DESIGN AND DEVELOPMENT OF A PROPELLER PUMP

By

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THESIS

Submitted in partial fulfilment of the requirement for the degree

MASTER OF SCIENCE IN AGRICULTURAL ENGINEERING

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Department of Agricultural Engineering
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1984

DECLARATION

I hereby declare that this thesis entitled "Design and Development of a Propeller Pump" is a bonafide record of research work done by me during the course of research and that the thesis has not previously formed the basis for the award to me of any degree, diploma, associateship, fellowship, or other similar title, of any other University or Society.

Vellanikkara

23.11. 1985.

(K. SASI)

CERTIFICATE

Certified that this thesis, entitled "Design and Development of a Propeller Pump" is a record of research work done independently by Sri.K.Sasi under my guidance and supervision and that it has not previously formed the basis for the award of any degree, fellowship, or associateship to him.

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We, the undersigned, members of the Advisory Committee of Sri.K.Sasi, a candidate for the degree of Master of Science in Agricultural Engineering with major in Farm Power and Machinery, agree that the thesis entitled "Design and Development of a Propeller Pump" may be submitted by Sri.K.Sasi in partial fulfilment of the requirements for the degree.

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ABBREVIATIONS

ASME	American Society of Mechanical Engineers
cm	centimetre(s)
Co.	Company
Ed.	Edition
<u>et al</u>	and other people
Fig.	Figure
HP, hp	horse power
hr	hour(s)
IRRI	International Rice Research Institute
kg	kilogram(s)
KW	Kilo Watt(s)
Ltd.	Limited
lit	litre(s)
lps	litres per second
m	metre(s)
M.S	Mild Steel
No.	Number
pp.	pages
Proc.	Proceedings
rpm	revolutions per minute
Rs.	Rupees
sec.	second(s)
Sl.	Serial
/	per
%	per cent
°	degree

NOMENCLATURE

A_m	Mechanical cross section (area)
B	Axial width
C	Absolute velocity (Ideal)
C'	Absolute velocity (Actual)
C_m	Meridional component of absolute velocity
C_u	Peripheral component of absolute velocity
C_{u2}	Actual tangential component of absolute velocity at outlet
C_{u3}	Theoretical tangential component of absolute velocity at outlet
C_p	Pfleiderer's coefficient
C_d	Coefficient of discharge
D, d	Diameter
D_o	Outside tip diameter
D_i	Hub diameter (Inner diameter)
D_m	Mean diameter
d_h	Hub diameter
d_2	Impeller outer diameter
E	Energy transfer
e_h	Hydraulic efficiency
e_{va}	Vane efficiency
e_v	Volumetric efficiency
e_m	Mechanical efficiency
e_o	Overall efficiency
f	Reduction factor or Multiplier
f_s	Allowable shear stress

g	Acceleration due to gravity
H	Total head, height over notch
H_d	Euler's head
H_h	Head at hub
H_o	Head at periphery
H_1	Inlet head
h_s	Specific head
H_{th}	Theoretical head
K_u	Speed constant
K_m	Capacity constant or coefficient of flow velocity
K	Energy meter constant
L	Length of notch
l	length of the blades
N	Speed
N_s	Specific speed
n	Speed, number of rotations of the energy meter disc
P	Horse power transmitted
Q	Quantity of flow or Discharge
Q_L	Quantity of leakage
Q'	Total quantity of water including leakage
R, r	Radius
s	Blade thickness
T	Torque
t	Pitch of the blades, time
U	Peripheral velocity

V	Velocity of flow
W	relative velocity (Ideal)
W'	relative velocity (Actual)
W_u	Peripheral component of relative velocity
w	Specific weight of water, width of blades
Z	Number of blades
α	angle between C and C_u
α_1	angle of antry
α_3	angle of outflow
β_1	vane angle at inlet
β_1'	Flow angle at inlet
β_2	Vane angle at outlet
β_2'	Actual outlet angle
β_3	Theoretical outlet angle
γ	Specific weight
μ	Slip factor
ϕ	Capacity coefficient
ψ	Head coefficient
ω	Angular velocity

Subscripts:

1 - Conditions at inlet

2 - Conditions at outlet

Chapter 1

INTRODUCTION

INTRODUCTION

The importance of drainage in increasing agricultural production is getting more recognition in our country. A substantial area of land in Kerala is lying below or nearly level with the nearby sea. In order to make use of this for cultivation, the area should be drained by pumping. Low areas may be drained more economically by pumping than by constructing large and deep outlet drains. In irrigated areas pumping is needed to lower the groundwater table that may rise to dangerous levels due to over irrigation and by seepage from canals and laterals.

The rice fields of Kuttanad and Kole lands are lying below mean sea level. These rice tracts are known as 'Punja Lands', where the main cultivation season begins in September-October. Punja crop can be raised only after dewatering the field.

The area of Kuttanad Punja Land is 52000 hectares and that of Kole lands is 7900 hectares. Continuous pumping for about 20 days and intermitent pumping for about 3 months is required for adequate drainage to avoid crop losses in these areas. On an average one metre of water from these lands has to be pumped out to make them fit for preparatory cultivation. Therefore a total quantity of 59900 x 10^4 m^3 water has to be pumped out against an average head of one metre including losses, within 20 days.

At present dewatering is carried out with the aid of "Petti and Para", which is a crude form of axial flow pump. The efficiency of a 30HP Petti and Para reported by Kerala Agricultural University drainage research centre (All India Co-ordinated Research Project on Agricultural Drainage) Karumadi (1984) is 26 per cent. Energy required to pump 59900 x10^4 m^3 of water against a total head of one metre is 6280112 KWhr. Intermitent pumping for 3 months during the crop season requires about the same amount of energy needed for initial pumping. Therefore the total energy requirement is 12560224 KWhr.

Although Petti and Para was introduced in the state more than 65 years ago, the device which is a crude form of axial flow pump has not undergone any change in design. Earlier works on the performance of Petti and Para showed, they worked at an efficiency of about 26 per cent while a well designed axial flow pump can operate at 70 per cent efficiency (IRRI, 1979). A scientific design for the Petti and Para is therefore of considerable importance in reducing the power requirement for a given job and to increase the output per unit power input.

The axial flow or propeller pump is used especially for low head pumping. Propeller pumps operate at high efficiencies against heads less than 3m. Its response in terms of efficiency against fluctuating heads is less predominant when compared to centrifugal pumps. This characteristic of operating at nearly maximum efficiency through a greater range of head is most vital in dewatering.

This project was undertaken with the following specific objectives.

(i) To make a study of various water lifting devices used in Kerala for agricultural purposes.

(ii) To evolve a systematic design for a propeller pump which can be used for dewatering and for lift irrigation purposes.

(iii) To fabricate the pump and evaluate its performance by test.

(iv) To optimize the design of the pump and recommend for commerical manufacturing.

The advantage of the scientifically designed axial flow pump over the conventional Petti and Para is that the power intake is less and thereby reducing the wastage of energy. Even by increasing the efficiency by 10 per-cent, power to the extent of 3488952 KWhr can be saved for the state during Punja crop season itself. Report from Kerala Agricultural University drainage research centre (All India Co-ordinated Research Project on Agricultural

Drainage) Karumadi (1984), shows that for additional crop the requirement of energy is high, nearly 80 percent more than that of the Punja crop season. So the total saving in energy per year is about 9769066 KWhr, ie. approximately ten million KWhr.

Light weight axial flow pumps will get more acceptance among Kerala farmers, because at present in the Kuttanad and Kole lands they are using huge Petti and Para to drain the crop fields and to pump water for irrigation from the nearby water sources. Individual farmers and those in groups organised for drainage will make increasing use of small axial flow pumps to drain submerged areas that cannot be drained by gravity drains, and to irrigate low lying fields, because light weight pumps are easier to transport and install and it needs only cheaper foundations in addition to substantial savings in energy.

Chapter 2

REVIEW OF LITERATURE

REVIEW OF LITERATURE

2. 1. Work done in axial flow pumps

Morelli *et al* (1953) designed and developed an axial flow pump. The test results showed that pump could deliver water at the rate of 3600 lit/min. against a head of 3.2m with a maximum efficiency of 68 per cent.

Wallis (1960) reported that he tested an axial flow pump and obtained a discharge of 68 lit/sec. against a head of 3m with an efficiency of 57 per cent. Subsequently he reported that axial flow pump having an efficiency of 70 per cent could be designed.

Samuel and Ramadevi (1968) reported that an axial flow pump could deliver water at the rate of 20,000 gallons per hour at not more than 15 ft head. A design to satisfy the above criteria had been made in accordance with the procedure laid by Addison (1956). The major dimensions are given in the following table.

Table 2.1.Major dimensions of an axial flow pump(Addison,1956)

Description	Design value
Specific speed	580
Shaft horse power	1.3
Impeller outer diameter	4.1 in.
Impeller inner diameter	2.05 in.
Inlet angle at hub	26.35°
Inlet angle at outer end	13.15°
Inner outlet angle	51.3°
Outer outlet angle	16.3°

Addison (1976) proposed that for a good axial flow pump the value of speed constant (Ku), which is defined as the ratio of outlet peripheral velocity to the free jet velocity, should be in between 2 and 2.7. He conducted a series of trials using this range of speed constants. The results revealed that when K_u=2 the efficiency was 56 per cent at a discharge of 86 lit/sec. at a head of 2.6m. Again he got an efficiency of 57.2 per cent for a discharge of 84 lit/sec. against a head of 2.6m with a speed constant 2.7.

Vasandani (1977) studied the flow pattern based on the assumption that (i) flow parameters change only along a stream line (ii) there is no variation in these parameters radially from one stream surface to another (iii) the flow is frictionless and (iv) flow takes place along the passage in a direction parallel to that of blades. During the year 1979 he reported that 66 per cent efficiency at a discharge of 98 lit/sec. against a head of 3m was possible with a perfectly fabricated axial flow pump.

Work done at IRRI

The IRRI (1979) designed and developed a portable, low cost, low head, high capacity axial flow irrigation pump. The portable axial flow pump consisted of an axial flow impeller, which was secured to a pump shaft and located inside a 150mm diameter steel discharge tube. It pumped water at the rates of 1500 to 3000 litres per minute at heads of 1 to 4 metres, when driven by a 5 HP gasoline or diesel engine. The pump inlet had a 30° cut along the tube axis to provide a large suction opening for low entrance losses. A wire mesh inlet strainer was used to protect the pump from the entry of foreign materials. The main bearing holder contained diffusion vanes which straightens the flow from the impeller for improved efficiency. The design is simple so that it can be fabricated by small shops with standard forming tools. The pump can be carried by two men. IRRI designed a jig to aid manufacturers in fabricating the blades to specification. They got a maximum efficiency of 69.1 per cent at a capacity of 2690 lpm against a head of 2.5m at 2890 rpm.

Department of Agricultural Engineering, College of Technology, Pantnagar (1982) designed and fabricated a propeller pump named as Pantnagar Propeller pump. The pump has a capacity of 45 to 65 lit/sec. at 1.0 to 2.0m. of head requiring a 5 HP 1440 rpm motor as a drive unit. The efficiency of the pump at this range varied from 65 to 50 per cent.

They designed and fabricated another pump for higher discharge. The propeller has 3 vanes and is 30cms in diameter. The shaft is made of 2.5cms diameter G.I. pipe inserted and welded with solid M.S. shaft on both ends. Pump was tested at a head of 1.9m to 2.8m at 1440 rpm. The discharge varied from 106 to 130 lit/sec and the pump efficiency varied from 27 per cent to 29 per cent.

The efficiency of a 30 HP Petti and Para reported by

Kerala Agricultural University Drainage Research Centre (All India Co-ordinated Research Project on Agricultural Drainage) Karumadi (1984) is 26 per cent. It worked at 26 per cent efficiency against a static head of 0.9m at a discharge of 526 lit/sec.

Work by Malithara Industries

Some work in Petti and Para was attempted by M/s Malithara Industries of Kerala and they are manufacturing improved version. A practical test has revealed that a Petti and Para fitted with 20 HP motor can pump at the rate of 2 lakh gallons of water per hour (250 lit/sec.). It was working against a head of 1m of water.

2. 2. Basic Theory

2. 2. 1. Velocity Triangles.

The velocity of a fluid element is represented by a vector. The length of the vector gives the magnitude of the velocity and the direction of the vector is tangential to the streamline. The fluid velocity in a stationary conduit is measured with reference to an earthbound co-ordinate system. Therefore it is called absolute velocity and denoted by C'. The prime signifies an actual flow velocity. Idealized velocity vectors are calculated by assuming perfect guidance of the flow by vanes or walls, are given no prime. Failure to distinguish clearly between actual and idealized velocities results an erroneous application of basic impeller theory and consequently a faulty design.

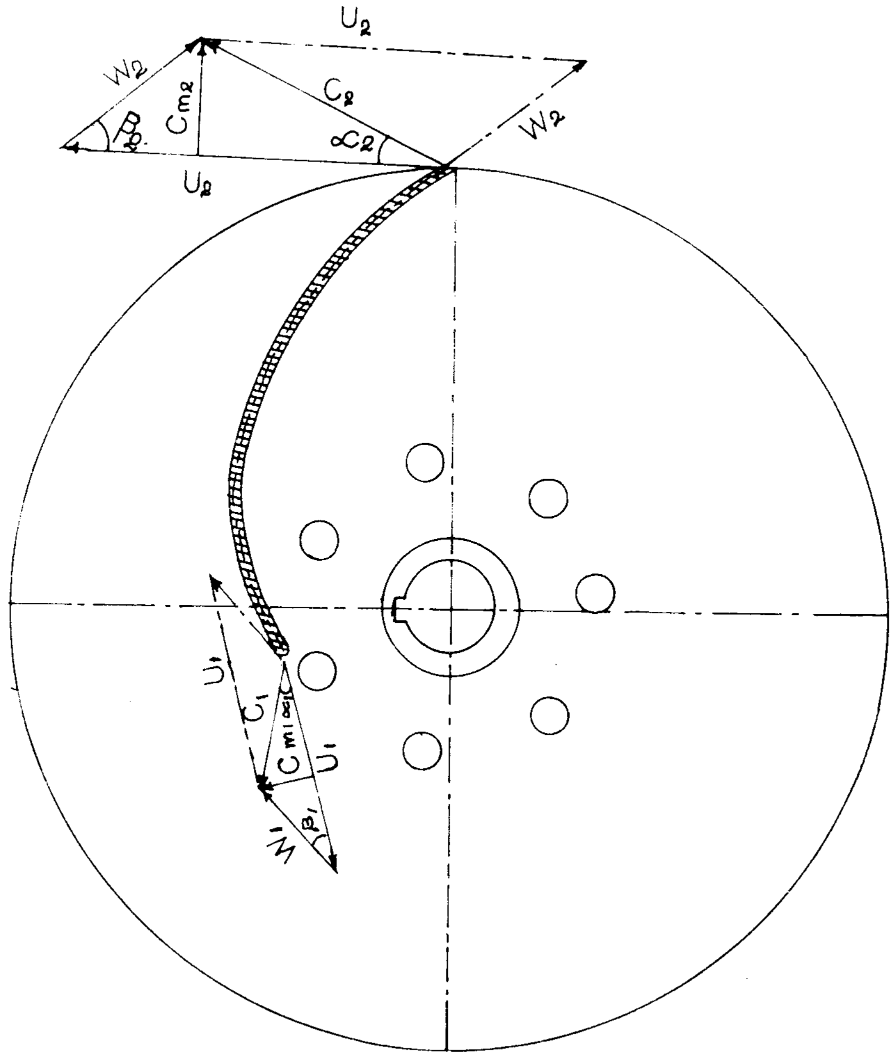
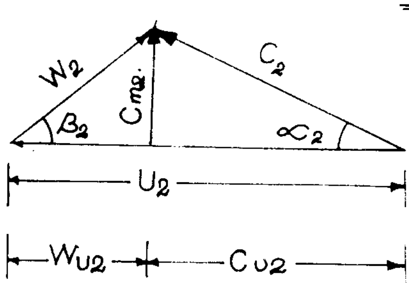
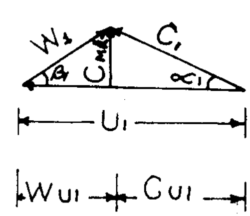


FIG.1 VELOCITY DIAGRAM
FOR CENTRIFUGAL PUMP



EXIT VELOCITY TRIANGLE



ENTRANCE VELOCITY TRIANGLE

The fluid velocity in an impeller blade can be represented by either the absolute velocity C' or the relative velocity W'. The co-ordinate system rotates with the impeller angular velocity ω. The idealized relative velocity W can be calculated by dividing the flow per vane channel by the cross sectional area of that channel.

The absolute velocity can be considered as the resultant of the relative velocity and the local impeller peripheral speed. Velocity triangles provide much information on the design under consideration and should be drawn for every calculated point. The vectors of the absolute velocity and the relative velocity end at the same corner of the triangle. The peripheral velocity vector U starts at the initial point of the absolute velocity C and ends at the starting point of the relative velocity W.

The meridional velocity C_m is the component in the meridional plane of the absolute as well as the relative velocity. It is always at right angle with U. For strictly radial flow, the meridional velocity is also the radial component of the absolute and relative velocity. Similarly, for strictly axial flow, it is the axial component.

The peripheral components of the absolute velocity C' and the relative velocity W' are denoted by C_u' and W_u'.

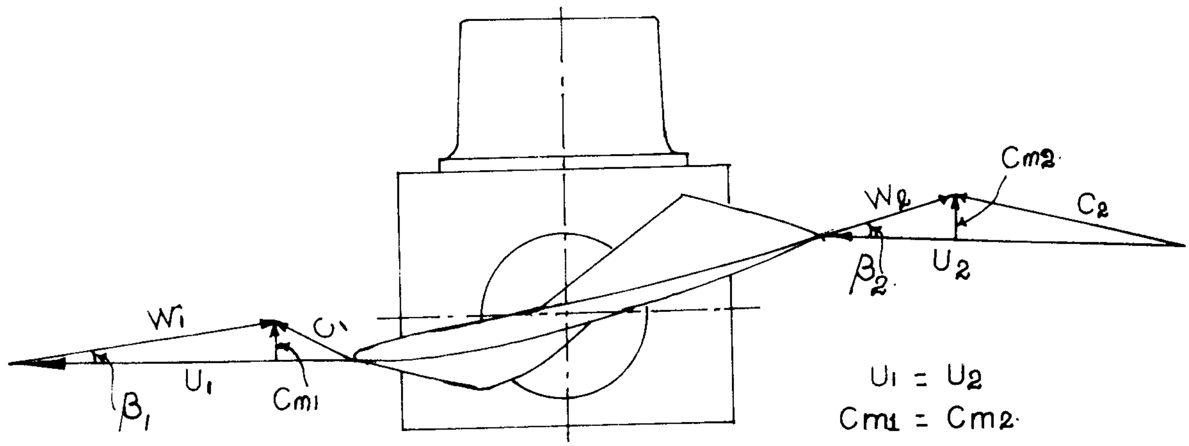


FIG. 2 VELOCITY DIAGRAM FOR AXIAL FLOW PUMP

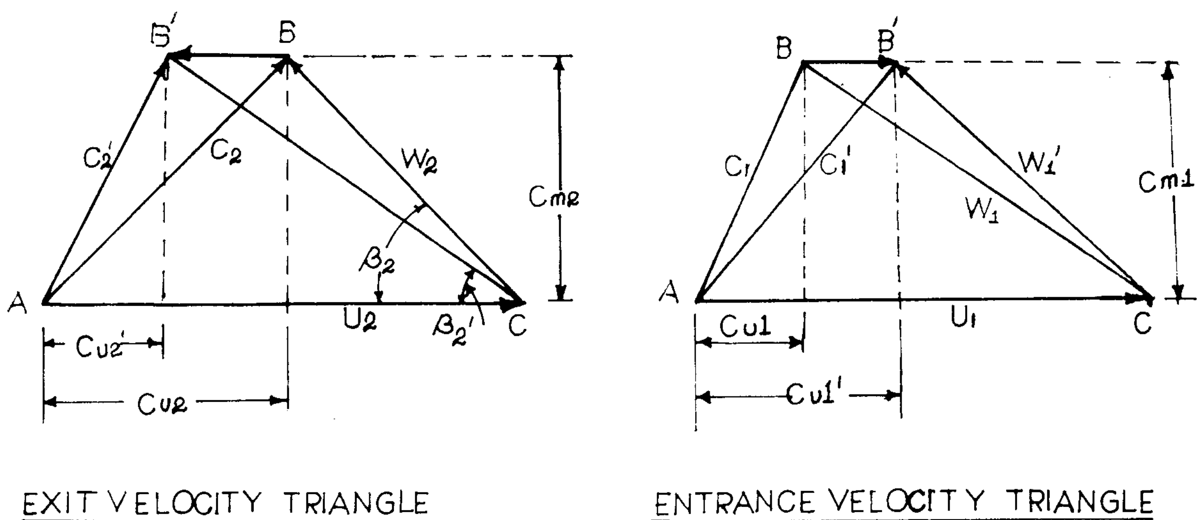


FIG. 3. EFFECT OF CIRCULATION IN OUTLET ANGLE

respectively. The prime is omitted if idealized velocities are used. The angle between the absolute velocity C' and peripheral direction is α'. The relative velocity W' forms with the peripheral direction, the angle β'. It is approximately equal to the vane angle β. The idealized relative velocity W forms with the peripheral direction the vane angle β. Figures 1and 2 are the velocity diagrams of centrifugal pump and axial flow pump respectively.

The main difference between the vector diagrams of centrifugal pump and that of axial flow pump are (i) The peripheral velocity at inlet and outlet are equal in axial flow pump, but in centrifugal pump both are different (ii) The meridional velocities at inlet and outlet are equal in axial flow pump whereas in the case of centrifugal pump both are different. Figure 2 shows the vector diagrams for a typical axial flow pump.

2. 2. 2. Slip Factor.

Often idealized velocity triangles are drawn by assuming perfect guidance of the flow by the vanes. The vane angle is substituted for the flow angle β'. The angle between the idealized absolute velocity C and the peripheral direction is denoted by α. The meridional Velocity C_m is usually assumed to remain the same. The relative velocity W and the absolute velocity C are not actual velocities, but they are utilized in the design of impellers since they are much easier to calculate. So the results must be corrected accordingly.

Deviation of the fluid from the vane direction is important at the impeller discharge since it reduces the peripheral component of the absolute impeller discharge velocity. This causes a proportionate reduction in head as wellas power input. Generally, the flow angle β_2' is smaller than the vane angle β_2 . This phenomenon is often called slip. The slip is represented by the ratio between the actual and theoretical tangential components of absolute velocity at outlet (C_{u2}/ C_{u3}). The slip is the outcome of nonuniform velocity distribution across the impeller channel, boundary layer accumulation and flow seperation due to circulation. Figure 3 will reveal this phenomenon. Practically it is very difficult to predict the slip. This causes a lot of difficulty in the proper design of an impeller.

The following formula for slip factor had been proposed by Stodola (1945).

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

where μ is the slip factor,

β_2 is the outlet vane angle,

Z is the number of blades.

2. 2. 3. <u>Basic Flow Equation.</u>

The fundamental Euler's equation for the operation of centrifugal pumps (radial, mixed and axial flow) is obtained from the principle of torque and angular momentum. As applied to fluid problems, the equation is

$$\Delta T = \frac{\gamma}{g} \left(C_{u2}R_2 - C_{u1}R_1\right)\Delta Q \text{ ....(1)}$$

where C_{u1} and C_{u2} are tangential components of absolute velocities at inlet and outlet respectively.

If the discharge is imagined to be divided into small elements ΔQ, the quantities $C_{u2}R_2$ and $C_{u1}R_1$ for the elements may vary both radially and with position between the blades at a constant radius. Equation (1) is often stated to be valid only for an infinite number of blades but the view is not correct (O'brien *et al.* 1936). It has the same validity as Newton's equations of motion and the reason for assuming an infinite number of blades is that the velocity component C_u can then be specified. Integrating equation (1),

$$T = \frac{\gamma Q}{g} \left(C_{u2}R_2 - C_{u1}R_1\right)$$

Multiplying by angular velocity, yields the energy transfer,

$$E = \frac{\gamma Q}{g} \left(\omega R_2 C_{u2} - \omega R_1 C_{u1}\right)$$

$$E = \frac{\gamma Q}{g} \left(U_2 C_{u2} - U_1 C_{u1}\right)$$

The head developed H_d = Energy/unit mass

Therefore $H_d = \frac{1}{g} \left(U_2 C_{u2} - U_1 C_{u1}\right)$ ......(2)

This head is known as Euler's head H_e.

The equation is known as Euler's equation in which the hydraulic losses are not considered. If the liquid enters the impeller without a tangential component, or if $C_{u1} = 0$ (radially for radial pumps and axially for axial pumps) Euler's equation reduces to

$$H_d = \frac{U_2 C_{u2}}{g} \quad \ldots\ldots\ldots(3)$$

From the velocity triangles

$$W_2^2 = C_2^2 + U_2^2 - 2U_2C_2 \cos \alpha 2$$
$$W_1^2 = C_1^2 + U_1^2 - 2U_1C_1 \cos \alpha 1$$

Making use of these, Euler's equation becomes

$$H_d = \frac{C_2^2 - C_1^2}{2g} + \frac{U_2^2 - U_1^2}{2g} + \frac{W_1^2 - W_2^2}{2g} \quad \ldots\ldots(4)$$

The first term represents a gain of kinetic energy of the flow through the impeller.

The second and third terms jointly represent an increase in pressure from the impeller inlet to the outlet. Stepanoff (1967) advocated that it is meaningless to attach any physical explanation to the second and third terms of equation (4) individually. Thus the second term does not represent entirely gain in pressure of the flow due to centrifugal force because there are no particles of the fluid moving with the peripheral velocities U_1 and U_2. Similarly, the third term does not represent an increase in pressure due to conversion of the relative velocity from w_1 to w_2, because it is observed that no diffusion can take place in a curved channel, stationary or moving. In the case of axial flow impellers there is no definite channel containing velocities w_1 and w_2.

2. 3. Vortex Theory of Euler's Head

2. 3. 1. Radial impeller.

Flow through the impeller can be considered as consisting of two components; namely, circular motion around the axis as a result of the impelling action of the vanes and through-flow or meridional flow caused by the energy gradient drop. The circular component of flow forms a vortex motion. The type of vortex depends on the velocity and pressure distribution and can be established from a study of Euler's equation. Considering a straight radial impeller in which the flow approaches the impeller without pre-rotation, Euler's equation can be written as

$$H_d = \frac{U_2^2}{g} - \frac{U_2 W_{u2}}{g} \quad \ldots\ldots\ldots (5)$$

Only tangential velocities appear in this equation, indicating that all head is produced by vortex action in planes normal to the axis of rotation. In general, this is true, for all centrifugal pumps, including straight axial flow pumps (Stepanoff, 1967).

When the flow is zero ($w_{u2} = 0$), Euler's head becomes

$$H_d = \frac{U_2^2}{g} = \frac{2U_2^2}{2g}$$

and the total head at any radius r is equal to

$$H = \frac{2U_2^2}{2g} \quad \ldots\ldots\ldots\ldots (6)$$

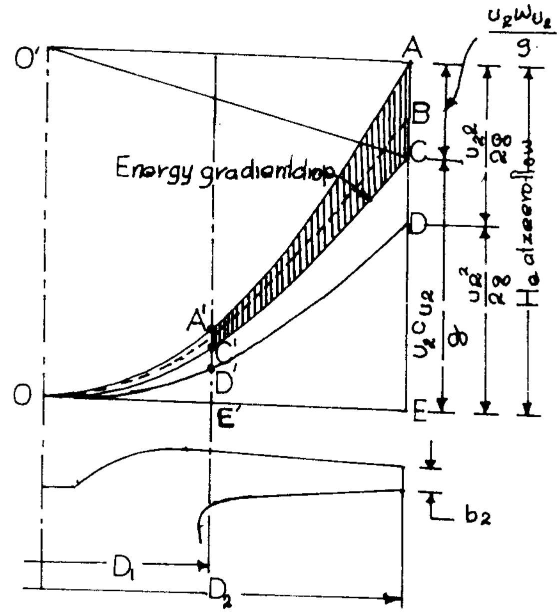


FIG.4. EULER'S HEAD, RADIAL FLOW IMPELLER. (STEPANOFF, 1967)

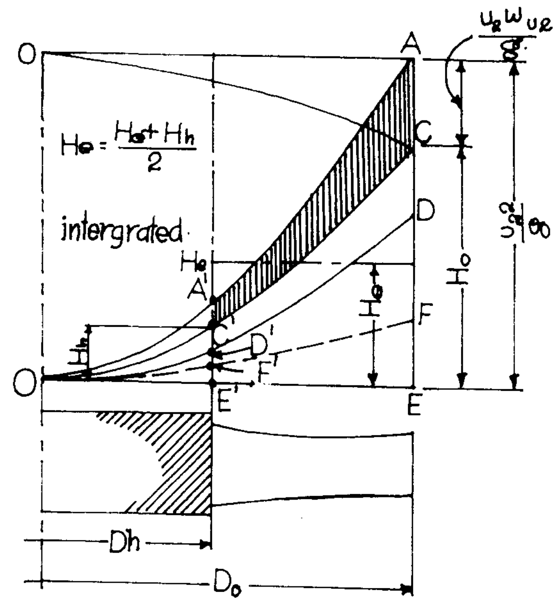


FIG.5. EULER'S HEAD, AXIAL FLOW IMPELLER. (STEPANOFF, 1967)

where U is the peripheral velocity at radius r. This head is equally divided between static and kinetic heads. Such energy distribution along the radius is typical for a forced vortex and is represented by a square parabola OA (Fig.4). When flow starts, the head drops by an amount $U_2 W_{u2}/g$, where W_{u2} is proportional to the flow. This head is the energy gradient drop necessary to produce flow, because even an idealized pump cannot start flow against a head higher than or equal to its zero flow head. It is also evident that a further drop of energy gradient is necessary to increase the flow or produce a higher capacity. Thus the total head drops from AE to CE. The value of $U_2 W_{u2}/g$ decreases with decreasing radii and the head variation along the radius is represented by a parabolic curve OC. Assuming W_{u2} to be constant along the radius, the energy gradient drop $U_2 W_{u2}/g$ will vary as U_2, or will increase directly as the distance travelled by the flow (O'C on Fig.4). This is analogous to the hydraulis gradient drop in a pipe flow of constant velocity. However, in this case the drop in hydraulic gradient represents hydraulic loss along the pipe, whereas in a centrifugal pump impeller the drop in energy gradient is a condition which is necessary to realize flow which results in an equal reduction of the impeller input. As the capacity increases, the energy gradient drop increases and Euler's head decreases correspondingly.

2. 3. 2. <u>Axial flow impeller.</u>

In an axial flow pump, the liquid particles leave the impeller at the same radius at which they enter. Applying Euler's equation to a point on the impeller periphery and noting that $U_2 = U_1$; we get

$$H_d = \frac{C_{u2}{}^2 - C_{u1}{}^2}{2g} + \frac{w_{u1}{}^2 - w_{u2}{}^2}{2g} \quad \text{........(7)}$$

Again assuming that the liquid approaches the impeller without pre-rotation ($C_{u1} = 0$ and $w_u{}^1 = U_1$) the above equation reduces to

$$H_d = \frac{U_2{}^2}{2g} + \frac{C_{u2}{}^2}{2g} - \frac{w_{u2}{}^2}{2g}$$

Substituting $C_{u2} = U_2 - w_{u2}$

$$H_d = \frac{U_2{}^2}{g} - \frac{U_2 w_{u2}}{g} \quad \text{.........(8)}$$

This is exactly the same equation for radial flow pump and indicates that the process of generating head is the same in axial flow pumps as it is in radial flow pumps. In both cases, head is generated through the vortex motion and the flow through the impeller is caused by the energy gradient drop $U_2 w_{u2}/g$. The head distribution along the radius is shown on fig.5, where curve AA' shows the head at different radii with zero flow. This is a square parabola. Curve CC' shows the head variation for one rate of flow (w_{u2}). AC is the energy gradient drop at the periphery. Ordinates between curves AA' and CC' represent the energy gradient drop for different radii. For a normal design both w_{u2} and U_2 vary directly as the radius. Therefore the energy gradient drop $U_2 w_{u2}/g$ varies directly as the square of the radius (curve O'C), and the curve OC is a square parabola. This is a characteristic feature of a forced vortex when all particles rotate with the same angular velocity.

Although the head distribution along the radii is similar for radial and axial flow pumps there is an important difference between the final head produced by the two. In a radial impeller all particles reach the same maximum head at the periphery of the impeller, whereas in an axial flow pump, liquid particles enter and leave at the same radii and the head produced at different radii are different, being a maximum at the periphery and a minimum at the hub. The pump total head is an integrated average. The integrated head of the impeller in Fig.(5) is equal to the average of the head at the hub (H_h) and the head at the periphery (H_o). This follows from the geometric properties of square paraboloid.

$$H_d = H_e = \frac{H_h + H_o}{2} \quad \ldots\ldots\ldots\ldots\ldots (9)$$

2. 4. Flow through Hydrodynamic Machine

2. 4. 1. Flow pattern.

Flow through a hydrodynamic machine is generally complicated in nature. Flow through a hydrodynamic machine is assumed to be axissymmetric in character ie. flow velocity and other parameters are constant along any circle at radius 'r' from machine axis. This obviously is not true at points where blades of machine cut the stream surface and the particular circle of stream surface under consideration. Thus, axissymmetric flow really can occur only in a vaneless space of revolution. However, without this assumption, no flow analysis is really possible (Vasandani, 1977).

The assumption of axissymmetric flow also reduces the three dimensional problem effectively into two dimensional problem. Parameters describing flow in such a case vary (i) when we move from one concentric circle to another on the same stream surface or (ii) when we move from one stream surface to another.

In addition to the above assumptions, we further assume that (i) flow parameters change only along a stream line (ii) there is no variation in these parameters radially from one stream surface to another (iii) the flow is frictionless and (iv) flow takes place along the passage in direction parallel to that of blades, ie. the flow is assumed to be one dimensional frictionless flow. Such a flow will be irrotational in character. This assumption involves in arriving at Euler's basic equation.

Simplification of the flow to one dimensional frictionless flow, parallel to the path of blades, not only makes it possible to use Euler's equation for transmission of torque or specific energy conversion but also it makes it easier to calculate the value of discharge passing through the runner. Thus for radial flow runner, flow cross section normal to meridional stream lines is the product of circumference and axial width. Thus we have

$$Q = 2\pi r B C_m = \pi D B C_m$$

where, B is the Axial width,

C_m is the meridional velocity.

The above equation does not take into account the blockage effect due to blade thickness. If this is taken into account, Q reduces to

$$Q = (2\pi r - Zs)\ BC_m \ldots\ldots\ldots\ldots\ldots (10)$$

where Z - number of blades.

s - blade thickness measured along circumference.

Similarly for an axial flow machine, neglecting the effect of blade thickness

$$Q = \frac{\pi}{4} (D_o^2 - D_i^2)\ C_m$$

where D_o is the outside tip diameter

D_i is the hub diameter.

This may also be written as

$$Q = \frac{\pi}{4} D_o^2 \left(1 - \frac{D_i^2}{D_o^2}\right) C_m \ldots\ldots\ldots\ldots (11)$$

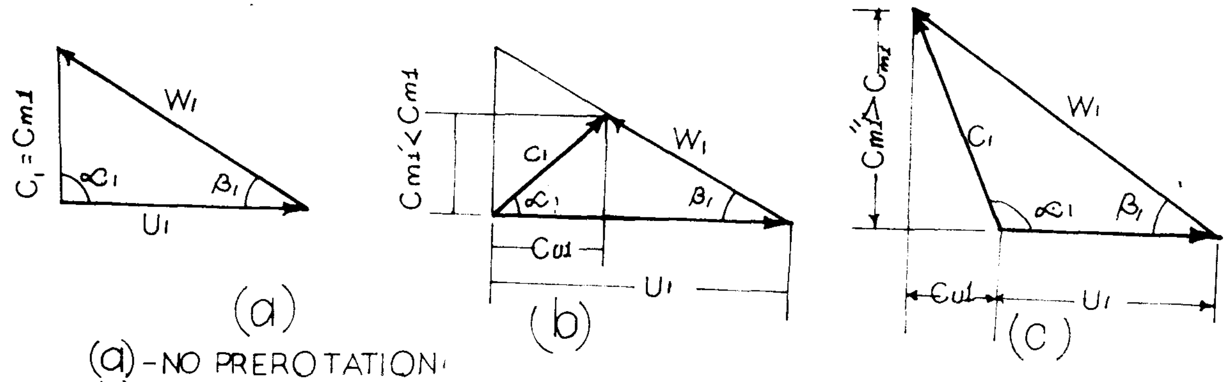
Equation (10) also presume that flow through the runner is steady. This is not strictly true because as every point in space within the runner is cut by the blades during the runner rotation and thus causing change in value of parameters at that point. However, since the flow is presumed to be axissymmetric and hence it may as well be presumed as steady.

The effect of friction on flow is two fold (i) It causes hydraulic losses and thus reduces the specific energy conversion inside the runner. This is taken care of by the term "hydraulic efficiency." (ii) effect of friction, and that is velocity profile across the flow passage along axial or circumferential directions can not be uniform as assumed in one dimensional flow.

2. 4. 2. Impeller approach and prerotation.

The flow towards the impeller and beyond the impeller is caused by the drop of the energy gradient below its level at zero flow. The drop in energy gradient permits liquid to proceed through the impeller against a gradually increasing head. Following the energy gradient the liquid selects a path of least resistance to get into and through the impeller and out of the pump. The liquid acquires prerotation to enter the impeller passage with a minimum disturbance and the direction depends on the impeller vane entrance angle β_1, the capacity going through and the impeller peripheral velocity. All these three factors determine the entrance velocity triangle (Stepanoff, 1967).

It is evident that resistance to flow is a minimum if the liquid enters the impeller at an angle approaching the vane angle β_1. For a given impeller speed, there is



(a) - NO PREROTATION.
(b) - PREROTATION IN THE DIRECTION OF IMPELLER ROTATION.
(c) - PREROTATION IN THE OPPOSITE DIRECTION.

FIG.6. EFFECT OF PRE-ROTATION

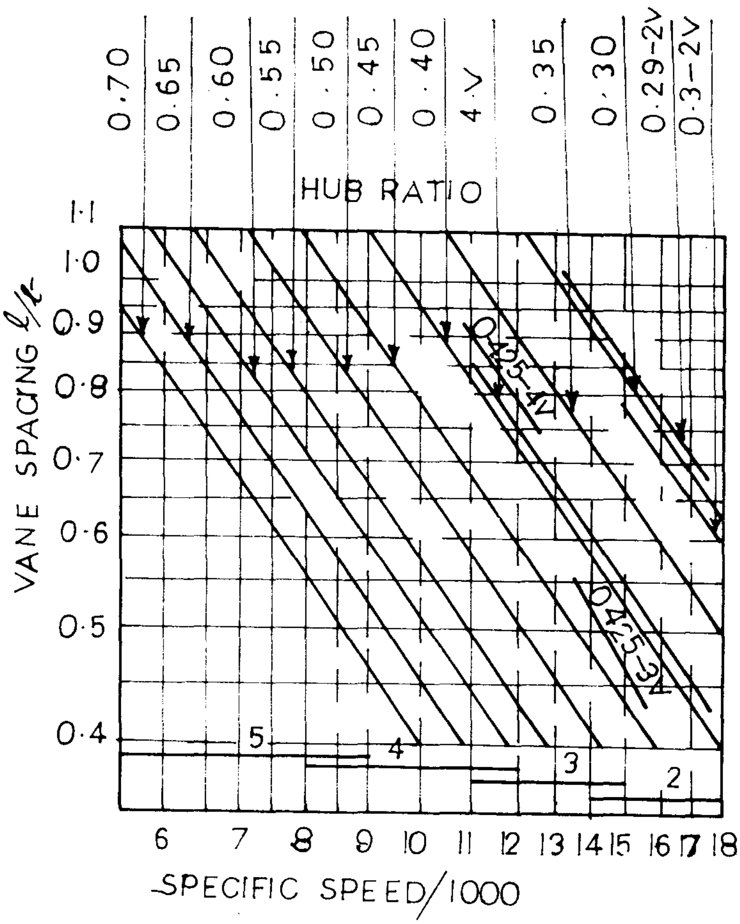


FIG.7. HUB RATIO, NUMBER OF VANES AND l/t RATIO FOR AXIAL FLOW PUMPS. (STEPANOFF, 1967)

only one capacity at which the liquid will approach the impeller meridionally, or without prerotation. At a capacity considerably smaller than normal the liquid should acquire prerotation in the direction of impeller rotation, to enter at an angle approaching β_1. But a capacity greater than normal a prerotation in the opposite direction is necessary for the liquid to satisfy the " least resistance " requirement. Figure6 shows the entrance velocity triangles for the above three conditions.

2. 5. Efficiencies

2. 5. 1. Hydraulic efficiency.

All the head in a pump is generated by the impeller. The rest of the parts contribute nothing to the head but incur losses - hydraulic, mechanical and leakage. All losses of head which take place between the points where the suction and discharge pressure are measured constitute hydraulic losses. Hydraulic efficiency is defined as the ratio of the available total dynamic head to the input head, or

$$e_h = \frac{H}{H_i} = \frac{H_i - \text{hydraulic losses}}{H_i}$$

The ratio of input head to Euler's head will be referred to as vane efficiency, or

$$\frac{H_i}{H_d} = e_{va}$$

2. 5. 2. Volumetric efficiency.

Besides losses of head there are losses of capacity in each pump known as leakage losses. These take place through the clearances between the rotating and stationary parts of the pump. The discharge at the exit of the pump is smaller than that passed through the impeller by the amount of leakage. Thus the volumetric efficiency is

$$e_v = \frac{Q}{Q_i} = \frac{Q}{Q + Q_L}$$

where Q_L is the amount of leakage.

2. 5. 3. Mechanical efficiency.

Mechanical losses include loss of power in bearings and stuffing boxes and the disk friction. The disk friction is hydraulic in nature but is grouped under mechanical losses since it is external to the flow through the pump and does not result in a loss of head. The mechanical efficiency is the ratio of the power actually absorbed by the impeller and converted into head to the power applied to the pump shaft

$$e_m = \frac{\text{Brake horsepower - mechanical losses}}{\text{Brake horsepower}}$$

$$e_m = \frac{\gamma Q_i H_i}{K}$$

where the value of K depends on the units. By substituting for Q its value $Q = e_v Q_i$, for H its value $H = e_h H_i$ and for K its value $K = \gamma Q_i H_i / e_m$, the relationship between the partial efficiencies and overall efficiency can be obtained as follows.

$$e_o = \frac{\gamma QH}{K}$$

$$e_o = \frac{\gamma\, e_v Q_i \times e_h H_i}{(\gamma Q_i H_i / e_m)}$$

$$e_o = e_v e_h e_m \quad \ldots\ldots\ldots\ldots (12)$$

2. 6. Design Procedure

The design of a pump impeller involves the following steps.

2. 6. 1. Selection of speeds.

To meet given head capacity conditions the rotative speed is selected first. This establishes the specific speed or type of the impeller. Selection of the speed is governed by a number of considerations.

i. Type of driver available for the unit

ii. Higher specific speed results in a smaller pump and cheaper driver

iii. Optimum hydraulic (and total) efficiency possible with each type varies with the specific speed.

iv. If the total head required cannot be produced in one stage, it is divided into two or more stages.

The head per stage also affects the final specific speed and hence the expected efficiency of the pump.

Having established the specific speed of the proposed impellers, the designer looks for suitable configuration from existing impellers of the same specific speed which have satisfactory hydraulic performance, ie. suitable slope of the head capacity curve and acceptable efficiency. Besides the required specific speed the model should be of the same class of pump and be of suitable mechanical type. For instance an impeller of a multistage pump would not be a suitable model for single-stage overhung construction with an end inlet. The reduction factor or multiplier to be added to the existing model is found by the use of affinity formulae. Design of an impeller for which no existing type is available is made from basic design constants.

2. 6. 2. <u>Reduction factor or Multiplier.</u>

If an impeller which is selected for a model is rated Q_1 lps at H_1 m head at n_1 rpm and its impeller diameter is D_1 and the new impeller is required to produce Q_2 lps H_2 m head at n_2 rpm with an impeller diameter D_2 the specific speed of both should be the same,

$$n_1 Q_1^{\frac{1}{2}} H_1^{\frac{3}{4}} = n_2 Q_2^{\frac{1}{2}} H_2^{\frac{3}{4}}$$

In addition, the following relationship between model and prototype with regard to capacities and heads can be established.

$$Q_2 = Q_1 f^3 (n_2/n_1)$$

$$H_2 = H_1 f^2 (n_2/n_1)^2$$

where $f = D_2/D_1$ is the reduction factor or multiplier. From equations above formula for reduction factor f is obtained:

$$f = \frac{n_1/\sqrt{H_1}}{n_2/\sqrt{H_2}} = \frac{n_1}{n_2}\left(\frac{H_1}{H_2}\right)^{\frac{1}{2}}$$

Expression $(n/\sqrt{H})$ is referred to as "unit speed" meaning revolutions per minute, needed to produce unit head by a given impeller. It is also possible to express the factor f in terms of capacities and speed,

$$f^3 = \frac{Q_2/n_2}{Q_1/n_1} = \frac{n_1 Q_2}{n_2 Q_1}$$

Q/n is referred to as unit capacity and represents litres per second per revolution for a given impeller.

2. 6. 3. New Impeller Design.

To design a new impeller for which no model is available designers use "design factors" established experimentally from successful designs that give direct relationship between the impeller total head and capacity at the design point and several elements of Euler's velocity triangles. These are dimensionless velocity ratios independent of the impeller size and speed which are correlated on the basis of specific speed for different impeller discharge angles. In addition a number of ratios of important linear dimensions, not directly related to velocities, are found helpful in perfecting hydraulic design of impellers. The ratios are entirely experimental. The degree of perfection of a design is measured by the value of the pump hydraulic efficiency.

The impeller profile and vane layout is possible if the following elements are known:

i. Meridional velocities at inlet and outlet

ii. Impeller outside diameter

iii. Impeller vane inlet and outlet angles.

These same quantities determine both Euler's entrance and discharge triangles. For straight radial vanes, all particles of fluid enter and leave the impeller at the same diameter, and the vane is "plain" or of single curvature. Thus only one entrance and one discharge triangle determine the impeller design. For mixed flow and axial flow impellers velocity triangles are drawn for several streamlines. Three streamlines will be usually sufficient for average mixed flow and axial flow impellers. Variation of vane angles along the radius determines the vane curvature and "twist".

2. 6. 4. <u>The Vane Discharge Angle.</u>

This angle is the most important design element. It has been shown that the theoretical characteristicsare determined by the vane angle alone. In practice, β_2 is still the deciding factor in design. All the design constants depend on the value of β_2 . Therefore a choice of β_2 is the first step in selecting impeller design constants. This selection is based on the consideration of the desired steepness of the head-capacity curve and whether or not a maximum output is desired from the impeller of a given diameter as both normal head and capacity increase with the angle β_2 . If there are no such limitations, selection of β_2 is made for an optimum efficiency, or normal design.

2. 6. 5. Speed constant.

A speed constant is a factor giving the relation between the pump total head and the impeller peripheral velocity.

The speed constant is defined as follows:

$$K_u = \frac{U_2}{\sqrt{2gH}}$$

This was originally introduced for hydraulic turbines and later adopted by pump engineers. In this definition, K_u is a ratio of U_2 to the free jet velocity under head H. It is used for calculation of the impeller diameter when the head H is given and the speed is selected. Herbert Addison (1976) proposed that for a good axial flow pump the value of K_u should be in between 2 to 2.7.

2. 6. 6. Specific Head and Head Coefficient.

The equation $h_s = gH/n^2D^2$ termed as "specific head" means input energy per unit mass per revolution and with an impeller of unit diameter. It remains constant for all similar impellers. Affinity formulae follow from this property of the specific head; for a given D, head varies directly as square of the speed to satisfy the above condition. Also if n is kept constant, these head H varies directly as the square of the impeller diameter.

As a dimensionless factor the specific head expression is slightly modified and is known as the "head coefficient". It is noted by

$$\psi = \frac{H}{U_2^2/g}$$

2. 6. 7. <u>The Capacity Constant.</u>

Capacity constant is defined by $K_{m2} = \dfrac{C_{m2}}{\sqrt{2gH}}$

Where C_{m2} is the meridional velocity at discharge.

2. 6. 8. <u>The Capacity Coefficient.</u>

This is used as a capacity design constant and is defined as $\phi = C_{m2}/U_2$

Where C_{m2} is the meridional velocity at impeller discharge, for the best efficiency point based on the net discharge area (excluding vanes and disregarding the leakage) The capacity coefficient increases for higher specific speeds at constant values of β_2. Also, ϕ increases with the angle β_2 for a constant specific speed. It is connected to K_{m2} as follows;

$$\phi = \frac{K_{m2}}{K_u}$$

2.6. 9. <u>The Entrance Velocity.</u>

In order to complete the impeller profile, the meridional velocity at entrance should be also known. This is given by the ratio

$$K_{m1} = \frac{C_{m1}}{\sqrt{2gH}}$$

This is calculated for the area at the vane entrance tips, omitting the leakage. The vane thickness can be disregarded as the vane tips are usually tapered, and C_{m1} can be assumed to be the velocity just ahead of the vanes.

Neglecting leakage introduces an error in vane angles β_1 and β_2 for several stream lines as determined from Euler's velocity triangles. Comparing to the inaccuracy resulting from the assumption of a uniform meridional velocity for several streamlines, this error is negligible.

2. 7. Axial flow impeller design procedure

The design procedure for a single-stage axial flow impeller is same as for a centrifugal impeller. The design procedure involves the following steps.

(i) To meet a given set of head-capacity requirements, the speed is selected; thus the specific speed of the impeller is fixed. Due consideration should be given to the head range, the proposed pump should cover in future applications under the most adverse suction conditions.

(ii) For the specific speed thus obtained, the hub ratio and vane spacing (l/t) are selected. The number of vanes is fixed at the same time.

(iii) The speed constant and the capacity constant are chosen next. These constants having been established, the meridional velocity and impeller diameter can be calculated and the impeller profile can be drawn.

(iv) The impeller vane profiles for both vane curvature and vane twist, are drawn after the entrance and discharge vane angles for several streamlines are established from Euler's entrance and exit velocity triangles. In drawing the vane profiles for several streamlines, airfoil shapes are good to follow, but the vane thickness should be kept to a minimum consistant with the vane mechanical strength.

2. 8. Experimental Design Factors

There are a number of design elements of axial flow pumps which do not enter into the theoretical discussion although they affect directly the performance of the axial impeller. These include: (i) hub ratio, (ii) number of vanes, (iii) vane thickness, (iv) turning of vanes on the hub as it occurs in adjustable vane impellers, and (v) pump casing, with or without diffusion vanes. Selection of any of these design elements depends upon experience. As more factors are involved it depends upon the skill of the designer to discern the effects of these several variables leading to the optimum hydraulic performance.

2. 8. 1. Impeller Hub Ratio.

The ratio of impeller hub diameter to the impeller outside diameter is directly connected with the specific speed of axial flow pumps. Pumps with higher specific speed have smaller hubs, which give a greater free area for the flow and a smaller diameter to the average streamline, resulting in a greater capacity and a lower head. Figure 7 gives the hub ratio for various specific speeds compiled (Stepanoff, 1967) from a number of modern axial flow pumps and blowers. The hub ratio is the most important design element controlling specific speed of the axial flow impeller.

2. 8. 2. Chord-Spacing Ratio.

The chord-spacing ratio (l/t) is another important design element which is selected on the basis of previous experience. For axial flow pumps of specific speeds of 10,000 rpm and higher, the ratio (l/t) is less than unity. It is possible to obtain the same values of (l/t) with a different number of vanes. The actual hub ratios and number of vanes for different specific speeds are marked for each point on Fig. 7.

2. 8. 3. Number of Vanes.

Kaplan (1935) found that for a given wetted area of the vane (l/t) the number of vanes should be minimum. This was also confirmed by Schmidt. He showed that a two-vane impeller was most efficient with a projected vane area of about 63 per cent.

With heavy vanes and a low chord angle, the maximum number of vanes is almost fixed since adding vanes will restrict the free area of the flow. The normal capacity will decrease and efficiency will drop.

2. 8. 4. Vane Curvature and Vane Setting.

Schlimbach (1935) showed that the head produced is essentially the same for all vane setting and thus is a function of the vane curvature ($\beta_2 - \beta_1$) alone.

2. 8. 5. Vane Thickness.

Eckert (1944) tested two impellers, one with airfoil vanes well streamlined and polished, the other of the same solidity and camber line but made of stamped steel sheet vanes welded to the hub. The performance of the two impellers proved identical.

Eckert (1944) also found that another impeller of the same airfoil pattern but made of cast iron with the trailing edge about 1/8 inch thick was lower in efficiency. The efficiency reduction was caused by the greater relative roughness of the cast iron vanes as compared to the polished alloy vane. Excessive vane thickness results in seperation and noise with high pressure high speed impellers. Thus the advantage of airfoil sections lie in the fact that they permit the desired mechanical strength with a minimum sacrifice of efficiency.

2. 9. Design for pump casing

The purpose of the diffusion casing of an axial flow pump is to convert the tangential component of the absolute velocity leaving the impeller into pressure. This is done by straightening the flow as it leaves the impeller and by reducing the velocity.

In addition to the reduction obtained by converting the tangential velocity component into pressure, the axial velocity is reduced by increasing the diffuser diameter at the discharge. A small divergence angle of the diffuser cone (8° total) is essential for an effective conversion (Morelli *et al.*, 1953; Stepanoff, 1967).

Chapter 3

MATERIALS AND METHODS

MATERIALS AND METHODS

3. 1. Objectives

The broad objective of the project was to develop an axial flow pump with improved efficiency which could be economically used for dewatering and for lift irrigation purposes under low head conditions.

This project was undertaken with the following specific objectives.

(i) To make a study of various water lifting devices used in Kerala for agricultural purposes.

(ii) To evolve a systematic design for a propeller pump which can be used for dewatering and for lift irrigation purposes.

(iii) To fabricate the pump and evaluate its performance by test.

(iv) To optimize the design of the pump and recommend for commercial manufacturing.

3. 2. Brief description of water lifting devices in Kerala

Water lifters range from age-old indigeneous water lifts to highly efficient pumps. Selection of a suitable water lifting device for a particular requirement depends on the characteristics of the source of water, the quantity of water to be lifted, the depth to the pumping water level from ground surface, type and amount of power available at that field and the economic status of the farmer.

3. 2. 1. Indigeneous water lifters.

Many types of indigeneous water lifts are used in Kerala. They may be mannually operated or animal operated. Based on the optimum range in the height of lift, they are grouped under devices for low lift, medium lift and high lift.

3. 2. 1. 1. Low head water lifts

The swing basket, Don, Archemedian screw and Water wheel are used when the depth to water level is not more than 1.2m.

Swing basket.

One of the most ancient water lifts is the swing basket consisting of a basket or shovel like scoop to which four ropes are attached. Two persons stand facing each other and swing the basket to fill in water. The basket is raised and discharged into the field channel.

Don.

The don is a mannually operated boat shaped trough, closed at one end and open at the other end. The closed end of the trough is tied with a rope to a long wooden pole which is pivoted as a lever on a post. A weight is fixed to the shorter end of the lever. The open end is hinged to the discharge point. The trough is dipped into the water by applying the body weight and the force of the operator. Water is lifted by the counter-weight on the beam and is emptied.

Archemedian screw.

The device is mannually operated and consists of a wooden or metal wheel drum with inner partition in the form of a screw. The screw is rotated by means of a handle fixed to a central spindle. The spindle projects from both ends and is supported near its ends by posts. The drum is placed at an angle less than 30° with its lower end in water. When the handle is turned, the water moves up through the drum and discharges through the upper end.

Water wheel.

The water wheel consists of small paddles mounted radially on a horizontal shaft. The wheel is fixed on a close fitting concave trough. The wheel when rotates, pushes the water to the field through the trough.

3. 2. 1. 2. Medium Head Water Lifts.

Medium head lifts are used when the height of lift is within the range of 1.2m to 10m. The Persian wheel, chain pump, circular two-bucket lift and the counter-poise bucket lift are categorised into medium head lifts.

Of these counterpoise bucket lift is very popular in Kerala. This device consists of a long wooden pole which is pivoted as a lever on a post. A weight is fixed to the shorter end of the pole. This weight serves as a counter weight to a bucket suspended by a rope or a rod attached to the long arm of the lever. To operate the lift a man pulls down the rope or rod until the bucket is filled. The bucket is drawn up by the counter weight. When the bucket reaches the ground level it is tipped into a trough.

3. 2. 1. 3. High Head Water Lift.

Rope and bucket lift.

The only indigeneous high head water lift in use is the rope and bucket lift. The device may be operated singly or in multiples of two or more working simultaneously powered by men or animals.

3. 2. 2. Windmill and Hydraulic Ram.

Windmill and Hydraulic ram are two promising devices to tap natural energy from wind and waterfalls respectively. Eventhough many farmers have been using wind powered reciprocating pumps for irrigation from years back, it is not so popular. Low velocity of wind during some months and high initial cost are the two main reasons for less popularity of windmill among Kerala farmers.

The hydraulic ram is an impulse pump which utilizes the momentum of falling water. Advantage is taken of a small fall in a running stream to lift water to a great height. The ram can be installed where a relatively large amount of water at a moderate head is available in a stream with rapid fall, where the pump can be installed in its bed to pump a small volume to a higher level than the supply. Hydraulic ram known as the "Pump without Power" is rarely used in Kerala.

3. 2. 3. Pumps.

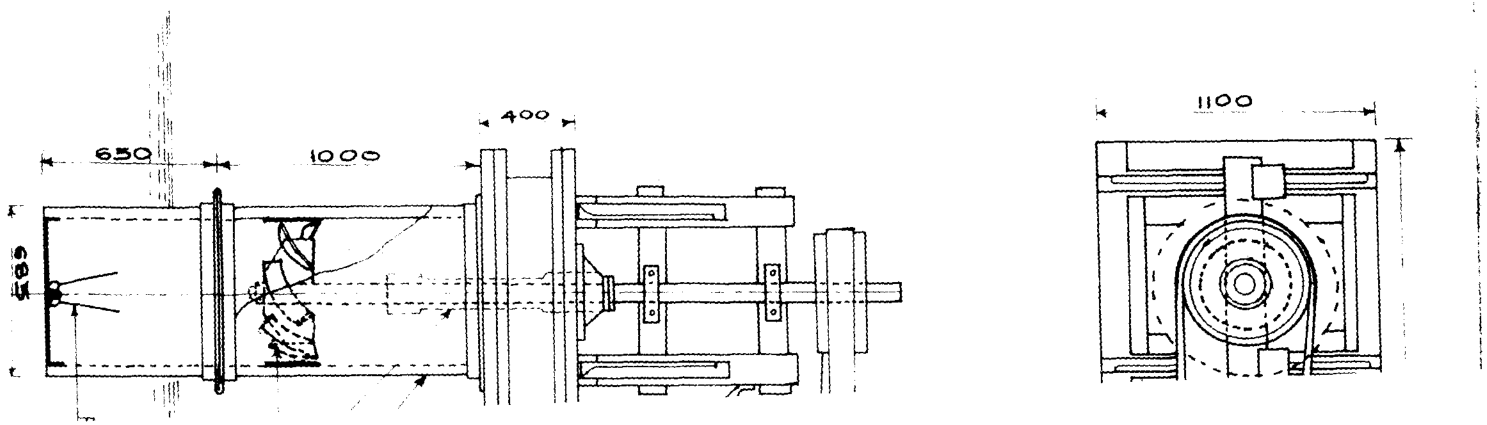
The pumps usually used by the farmers of Kerala are centrifugal pumps , piston pumps, mixed flow pumps and axial flow pumps. The primemover used is either oil engine or electric motor.

The specific speed of an impeller is the most valuable index of the type of pump, which is important in determining the maximum head. The specific speed of an impeller may be defined as the revolutions per unit time to which a geometrically similar impeller would run if it were of such size as to discharge unit quantity per unit time against unit head.

An indigeneously fabricated axial flow pump with a local name "Petti and Para" is commonly used in Kerala for dewatering.

3. 2. 4. Petti and Para.

A Petti and Para consists of a cylindrical wooden drum (Para) with a horizontal outlet (Petti). The two long sides of the Petti should be made of single piece wooden planks. The rear end of the box is completely closed and is provided with an inspection opening with water tight sliding shutter. The front end of the box is provided with a one way valve fitted on hinges with suitable inclination so that water from outside will not enter the Petti when the pump is not working. The top


650
1000
400
1100

and bottom sides of the box is tongued and grooved to make it water tight with wooden planks.

The Para is of wooden planks with joints tongued and grooved. Para is strengthened by providing angle iron rings at the top and bottom, and by providing a metal band at the middle. Holes are provided on angle iron to join Para with another Para and Para with Petti. Para is provided with a protective inside cover where the impeller works, made of sheet metal. A metal ring is provided where the Para is connected to the Petti. All rivets used for fixing angle iron, protective cover etc. are of copper.

The impeller consists of a cast iron hub to which suitably shaped leaves are fixed. The impeller shaft is made of mild steel. A suitable cast iron "stand tube" is provided on which a heavy duty ball bearing works. A collar is rigidly bolted to the shaft and it rests on the ball bearing. Two gun metal bush bearings are provided inside the "stand tube", one at the top and one at the bottom. The pulley is fixed in between two gun metal bearings.

Figure 8 shows the different parts of Petti and Para. At present Petti and Para is being manufactured by local blacksmiths based on their practical experience and so the performance of each unit vary widely.

3. 3. Impeller Design

3. 3. 1. Specific speed.

The requirement of the pump to be designed was studied and the capacity, head, and speed of the pump were fixed. The capacity fixed was 250 lit/sec and head was 1.5m. From the data available, for optimum conditions speed of the pump is in between 600 rpm and 1000 rpm. So the speed of the pump was taken as 700 rpm. From these data, the specific speed of the pump was determined.

Stepanoff from his experimental studies, found out that high specific speed pumps are having more hydraulic efficiency. If the total head cannot be produced in one stage, it has to be divided into two or more stages. So the head was limited to 1.5m. The capacity was fixed as 250 lit/sec, because it is easy to handle a medium size pump. At the same time it needs only cheaper foundation.

$$N_s = N \sqrt{Q}/H^{3/4}$$

where N_s is the specific speed, in rpm,

N is the speed of the pump, in rpm,

H is the head, in feet,

Q is the discharge in gallons/minute.

For the easiness of comparing the value of N_s with the values suggested by Stepanoff (Fig.7) the unit of H and Q are taken in FPS units.

$$N_s = N\sqrt{Q}/H^{3/4}$$

$$N_s = \frac{700\sqrt{\frac{250 \times 60}{4.546}}}{(150/2.54 \times 12)^{3/4}}$$

$$N_s = 12169.95 \text{ rpm}$$

N_s was approximately taken as 12200 rpm

Again in metric units, Kinematic specific speed

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

$$N_s = \frac{700 \times \sqrt{0.250}}{1.5^{3/4}}$$

$$N_s = 258.23 \text{ rpm}$$

$$N_s = 260 \text{ rpm (approximately)}$$

Herbert Addison (1976) suggested that optimum number of blades is 3 to 5. Therefore number of blades was fixed as 3. This value is in good agreement with the value suggested by Stepanoff (Fig.7).

3. 3. 2. <u>Horse power of the pump.</u>

The horsepower of the pump was found out by using the formula.

$$HP = \frac{wQH}{75 \times e_o}$$

where Q is the quantity in m^3/sec.,

W is the specific weight in kg/m^3,

H is the head in m,

e_o is the overall efficiency.

$$HP = \frac{1000 \times 0.250 \times 1.5}{75 \times 0.50}$$

$HP = 10$

The overall efficiency assumed was 50 per cent.

3. 3. 3. Design for impeller dimensions.

Mechanical cross section is given by

$$A_m = Q'/C_{m1}$$

where A_m is the mechanical cross section in m^2,

Q' is the quantity to be pumped in m^3,

(considering the volumetric efficiency)

C_{m1} is the meridional velocity in m/sec.

Q' is given by

$$Q = \frac{Q}{e_v}$$

where e_v is the volumetric efficiency.

C_{m1} is given by

$$C_{m1} = K_{m1}\sqrt{2gH}$$

where K_{m1} is the coefficient of flow velocity or capacity constant at inlet,

g acceleration due to gravity in m/sec^2,

$$Q' = Q/e_v$$

$$Q' = \frac{0.250}{0.90} = 0.278 m^3/sec.$$

Volumetric efficiency assumed was 90 per cent

$$C_{m1} = K_{m1}\sqrt{2gH}$$

K_{m1} was assumed as 0.55

$$C_{m1} = 0.55\sqrt{2 \times 9.81 \times 1.5}$$

$$C_{m1} = 2.98 \text{ m/sec.}$$

$$A_m = Q'/C_{m1}$$

$$A_m = \frac{0.278}{2.98}$$

$$A_m = 0.0932m^2$$

3. 3. 4. Hub Ratio.

Hub ratio = (d_h/d_2)

where d_h is the diameter of the hub.

d_2 is the impeller outer diameter.

Addison (1976) suggested that for good performance of the pump, hub ratio should be between 0.4 and 0.55. Therefore hub ratio was fixed as 0.46.

The mechanical cross section is the cross section at impeller, through which water flows.

$$A_m = \frac{\pi}{4}(d_2^2 - d_h^2)$$

$$A_m = \frac{\pi}{4} d_2^2(1 - (d_h/d_2)^2)$$

$$d_2 = (4A_m/\pi(1 - (d_h/d_2)^2))^{\frac{1}{2}}$$

$$d_h/d_2 = 0.46$$

$$d_2 = (4 \times 0.0932/\pi(1 - 0.46^2))^{\frac{1}{2}}$$

$$d_2 = 0.3879m$$

$$d_2 = 39cm$$

$$d_h = 0.46d_2$$

$$d_h = 0.46 \times 39$$

$$d_h = 17.85$$

$$d_h = 18cm$$

3. 3. 5. Inlet angle.

Inlet angle β_1 is given by,

$$\mathrm{Tan}\,\beta_1 = \frac{C_{m1}}{U_1}$$

where C_{m1} is the meridional velocity at inlet,

U_1 is the absolute velocity at inlet,

$$U_1 = \frac{D_m N \pi}{60}$$

where D_m is the mean diameter given by $(d_h+d_2)/2$ in m.

N is the speed of the impeller in rpm.

$$C_{m1} = 2.98$$

$$D_m = \frac{18 + 39}{2} = 28.5\text{cms}$$

$$D_m = 0.285\text{m.}$$

$$U_1 = \frac{\pi \times 0.285 \times 700}{60}$$

$$U_1 = 10.45\text{m/sec}$$

$$\mathrm{Tan}\,\beta_1 = \frac{2.98}{10.45}$$

$$\mathrm{Tan}\,\beta_1 = 0.285$$

$$\beta_1 = \mathrm{Tan}^{1}(0.285)$$

$$\beta_1 = 15.92°$$

Therefore inlet angle was fixed at 16°

3. 3. 6. Outlet Angle.

Always it is better to fix the outlet angle by trial and error. Brunoeck (1961) practically proved that outlet angle from 22° to 27° is good for efficient and effective working of the pumps.

From the hydraulic point of view the theoretical angle is given by

$$\tan\beta_3 = \frac{C_{m2}}{U_2 - C_{u3}}$$

where β_3 is the theoretical outlet angle ,

C_{m2} is the meridional velocity at outlet ,

U_2 is the absolute velocity at outlet.

C_{u3} is the tangential component of the absolute velocity (theoretical) at outlet.

In axial flow pumps as water enter and leave the blade at the same radius U_1 is taken as U_2 and C_{m1} is taken as C_{m2} or

$$U_1 = U_2$$

$$C_{m1} = C_{m2}$$

C_{u3} is given by the formula,

$$C_{u3} = g.H_{th}/U_2$$

where g is the acceleration due to gravity,

H_{th} is the theoretical head,

U_2 is the absolute velocity at outlet ,

$$H_{th} = \frac{H}{e_h}$$

where e_h is the hydraulic efficiency.

From these data the constructional angle β_2 was fixed

$$\tan\beta_3 = \frac{C_{m2}}{U_2 - C_{u3}}$$

$$C_{u3} = gH_{th}/U_2$$

$$H_{th} = H/e_h$$

$$H_{th} = \frac{1.5}{0.8}$$

$$H_{th} = 1.875\text{m.}$$

Hydraulic efficiency was taken as 80 per cent

$$C_{u3} = \frac{9.81 \times 1.875}{10.45}$$

$$C_{u3} = 1.76 \text{ m/sec.}$$

$$\text{Tan}\,\beta_3 = \frac{2.98}{10.45 - 1.76}$$

$$\text{Tan}\,\beta_3 = 0.3429$$

$$\text{Tan}\,\beta_3 = \text{Tan}^{-1}(0.3429)$$

$$\beta_3 = 18.93°$$

$$\beta_3 = 19° \text{ (approximately)}$$

Theoretically angle β_2 will be less than 19°.

But from the literatures it is clear that the performance of the pump is based on the difference between the inlet and outlet angles and β_2 is good in between 22° and 27°.

Therefore β_2 was selected as 24°

By Stodola formula slip factor μ is given by

$$\mu = 1 - \frac{\pi \sin \beta_2}{z}$$

where z is the number of blades

$$\mu = 1 - \frac{\pi \sin 24}{3}$$

$$\mu = 0.574$$

μ is again given by $\frac{C_{u2}}{C_{u3}} = \mu$

$$C_{u2} = C_{u3} \times \mu$$

$$C_{u2} = 0.574 \times 1.76$$

$$C_{u2} = 1.01$$

again $\operatorname{Tan} \beta_2' = \dfrac{C_{m2}}{U_2 - C_{u2}}$

$$\operatorname{Tan} \beta_2' = \frac{2.98}{10.45 - 1.01}$$

$$\operatorname{Tan} \beta_2' = 0.3157$$

$$\beta_2' = \operatorname{Tan}^{-1} (0.3157)$$

$$\beta_2' = 17.5°$$

The actual outlet angle β_2' is less than the theoretical outlet angle β_3; So the angle β_2 practically selected as 24° is reasonable. It also gives an allowance for slip.

The validity of angle β_2 was again checked by Pfleiderer's semi-empirical relation.

$$C_p = \frac{0.16\ \psi' (t/1)}{\sin \left(\frac{\beta_1 + \beta_2}{2}\right)}$$

where C_p is the pfleiderer's coefficient,

ψ' is a coefficient given by

$\psi' =$ (1 to 1.2) of $(1 + \sin \beta_2)$

(t/1) is the pitch to length ratio.

β_1 and β_2 inlet and outlet angles respectively.

$\psi' =$ 1 to 1.2 of $(1 + \sin \beta_2)$

ψ' was taken as $1.1\ (1 + \sin \beta_2)$

$\psi' = 1.1\ (1 + \sin 24)$

$\psi' = 1.55$

$$C_p = \frac{0.16 \times 1.55\ (1/0.5)}{\sin \left(\frac{16 + 24}{2}\right)}$$

$$C_p = 1.45$$

The vane spacing ratio (l/t) used is 0.5 (from the graph)

Andrewkovats (1964) pointed out that the value of C_p around 1.5 is reasonable. Therefore β_2 selected is also reasonable.

3. 3. 7. Vane spacing ratio.

Pitch of the blades is given by,

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

where t is the pitch,

D is the diameter of the hub,

Z is the number of blades.

The vane spacing ratio (l/t) was assumed as 0.5 This value is in agreement with the values given by Stepanoff, (Fig.7).

$$t = \frac{\pi \times 18}{3}$$

$$t = 18.85\text{cm}$$

$$t = 19\text{cm (approximately)}$$

$$(l/t) = 0.5$$

$$l = t \times 0.5$$

$$l = 19 \times 0.5$$

$$l = 9.5\text{cm}$$

Therefore length of the blades was taken as 10cm.

3. 3. 8. Width of the blades.

Width of the blades is given by

$$w = \frac{d_2 - d_h}{2}$$

where d_2 is the impeller outer diameter,

d_h is the hub diameter,

$$w = \frac{39 - 18}{2}$$

$$w = 10.5\text{cm}$$

A clearance of 5mm was given and the width was taken as 10cm.

3. 3. 9. Design for the shaft.

The diameter of the shaft was fixed by using the Torsion formula,

$$T = \frac{\pi}{16} (f_s) d^3$$

Horse power transmitted by the shaft is

$$P = \frac{2 \pi NT}{4500}$$

where P is the horse power,

T is the torque transmitted in kg-m,

N speed in rpm,

(f_s) is the allowable shear stress in kg/cm^2.

The horse power P of the pump is 10 and speed of the pump is 700 rpm.

Therefore $T = \frac{10 \times 4500}{2\pi \times 700}$

$$T = 10.23 \text{ kg-m}$$

$$T = 1023 \text{ kg-cm}$$

The shear stress (f_s) was taken as 450 kg/cm^2

$$d^3 = \frac{16 \times 1023}{\pi \times 450}$$

$$d^3 = 11.58cm$$

$$d = 2.26cm$$

For safety the shaft selected was of 3.2cm diameter

3. 3. 10. Design for Pump Casing.

In Petti and Para usually the area of cross section of the Petti is always less than the area of cross section of the Para. Here the open space in between the impeller hub and casing (including blades) is made equal to the exit cross sectional area.

So in the design of the pump casing mainly the above cross sectional areas were taken into consideration.

The diameter of the casing at impeller is the impeller outer diameter. Impeller outer diameter was found out as 39cm.

A total length of 46cm was given for effective conversion of tangential velocity component into pressure. Then the diameter was reduced to 34.5cm for continuous uniform flow. At the impeller discharge there is loss caused by high rate of shear due to low average velocity in the casing and high velocity at impeller discharge (Stepanoff, 1967). To avoid this loss the diameter of the impeller was reduced to 34.5cm after giving a length of 46cm for effective conversion of tangential velocity component into pressure, with a small divergence.

The mechanical cross sectional area,

$$A_m = 0.0932 m^2$$

But, $$A_m = \frac{\pi d^2}{4}$$

where d is the diameter of the pipe.

$$0.0932 = \frac{\pi d^2}{4}$$

$$d^2 = \frac{4 \times 0.0932}{\pi}$$

$$d = 0.345 m$$

$$d = 34.5 cm.$$

The advantage of making the area of cross section of the casing equal to the area of cross section at the impeller is double fold. (i) By giving a velocity gradient after making the flow well established, the loss due to shear can be reduced. (ii) By making the area of cross section of outlet (exit) equal to the area of cross section at the impeller the rate of flow at these two points can be kept almost equal and thereby reducing the shock.

3. 4. Fabrication

The pump was designed and fabricated in the Agricultural Engineering Research Workshop at Mannuthy. The details are given under different sub sections.

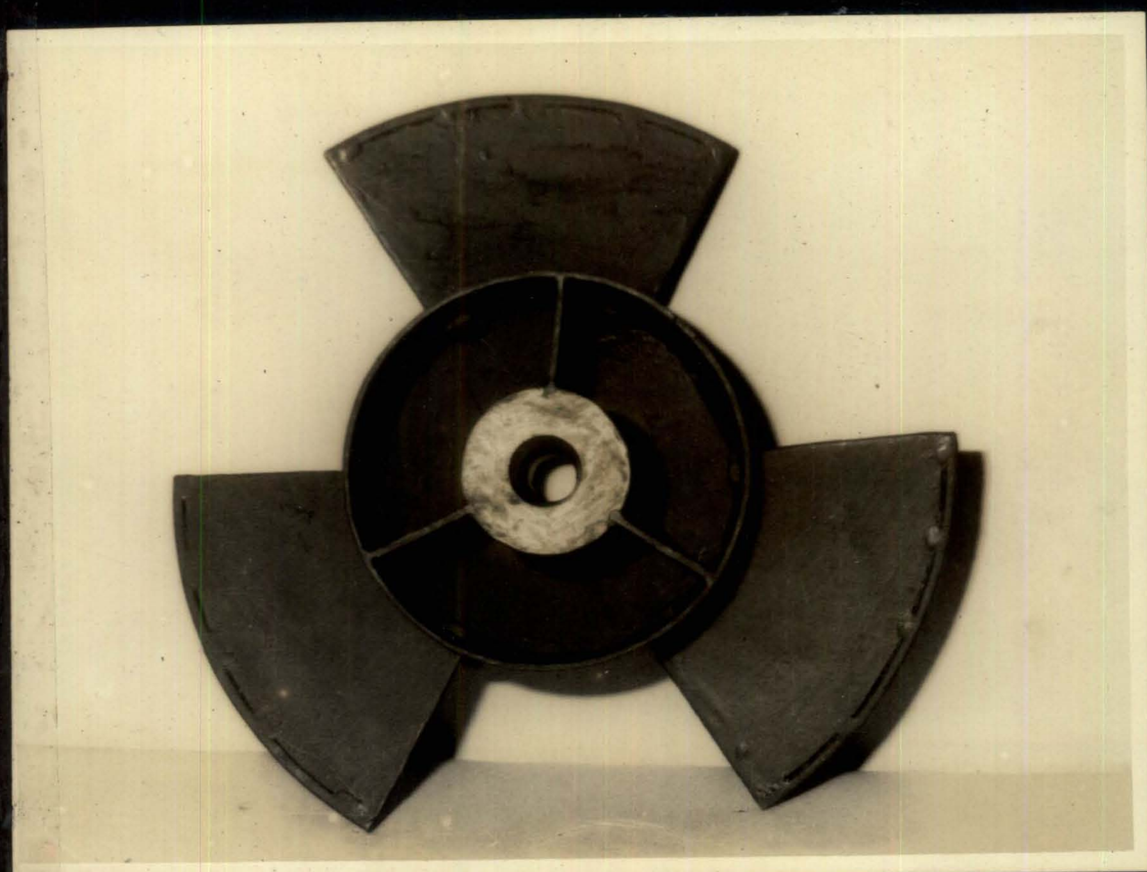
3. 4. 1. Hub.

The hub was made from 8 gauge M.S sheet. The sheet was cut in correct dimensions and bent. The edges were

DETAILS OF HUB

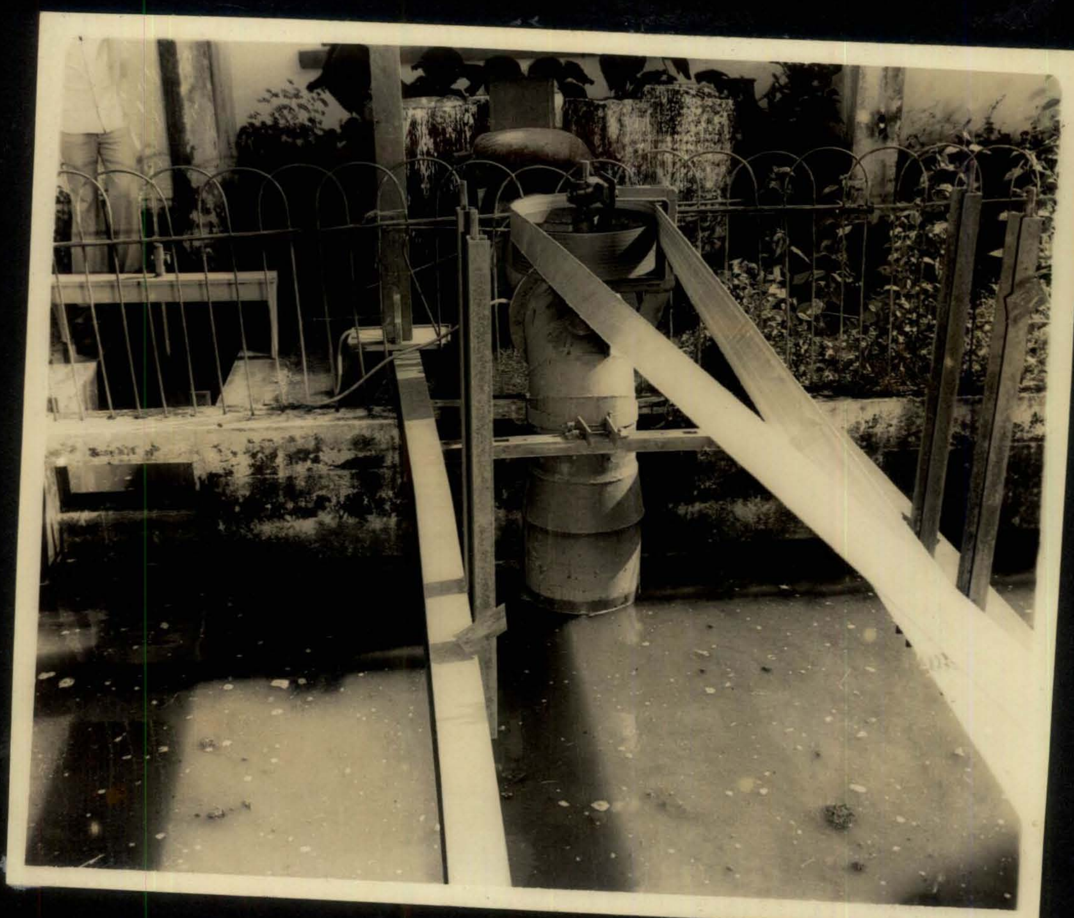
ASSEMBLY OF BLADES ON HUB

PLATE-I



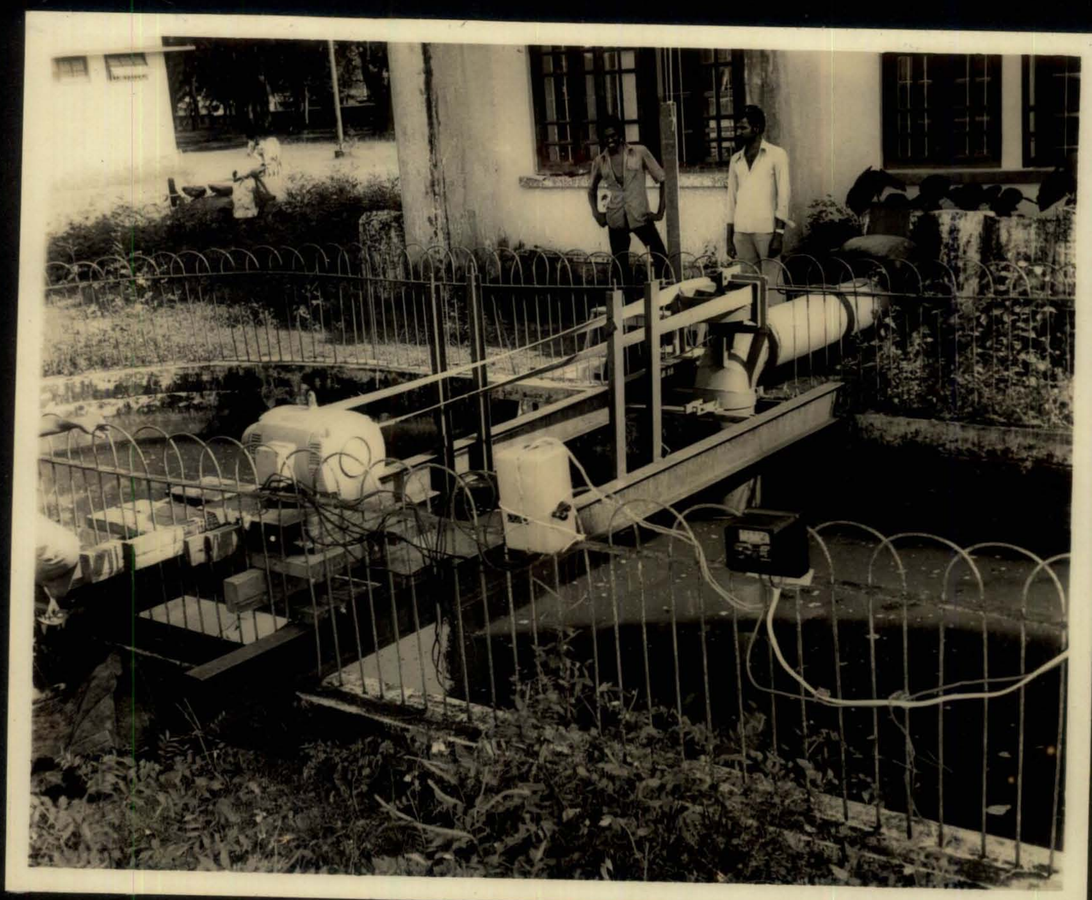
FULL VIEW OF THE CASING

PLATE-III



EXPERIMENTAL SET UP - INPUT POWER MEASUREMENT

PLATE IV



EXPERIMENTAL SET UP - MEASUREMENT OF PUMP OUTPUT welded together and the surface was finished. The bottom side was covered with M.S sheet in conical fashion. Inside the hub there is a M.S bush inside the hub through which the shaft passes. The bush and the hub covering were joined together by three M.S sheet pieces, which were welded in between the two. The bottom half of the bush hole was made 30° conical. Plate 1 shows the details of the hub.

3. 4. 2. <u>Blades.</u>

The blades were cut in correct dimensions. Then the blades were twisted from 16° to 24° in uniform variation. Twisting was done by simple blacksmithy. Blades were then welded to M.S flats, which were shaped in the correct curvature to suit on the hub. On each flats two holes were drilled in order to bolt the blades on the hub. Plate 2 shows the details of blades.

3. 4. 3. <u>Shaft.</u>

The shaft used is 32 mm M.S. The bottom of shaft was turned to 30° cone in order to suit the hub. Below that, shaft was threaded to check the impeller by check nut.

3. 4. 4. <u>Bush bearing to support the impeller.</u>

This was made from teak wood. The wooden bush was inserted in a M.S bush. This bush was fixed to the casing by welding three M.S flats of 26 mm width and 13 mm thick. Figure 9 shows the details.

3. 4. 5. <u>Casing.</u>

Casing was fabricated from 8 gauge and 12 gauge

M.S sheets. The casing upto a height of 66cm was made from 8 gauge sheet. The other parts were made from 12 gauge sheet. The sheet was cut to its size, rolled and welded. The bend of the pump casing was fabricated by drawing the development on the M.S sheet, cutting into pieces and welding together. Two outlet pipes of length 1m each were fabricated from 12 gauge sheet. Plate 3 shows the casing.

3. 4. 6. Pulley.

A pulley of 46cm diameter and 15cm width was made from 8 gauge M.S sheet and M.S flats of 52 mm width and 26 mm thick. The pulley hub was made from M.S bar. Pulleys of diameter 10cm, 13cm, and 15cm were also fabricated to suit the motor shaft.

3. 4. 7. Bearings.

Two 306 roller bearings were used. One was fixed above the pulley and other below the pulley. The bearing blocks were supported by a support made in angle iron which was joined on the bend of the pump casing by welding.

3. 4. 8. Belt.

The belt used was 4 inch width ¼ inch thick 4 ply Dunlop flat belt.

3. 4. 9. Assembly.

The pump was assembled in the Agricultural Engineering Workshop at Mannuthy. At the bend of the exit pipe, a sealing was used to avoid leakage. The impeller was fitted to the shaft and checked by a washer and nut. The total height of the pump is 2m.

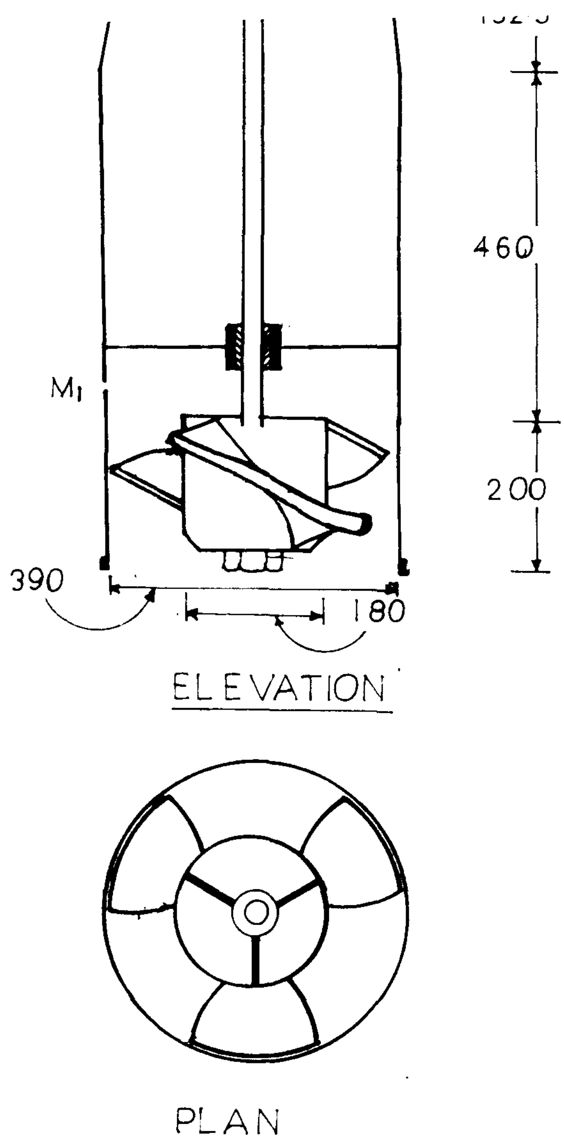
3. 5. Testing

Testing was done at Government Engineering College, Trichur. Pump was taken to the hydraulicslaboratory of the Engineering college. It was installed in a large tank (pond) attached to the hydraulics laboratory. The power unit used was a 15 HP, 3 phase induction motor. Figures4 and 5 show the details of experimental set up.

3. 5. 1. Instruments and measuring apparatus used.

The equipments used to measure the flow rate was a rectangular notch and a Hook gauge. The dynamic head was measured by water manometer. The input energy to the motor was taken by using an energy meter of energy meter constant 60. Time taken by the energy meter disc for 2 revolutions was taken by a stop watch. A tachometer was used to measure the speed of the motor and pump.

Before starting the testing, pump was run without water and noted that there was no objectionable sound. Then the tank was filled. Test was conducted for two levels of water above the impeller, one 20cm above the impeller and the other 10cm above the impeller. The accessory used to change the quantity of flow was a shutter at the exit. Total dynamic head at two points were measured using water manometers, one after the bend and the other just above the impeller. The manometer



B, B_1 BEARINGS
M_1, M_2 MANOMETER POINTS

SCALE 1:100

FIG.9. SECTION OF THE AXIAL FLOW PUMP

points are shown in Fig.9. The second manometer helped to know the measurable hydraulic loss in between the two manometer points.

Testing was performed at constant sump level by pumping the water to a channel and leading the water to the same tank. For various quantities of flow, manometer reading, height of water over notch, time for 2 revolutions of energy meter disc were noted. The difference in speed of pump from full flow to minimum flow was found as 10 rpm, which may be due to the slip of electric motor. The quantity of flow was calculated by using the formula $(2/3)\ C_d \sqrt{2g}\ L\ H^{3/2}$ where

C_d is the coefficient of discharge,

L is the length of notch,

H is the height of water over notch.

The input to the motor was calculated by the formula

$$(n/t) \times \frac{3600}{K} \times \frac{1000}{746}$$

Where n is the number of revolutions,

t is the time for n revolutions,

K energy meter constant.

A sample calculation is shown in Appendix 1.

3. 6. Cost of the pump

The cost of the pump is calculated as Rs.3275/- (Rupees three thousand two hundred and seventy five only). The details of calculation is shown in Appendix 2.

Chapter 4

RESULTS AND DISCUSSION

RESULTS AND DISCUSSION

4. 1. Test data

The pump was tested at two levels of water above the impeller; (i) 20 cm above the impeller (ii) 10 cm above the impeller. For these two cases, the head was measured at two points; one after the bend and the other just above the impeller. In both cases efficiency was found out, without considering the loss in between the two manometer points at first, and then by considering the loss. In the latter case loss due to velocity ($V^2/2g$) was also added. The tables 4.1, 4.2, 4.3, and 4.4 illustrate test data of discharge capacities at different heads.

4. 2. Operating Characteristics and results

In the first case (water level 20 cm above the impeller) it was seen from the test result that discharge varied from 165.19 lit/sec. to 39.53 lit/sec. against the heads 1 m to 2.65 m. The efficiency varied from 31.95 per cent to 15.36 per cent. (Table 4.1).

In the second case (water level 10 cm above impeller) the discharge varied from 147.46 lit/sec. to 30.31 lit/sec. against the heads 1.1 m to 2.78 m. In this case the efficiency varied from 28.69 per cent to 11.66 per cent (Table 4.3).

When the efficiency was calculated by considering the

Table 4. 1. Performance of pump without considering losses
(Water level 20cm above the impeller)

Sl.No.	Total head in m	Ht. of water over notch in cm	Time for 2 revolutions of energy meter disc in seconds	Q(discharge) in lit/sec	Water horse power QH/75	HP input 80% motor efficiency and 75% transmission efficiency	Overall efficiency e_o%
1.	2.65	11.10	10.60	39.53	1.40	9.11	15.36
2.	2.48	12.30	11.00	46.11	1.52	8.77	17.35
3.	2.25	14.60	11.20	59.63	1.78	8.62	20.73
4.	1.98	15.90	11.80	67.76	1.79	8.17	21.89
5.	1.92	18.70	12.40	86.43	2.21	7.78	28.41
6.	1.71	21.10	13.00	103.59	2.37	7.42	31.89
7.	1.41	23.90	13.60	124.88	2.35	7.09	33.07
8.	1.00	28.80	14.00	165.19	2.20	6.89	31.95

Table 4. 2. Performance of pump considering losses

(Water level 20cm above the impeller)

Sl. No.	Mano meter head m.	Ht. over notch in cm	Time for 2 revolutions of energy meter disc in sec.	Q(discharge lit/sec	$\frac{V^2}{2g}$ m	Total head in m	Water horse power QH/75	HP input 80% motor efficiency 75% transmission efficiency.	Overall efficiency e_o%
1.	2.90	11.10	10.60	39.53	0.0092	2.90	1.53	9.11	16.83
2.	2.73	12.30	11.00	46.11	0.0125	2.73	1.68	8.77	19.20
3.	2.50	14.68	11.20	59.63	0.0210	2.52	2.01	8.62	23.32
4.	2.24	15.90	11.80	67.76	0.0269	2.26	2.05	8.17	25.09
5.	2.17	18.90	12.40	86.43	0.0438	2.22	2.56	7.78	32.90
6.	1.98	21.10	13.00	103.59	0.0630	2.04	2.83	7.42	38.09
7.	1.68	23.90	13.60	124.88	0.0920	1.77	2.96	7.09	41.68
8.	1.28	28.80	14.00	165.19	0.1600	1.45	3.18	6.89	46.19

Table 4. 3. Performance of pump without considering losses

(Water level 10cm above the impeller)

Sl. No.	Total head in m	Ht. of water over notch in cm	Time for 2 revolutions of energy meter disc in seconds	Q(discharge) in lit/sec	Water horse power QH/75	HP input 80% motor efficiency and 75% transmission efficiency	Overall efficiency e_o%
1.	2.78	9.30	10.00	30.31	1.13	9.65	11.66
2.	2.63	10.30	10.20	35.33	1.24	9.46	13.11
3.	2.48	14.00	10.60	55.98	1.85	9.11	20.31
4.	2.15	16.80	10.80	73.59	2.11	8.90	23.71
5.	1.95	19.00	11.20	88.52	2.29	8.62	26.64
6.	1.74	20.95	11.60	102.50	2.38	8.32	28.61
7.	1.54	22.50	12.20	114.10	2.34	7.91	29.61
8.	1.10	26.70	12.80	147.46	2.16	7.54	28.69

Table 4. 4. Performance of pump considering losses
(Water level 10cm above the impeller)

Sl. No.	Mano meter head m.	Ht. over notch in cm	Time for 2 revolutions of energy meter disc in sec.	Q (discharge) lit/sec	$\frac{V^2}{2g}$ m	Total head in m	Water horse power QH/75	HP input 80% motor efficiency 75% transmission efficiency	Overall efficiency e_o%
1.	2.98	9.30	10.00	30.31	0.0054	2.99	1.21	9.65	12.50
2.	2.83	10.30	10.20	35.33	0.0073	2.84	1.34	9.46	14.16
3.	2.68	14.00	10.60	55.98	0.0180	2.70	2.02	9.11	22.17
4.	2.35	16.80	10.80	73.59	0.0320	2.38	2.35	8.90	26.40
5.	2.15	19.00	11.20	88.52	0.0460	2.20	2.59	8.62	30.14
6.	1.94	20.95	11.60	102.50	0.0620	2.00	2.74	8.32	32.93
7.	1.74	22.50	12.20	114.10	0.0764	1.82	2.78	7.91	35.15
8.	1.32	26.70	12.80	147.46	0.1280	1.45	2.85	7.54	37.79

losses, it varied from 16.83 to 46.19 against a head of 2.90 m to 1.45 m (Table 4.2) and 12.5 per cent to 37.79 per cent against a head of 2.99 m to 1.45m (Table 4.4), respectively.

In the original design, discharge was 250 lit/sec. against a head of 1.5 m for 10 HP. The overall efficiency assumed was 50 per cent. The maximum discharge obtained from the test result was 165.19 lit/sec. against a head of 1.45 m (Table 4.2, Sl.No.8), and the maximum working capacity was 165.19 lit/sec. against a head of 1 m. The input HP taken was 6.89 (Table 4.1, Sl.No.8). The discharge obtained at designed head (1.5 m) was 121 lit/sec. with an efficiency of 33.00 per cent (from Fig.10).

When the water level was 20 cm above the impeller the overall efficiency increased from 15.36 per cent to 33.07 per cent and then decreased (without considering losses). Similarly, when the water level was 10 cm above the impeller the overall efficiency increased from 11.55 to 29.61 per cent and then decreased. Discharge obtained at designed head (1.5 m) was 114 lit/sec. with an efficiency of 29.5 per cent (Fig.12) illustrates the result. In these two cases (two levels of water) the difference in maximum efficiencies was (33.07 - 29.61) = 3.46 per cent and the difference in the efficiencies at maximum discharge was (31.95 - 28.69) = 3.26 per cent.

From the test results it is obvious that the hydraulic loss is very high. In both cases, when hydraulic losses (measured) are added, the overall efficiencies vary widely.

4. 3. Causes of Loss and Anticipated Rectifications Carried out in the Design

The hydraulic loss is not fully measurable. The reason for this is that there are so many factors contribute to the hydraulic losses. Even the combined effect of all these factors cannot be measured accurately. Hydraulic losses are caused by (i) skin friction (ii) Eddy and separation losses due to change in direction and magnitude of the velocity of flow. Because of the latter reason the casing design was done mainly for the volumetric capacity. If the casing design would have been done only for converting the tangential component of velocity into pressure, then the flow would be disturbing the velocity distribution. If the case is as above, in the channel from the suction to the discharge point there is no single stretch flow path where either the direction of flow or the area and shape of the channel is constant. Moreover part of the channel is rotating. It is not easy to predict that whether the design of casing for volumetric capacity is efficient or not, because the head-capacity, efficiency and power input are so correlated that a change in one is followed by a change in the other two.

Losses at the impeller entrance and exit are usually called shock losses. Liquid flow in a pump tends to avoid shock by acquiring prerotation at the impeller inlet and by establishing a velocity gradient in the volute casing at the impeller discharge, thereby cushioning the shock. The nature of the hydraulic loss at the impeller entrance, when liquid approaches at a high entrance angle, is that caused by a sudden expansion after separation. At the impeller discharge the loss is mostly caused by a high rate of shear due to low average velocity in the casing and high velocity at the impeller discharge. But in this design all these factors were considered judiciously.

High specific speed pumps, particularly of the axial flow type, have input horse power curves which rise sharply towards zero capacity. In some cases the zero capacity may be twice or more that at the maximum capacity. This is a very undesirable feature because the head on propeller pumps varies with water level variations in the suction or discharge reservoirs, and hence requires an oversize motor. The test results revealed that the variation in input HP was not high. In the first case it varied only from 6.89 to 9.11 HP (Table 4.1) and in the second case it varied from 7.54 to 9.65 HP (Table 4.3). This is a commendable characteristic of the pump, because attention of designers hasbeen directed towards developing, types of propeller pumps having a lower value of HP at shut-off.

If at zero capacity the liquid in the impeller rotates at the same speed as the impeller, the input horse power is a minimum. These conditions are approached with low specific speed centrifugal impellers, where because of narrow impeller passages, the exchange of momentum between the liquid inside the impeller and that in the casing is limited. On the other hand, in mixed flow and axial flow pumps the exchange of momentum at zero capacity between the liquid in contact with the impeller vanes and liquid in the pump casing takes place most freely. As a result power is wasted in eddies and the input HP increases towards shut-off.

As the blades were made from M.S sheets by simple blacksmithy, the hydraulic loss may be very high, because the curvature and finish of the blades may not be perfect. By using jigs to produce blades the hydraulic and overall efficiencies can be increased. Directly connecting a low speed electric motor to the pump, efficiency can be increased to a noticeable extent.

During testing of the pump the quantity of flow obtained was about half of the designed quantity, because the volumetric efficiency assumed for the design was very high (90%). The overall efficiency of the pump could be increased if a low value of volumetric efficiency was assumed.

Only after redesigning and testing the pump atleast for three times the design can be recommended for commercial manufacturing. Because of lack of time redesinging was not done.

Eventhough the capacity fixed was 250 lit/sec. against a head of 1.5 m the capacity obtained was 121 lit/sec. (considering the losses when water level was 20 cm above the impeller). Stepanoff (1967) found out that leakage loss decreases rapidly with increasing specific speed. Therefore by increasing the specific speed leakage loss can be minimised.

The important design values for the pump is given in the following table.

Table 4.5. Design values.

Description	Designed values
Specific speed	250 rpm
Number of blades	3
Impeller outer diameter	39 cm
Hub diameter	18 cm
Inlet blade angle	16°
Outlet blade angle	24°
Slip factor	0.574
Speed constant	1.92
Head coefficient	0.136
Capacity constant	0.55
Capacity coefficient	0.287

Chapter 5

SUMMARY

SUMMARY

The objective of the project was to develop an axial flow pump with improved efficiency which could be economically used for dewatering and for lift irrigation purposes under low head conditions. At present, "Petti and Para" a crude from of axial flow pump is used in Kerala for dewatering agricultural fields. Petti and Para is usually fabricated by local blacksmiths using their practical experience, and there is no scientific design for the pump. Petti and Para using in Kerala is found to be less efficient. In general, axial flow pump is more efficient at low head high discharge conditions.

In the present investigation the basic requirements of the pump to be considered for the dewatering of Kuttanad and Kole lands of Trichur were studied first. Then the specific speed was found out, and designed the pump. A thorough theoretical study including fluid dynamics inside the axial flow pump was done before designing the pump. The effect of various parameters involved in the design of an axial flow pump was carefully studied.

After designing and fixing the dimensions, the pump was fabricated in the Kerala Agricultural University Research Workshop at Mannuthy. The blades were made from 8 gauge M.S. Sheet by simple blacksmithy. The

inlet and outlet angles of the blades selected were 16° and 24° respectively. The casing was also fabricated from M.S. sheets. The kinematic specific speed selected was 260 rpm, and the hub diameter and outer diameter of the impeller were fixed as 18 cm and 39 cm respectively. All the designed dimensions were compared with the values given by the pioneers in pump design and found that the designed values are in agreement with their designs.

Because of the lack of facility in the research work shop, the pump was tested in the hydraulics laboratory at Government Engineering College, Trichur. The power unit used was a 15 HP electric motor. The input energy to the motor was measured by using an energy meter, of energy meter constant 60. The accessories used to measure discharge and head were a rectangular notch with hook gauge and water manometers respectively.

During testing two manometers were used, one was fixed just above the impeller and the other was fixed after the bend. This was done to note the measurable hydraulic loss in between the two manometer points. Testing was carried out at two levels of water above the impeller, one 20 cm above the impeller and the other 10 cm above the impeller. For the above two conditions at designed head (1.5 m) the efficiencies obtained were

33 per cent and 29.5 per cent respectively at discharges of 121 lit/sec. and 114 lit/sec. The maximum efficiencies obtained at these two water levels were 33.07 per cent and 29.61 per cent against heads of 1.41 m and 1.54 m respectively at discharges of 124.88 lit/sec. and 114.1 lit/sec. The maximum working capacity was 165.19 lit/sec. against a head of 1 m with an efficiency of 31.95 per cent.

Comparing with the performance of Petti and Para and the Axial flow pump designed and developed by the Department of Agricultural Engineering, College of Technology Pantnagar, this design is satisfactory.

Cavitation is an important factor in the operation of propeller pumps. But in this design cavitation phenomenon was not considered. Mo[illegible] the design by considering the effect of cavitation, the efficiency can be further increased.

Conclusion

Usually the efficiency of Petti and Para is around 25 per cent. The maximum efficiency obtained in the newly designed axial flow pump is 33.07 per cent. Again comparing with the performance of the three bladed high capacity pump designed by the Department of Agricultural Engineering, College of Technology Pantnagar (1982) this design is more perfect. The following table makes it clear.

Table 5. 1. Comparison of performance

Pump	Head Variation	Discharge Variation	Efficiency Variation
Pantnagar Pump	1.8 m to 2.8 m	106 lit/sec. to 130 lit/sec.	27% to 29%
Kerala Agricultural University Pump	1 m to 2.65 m	39.53 lit/sec. to 165.19 lit/sec.	15.36% to 33.07%

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APPENDICES

APPENDIX - I

SAMPLE CALCULATION

Data taken is 8th set readings in Table 4.1.

Height of water over notch is 28.8 cm.

Discharge, $Q = (2/3)\ C_d\ L\ \sqrt{2g}\ \ H^{3/2}$

where L is the length of notch $= 48.26$ cm

$= 0.4826$ m

H is the height over notch $= 0.288$ m

C_d is the coefficient of discharge $= 0.75$ (assumed)

$Q = (2/3) \times 0.75 \times (.4826) \sqrt{2 \times 9.81}\ (0.288)^{3/2}$

$Q = 0.16519\ m^3/sec.$

$Q = 165.19$ lit/sec.

Total Head $=$ 1m

Water horse power $= \frac{QH}{75}$

$= \frac{165.19 \quad \times 1}{75}$

$= 2.20$

Input Horsepower

The input to the motor

$= (\frac{n}{t})\ (\frac{3600}{K})\ \ (\frac{1000}{746})$

where n is the number of revolutions of the energy meter disc $= 2$

t is the time taken for n revolutions in seconds $= 14$ sec.

K is the energy meter constant $= 60$.

Input HP to motor

$$= \left(\frac{2}{14}\right)\left(\frac{3600}{60}\right)\left(\frac{1000}{746}\right)$$

$$= 11.49$$

The efficiency of the 15 HP motor was taken as (assumed) 80 per cent and the transmission efficiency was taken as 75 per cent

So the power input to the pump

$$= 11.49 \times 0.8 \times 0.75$$

$$= 6.89 \text{ HP}$$

$$\text{Overall efficiency} = \frac{\text{Water horsepower}}{\text{Power input}}$$

$$e_o = \frac{2.20}{6.89}$$

$$e_o = 31.95\%$$

APPENDIX - II

COST OF PUMP

			Rs.	Ps.
1.	Cost of impeller material	=	76	00
2.	Cost of casing	=	737	00
3.	Cost of bush bearing (including teak bush) and its supports	=	32	00
4.	Cost of shaft	=	120	00
5.	Cost of sealing	=	5	00
6.	Cost of two roller bearings and bearing blocks	=	230	00
7.	Cost of support for bearing blocks	=	50	00
8.	Cost of pulleys	=	300	00
9.	Cost of belt	=	370	00
10.	Cost of checknut	=	5	00
11.	Cost of packing materials, nuts, bolts etc.	=	100	00
12.	Cost of welding rods	=	250	00
	Labour cost	=	1000	00
	Total cost	=	3275	00

DESIGN AND DEVELOPMENT OF A PROPELLER PUMP

171147

By

K. SASI

ABSTRACT OF THE THESIS

Submitted in partial fulfilment of the requirement for the degree

MASTER OF SCIENCE IN AGRICULTURAL ENGINEERING

Faculty of Agriculture
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Department of Agricultural Engineering
COLLEGE OF HORTICULTURE
Vellanikkara-Trichur

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ABSTRACT

"Petti and Para" a crude form of Axial flow pump is used in Kerala for dewatering agricultural fields. It is being fabricated by local blacksmiths and there is no scientific design for the pump. For low head high discharge conditions Petti and Para is very effective, but less efficient because of faulty construction.

The broad objective of the project was to develop an axial flow pump with improved efficiency which could be economically used for dewatering and for lift irrigation purposes under low head conditions.

In this investigation, a good amount of attention was given to the fluid dynamics inside the axial flow pump and theoretically studied the combined effect of various parameters involved in the design of an axial flow pump. Then the parameters were compared with the approximate values given by the various researchers in the field. The requirement of the pump to be designed, at field conditions, were studied thoroughly and an one dimensional design was done.

After designing and fixing the dimensions, the pump was fabricated in the Kerala Agricultural University Research Workshop at Mannuthy. Because of the lack of facilities in the Research Workshop, it was tested in the hydraulics laboratory at the Government Engineering

College, Trichur. The power unit used was a 15 HP electric motor. The accessories used to measure discharge and head were a rectangular notch with Hookgauge and water manometer respectively. Testing was done with two levels of water above the impeller, one 20 cm above the impeller and the other 10 cm above the impeller. For the above two conditions, at designed head (1.5m) the maximum efficiencies obtained were 33 per-cent and 29.5 per cent at discharge 121 lit/sec. and 114 lit/sec. respectively. The maximum working capacity was 165.19 lit/sec. against a head of 1 m with an efficiency of 31.95 per cent (20 cm above the impeller)

In the three bladed pump, the blades were fixed to the hub by welding it to the suitably shaped (curved) M.S flats and then bolting the curved M.S. flats to the hub. The blades were twisted from 16° to 24° in a uniform variation by simple blacksmithy. The efficiency of the pump can be increased by using perfectly curved blades, which reduces eddies and skin friction. Comparing with the existing Petti and Para made by local blacksmiths, it is found that the pump is cheap as well as efficient.