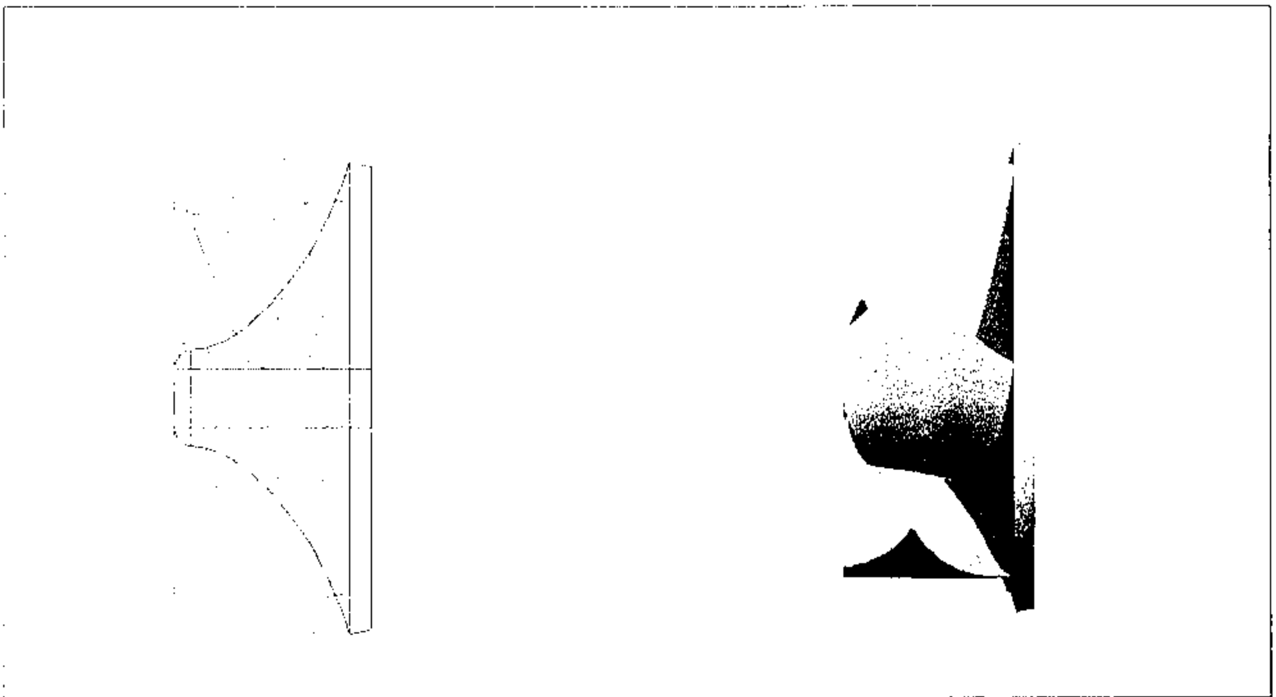

Fig 7.9 Three Dimensional side view of double curvature blade

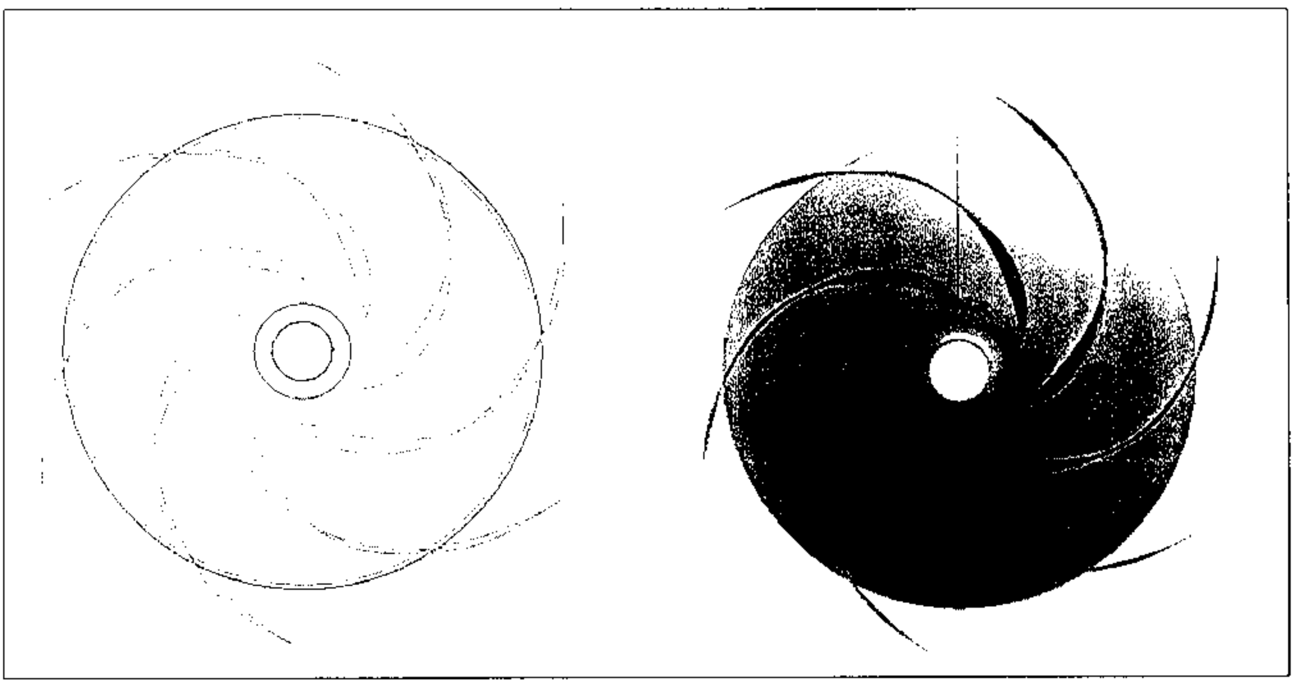

Fig 7.10 Three Dimensional front view of double curvature blade

7.7 IMPELLER WITH AND WITHOUT FRONT SHROUD

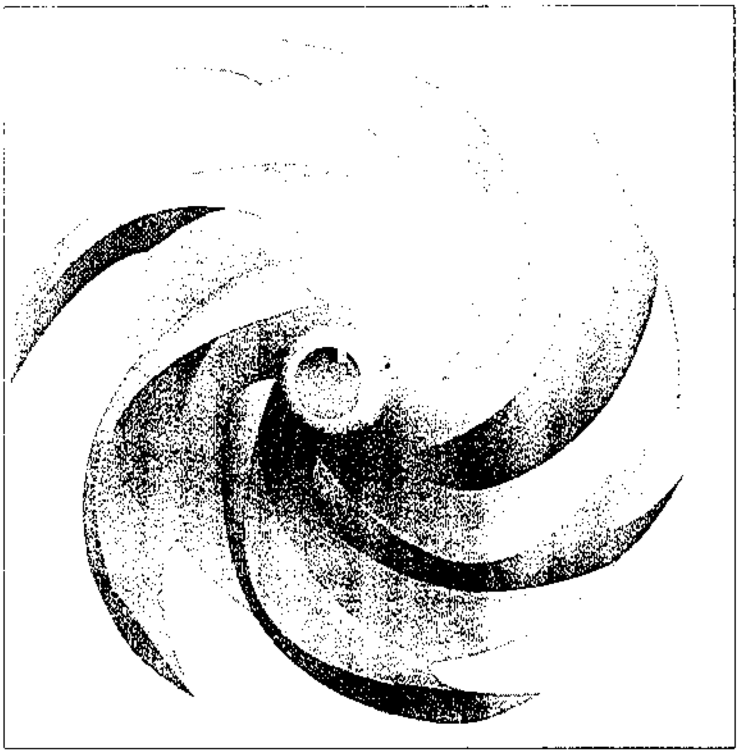

Fig 7.11 Impeller without front shroud

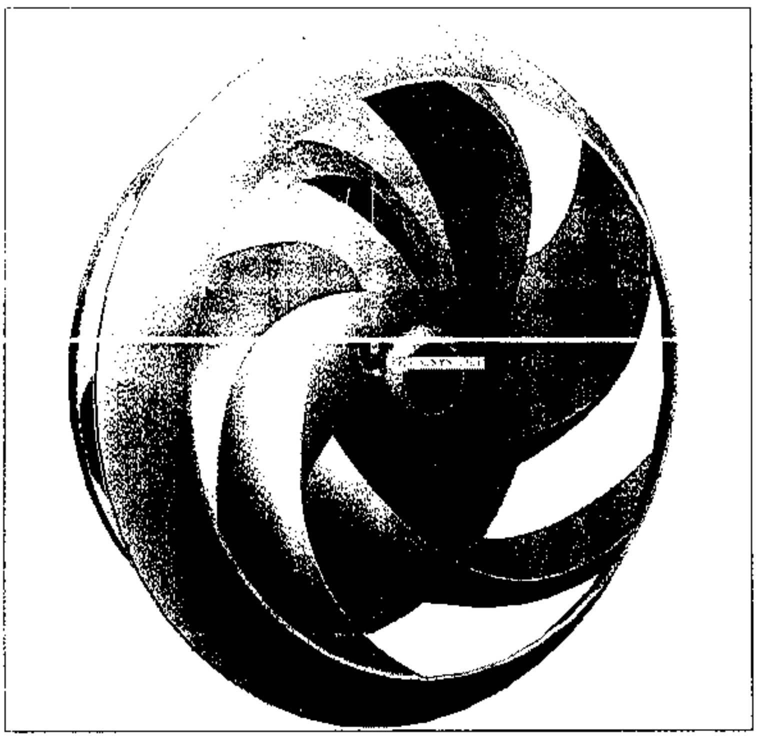

Fig 7.12 Impeller with front shroud

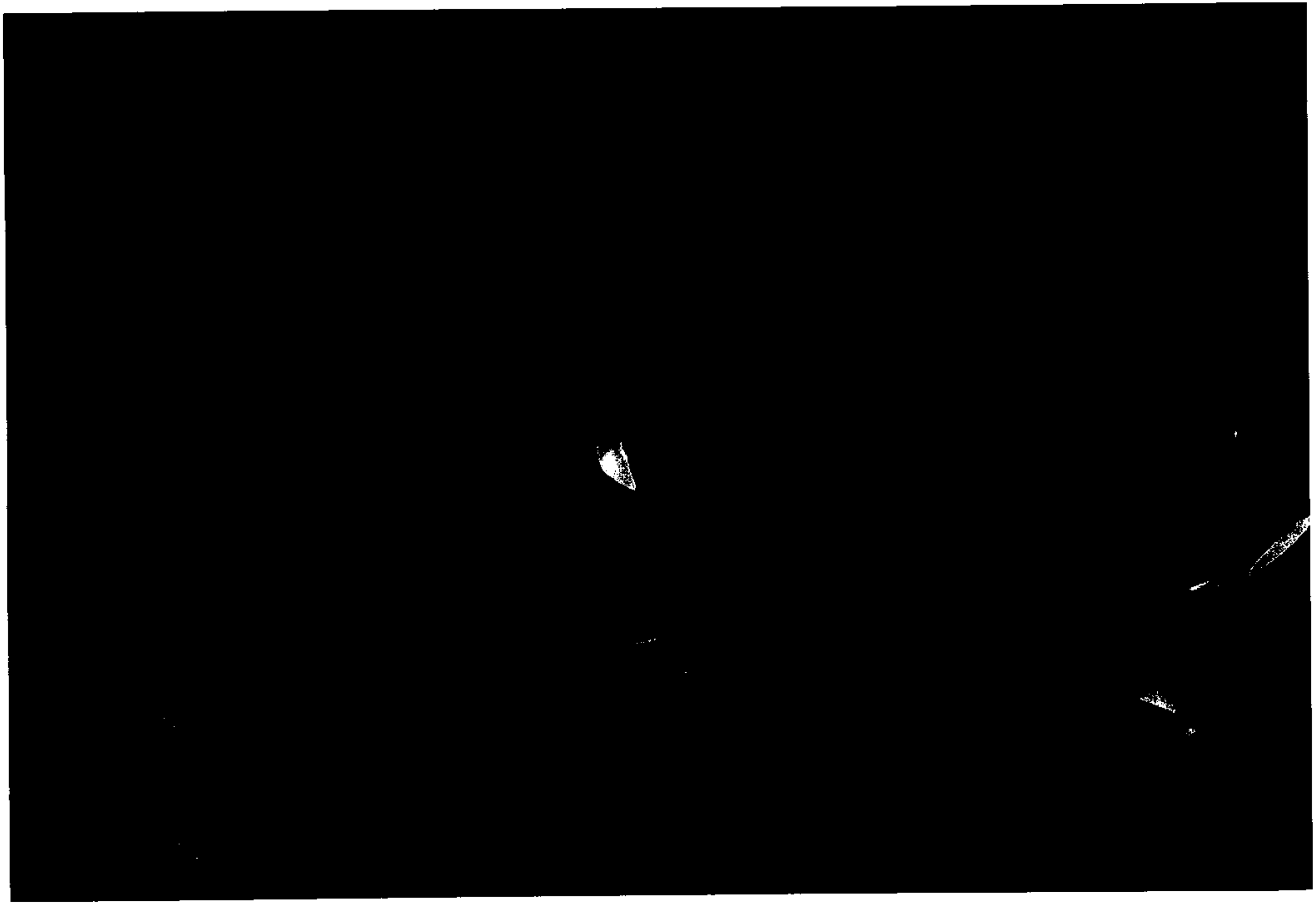

Fig 7.13 Final impeller design

CHAPTER 8

RESULT AND CONCLUSION

CHAPTER 8
RESULT AND CONCLUSION

This design of a double curvature type of impeller has been completed in PRO-E Wildfire 4.0 - CAD software. Its efficiency value has been found very high when comparable with the normal single curvature blade design followed in the industries. The efficiency of the impeller with double curvature design is 81% while that of the normal mixed flow impeller is only 49 %. Also the double curvature concept in blade design overcomes the problem of blade loading and increases the performance level of the pump.

The only difficulty in the double curvature blade design is the complexity in the design procedure. Also the manufacturing of this design, particularity the pattern making requires a skilled labour, since the profile has to be developed for each streamlines. In the impeller, designed in this project, thickness has been added above the double curvature profile. A thickness of 1 to 2 mm is given at the leading and trailing edges and the original thickness of 5 mm has been given at the centre. But for effective functioning, an airfoil profile for the specification can be developed and placed over the initial double curvature design.

REFERENCES

REFERENCES:

1. K.M.Srinivasan , (2008) 'Rotodynamic pumps (Centrifugal and axial)', pp. 1- 129.

2. G.K.Sahu , 'Pump Rotodynamic and positive displacement type(theorey, design, application)', pp. 127-140.

3. V.M.Cherkassky, (1985) 'Pumps, Fans, Compressors', pp. 35 – 217.

4. Johann Friedrich Gülich, 'Centrifugal Pumps' , pp . 348 – 356.

5. George F. Round, 'Incompressible Flow Turbomachines (Design, Selection, applications,and Theory)' pp. 210 – 230.

6. Larry Bachus and angelCustodio, 'Know and Understand centrifugal pumps', pp. 76 – 85.

7. Igor J. Karassik, Joseph P. Messina, Paul Cooper, Charles C. Heald, 'Pump Handbook', pp. 2.3 – 2.97.

8. Val S.Lobanoff, Robert R. Ross, 'Centrifugal pumps Design and Application', pp. 28 -44.

9. Ron Palgrave, 'Troubleshooting Centrifugal pumps and their systems', pp. 1 – 20.