EVALUATION OF THE CHARACTERISTICS OF PETTI AND PARA' (AXIAL FLOW 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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1988

DECLARATION

I hereby declare that this thesis entitled "Evaluation of the characteristics of 'Petti and Para' (Axial flow 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.

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CERTIFICATE

Certified that this thesis, entitled "Evaluation of the characteristics of 'Petti and Para' (Axial flow pump)" is a record of research work done independently by Sri.Jose Abraham 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.Jose Abraham, a candidate for the degree of Master of Science in Agricultural Engineering with major in Soil and Water Engineering, agree that the thesis entitled "Evaluation of the characteristics of 'Petti and Para' (Axial flow pump)" may be submitted by Sri.Jose Abraham in partial fulfilment of the requirements for the degree.

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ACKNOWLEDGEMENTS

The author wishes to place on record his profound gratitude and indebtedness to,

Prof. K. John Thomas, Professor and Head, Department of Irrigation and Drainage, Chairman of the Advisory Committee, for the valuable guidance and encouragement during the course of this investigation and preparation of the thesis,

Prof. T.P. George, Dean i/c., Faculty of Agricultural Engineering for his constructive suggestions, encouragement, and help in the preparation of the manuscript,

Prof. T.F. Kuriakose, Co-ordinator of Rice, Agricultural Research Station, Mannuthy and Sri.E.K. Mathew, Assistant Professor, Agricultural Drainage Research Centre, Karumadi, members of advisory committee, for their help during the investigation,

Sri.M.R. Sankaranarayanan and Smt.K.P. Visalakshi, Assistant Professors, Faculty of Agricultural Engineering for their unfailing help,

Prof. U. Muhammed Kunju, Professor, Research Co-ordinator and former Head of Regional Agricultural Research Station, Kumarakom; Dr.R.R. Nair, Associate Director, Regional Agricultural Research Station, Kumarakom and Dr.C.A. Joseph, Head, Rice Research Station, Moncombu for their timely help, the technicians involved in the fabrication and testing works,

Kerala Agricultural University for providing a part time job and

to all his friends and relatives for their love and encouragement through out the study.

JOSE ABRAHAM

LIST OF TABLES

LIST OF FIGURES

LIST OF PLATES

ABBREVIATIONS

ASME	American Society of Mechanical Engineers
cm	Centimetre(s)
Ed.	Edition
En.	Energy
et al	and other people
Fig.	Figure
gpm	gallons per minute
ha	hectare(s)
HP, hp	Horse power
h	hour(s)
IRRI	International Rice Research Institute
kg	kilogram(s)
KW	Kilo Watt(s)
l/s	litres per second
m	metre(s)
M.S.	Mild Steel
No.	Number
pp.	pages
Proc.	Proceedings
rpm	revolutions per minute
s	Second(s)
Sl	Serial
/	per
%	per cent
°	degree

NOMENCLATURE

A_m	Mechanical cross section (area)
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_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_o	Overall efficiency
f_s	Allowable shear stress
g	Acceleration due to gravity
H	Total head, height over notch
H_e	Euler's head
H_h	Head at hub
H_o	Head at periphery
H_1	Inlet head

H_{th}	Theoretical head
K_u	Speed constant
K_m	Capacity constant or coefficient of flow velocity
K	Energy meter constant
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'	Total quantity of water including leakage
R, r	Radius
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 entry
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
ω	Angular velocity

Subscripts:

1 - Conditions at inle

2 - Conditions at outlet

Introduction

INTRODUCTION

Adequate supply of water is one of the basic inputs to agriculture for stable and sustained production. But unlike in other rice growing areas in the state or in the country paddy cultivation in Kuttanad and Kole lands is made possible only after large scale dewatering operations. Both Kuttanad and Kole lands are lying below mean sea level. These rice tracts are known as punja lands.

The area of Kuttanad punja land is 52,000 hectares and that of Kole lands is 7900 hectares. It is observed that continuous pumping for about 20 days and intermittent pumping for about 3 months is required for adequate drainage to avoid crop losses in these areas. It is estimated that on an average one mater of water from these areas has to be pumped out to make them fit for cultivation.

The popular pumping equipment used in Kuttanad and Kole land for dewatering is known as 'Petti and Para' which is a local adaptation of the class of pump called axial flow or propeller pumps. 'Petti and Para' was orginally developed by a British Engineer George Brendan soon after the first world war (1914-1918), and was installed in one of the 'Padasekharams' of Kumarakom Village. It was first operated by kerosene engines imported from England and was shortly replaced by crude oil and diesel engines. Virtually all these prime movers were replaced by electric motors in 1950's which have lesser operating cost and greater convenience of operation (Abraham, 1980). Inspite of the fact that these electric motors have higher rated horse power values, the discharge output has not increased in proportion to the increase in the input power.

Although 'Petti and Para' was introduced in the state more than 65 years ago, this has not been subjected to serious scientific evaluation except probably by Kerala Agricultural University. The efficiency of a 30 hp 'Petti and Para', reported by the Kerala Agricultural University drainage research centre at Karumadi is only 15-20% while a well designed axial flow pump can operate at 70-75% efficiency (IRRI, 1979). Hence a detailed scientific study will be of considerable importance in reducing the power requirements and to increase the output per unit power input.

The advantage of scientifically designed axial flow pumps over the conventional 'Petti and Para' is that the power intake is less and this reduces the wastage of energy. Even by increasing the efficiency by a small percentage a very large amount of power can be saved for the punja crop alone. For additional crop the requirement of energy is much more than that of the punja crop. Hence the total saving of energy will be several million kWh.

The project was under taken with the following specific objectives.

1) To find out energy requirement of drainage pumping in Kuttanad area.
2) To find out the efficiency of 'Petti and Para' already in operation in Kuttanad area.
3) To find out reasons for low efficiency.
4) To suggest improvements on pump design so as to increase efficiency.

These objectives can be achieved by conducting field survey, field pumping tests, and by testing scientifically developed propeller pumps developed specifically for Kuttanad conditions, and applying the positive results to the existing pump design.

Review of Literature

REVIEW OF LITERATURE

'Petti and Para' a crude form of axial flow pump used for dewatering agricultural fields of Kerala, is being fabricated by local blacksmiths. Very little attention has been given by researchers to study the characteristics of this crude form of pump. Therefore an attempt has been made to evaluate the characteristics of 'Petti and Para' and to suggest modifications accordingly. Literature relative to this has been cited here.

According to Stepanoff (1957), Ingersoll-Rand Company developed a 14 inch elbow type axial flow pump running at an rpm of 690. When the pump was tested without vaned diffuser the discharge varied from 2000 gpm to 5000 gpm under a head of 3.8 m to 8 m, with a variation in efficiency from 48 to 77 per cent and bhp from 6 to 9.1 hp. When the pump was tested with a vaned diffuser, the discharge varied from 2000 gpm to 5000 gpm, under a head of 4 m to 10 m with an efficiency of 50.2 to 80.5 per cent and the bhp varied from 6.2 to 9.8 hp.

Sterling pump corporation manufactures propeller pumps with following performance parameters (Finch, 1967).

Table-1.

Pump size (Inch)	Dynamic head (feet)	Capacity (US gpm)	Power unit	
			hp	rpm
8	2 to 16	1000 to 2000	3 to 7.5	1760
10	2 to 11	1000 to 3000	3 to 7.5	1160
10	6 to 25	2000 to 4000	7.5 to 20	1760
14	4 to 12	3000 to 6000	7.5 to 15	875
14	8 to 20	3000 to 8000	15 to 40	1160
20	8 to 22	8000 to 16000	40 to 75	875
24	5 to 15	10000 to 20000	25 to 60	580
24	8 to 22	12000 to 24000	50 to 100	705

International Rice Research Institute, Manila (1979) designed and developed a portable and cheap axial flow irrigation pump which may be coupled to a 5 hp engine or on a power tiller. The pump consists of an axial flow impeller, rotating inside a 15 cm diameter casing and is capable of pumping water at a rate of 1500 to 3000 litre/minute at heads of 1 m to 4 m. The pump inlet was flared to 30° to reduce suction losses. Duffusion vanes were provided to increase the efficiency of the pumping unit. Pumping test revealed that the efficiency of the unit was 69.1 per cent at a discharge capacity of 2690 lpm against a head of 2.5 m while operating at a speed of 2890 rpm.

The Department of Agricultural Engineering, College of Technology, Pantnagar (1982) developed a propeller pump for the range of discharge of 30 to 65 l/s at a head of 1 m to 3 m. The pump consists of 22 cm diameter propeller and 45 cm long diffuser tapering from 22 cm to 30 cm in diameter. The efficiency of the pump varied from 65 to 50 per cent at a discharge rate of 45 to 65 l/s against a head of 1 to 2 m at 1440 rpm. The power unit was a 5 hp motor.

Later, the work was further extended to develop a pump of higher discharge. A propeller with 3 vanes was designed. The hub diameter was 13.5 cm. The outer diameter of the propeller was 30 cm. A diffuser having 7 vanes was designed and fabricated from aluminium alloy. The diffuser was 45 cm long with a tapered diameter of 30 cm on the lower end to 37.5 cm on the other end. A uniform diameter of 37.5 cm was provided above the tapered portion. The length of the casing was 3 meter. The pump was tested at 1440 rpm with a 15 hp electric motor with belt and pulley arrangement. The pumping test revealed that the pump had a capacity of 87 to 122 l/s at a head of 2 to 3m, and the efficiency varied from 29 to 32 per cent.

Sasi (1984) designed and developed a propeller pump with the following design values.

Table-2.

Description	Design values
Specific speed (rpm)	260
Number of blades	3
Impeller outer diameter (cm)	39
Hub diameter (cm)	18
Inlet blade angle (degree)	16
Outlet blade angle (degree)	24
Slip factor	0.574
Speed constant	1.92
Head coefficient	0.136
Capacity constant	0.55
Capacity coefficient	0.287

The pump was tested at two levels of water 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 efficiency obtained were 33 per cent and 29.5 per cent respectively at discharges of 121 l/s and 114 l/s. The maximum efficiency 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 discharge of 124.88 l/s and 114.1 l/s. The maximum working capacity was 165.19 l/s against a head of 1 m with an efficiency of 31.95 per cent. The power unit was a 15 hp electric motor.

The U.S.S.R. industry manufactures axial flow pumps of types O and O 11 (Cherhasshy, 1985). In the type O pumps the vanes were rigidly fixed on the hub, while the type O 11 pumpswere equipped with adjustable vanes. For impeller diameter ranging from 295 to 1850 mm, the performance parameters of type O and O 11 pumps were as follows:

Rotative speed, rpm	960 to 2,100
Capacity, m^3/h	4,450 to 54,700
Head, m	1.9 to 20.9
Power, KW	44 to 3,000
Efficiency, %	81 to 86

2.1. Principle behind the operation of a 'Petti and Para'

2.1.1. Velocity triangles

The study of the several component velocities of flow through an impeller can be carried out graphically by means of velocity vectors Fig. 1. In the design of a pump attention is focused on the entrance and discharge part of the impeller vanes, and the velocity triangles are called entrance and discharge triangles. The relative velocity of flow is considered relative to the impeller. The absolute velocity of flow is taken with respect to the pump casing and is always equal to the vectorial sum of the relative velocity and the peripheral velocity of the impeller.

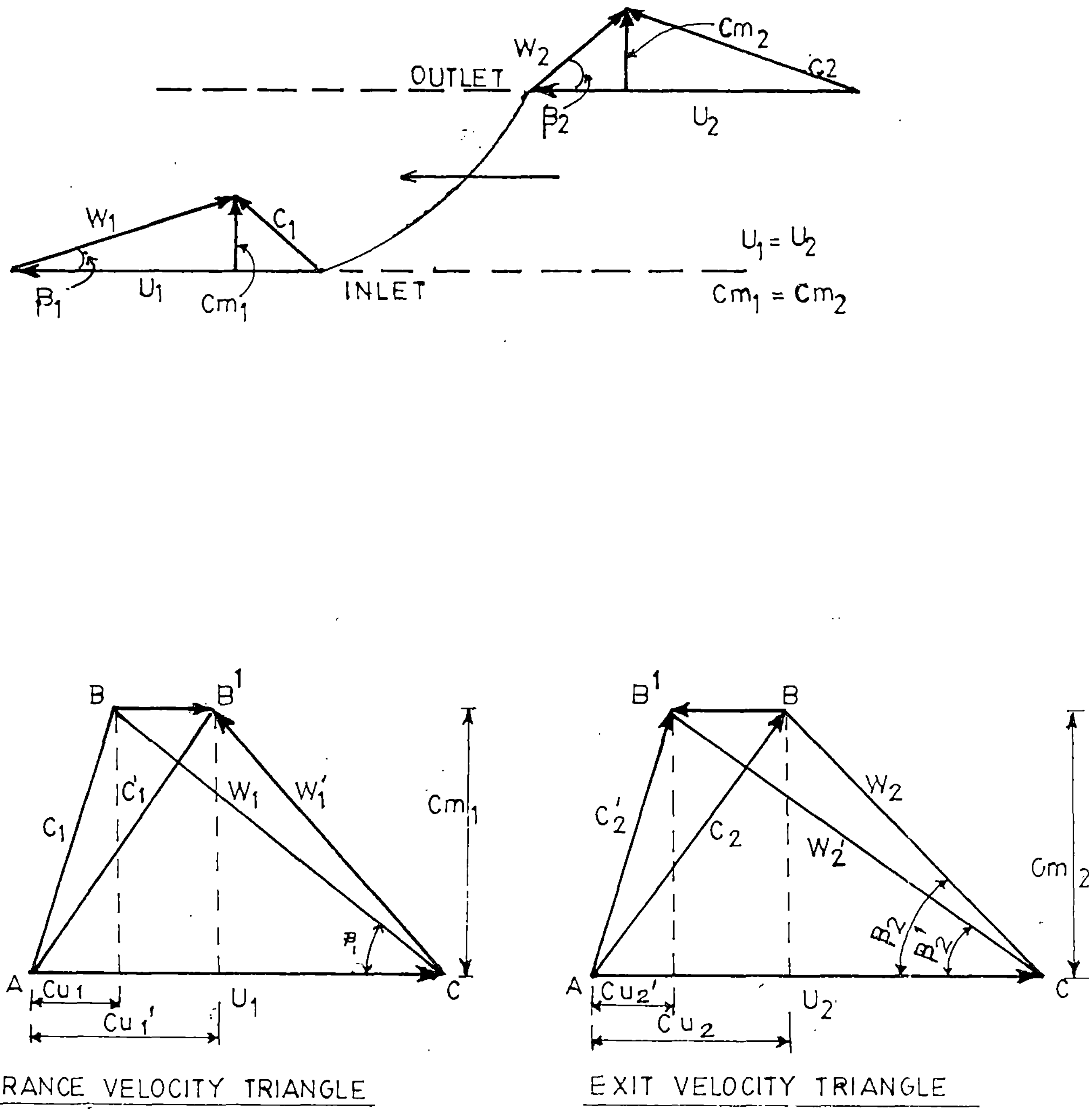


FIG:1 PROPELLER PUMP VELOCITY DIAGRAMS

2.1.2. Theoretical head

An expression for the theoretical head of a pump can be derived by applying principle of angular momentum to the mass of liquid going through the impeller passage. This principle states that the time rate of change of angular momentum of a body with respect to the axis of rotation is equal to the torque of the resultant force on the body with respect to the same axis. The change of moment of momentum is equal to the moment of all external forces applied to the liquid. Let 'T' denote the moment of external forces, dm the mass of a small element of liquid, dt - the time interval, c - absolute velocity of flow, u - the peripheral velocity of impeller and w - relative velocity of flow, then

$$T = \frac{dm}{dt}\,(r_2c_2\cos\alpha_2 - r_1c_1\cos\alpha_1)$$

This equation is often stated to be valid only for an infinite number of blades but the view is not correct (O'brien *et al.*, 1936).

The term $\frac{dm}{dt}$, when applied to all impeller channels, represents the constant time rate of mass flow through the impeller which is Qr/g.

$$\text{ie.}\quad T = \frac{Qr}{g}\,(r_2c_2\cos\alpha_2 - r_1c_1\cos\alpha_1)$$

Multiplying both sides of the equation by angular velocity of the impeller gives the power input 'P' applied to the liquid by the impeller vanes. Also from the velocity triangles $U_2 = wr_2$, $C_2 \cos\alpha_2 = Cu_2$, $U_1 = wr$, and $C_1 \cos\alpha_1 = Cu$.

$$P = \frac{Qr}{g}\left(U_2\ Cu_2 - U_1\ Cu_1\right)$$

Assuming that there is no loss of head between the impeller and the point where the total dynamic head is measured, this power is available as the pump output of an idealised pump. Eliminating Qr, gives Euler's equation.

$$Hi = \frac{U_2\ Cu_2 - U_1\ Cu_1}{g}$$

Euler's equation gives the theoretical head, measured disregarding all hydraulic losses between the points where the actual total dynamic head of a pump is measured. If the liquid enters the impeller without a tangential component, or if $Cu_1 = 0$, ie. axially for a axial flow pump, Euler's equation reduces to

$$Hi = \frac{U_2\ Cu_2}{g}$$

By geometric substitution from the velocity triangles

$$W_2^{\ 2} = C_2^{\ 2} + U_2^{\ 2} - 2\ U_2 C_2 \cos\alpha_2$$

$$W_1^{\ 2} = C_1^{\ 2} + U_1^{\ 2} - 2\ U_1 C_1 \cos\alpha_1$$

$$\text{ie. } Hi = \frac{C_2^{\ 2} - C_1^{\ 2}}{2g} + \frac{U_2^{\ 2} - U_1^{\ 2}}{2g} + \frac{W_1^{\ 2} - W_2^{\ 2}}{2g}$$

The first term represents a gain of the 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. It is futile to attach any physical meaning to the second and third terms 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, as 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, or W_2, and actually the third term is subtractive term. Euler's velocity triangles can be utilised for graphical determination of the impeller vane shape of axial flow impellers.

2.1.3. Vortex theory of Euler's head

Flow through the impeller can be considered as consisting of two components; a circular motion around the axis as a result of the impelling action of the vanes, and through flow or meridianal flow caused by the energy gradiant drop. The circular component of flow forms a vortex motion. The type of vortex depends on the velocity and pressure distribution. In an axial flow pump, 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 $U_2 = U_1$, the Euler's equation reduces to

$$He = \frac{Cu_2^{\,2} - Cu_1^{\,2}}{2\ g} + \frac{Wu_1^{\,2} - Wu_2^{\,2}}{2\ g}$$

Again assuming first that the liquid approaches the impeller without prerotation ($Cu_1 = 0$ and $Wu_1 = U_1$) the above equations reduces to

$$He = \frac{U_2^{\,2}}{2\ g} + \frac{C_2^{\,2}}{2\ g} - \frac{Wu_2^{\,2}}{2\ g}$$

But $Cu_2 = U_2 - Wu_2$

$$He = \frac{U_2^{\,2}}{g} - \frac{U_2 Wu^2}{g}$$

In this equation only tangential velocities appear, indicating that all heads are produced by vortex motion in planes normal to the axis of rotation, and flow through the impeller is caused by the energy gradient drop $\frac{U_2 Wu_2}{g}$.

The head distribution along the radius is shown in the Fig. 2; 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 (Wu_2). AC is the energy

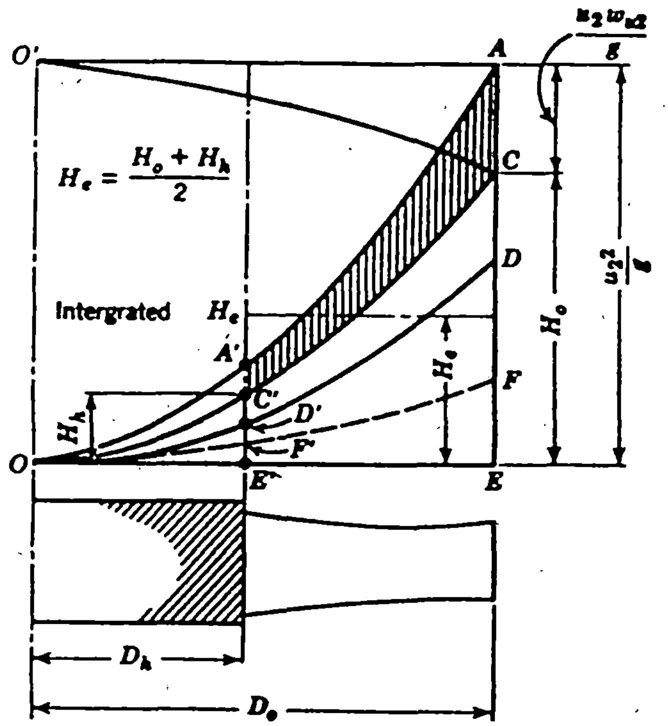


FIG. 2 Euler's head, axial flow impeller.
(Stepanoff, 1957)

gradient drop at the periphery. Ordinates between curves AA' and CC' represent the gradient drop for different radii. For a normal design both Wu_2 and U_2 vary directly on the radius. Therefore, the energy gradient drop $\frac{U_2 Wu_2}{g}$ varies directly as the square of the radius (Curve O'C), and the curve of heads OC is a square parabola. This is a characteristic of a forced vortex when all particles rotate with the same angular velocity.

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 hydraulic integration of the head over the whole impeller area takes place in the discharge casing, where the tangential component of the absolute velocity is converted into pressure and the pressure is equalised over the whole area of the discharge pipe.

In an efficient casing this equalisation of pressure occurs without mixing of streamlines and the pressure equalisation takes place by conduction. The integrated head of the impeller is equal to the average of the head at the hub (Hu) and the head at the periphery (Ho).

$$He = \frac{Hu + Ho}{2}$$

If the liquid approaches the impeller with prerotation, Euler's head for an axial flow impeller is given as

$$He = \frac{U_2 \; Cu_2 \; - \; U_1 \; Cu_1}{g}$$

The substantive term is of the same appearance as the first term and represents a square parabola of suction heads at different radii as shown by curve FF' (Fig.2). The net Euler's head at different radii is represented by ordinates between the curves CC' and FF'. The curve of the net Euler's head will remain a square parabola.

2.1.4. Operating conditions peculiar to long vanes

Usually impellers of a 'Petti and Para' are having long vanes. The elements of an axial flow machine vane at different radii are rotating at different speeds. Because of this, the head produced by a vane of constant width, invariable entry and discharge angles changes along the vane length. This phenomenon makes liquid particles shift in a radial direction in the flow passage of impellers and down stream of the vane and decreases the efficiency of the machine. The radial shift effect is particularly pronounced in a machine with relatively long vanes. Therefore a propeller pump with long vanes are normally designed on the basis that there should not be any radial liquid flow. The condition to be satisfied to avoid radial liquid flow is given as r Cu = Constant (Cherhasshy, 1985). Radial flow can be avoided only if circulation is constant over the vane length. In this case each particle of the flow is moving over a cylindrical surface of appropriate radius. When fulfilled, This condition ensures a considerable gain in the efficiency of axial flow pumps.

2.2. Effect of experimental design factors on the performance of a propeller pump.

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 hub ratio, number of vanes, vane thickness and pump casing.

2.2.1. 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. This ratio is established experimentally. Higher specific speed pumps 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. Addison (1956) proposed that for good performance the hub ratio should be between 0.4 and 0.55. Fig. 3 gives

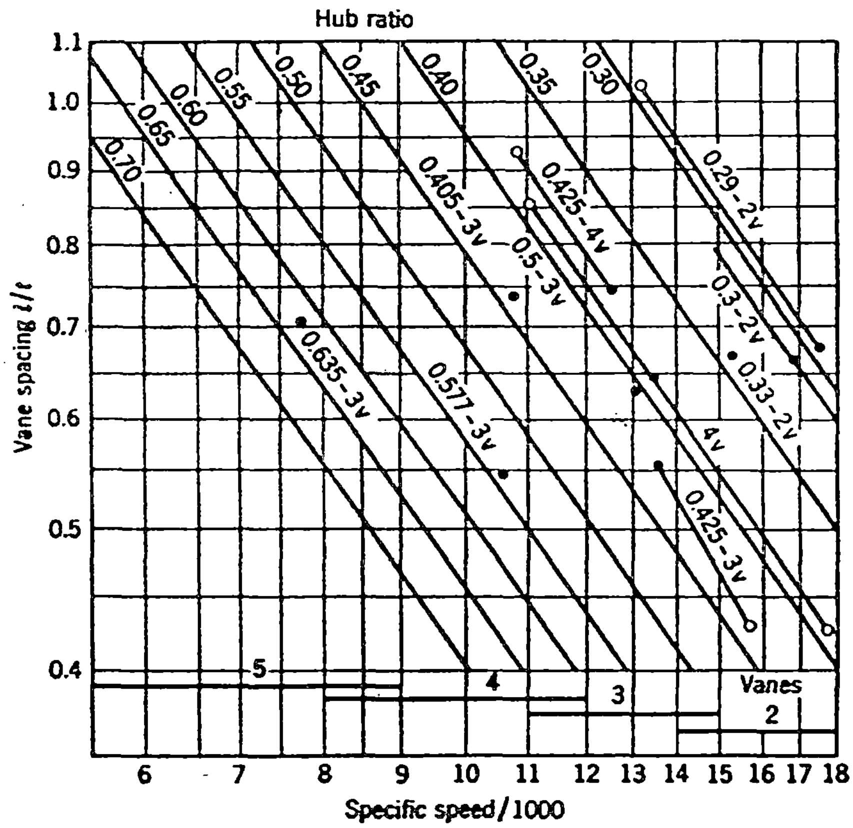


FIG. 3 Hub ratio, number of vanes, and l/t ratio for axial flow pumps. (Stepanoff, 1957)

hub ratios for various specific speeds compiled 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. The hub ratio depends also upon the selection of the capacity coefficient, which in turn depends upon the impeller discharge angle and specific speed.

2.2.2. Chord spacing ratio

The chord spacing ratio is an 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 is less than unity. The chord spacing ratio varies along the radius, increasing toward the hub. This increase in chord spacing ratio at the hub is desirable for mechanical reasons. Figure 3 shows values of the chord spacing ratio for the section at the periphery of the impeller for various hub ratios. The value of chord spacing ratio at the hub is 1.25 to 1.30 times that at the outside diameter of the impeller, depending on the hub ratios.

2.2.3. Number of vanes

Kaplan (1931) after extensive tests with hydraulic turbines, found that for a given wetted area of the vane the number of vanes should be a minimum. Schmidt (1928) also

confirmed the above, and showed that a two vane impeller was most efficient with a projected vane area of about 63 per cent. With heavy vanes and low chord angle, the maximum number of vanes was almost fixed since adding vanes would restrict the free area of the flow. Schlimback (1935) in his experiment with vanes of 3 to 5 numbers observed that:
(1) the capacities at normal and zero head were the same for several impellers. This was determined mainly by the vane entrance angle. (2) the heads increased with the number of vanes. This was entirely due to the increase in the chord spacing ratio.

2.2.4. Vane curvature and vane setting

Schlimback (1935) in his experiment with a four vane impeller with different vane settings, or by keeping the vane curvature $\beta_2 - \beta_1$ or vane camber equal, and altering the discharge angle β_2 and the inlet angle β_1 showed that the head produced was the same for all vane settings and thus was a function of the vane curvature $(\beta_2 - \beta_1)$ alone. This means that although the tangential component at the impeller discharge (Cu_2) is higher at higher values of β_2, the tangential component at inlet (Cu_1) is increased by approximately the same amount. The peripheral velocity being the same at inlet and outlet no change in heads results. Schmidt (1928) and Schlimback (1935) observed that capacity, varies approximately directly with the inlet pitch.

2.2.5. Vane thickness

Eckert (1944) found that with two impellers, one with airfoil vanes well streamlines 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 were identical. He 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 five points lower in efficiency. Part of the efficiency reduction was caused by the greater relative roughness of the cast iron vane as compared to the polished alloy vane. Excessive vane thickness resulted in separation and noice with high pressure high speed impellors. The advantages of airfoil sections lie in the fact that they permit the desired mechanical strength with a minimum sacrifice of efficiency.

2.2.6. Pump casing

The purpose of diffuser casing of propeller pump is to convert the tangential component of the absolute velocity leaving the impeller into pressure. This was achieved 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 was reduced by increasing the diffuser diameter at the discharge. A small divergence angle of 8° was essential for an effective conversion (Morelli *et al.*, 1953).

Materials and Methods

MATERIALS AND METHODS

A 'Petti and Para' consists of a cylindrical wooden drum ('Para') with a horizontal rectangular outlet ('Petti'). The two long sides of the 'Petti' are made up of single piece wooden planks which takes up the whole self load and dynamic load of the unit. The rear end of the box is completely closed and is provided with an inspection opening with water tight sliding shutter to achieve easy maintenance and clearing of impeller. The discharge end of the 'Petti' was provided with a one way valve fixed on hinges with suitable inclination, to avoid entrance of water from the outer discharge channel when the pump is not working. The top and bottom sides of the box is tongued and grooved to make it water tight with wooden planks.

The 'Para' is made up 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 rings to join the bottom 'Para' with the top 'Para' and top 'Para' with 'Petti'. A metal ring is provided where the 'Para' is connected to the 'Petti'. The impeller is housed inside the top 'Para'. The top 'Para' is provided with a protective inside cover made of sheet metal, where the impeller works.

The impeller consists of a cast iron hub to which suitably shaped vanes are fixed. The shaft is made of mild steel. A suitable cast iron 'stand tube' was provided on which a heavy duty thrust bearing works. A collar is rigidly bolted to the shaft and it rests on the thrust bearing. Two gun metal bush bearings are provided inside the 'stand tube'; one at the top and one at the bottom. Two vertical, and horizontal brackets along with angle iron stiffners holds the rotating elements in position. The pulley is fixed in between the two horizontal brackets. Two ball bearings fixed in the horizontal brackets, hold the shaft in position. The power transmission is through quarter turn flat belts. The different parts of 'Petti and Para' are shown in Fig. 4, 5 and Plate 1, 2 and 3.

The bottom 'Para' has a flap type foot valve. The impeller is situated in the middle of the top 'Para'. Usually the discharge end of the 'Petti' is kept submerged, so that priming of the pump is achieved by simply opening the outlet flap valve. Unlike the other propeller pumps, the 'Petti and Para' do not possess a submerged impeller. The pump is started when the water level in the sump covers the impeller, or by priming with the help of foot valve.

The research programme comprise of three parts, (1) field survey, (2) field pumping test and (3) design,

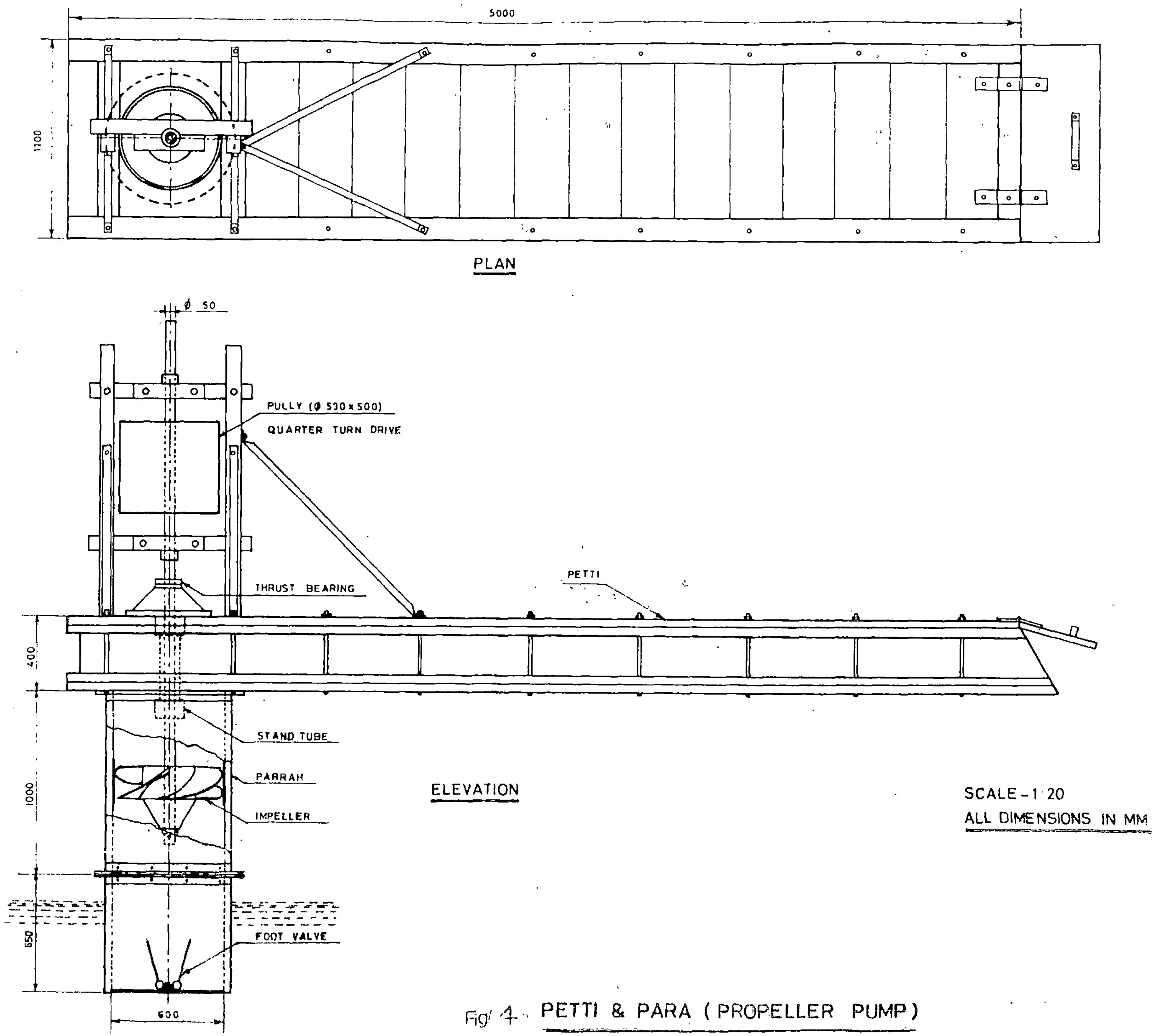


Fig 4 PETTI & PARA (PROPELLER PUMP)

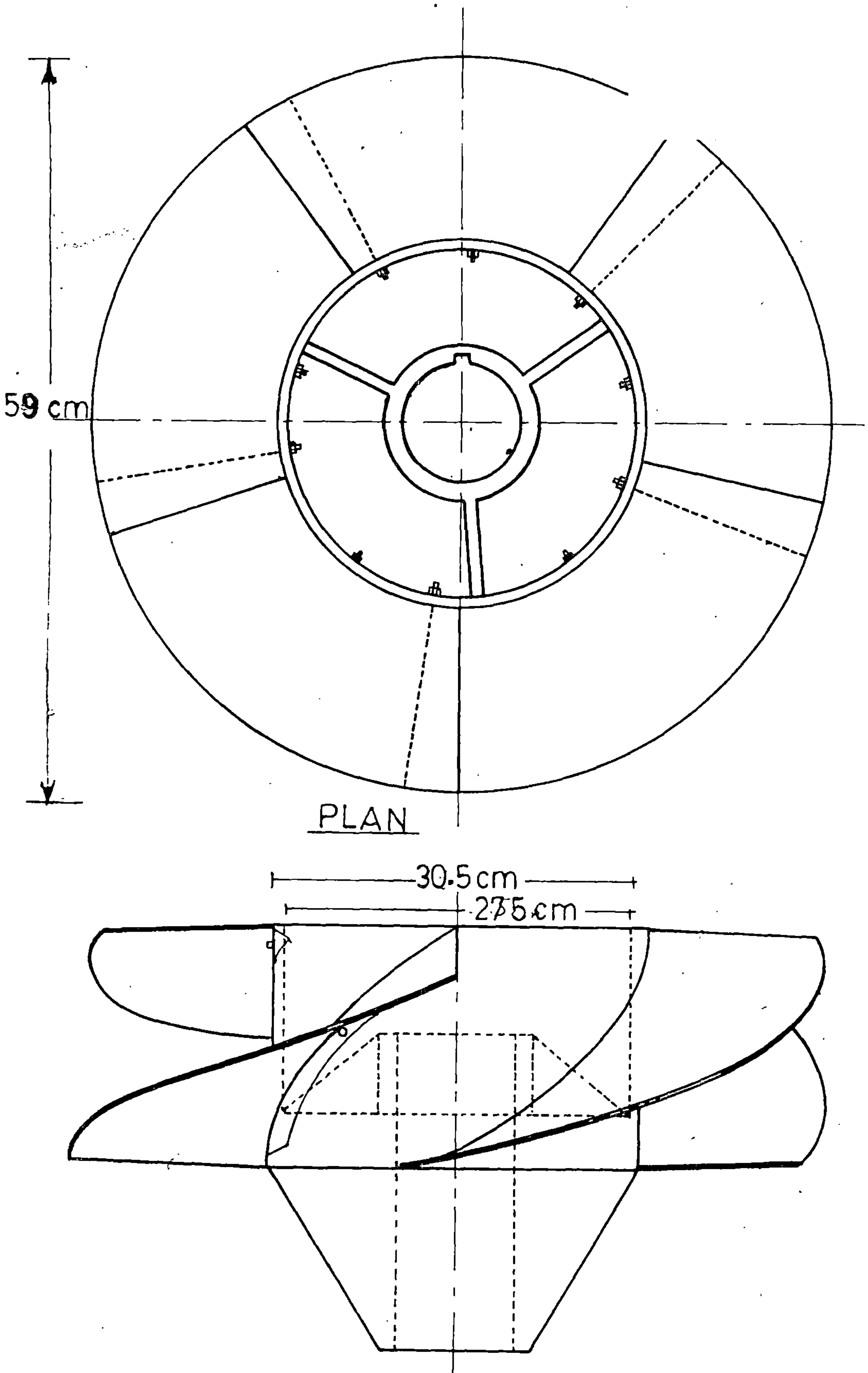


FIG. 5 20 HP PETTI AND PARA IMPELLER

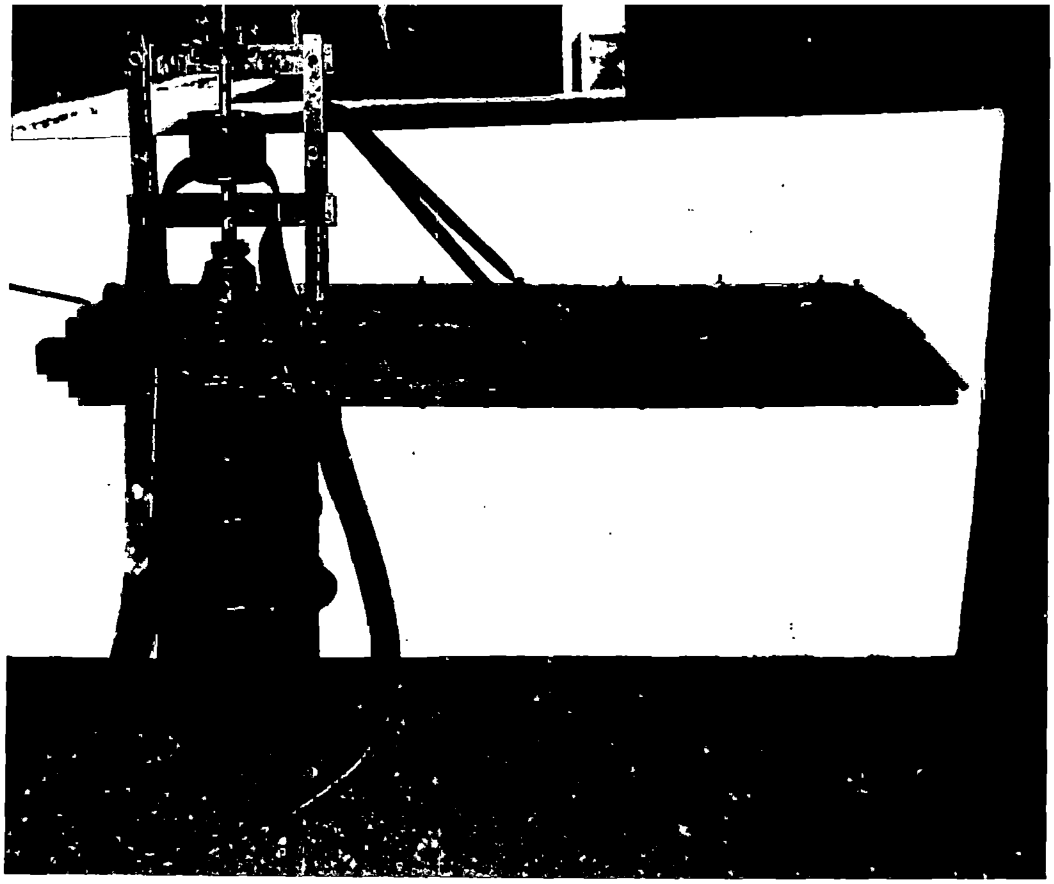


Plate 1. Model of a 'Petti and Para'

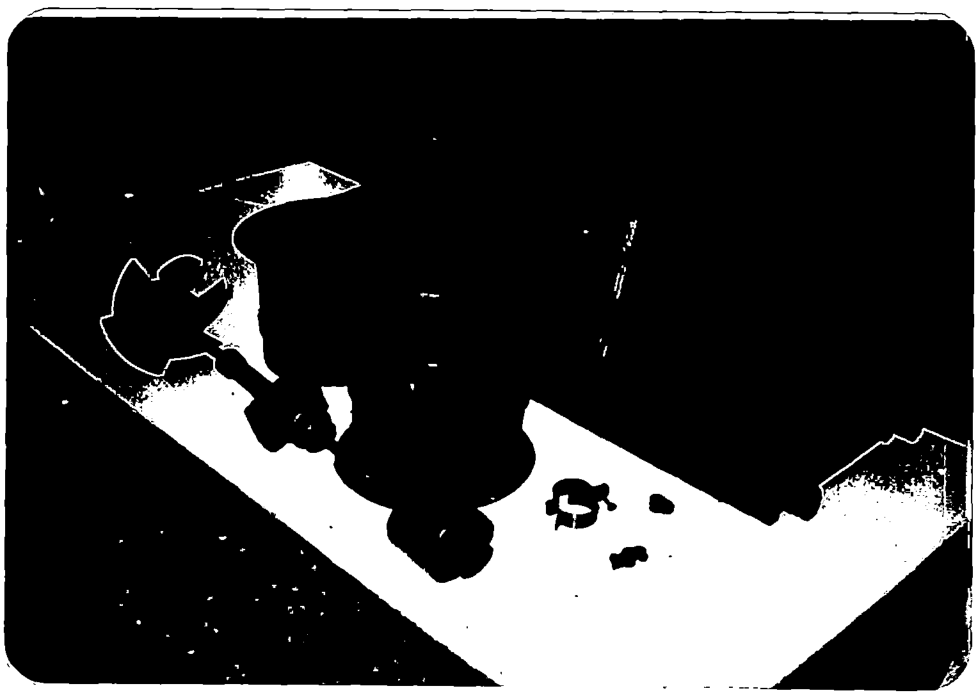


Plate 2. Different parts of 'Petti and Para'

Plate 3. 'Petti and Para'

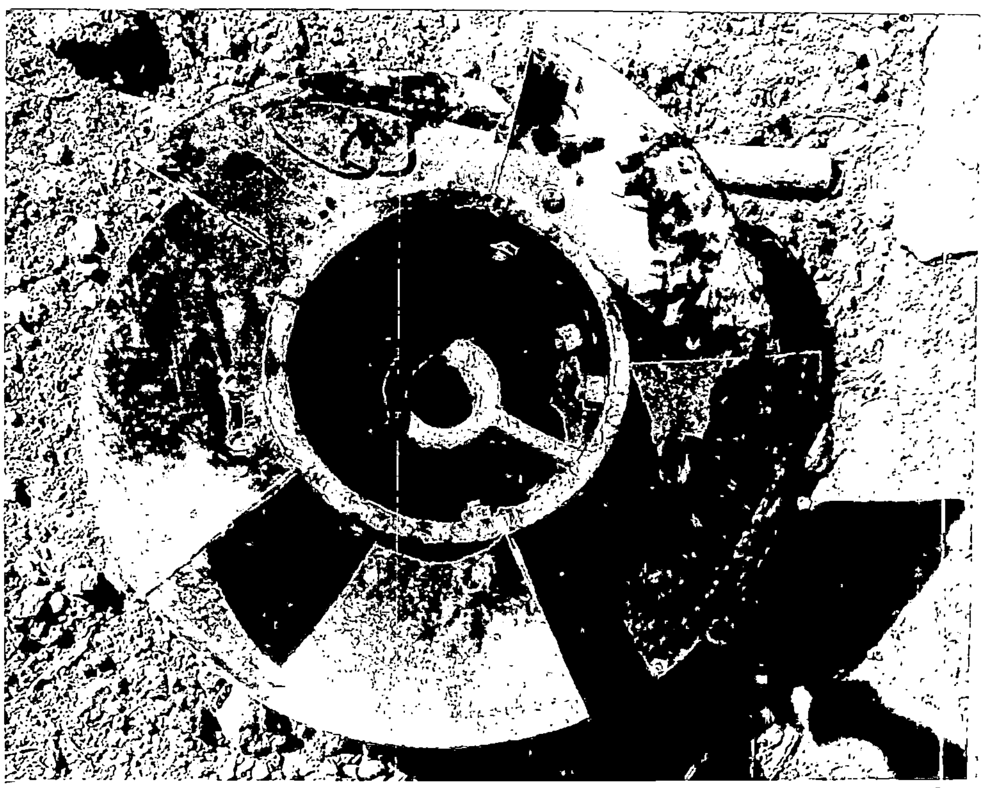


Plate 4. Impeller of a 15 hp 'Petti and Para' (Plan)

development and testing of a propeller pumps to recommend necessary modification to the existing 'Petti and Para'.

3.1. Field survey

A survey was conducted to collect information on the general characteristics of 'Petti and Para' commonly fabricated in Kuttanad for dewatering. All the available informations like area of the field, mean depth of flooding, number of pumpsets installed, motor hp and pump size and energy consumption in different seasons were collected from different agencies. The data collected were analysed by grouping the 'Padasekharams' into three groups viz. (1) 'Padasekharams' below 20 ha (Category I), (2) 'Padasekharams' between 20 and 100 ha (Category II) and (3) 'Padasekharams' above 100 ha (Category III). From these date the following observations were recorded.

(1) Type of pump, size, pump speed and other general characteristics.
(2) Mean installed hp/ha
(3) Mean energy consumption/ha
(4) Mean hours of operation

3.2. Preliminary studies

Due to the absence of reliable data on the characteristics of 'Petti and Para', a preliminary study was undertaken to evolve a suitable device to measure accurately discharge of water and to get a general picture of the characteristics of 'Petti and Para' type of pump sets. The preliminary study was conducted on a 15 hp 'Petti and Para' at Regional Agricultural Research Station, Kumarakom, Kerala Agricultural University. This 15 hp pump has the following general characteristics (Plate 4 and 5).

Diameter of pump (cm)	-	53.00
Number of blades	-	5
Pump rpm	-	328
Cross section of 'Petti' cm	-	80 x 20
Size of upper 'Para' Length x thickness (cm)	-	100 x 3.5
Impeller outer diameter(cm)	-	52.00
Hub diameter (cm)	-	26.00
Hub ratio	-	0.50
Inlet blade angle (degree)	-	22
Outlet blade angle (degree)	-	29

The discharge was measured by velocity area method. The discharge measurement was made in the channel leading to the pump. The channel was a exactly straight one having a length of 125 m. A length of 60 m was taken for discharge measurement, and divided into 10 equal longitudinal sections. Area of cross section was found out by simple segments method

Plate 5. Impeller of a 15 hp 'Petti and Para'
(End view)

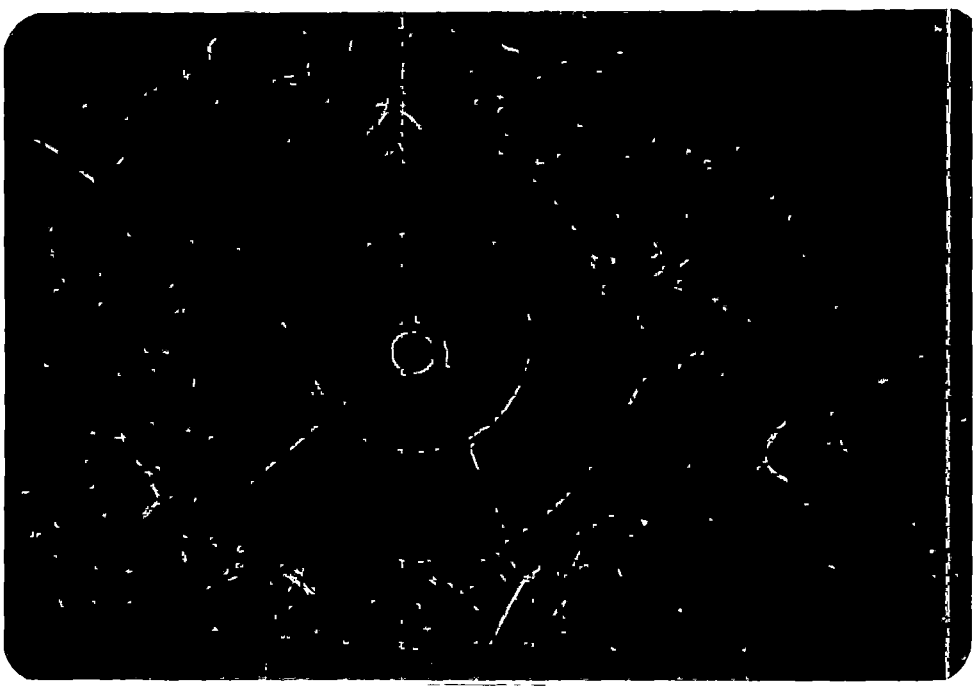


Plate 6. 20 hp 'Petti and Para' impeller (Plan)

after removing the acquatic weeds. The cross sections were plotted by keeping the sump level at constant level. The velocity was found by surface float. The delivery head was measured with help of a manometer fitted to the 'Petti' and the input power by energy meter.

3.3. Field pumping test on a 20 hp 'Petti and Para'

The testing was conducted on a 20 hp 'Petti and Para' installed at the Rice Research Station, Moncombu, Kerala Agricultural University. The pumping unit has the following characteristics (Plate 6 and 7).

Diameter of pump (cm)	-	60.00
Number of blades	-	5
Cross section of 'Petti' (cm)	-	80.00 x 22.00
Size of old upper 'Para', Length x thickness (cm)	-	90.00 x 3.5
Size of old bottom 'Para', Length x thickness (cm)	-	65.00 x 3.50
Size of new top 'Para', Length x thickness (cm)	-	100.00 x 3.50
Diameter of shaft (cm)	-	5.00
Diameter of pulley on motot(cm)	-	17.50
Diameter of pulley on pump(cm)	-	53.00
Top and bottom plank thickness of 'Petti' (cm)	-	3.50

Side plank thickness of 'Petti' (cm)	-	5.00
Belt size (width in cm x ply rating)	-	12.50 x 5
Impeller outer diameter (cm)	-	59.00
Hub diameter (cm)	-	30.50
Pump speed (rpm)	-	317
Inlet blade angle (degree)	-	19
Outlet blade angle (degree)	-	27
Width of blade (cm)	-	14.00
Hub ratio	-	0.52
Length of blade at hub (cm)	-	24.00
Length of blade at outer periphery (cm)	-	42.00

3.3.1. Test site

The pumping unit was installed at one end of the paddy field. The sump had a cross sectional area of 10 m^2 and a depth of 2.5 m. The pump discharged into a channel having a cross section of 4.5 m x 1 m. The discharge end of the 'Petti' was held in position by a clay embankment, which also prevented entry of water from the discharge channel into the paddy field. The other end of the pumping unit was supported by a coconut beam having a diameter of about 20 cm. The whole pumping unit was housed in a temperary shed. The motor had an adjustable wooden foundation.

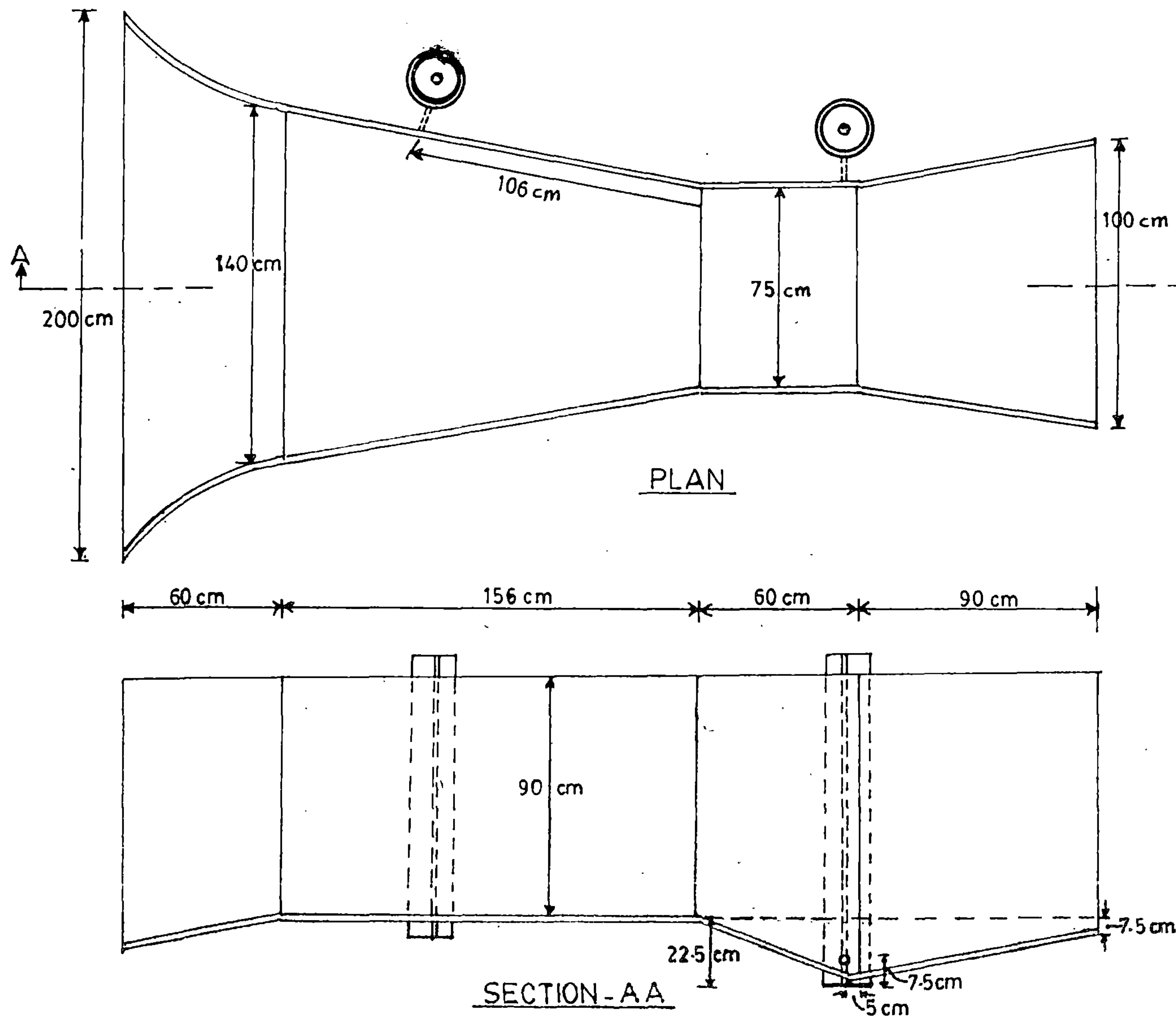
A mat made of split bamboo, fixed infront of the discharge end of the 'Petti' prevented erosion from the embankment, and also acted as an energy dissipater.

The water discharged from the pump flows into the river 100 m away through the channel. There was a parallel channel of the same size from the river, through which fresh water was taken to the paddy field.

3.3.2. Discharge measurement

Based on prelimanary studies the instruments and programme of testing were fixed. Accordingly a parshall flume was fabricated with sheet metal, having a throat width of 75 cm at Agricultural Engineering Workshop, Mannuthy (Fig. 6). It had a capacity to measure a maximum discharge of 1000 l/s. The system was made portable by fabricating it into 4 units. To install the flume a small channel was constructed by joining the two parallel channels at the test site and discharge measurement was made by diverting flow into the parshall flume.

Discharge through a parshall flume can occur for two conditions of flow (1) free flow and (2) submerged flow. For



SECTION - AA

FIG. 6 PARSHALL FLUME

SCALE 1 : 20

free flow, only the flume head Ha at the upstream guage locations is needed to determine the discharge from a standard table. The free flow range includes some of the range which might ordinarily be considered submerged flow because parshall flumes tolerate 50 to 80 per cent submergence before the free flow rate is measurably reduced. For submerged flows, both upstream and downstream heads ha and hb are needed to determine the discharge. Equation for discharge through a flume under free flow condition is given by $Q = 4\ WHa^{1.522} \times w^{0.026}$, where 'W' is the throat width in feet, Ha is the upstream head and w tolerance for throat width (Karassik et al., 1975). This formulae is applicable only to flume of throat width 1 to 8 feet. When the ratio of the two heads Hb and Ha exceeds the limits for free flow conditions, it becomes necessary to apply a negative correction to the free flow discharge in order to determine rate of submerged flow. For flumes with throat width between 1 and 8 feet, the submerged discharge is determined by using a correction diagram given in Fig. 7. The diagram is for a 1 foot throat width and was made applicable to the larger flumes by multiplying the correction for a 1 foot flume by the factor (M) for the size of flume in use. This correction is then substracted from the free flow discharge for the measured head Ha. The multiplying factor for a 75 cm throat width flume is 2.1 and upto 70% submergence discharge is not reduced.

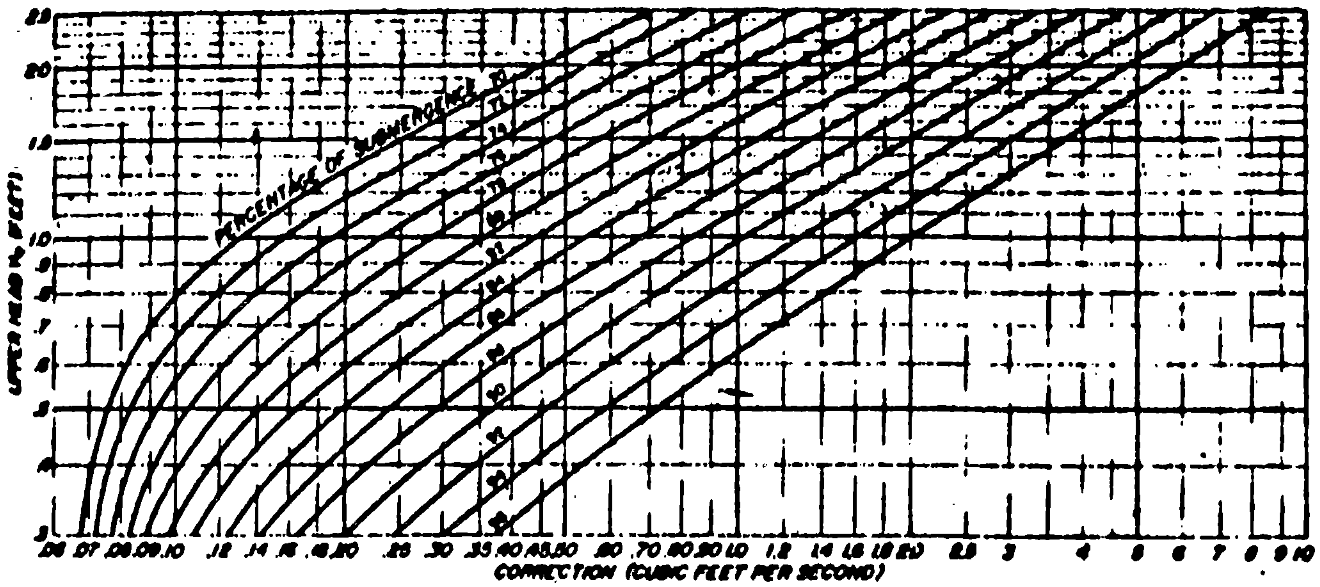


Figure-7 -- Diagram for computing the rate of submerged flow, in cubic feet per second, through a 1-foot Parshall measuring flume

3.3.3. Head measurement

The delivery head was measured with help of a water manometer fitted on the side of the 'Petti'. To ensure steady flow conditions, the orifice point was selected at a distance of 5 times the diameter of 'Para' from the elbow joint of 'Petti and Para'. The following precautions were taken while fixing the manometer (1) the orifice in the pipe was flush and normal to the wall of the water passage (2) the wall of the water passage was smooth and of unvarying cross section.

To keep the sump level constant the discharge from the pump was circulated back into the paddy field. The total head was calculated as $H = Hd + Hs + Vd^2/2g$, where Hs was the distance from the sump level to the centre line of the discharge conduit and Hd was the discharge head referred to the centre of the discharge conduit. $Vd^2/2g$ was the velocity head at the discharge. The loss in the suction end and the discharge column up to the point where the discharge head measured was charged against the pump.

3.3.4. Power measurement

The input power of the unit was determined with the help of energy meter. For this calibrated energy meters were collected from the Kerala State Electricity Board. The time taken by the energy meter disc for fixed number of rotations were noted by a stop watch, and the input to the motor was calculated by the formula, Power input = $\frac{n}{t} \times \frac{3600}{K} \times \frac{1000}{746}$, where 'n' is the number of revolutions, 't' is the time taken for 'n' revolutions, K - energy meter constant.

3.3.5. Speed measurement

Speed of the pump under test was determined by tachometer.

3.3.6. Programme of testing

The experiment was undertaken before and after conducting repairs to the pumping unit. Observations were taken (1) by keeping the sump level constant by recirculating the discharge back, (2) by applying pressure by clossing the outlet flap valve partially, (3) by varying the sump level as in the case of usual pumping operation, (4) by opening the flap valve fully and (5) by removing the bottom 'Para'. Repair works conducted on the pumping unit were:

1) The two contact points between the 'stand tube' and shaft was worn out due to constant rubbing action between them. Necessary repairs were conducted to bring it back into original shape.

2) Due to the above problem, the impeller outer tip also was worn out by 8 mm at one side due to constant rubbing action with the metal sheet protecting the wooden 'Para'. The worn out part was built in with metal pieces of same thickness and rough parts were given filling, so that relative roughness was reduced to a minimum. Thus the clearance between the casing and impeller outer tip was brought back to 5 mm.

3) The old 'Para' housing the impeller was replaced with a new one.

4) Leakage through the 'Petti' was reduced by carpentry work.

3.4. Design of a propeller pump

3.4.1. Axial flow impeller design procedure

The design procedure of single stage axial flow impeller are given below:

1. To meet a given set of head capacity requirements, the speed was selected; thus the specific speed of the impeller was fixed giving due consideration to the head range the proposed pump should cover in future applications under the most adverse suction conditions.

2. For the specific speed thus obtained, the hub ratio and vane spacing l/t were selected. The number of vanes were

fixed at the same time.

3. The speed constant and the capacity constants were chosen next. Thus meridional velocity and impeller diameter were determined.

4. The impeller vane profiles, both vane curvature and vane twist, were drawn after the entrances and discharge vane angles for several streamlines were established from Euler's entrance and exit velocity triangles, following airfoil shapes.

The pump was designed as a high specific speed pump, operating at high rotational speed, so that its physical dimensions were small. A small portable unit was designed initially, so that the work could be extended to develop bigger units utilising the experience and results obtained from the first unit.

Initially the requirement like capacity, head, speed, etc. of the pump to be designed were fixed. The capacity was fixed as 40 l/s and the head as 1.5 m. The speed of the pump was taken as 1900 rpm. From these data, the specific speed of the pump was determined. In metric unit

$$Ns = N\frac{\sqrt{Q}}{H^{3/4}} = \frac{1900\sqrt{0.04}}{1.5^{3/4}} = 280.35 \text{ rpm}$$

$$= 280 \text{ rpm}$$

In order to compare the value of Ns with the values suggested by Stepanoff (1957), specific speed was also calculated in FPS unit.

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

$$Ns = \frac{1900\sqrt{\dfrac{40 \times 60}{4.546}}}{\left(\dfrac{150}{2.54 \times 12}\right)^{\frac{3}{4}}}$$

$$Ns = 13212.5 \text{ rpm in FPS system}$$

The specific speed was approximately taken as 13000 rpm.

3.4.2. Number of vanes

Based on the specific speed the number of vanes was fixed as 3 (Fig. 3).

3.4.3. Design for impeller dimensions

Volumetric efficiency of the pump assumed was 90%

$$Q^1 = Q/ev$$

where Q^1 was the quantity to be pumped in m^3 (considering the volumetric efficiency) and 'ev' was the volumetric efficiency.

$$Q^1 = \frac{0.040}{0.90} = 0.044 \ m^3/s$$

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

Mechanical cross section was given by $Am = \frac{Q^1}{Cm_1}$ where $Cm_1 = Km_1 \sqrt{2gH}$, and Km_1 was the capacity constant at inlet or the coefficient of flow velocity.

Addison (1955) proposed that capacity constant should be within 0.25 to 0.6.

Km_1 was assumed as 0.55

$$Cm_1 = Km_1 \sqrt{2\ gH}$$

$$Cm_1 = 0.55 \sqrt{2 \times 9.81 \times 1.5}$$

$$Cm_1 = 2.98\ m/sec$$

$$Am = Q^1/Cm_1$$

$$Am = \frac{0.044}{2.98}$$

$$Am = 0.0148$$

3.4.4. Hub ratio

$$\text{Hub ratio} = dh/d2$$

where dh was the diameter of the hub, d2 is the impeller outer diameter.

Combining factors such as specific speed, vane spacing ratio, number of vanes, hub ratio was fixed as 0.325 (Fig. 3).

$$Am = \pi/4 \times \left(d2^2 - dh^2 \right)$$

$$Am = \pi/4 \times d2^2 \left(1 - (dh/ds)^2 \right)$$

$$d2 = \left(\frac{4\ Am}{\pi \left(1 - (dh/d2)^2\right)} \right)^{\frac{1}{2}}$$

$$dh/d2 = 0.325$$

$$d2 = \left(\frac{4 \times 0.0148}{\pi \ (1 - 0.325^2)} \right)^{\frac{1}{2}}$$

$$d2 = 14.5 \text{ cm}$$

$$dh = 14.5 \times 0.325$$

$$dh = 4.71 \text{ cm} \qquad 5 \text{ cm}$$

3.4.5. Chord spacing ratio

For axial flow pumps of specific speed 10,000 and more, the chord spacing ratio was less than unity. The chord spacing ratio was fixed as 0.9 (Fig. 3). Also the chord spacing ratio at hub was made 1.3 times the outside diameter of the impeller.

Pitch of the blades was given by

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

where D was the diameter of the hub,

Z was the number of blades

$$\text{Pitch at hub} = \frac{\pi \times 5}{3} = 5.23 \text{ cm}$$

Length of blade at hub $= 0.9 \times 5.23 = 4.71$ cm

Length of blade provided $= 1.3 \times 4.71 = 6.123$

$= 6.2$ cm

Pitch at outer diameter $= \frac{\pi \times 14.5}{3} = 15.18$ cm

Length of blade at outer diameter $= 0.9 \times 15.18 = 13.662$

$= 14$ cm

Therefore the length of the blades was taken as 6.2 cm at hub and 14 cm at outer diameter.

3.4.6. Width of the blades

Width of the blades was given by

$$W = \frac{d2 - dh}{2} = \frac{14.5 - 5}{2} = 4.75$$

The width of blades was taken as 4.75 cm. The inside diameter of casing was fixed as 153 mm.

3.4.7. Inlet angle

Inlet angle β_1 was given by

$$\text{Tan } \beta_1 = \frac{Cm_1}{U_1}$$

Where Cm_1 was the meridional velocity at inlet

U_1 was the absolute velocity at inlet

$$U_1 = \frac{\pi \, Dm \, N}{60}$$

Where Dm was the mean diameter given by $\left(\frac{dh + d2}{2}\right)$ in m and N was the speed of the impeller in rpm.

$$Cm = 2.98 \text{ m/s}$$

$$Dm = \frac{5 + 14.5}{2} = 0.75 \text{ cm}$$

$$U_1 = \frac{\quad x\ 0.0975 \times 1900}{60}$$

$$U_1 = 9.699 = 9.7 \text{ m/s}$$

$$\text{Tan } \beta_1 = \frac{2.98}{9.7} = 0.3072$$

$$\beta_1 = \tan^{-1} (0.3072)$$

$$\beta_1 = 17^\circ\ 4'\ 40'' \approx 17^\circ$$

Therefore the inlet angle was fixed at 17°

3.4.8. Outlet angle

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

$$\text{Tan } \beta_3 = \frac{Cm_2}{U_2 - Cu_3}$$

where β_3 was the theoretical outlet angle

Cm_2 was the meridional velocity at outlet.

U_2 was the peripheral velocity at outlet.

Cu_3 was the tangential component of the absolute velocity (theoretical) at outlet.

$$Cu_3 = \frac{g\ Hth}{U_2}$$

$$Hth = \frac{H}{eh}$$

where eh was the hydraulic efficiency.

Stepanoff (1957) from his experimental studies found that high specific speed pumps had more hydraulic efficiency. The hydraulic efficiency was taken as 80 per cent.

$$Hth = \frac{1.5}{0.8}$$

$$Hth = 1.875 \text{ m}$$

$$Cu_3 = \frac{9.81 \times 1.875}{9.7}$$

$$Cu_3 = 1.896 \text{ m/s}$$

$$\text{Tan } \beta_3 = \frac{2.98}{9.7 - 1.896} = 0.382$$

$$\beta_3 = \tan^{-1} (0.382)$$

$$\beta_3 = 20° \; 53' \; 58'' \qquad 21°$$

Theoretically angle β_2 will be less than $\beta_3 = 21°$

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. The performance of the pump was based on the difference between inlet and outlet angles.

Therefore β_2 was selected as 24°.

By Stodola formula slip factor ' μ ' was given by

$$= 1 - \frac{\pi \text{ Sin } B_2}{Z} = \frac{Cu_2}{Cu_3}$$

where Z was the number of blades

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

$$= 0.575$$

$$Cu_2 = Cu_3 = 0.575 \times 1.896$$

$$Cu_2 = 1.0902$$

again $$\tan \beta_2^1 = \frac{Cm_2}{U_2 - Cu_2}$$

$$\tan \beta_2^1 = \frac{2.98}{9.7 - 1.0902}$$

$$\beta_2^1 = \tan^{-1}(0.346)$$

$$= 19^\circ\ 5'\ 29''$$

The actual outlet angle β_2^1 is less than the theoretical outlet angle β_3. So the angle β_2 practically selected is reasonable.

3.4.9. Horce power of the pump

The horse power of the pump was found out by using the formula

$$HP = \frac{WQH}{75 \times eo}$$

where Q is the quantity in m^3/s

W is the specific weight in kg/m^3

H is the head in m

eo is the overall efficiency

$$HP = \frac{1000 \times 0.04 \times 1.5}{75 \times 0.50} = 1.6 \text{ hp}$$

The overall efficiency assumed was 50 per cent.

3.4.10. Design for the shaft

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

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

Horse power transmitted by the shaft was

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

where P is the horse power

T is the torque transmitted in Kg-m

N speed in rpm

'fs' is the allowable shear stress in Kg/cm^2

The horse power P of the pump was 1.6 and speed of the pump was 1900 rpm.

$$\text{Therefore} \quad T = \frac{1.6 \times 4500}{2 \times \pi \times 1900}$$

$$T = 0.6031 \text{ kg-m}$$

$$T = 60.31 \text{ kg-cm}$$

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

$$d^3 = \frac{16 \times 60.31}{\pi \times 450}$$

$$d = 0.88 \text{ cm}$$

The size of shaft provided was 2.5 cm.

3.4.11. Design for pump casing

In a 'Petti and Para' the cross sectional area of the 'Para' remains the same. Also the cross section of the 'Petti' is always less than the area of cross section of the 'Para'. So in the design of the pump casing the above feature was taken into consideration, and hence the same cross sectional area was maintained through out its length. A flared inlet was also provided to reduce entrance lose, and a strainer below it to avoid entry of foreign materials.

The mechanical cross sectional area was made equal to the cross sectional area of casing.

$$Am = 0.0148$$

$$\text{But} \quad Am = \frac{\pi d^2}{4}$$

$$\text{where} \quad d^2 = \frac{4 \times 0.0148}{\pi}$$

$$d = 13.72 \text{ cm} \approx 14 \text{ cm}$$

So the same diameter was provided for the pump casing through out its length. The diameter of the casing was 15.3 cm. The diameter of the outlet pipe was fixed as 12.5 cm.

The advantages 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 there by reducing the shock.

3.5. Fabrication

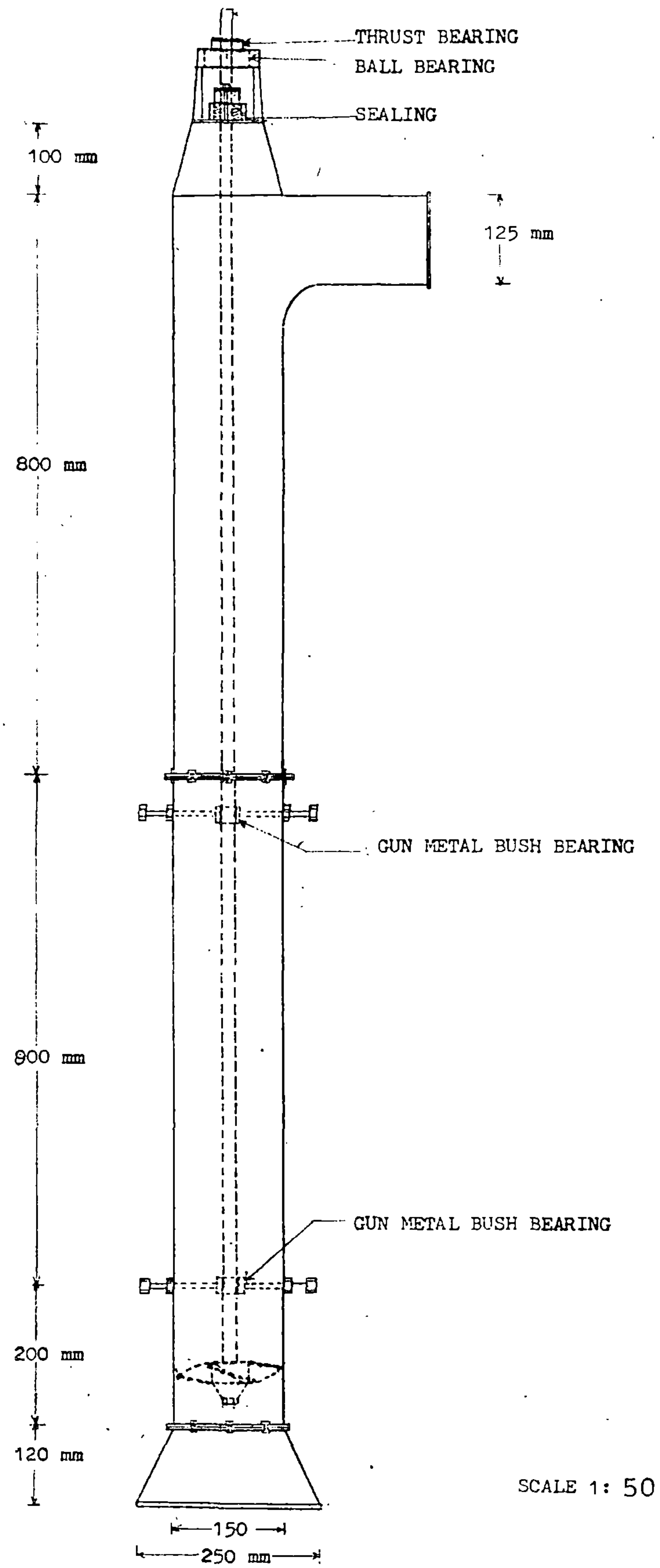
The pump was fabricated in a local machine shope at Kumarakom, Kottayam. The details are given below (Fig. 8).

3.5.1. Hub

The hub was made from mild steel. There is a MS bush inside the hub through which the shaft passess. The bottom half of the bush hole was made conical.

3.5.2. Blades

The blades were cut in correct dimension. Then the blades were twisted from 17° to 24° in uniform variation. Twisting was done by simple blacksmithy. Blades were then welded to MS pieces, 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 8 shows the details of blades.



ELEVATION

Fig-8 PROPELLER PUMP

3.5.3. Shafts

The shaft used was 25 mm mild steel shaft. The bottom of shaft was turned to conical shape in order to suit the hub. Below that, shaft was threaded to check the impeller by check nut.

3.5.4. Bush bearing to support the impeller

Gun metal bush bearing was utilised. The gun metal bush was inserted in a MS bush. This bush was fixed to the casing by three adjustable type bolts having a diameter of 1.4 cm.

3.5.5. Casing

Casing was fabricated from 12 gauge MS sheets. The sheets were cut to its size, rolled and welded. The outlet pipes were also fabricated from 12 gauge metal sheet (Plate 9).

3.5.6. Power transmission

The power transmission was through a bevel gear and V pulley. The pumping unit can be driven by a power tiller or a electric motor.

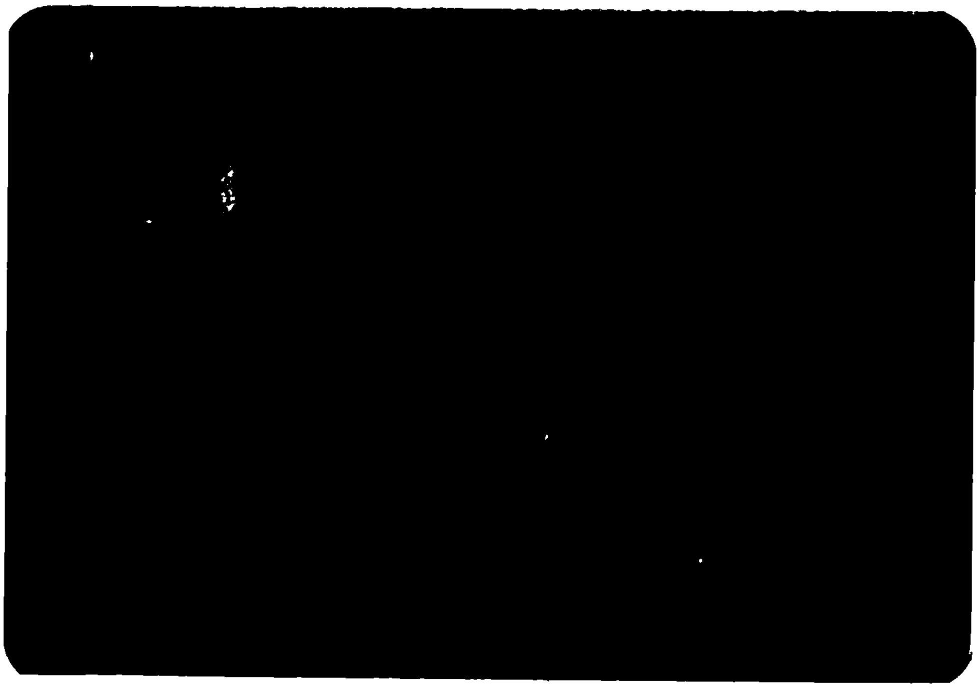


Plate 7. 20 hp Impeller inside the 'Para'

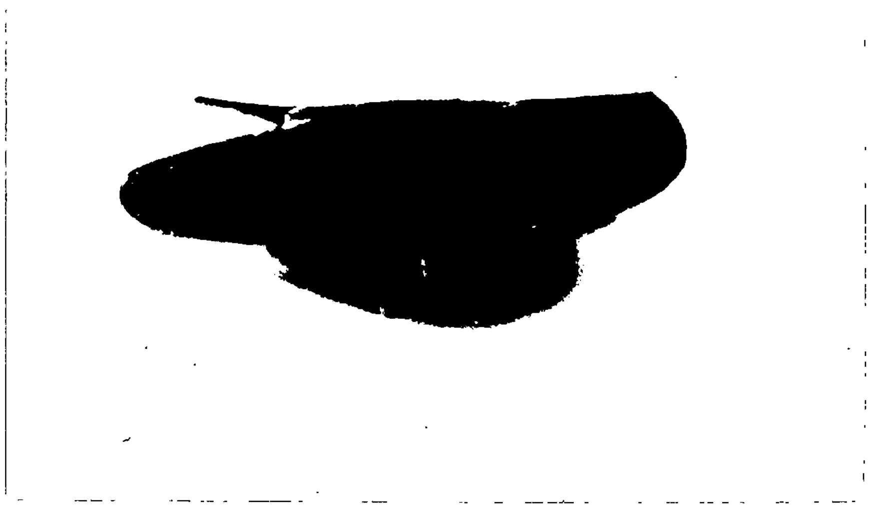


Plate 8. 3 vane impeller of 153 mm propeller pump

Plate 9. 153 mm Propeller pump

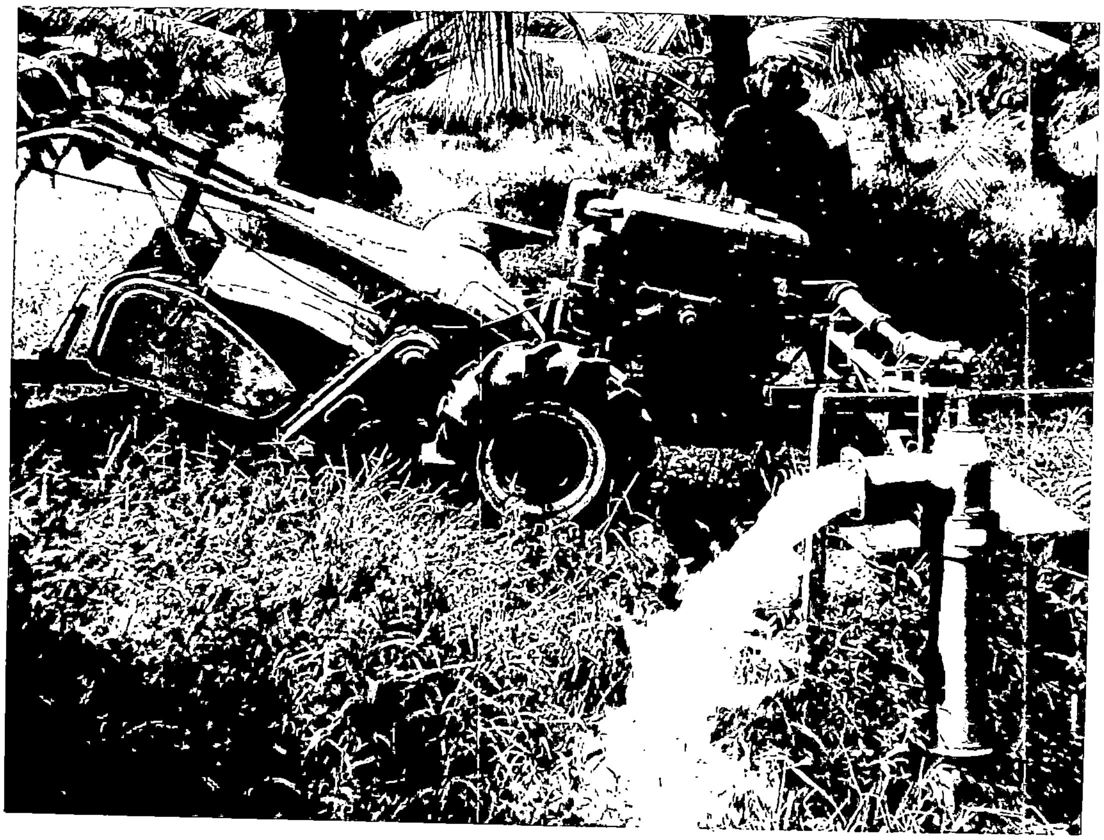


Plate 10. Testing of Propeller pump

3.5.7. Bearing

One thrust bearing along with a roller bearing was used in the pump.

3.5.8. Testing

The testing was done at Regional Agricultural Research Station, Kumarakom, Kerala Agricultural University. The power unit used was a 10 hp, 3 phase induction motor.

3.5.9. Instruments and measuring apparatus used

A fibre glass tank having a capacity of 1050 litres was used to measure the flow rate. The input energy to the motor was taken by using an energy meter of energy meter constant of 60. Time taken by the energy meter disc for 2 revolutions was taken by a stop watch.

The test was conducted at a constant static head of 120 cm. Flow rate was controlled by a shutter fitted at the exit. The head was measured using a water manometer. The manometer was fitted in the delivery pipe, at a distance of 6 times the diameter of pump, from the bend so as to reduce turbulence. The input to the motor was calculated by the formula

$$\text{Input} = \frac{n}{t} \times \frac{3600}{K} \times \frac{1000}{746}$$

where 'n' is the number of revolutions

't' is the time for 'n' revolutions

K energy meter constant

Results and Discussion

RESULTS AND DISCUSSION

4.1. Field survey

The major specifications of 'Petti and Para' commonly fabricated in Kuttanad area were given in Table 3. The most common type of 'Petti and Para' utilised are 10 hp, 15 hp, 20 hp, 25 hp, 30 hp, 40 hp and 50 hp units. During the survey, it was seen that the speed of the pump showed a declining tendency with the increase in size of the pump. The expected speed of a 10 hp 'Petti and Para' was 362 rpm and that of a 50 hp pumping unit was only 312. The number of vanes for a 10 hp unit was 4, for 15 and 20 hp unit it was 5, and for 30 hp, 40 hp and 50 hp unit it was 6. The shaft size also varied with the amount of power transmitted. For a 10 hp unit, the shaft diameter was 4.5 cm, and that of a 50 hp was 7.5 cm. To prevent slippage of belt from pulley, the pulley length was greater than the required length and the pulley was not given any crowning. The thickness of the single piece side plank was 5 cm for 10 hp, 15 hp and 20 hp unit and 6 cm for other types. The cross sectional area of the 'Petti' was less than that of the 'Para'. The ratio of cross sectional area of 'Petti', to cross sectional area of 'Para' was 0.883 for a 10 hp unit and 0.557 for a 50 hp unit. The purpose of reduction in cross sectional area was to equalise the cross

Table 3. Major specifications of 'Petti and Para' commonly fabricated in Kuttanad area (Motor rpm = 960)

	Description of item	10 hp	15 hp	20 hp	25 hp	30 hp	40 hp	50 hp
1	Expected rpm of pump	362	340	328	328	316	312	312
2	Diameter of pump (cm)	45.0	53.0	60.0	65.0	70.0	76.0	80.0
3	Number of blades	4	5	5	5	6	6	6
4	Size of shaft (cm)	4.5	5.0	5.0	6.4	6.4	6.4	7.5
5	Size of pulley on motor (cm)	18.0	18.0	18.0	18.0	20.0	23.0	23.0
6	Sixe of pulley on pump (cm)	48.0	51.0	53.0	53.0	61.0	71.0	71.0
7	Size of 'Petti' (cm)	78.0 x 18.0	80.0 x 20.0	80.0 x 22.0	85.0 x 24.0	90.0 x 26.0	95.0 x 27.0	100.0 x 28.0
8	Plank thickness of 'Petti'(cm)	3.5	3.5	3.5	4.0	4.0	4.5	4.5
9	Size of 'Para'							
	(a) top 'Para' (length x thickness(cm)	100.0 x 3.5	100.0 x 3.5	100.0 x 3.5	100.0 x 3.5	100.0 x 3.5	100.0 x 3.5	100.0 x 3.5
	(b) Bottom 'Para' (length x thickness (cm)	65.0 x 3.5	65.0 x 3.5	65.0 x 3.5	65.0 x 3.5	65.0 x 3.5	65.0 x 3.5	65.0 x 3.5
10	Belt size (Width in cm x ply rating)	10.0 x 4	10.0 x 4	12.5 x 5	12.5 x 5	15.0 x 6	15.0 x 6	15.0 x 6
11	Cross section of 'Para' (cm^2)	1590.4	2206.2	2827.4	3318.3	3848.5	4536.5	5026.5
12	Cross section of 'Petti'(cm^2)	1404.0	1600.0	1760.0	2040.0	2340.0	2565.0	2800.0
13	Ratio of cross section of 'Petti' to cross section of 'Para'	0.88279	0.7252	0.6223	0.6147	0.6080	0.5654	0.5570

sectional area of the 'Petti' almost equal to the flow area (mechanical area) through the impeller.

The information on land area, hp of the motor installed and the energy consumption of some of the 'Padasekharams' in Kuttanad in punja season and additional crop season for 1981-82 and 1982-83 were gathered (Table 4, 5 and 6)

For paddy fields having an area less than 20 ha, the mean hp installed per hectare was 1.585. The mean energy consumption per hectare were 254.8 kWh for punja and 454.9 kWh for additional crop. In case of additional crop the energy consumption was about 200 kWh/ha more than that for punja crop (78.5% increase).

For paddy fields having an area between 20 and 100 ha, the mean installed hp/ha was 0.792 and the mean energy consumption were 276.2 kWh/ha and 427 kWh/ha respectively for punja and additional crop. The increase in energy consumption for additional crop was found to be 55 per cent.

For paddy fields having an area greater than 100 ha. the hp/ha was 0.777 and mean energy consumption per hectare were 398.3 kWh and 452.5 kWh. The variation in energy requirement of punja crop was relatively lower and was about 13.6 per cent. The per hectare energy consumption of paddy fields

Table 4. Table showing the energy requirement of Padasekharams below 20 ha (Category - I)

Sl. No.	81-82 Punja crop			82-83 Punja crop			81-82 Additional crop			82-83 Additional crop			Area ha
	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	1.739	395.2	2272.4	1.739	303.5	1745.3	1.739	372.9	2144.4	-	-	-	5.75
2	1.132	221.92	1076.5	1.132	75.9	670.7	1.132	173.2	1529.3	-	-	-	8.83
3	1.385	238.6	1722.3	1.385	217.8	1572.4	1.385	707.1	5104.9	1.385	361.7	2611.5	7.22
4	4.395	653.1	1489.1	4.395	298.9	681.5	4.395	220.6	503.0	-	-	-	2.28
5	2.490	341.3	1371.9	2.490	306.9	1233.9	-	-	-	2.490	528.8	2125.8	4.02
6	1.346	294.6	2185.7	1.346	185.8	1378.7	1.346	715.3	5307.6	1.346	543.9	4036.4	7.42
7	1.219	269.9	3317.3	1.219	269.9	3317.3	1.219	436.6	5365.8	1.219	308.9	3796.9	12.29
8	1.159	359.4	3097.7	1.159	284.2	2450	1.159	419.7	3618.4	1.159	695.9	5999.5	8.62
9	1.040	135.7	1304.8	-	-	-	1.040	534.1	5132.3	1.040	516.9	4968.3	9.61
10	0.898	258.6	2878.0	0.898	127.2	1415.5	0.898	721.1	8026.3	-	-	-	11.13
11	0.781	262.9	5050.9	0.781	332.2	6458.7	0.781	297.8	5720.8	0.781	287.1	5514.9	19.21
12	2.445	475.8	970.7	2.445	466.1	950.7	2.445	590.5	1204.7	-	-	-	2.04
13	2.021	163.7	810.5	2.021	134.4	665.4	2.021	197.3	976.5	2.021	776.6	3844.0	4.95
14	2.312	734.1	3171.1	2.312	240.7	1040.0	2.312	256.6	1108.5	-	-	-	4.32
15	1.950	273.0	2099.4	1.950	344.6	2650.2	-	-	-	1.950	341.1	2623.3	7.69

(Contd.)

Table 4. Continued

1	2	3	4	5	6	7	8	9	10	11	12	13	14
16	1.950	315.1	2423.3	-	-	-	1.95	628.7	4834.6	1.95	258.8	1190.4	7.69
17	1.195	171.78	2155.94	1.195	232.1	2912.9	-	-	-	-	-	-	12.55
18	0.993	111.0	1118.0	0.993	212.8	2143.3	-	-	-	-	-	-	10.07
19	1.249	266.6	1268.8	1.249	145.0	690.4	-	-	-	-	-	-	4.76
20	0.995	193.3	1948.7	0.995	220.3	2216.2	-	-	-	-	-	-	10.06
21	1.210	248.8	2057.8	1.210	137.3	1135.3	-	-	-	-	-	-	8.27
22	0.819	109.6	1338.4	0.819	133.5	1629.5	-	-	-	-	-	-	12.21
23	1.580	302.6	1909.6	1.580	150.1	946.8	-	-	-	-	-	-	6.31
24	1.358	383.39	4236.5	1.358	331.2	3659.5	-	-	-	-	-	-	11.05
Mean	1.57hp	295kWh		1.573hp	214.6kWh		1.70hp	447.96kWh		1.53hp	461.97kWh		

Table 5. Table showing the energy requirement of Padasekharams between 20 to 100 ha (Category - II)

Sl. No.	81-82 Punja crop			82-83 Punja crop			81-82 Additional crop			82-83 Additional crop			Area ha
	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	0.655	183.3	4195.0	0.655	310.9	7115.9	0.655	303.4	6944.6	0.655	295.9	6772.7	22.89
2	-	-	-	0.454	154.5	5097.6	0.454	109.1	3599.5	-	-	-	33.00
3	0.929	284.6	7660.6	0.929	213.3	5741.6	0.929	407.8	10979.2	0.929	625.2	16829.9	26.92
4	1.124	508.6	13575.8	1.124	288.3	7697.0	1.124	611.4	16318.9	0.124	644.9	17214.2	26.69
5	0.943	313.9	9988.7	-	-	-	0.943	627.5	19967.9	0.943	309.9	9863.7	31.82
6	0.863	546.3	18978.9	0.863	450.2	15640.0	0.863	586.03	20358.8	0.863	313.6	10893.3	34.74
7	0.719	85.4	3224.4	0.719	161.7	6102.9	0.719	437.6	16513.5	-	-	-	37.74
8	-	-	-	0.798	194.3	6085.2	-	-	-	0.798	332.9	10426.6	31.32
9	-	-	-	0.604	159.5	5279.1	-	-	-	0.604	592.5	19605.1	33.09
10	1.249	438.7	10528.0	1.249	402.3	9656.1	-	-	-	1.249	592.5	14219.5	24.00
11	0.719	299.2	10403.7	0.719	238.8	8304.8	0.719	444.7	15463.8	0.719	427.2	14852.4	34.77
12	0.534	152.8	13592.6	0.534	80.22	7136.9	0.534	249.9	22235.9	0.534	286.6	25495.0	88.96
13	0.973	187.2	11536.0	0.973	156.2	9623.7	0.973	588.5	36275.3	0.973	199.7	12312.5	61.64
14	1.005	291.1	17383.9	1.005	532.9	31816.7	1.005	367.4	21935.3	1.005	295.6	17648.0	59.71

(Contd.)

Table 5. Continue.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
15	0.495	305.5	15445.0	-	-	-	0.495	359.8	18187.9	0.495	322.1	16280.3	50.55
16	0.749	330.2	22030.2	0.749	166.7	11123.8	0.749	680.6	45401.7	0.749	295.6	19716.8	66.71
17	-	-	-	0.8020	129.9	6477.9	0.8020	452.7	22577.5	0.8020	381.6	19032.3	49.87
18	0.726	155.8	11805.2	0.726	284.4	21553.4	0.726	404.5	30650.2	0.726	425.6	32254.0	75.78
19	0.630	502.6	47851.8	0.630	386.3	36771.8	0.630	401	38175.6	0.630	652.6	62127.4	95.20
20	-	-	-	0.702	185.1	10540.6	0.702	417.9	23803.5	-	-	-	56.96
21	0.669	263.8	11820.3	0.669	241.8	10829.9	0.669	516.6	23144.8	-	-	-	44.80
Mean	0.8113 hp	303.1 kWh		0.7844 hp	249.33 kWh		0.761 hp	442.6 kWh		0.811 hp	411.4 kWh		

Table 6. Table showing the energy requirements of Padesekharams having an area greater than 100 ha (Category - III)

Sl. No.	81-82 Punja crop			82-83 Punja crop			81-82 Additional crop			82-83 Additional crop			Area ha
	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	Hp/ha	En/ha	Total energy	
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	0.732	213.5	42307.3	0.732	233.3	46243.3	0.732	297.5	58971.6	0.732	327.8	64969.9	198.20
2	0.834	535.1	1321.7	0.834	420.9	65578.2	0.834	469.9	73213.7	0.834	530.7	82676	155.80
3	0.786	601.9	245179.9	0.786	673.3	274233.9	0.789	850.7	346517.0	-	-	-	407.30
4	0.676	260.5	26982.6	0.676	230.6	23890.5	0.676	453.9	47016.2	0.676	268.1	27774.2	103.58
5	0.683	237.8	73064.3	0.683	188.6	57957.6	0.683	386.3	118720.0	0.683	250.6	76996.1	307.30
6	0.479	180.5	30162.8	0.479	114.2	19080.8	0.479	218.7	36436.4	-	-	-	167.10
7	0.774	362.5	59326.4	0.774	443.2	68742.9	0.774	610.8	94732.0	0.774	445.6	69110.7	135.10
8	0.907	338.2	59682.7	0.907	257.9	45522.6	0.907	371.9	65635.6	0.907	270.0	47652.2	176.43
9	0.937	319.1	44259.1	0.937	416.5	57767.4	0.931	502.2	69651.7	0.937	335.2	46485.9	138.70
10	0.866	530.1	79536.3	0.866	493.6	74059.0	0.866	754.5	113195.5	0.866	268.6	402962.0	150.03
11	0.564	124.3	13215.7	0.564	73.6	7830.6	0.564	167.4	17804.7	0.564	163.9	17434.0	106.35
12	0.619	432.2	45363.7	0.619	168.6	17699.1	-	-	-	0.619	112.3	11782.9	104.96
13	0.621	438.6	134174.7	0.621	310.3	94915.1	0.621	428.8	131175.0	-	-	-	305.90
14	-	-	-	1.173	510.9	82724.1	-	-	-	-	-	-	161.92
15	0.418	607.9	145419.5	0.418	666.1	159333.0	-	-	-	0.418	601.9	143983.0	239.20

(Contd.)

Table 6. Continued

1	2	3	4	5	6	7	8	9	10	11	12	13	14
16	0.778	431.5	52681.2	0.778	139.6	17048.7	0.778	380.8	46501.7	0.778	258.9	31621.0	122.10
17	1.100	315.7	43050.9	1.100	538.3	73400.8	1.100	832.8	113561.9	1.100	642.4	87604.1	136.36
18	0.982	831.3	84659.9	0.982	582.9	59364.5	0.982	776.1	79040.4	0.982	740.0	75365.0	101.84
19	0.693	263.9	26663.4	0.693	340.3	34374.5	0.693	454.8	45934.9	-	-	-	101.00
20	0.655	265.5	28384.6	0.655	456.3	48779.3	0.655	353.6	37797.7	0.655	342.1	36567.3	106.90
21	0.567	349.1	36898.3	0.567	179.1	18930.8	0.567	429.6	45404.2	0.567	276.9	29266.9	105.69
22	0.813	638.3	125626.6	0.813	682.9	134410.4	0.813	759.7	14952.3	0.813	508.9	100152.0	196.80
23	-	-	-	0.747	339.3	53423.1	-	-	-	-	-	-	133.80
24	1.549	1201.5	372219.2	1.549	947.7	293609.5	-	-	-	1.549	947.7	293586	309.80
25	0.592	261.7	37556.9	0.592	185.0	26547.9	-	-	-	-	-	-	143.50
26	0.607	324.9	136545.3	0.607	325.5	136794.4	-	-	-	-	-	-	420.20
27	0.662	305.8	60072.2	0.662	450.2	88434.5	-	-	-	-	-	-	196.42
28	0.968	495.3	76525.3	0.968	467.6	72239.7	-	-	-	-	-	-	154.50
29	-	-	-	0.827	123.5	17912.7	-	-	-	-	-	-	145.20
Mean	0.764 hp	418.7 kWh		0.779 hp	377.9 kWh		0.760 hp	499.9 kWh		0.803 hp	405.1 kWh		

having an area greater than 100 ha during punja crop season is substantially more than the energy requirement of paddy fields having a total area of less than 100 ha. In all the three catagories the energy requirement for additional crop season was almost the same. Since the additional crop was raised during the south west monsoon period of the year, the energy consumption for pumping was found to be greater than that of punja crop.

4.2. Field pumping test

4.2.1. Preliminary studies on 15 hp 'Petti and Para'

The experiments on a 15 hp 'Petti and Para' revealed that the efficiency of this pumping system was low. The discharge rate varied from 217.75 to 143.60 l/s with the increase in total head from 65.44 to 100.11 cm. The corresponding static head variation was from 43 to 85 cm. The efficiency of the unit droped from a maximum of 21.19 to 18.16% with drop in sump level (Table 7 and Fig. 9).

4.2.2. Testing of 20 hp 'Petti and Para'

The pump was tested before conducting repair work and without adding bottom para to the unit. The sump level was kept constant at 34 cm above the mid point of the impeller and

Table 7. Performance of 15 hp 'Petti and Para'

Sl. No.	Static head Hs (cm)	Delivery head Hd (cm)	Time for 2 revolutions of energy disc(s)	Input power (hp)	Time taken by float to travel 60 m(s)	Float velocity v (cm/s)	0.8 v (cm/s)	Average area (cm^2)	Q (l/s)	Vd (cm/s)	$\frac{Vd^2}{2g}$ (cm)	Total head H (cm)	$\frac{QH}{75}$ (hp)	(%)
1	43.00	13.00	17.90	8.96	427	14.04	11.23	19388.20	217.75	136.10	9.44	65.44	1.899	21.19
2	56.00	13.00	17.20	9.38	415	14.46	11.57	16467.30	190.49	119.10	7.23	76.23	1.936	20.63
3	67.00	13.00	16.50	9.74	364	16.51	13.21	12973.60	171.34	107.10	5.85	85.85	1.960	20.11
4	73.00	12.00	16.40	9.83	346	17.32	13.86	11830.90	163.94	102.50	5.36	90.36	1.975	20.1
5	80.00	11.00	15.60	10.32	326	18.44	14.75	10726.40	158.22	98.89	4.99	95.99	2.025	19.63
6	85.00	11.00	15.20	10.57	306	19.63	15.70	9141.20	143.60	89.75	4.11	100.11	1.920	18.16

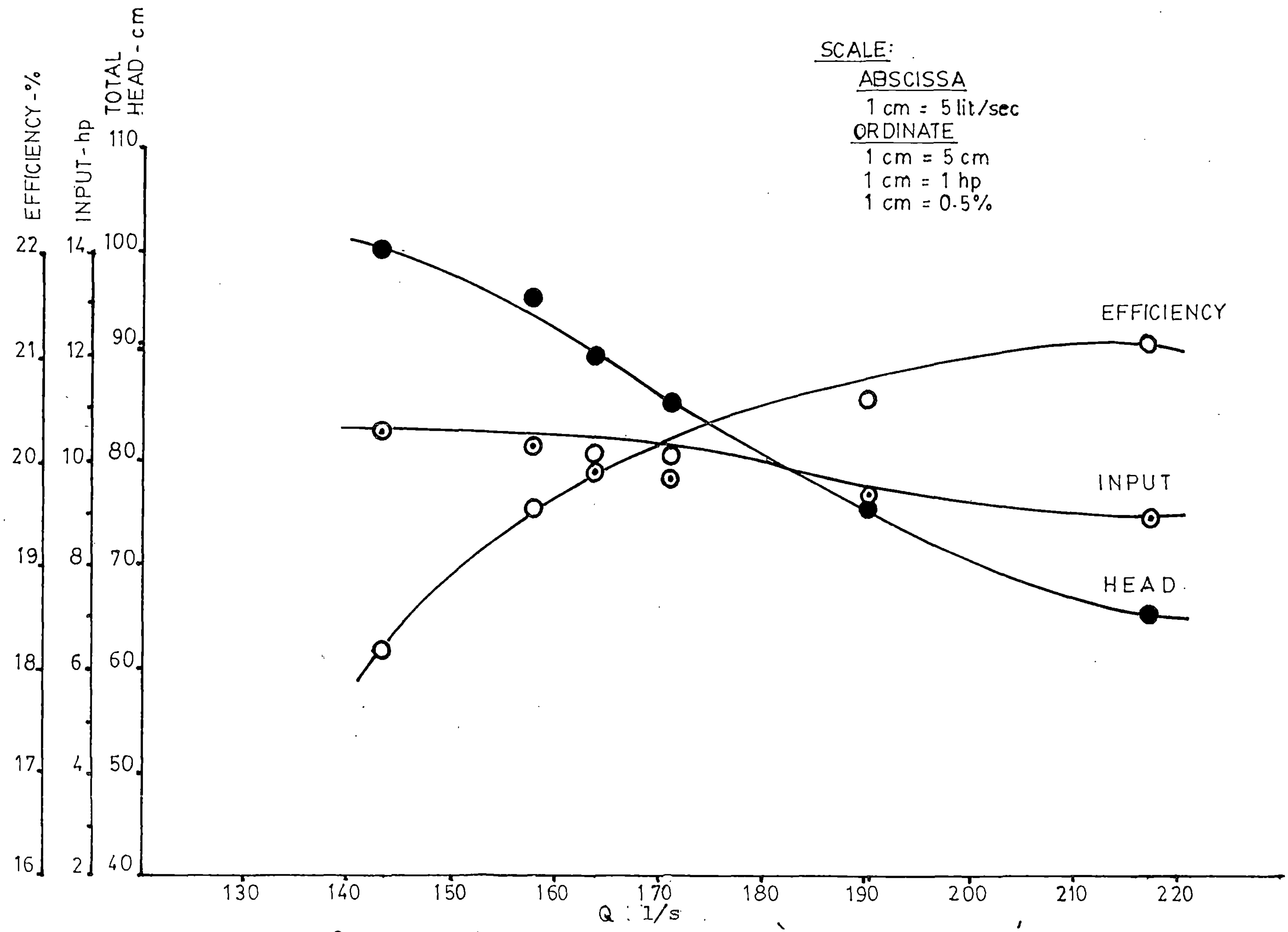


FIG. 9 CHARACTERISTICS OF 15 HP 'PETTI AND PARA'

the corresponding static head was 28 cm. Observations were taken by partially closing the outlet flap valve. The discharge head varied from 13 to 50 cm. The total head variation was from 63.15 to 97.48 cm. The discharge rate varied from 369.5 to 344.1 l/s. The water horse power gradually increased from 3.11 to 4.47 hp. The overall efficiency of the unit steadily increased from a value of 18.32 to 24.42 per cent (Table 8, Fig. 10).

The pump was tested by keeping the sump level 4 cm above the midpoint of the impeller. This test was done before conducting repair work and without adding bottom 'Para' to the unit. The static head was kept constant at 58 cm. The discharge head variation was from 11 to 48 cm, and the corresponding total head variation was from 81.1 to 114.1 cm. The discharge varied from 270.9 to 221.7 l/s. The input power varied from 17.98 to 19.98 hp with increase in discharge head. The efficiency curve was not showing a definite trend (Table 9, Fig. 11).

Table 10 and Fig. 12 shows the results of experiment conducted before the repair works on the 'Petti and Para', and without adding the bottom 'Para' to the unit. With a total head variation from 82.18 to 121.1 cm, the discharge rate also varied from 266.4 to 221.77 l/s. The efficiency

Table 8. Performance of a 20 hp 'Petti and Para'

Sl.No.	Time for 5 revolutions of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)					
1	18.00	16.98	39.50	30.00	76.00	369.50	28.00	13.00	22.15	63.15	3.1
2	17.70	17.21	40.00	32.00	80.00	366.90	28.00	18.00	22.15	68.15	3.3
3	17.40	17.57	40.00	32.50	81.25	362.20	28.00	30.00	21.59	79.59	3.8
4	17.30	17.67	39.50	31.50	79.70	360.60	28.00	38.00	21.39	87.39	4.2
5	17.10	17.88	39.00	31.50	80.70	349.90	28.00	45.00	20.46	93.46	4.3
6	16.70	18.31	39.00	32.00	82.00	344.10	28.00	50.00	19.48	97.48	4.4

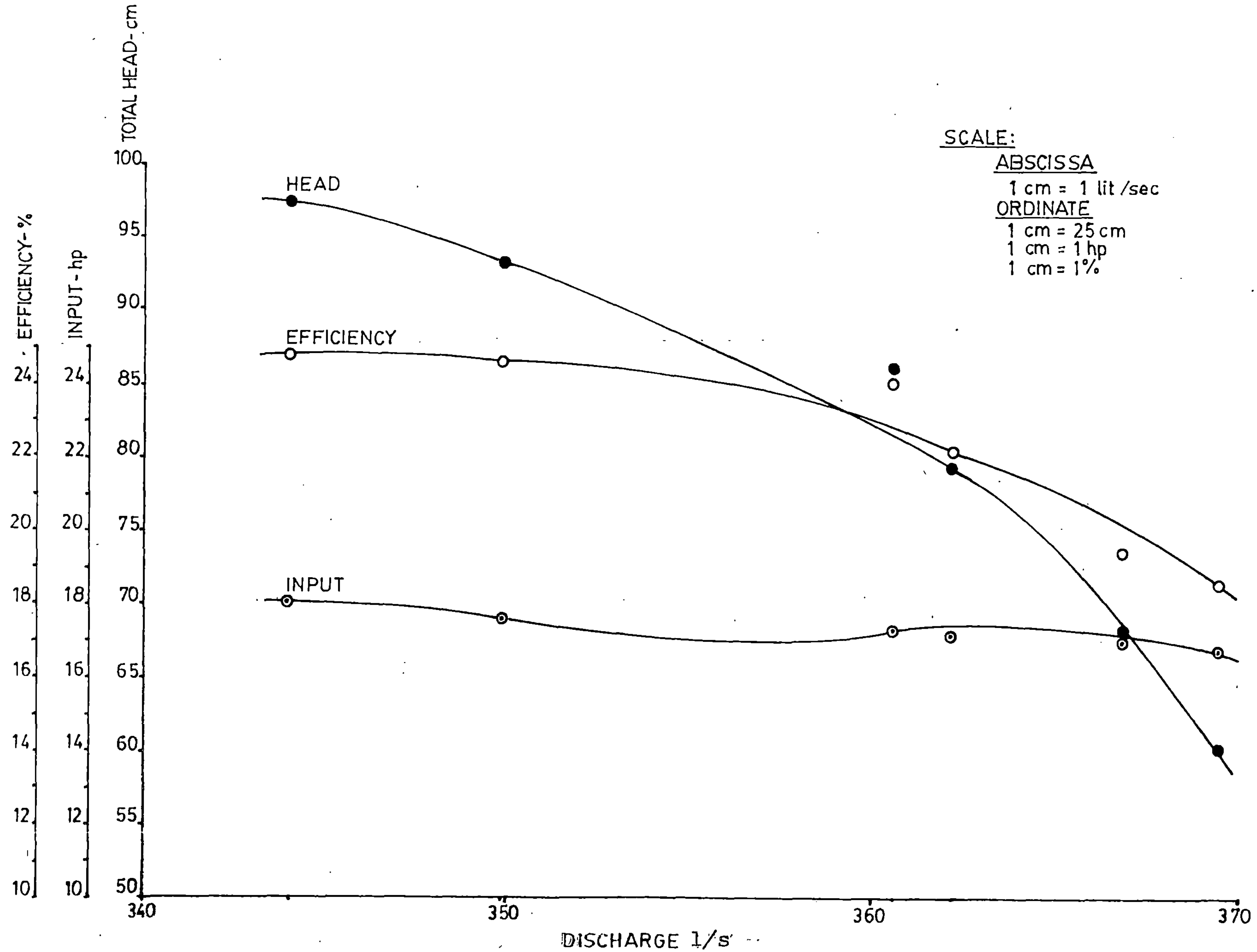


FIG. 10 CHARACTERISTICS OF 20 HP 'PETTI AND PARA'

Table 9. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time for 5 revolution of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)						
1	17.00	17.98	37.50	34.00	90.60	270.90	58.00	11.00	12.10	81.10	2.92	16.23
2	16.70	18.31	37.50	34.50	92.00	255.90	58.00	14.00	10.77	82.77	2.82	15.40
3	16.30	18.76	37.80	35.00	92.50	245.60	58.00	23.00	9.93	90.93	2.98	15.89
4	16.00	19.12	37.50	35.30	96.00	244.10	58.00	24.00	9.80	91.80	2.99	15.65
5	15.70	19.47	37.50	35.50	93.30	229.20	58.00	33.00	8.64	99.64	3.04	15.61
6	15.30	19.98	37.00	35.00	94.50	221.70	58.00	48.00	8.08	114.10	3.37	16.87

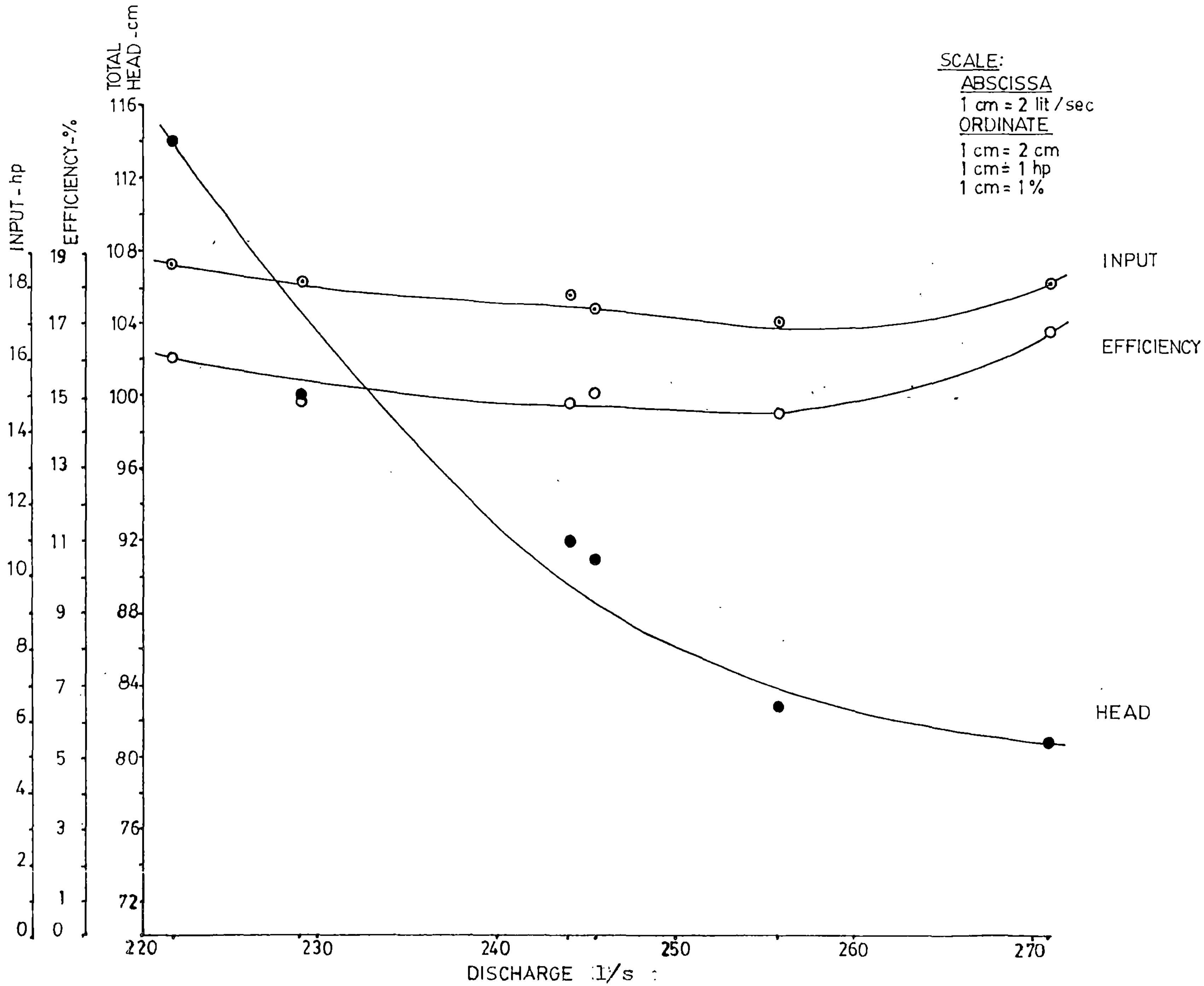


FIG.11 CHARACTERISTICS OF A 20 HP 'PETTI AND PARA'

74

Table 10. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time for 5 revolutions of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)						
1	16.70	18.31	36.00	32.00	88.80	266.40	58.50	12.00	11.68	82.18	2.92	15.95
2	16.70	18.31	35.80	32.00	89.30	263.60	59.00	15.00	11.43	85.43	3.06	16.71
3	16.30	18.75	35.50	32.00	90.10	259.24	61.00	23.00	11.10	95.10	3.29	17.54
4	16.00	19.11	35.20	32.00	91.00	243.02	62.00	33.00	9.72	104.70	3.39	17.74
5	15.80	19.35	35.50	32.50	91.50	241.18	63.00	43.00	9.57	115.57	3.72	19.22
6	15.30	19.98	37.00	35.00	94.50	221.77	65.00	48.00	8.10	121.10	3.58	17.92

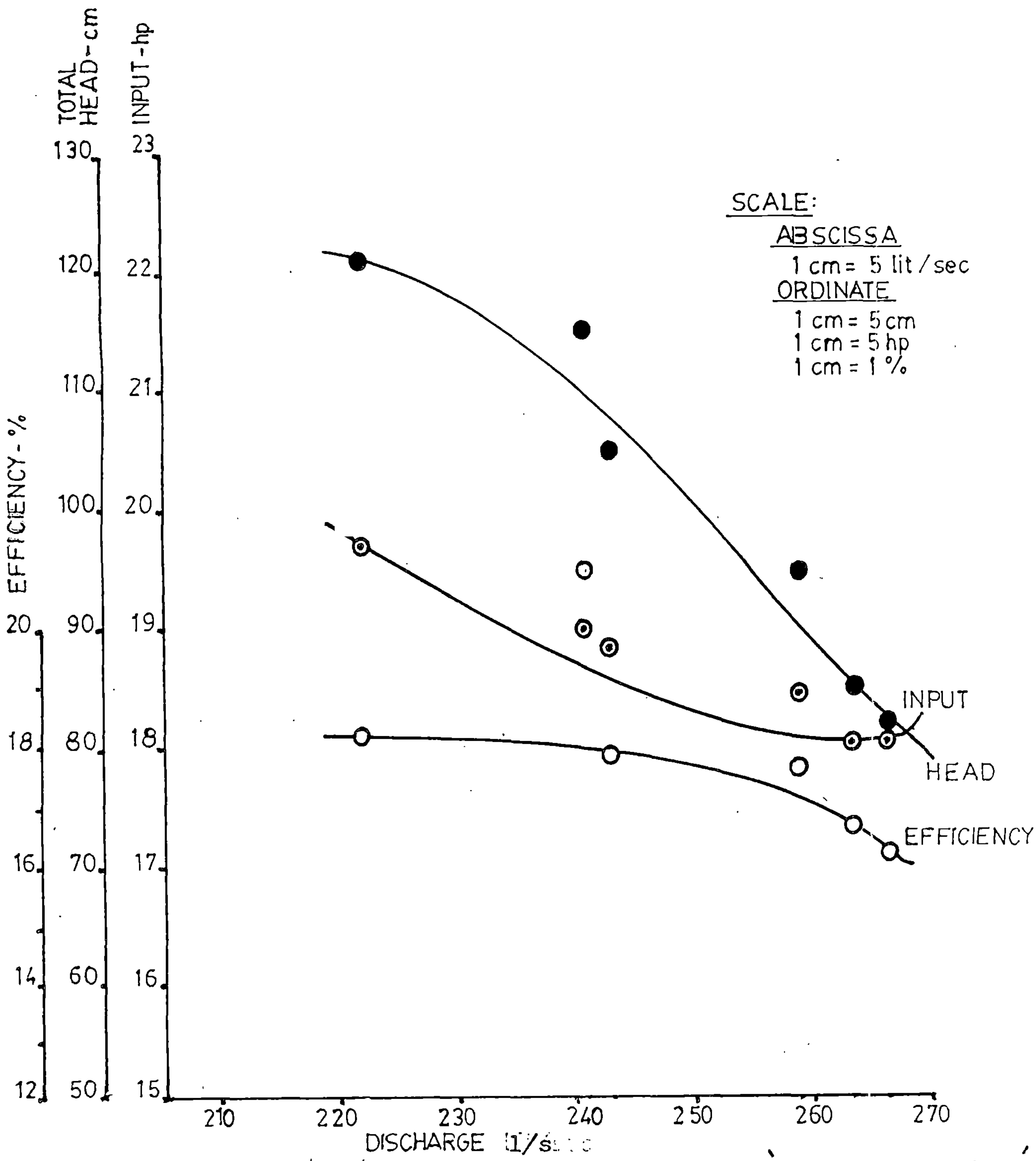


FIG.12 CHARACTERISTICS OF A 20 HP 'PETTI AND PARA'

of the unit increased with increase in total head. The input power variation was not very steep.

As the static head varied from 73 to 87 cm the discharge also varied from 257.5 to 215.1 l/s with increase in total head of 96.9 to 142.6 cm utilising a power of 18.76 to 21.84 hp and the efficiency of the unit dropped to 16.95 from 17.76 and then increased to 18.73 (Table 11, Fig. 13).

When the pump was tested with bottom 'Para' before conducting repair works for the unit, the discharge varied from 278.27 to 215.1 l/s with a variation in static head from 77 to 97 cm and the corresponding variation in total head was from 99.74 to 154.61 cm, utilising a power of 19.47 to 21.08 hp. The efficiency of the unit remained almost constant at about 20% (Table 12, Fig. 14).

When the static head varied from 80 to 127 cm before conducting repairs and with the bottom 'Para' in position, the discharge droped to 208.6 from a maximum of 278.3 l/s and the corresponding variation in total head was from 108.7 to 146.16 cm and the power consumption was from 19.57 to 21.84 hp with a drop in efficiency from 20.86 to 18.9% (Table 13, Fig. 15).

The experiment was later conducted after undertaking major repair works to the whole unit. The clearance between

Table 11. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time for 5 revelutions of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)						
1	16.30	18.76	36.00	32.50	90.30	257.50	73.00	13.00	10.90	96.90	3.33	17.76
2	16.40	18.64	33.50	32.50	91.50	241.40	75.00	15.00	9.59	99.60	3.21	17.22
3	16.30	18.76	35.20	32.50	92.30	225.18	78.00	20.00	8.34	106.34	3.18	16.95
4	15.20	20.38	35.10	32.00	91.16	223.75	81.00	30.00	8.24	119.28	3.56	17.46
5	14.50	21.08	35.00	32.50	92.82	222.31	85.00	35.00	8.13	128.13	3.79	17.98
6	14.00	21.84	34.50	32.00	92.75	215.10	87.00	48.00	7.60	142.60	4.09	18.73

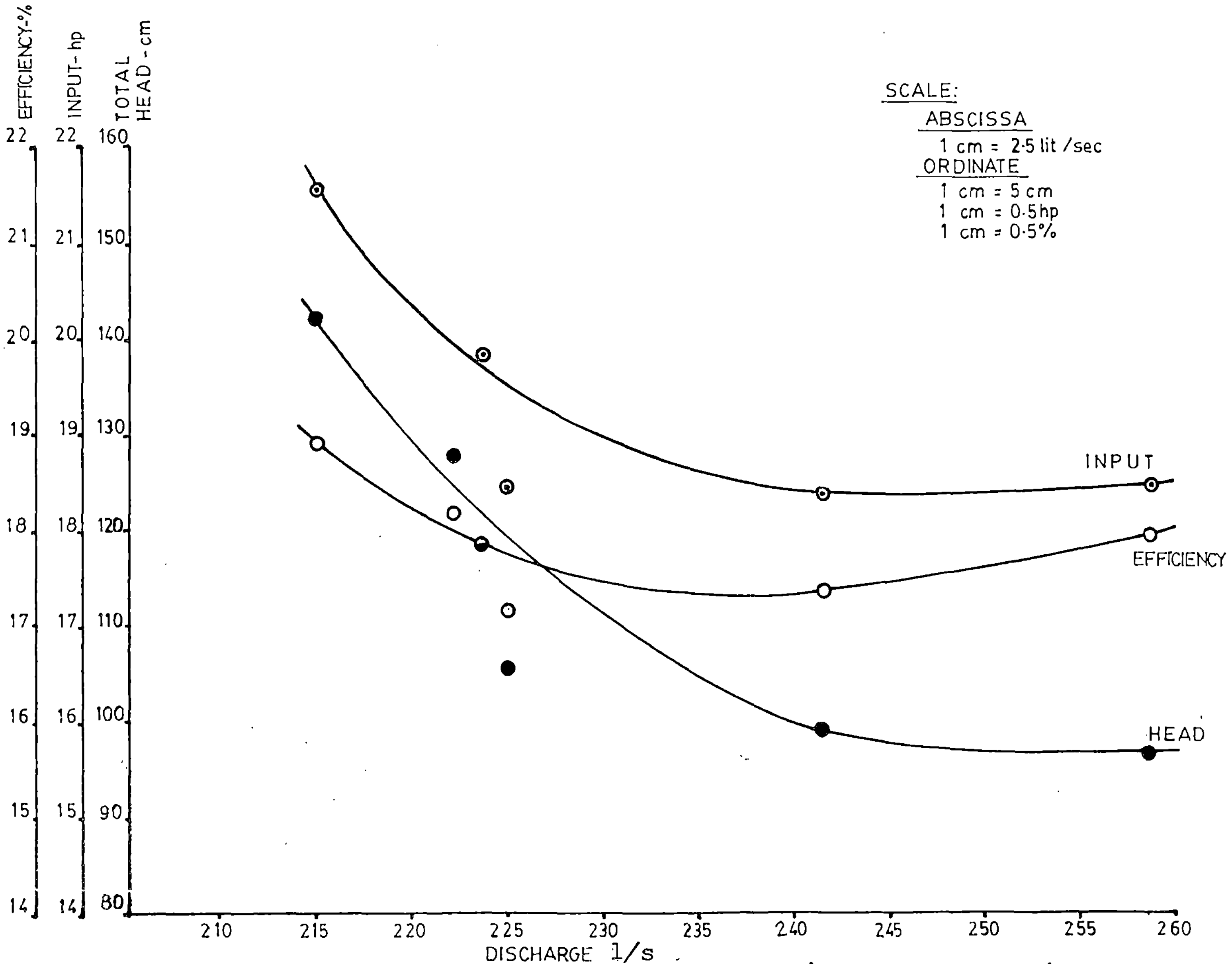


FIG. 13 CHARACTERISTICS OF A 20 HP 'PETTI AND PARA'

Table 12. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time of 5 revolutions of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (1/s)						
1	15.70	19.47	37.00	32.50	87.80	278.27	77.00	10.00	12.74	99.74	3.70	19.00
2	15.40	19.85	36.00	32.00	88.80	266.40	80.00	16.00	11.68	108.70	3.86	19.44
3	15.00	20.38	35.00	32.00	91.42	230.70	82.00	28.00	8.75	118.75	3.65	17.90
4	14.70	20.79	35.00	32.50	92.80	222.31	87.00	33.00	8.13	128.13	3.79	18.22
5	14.50	21.08	34.90	32.50	93.12	220.87	92.00	46.00	8.03	146.03	4.29	20.34
6	14.50	21.08	34.50	32.00	92.75	215.10	97.00	50.00	7.61	154.61	4.43	21.01

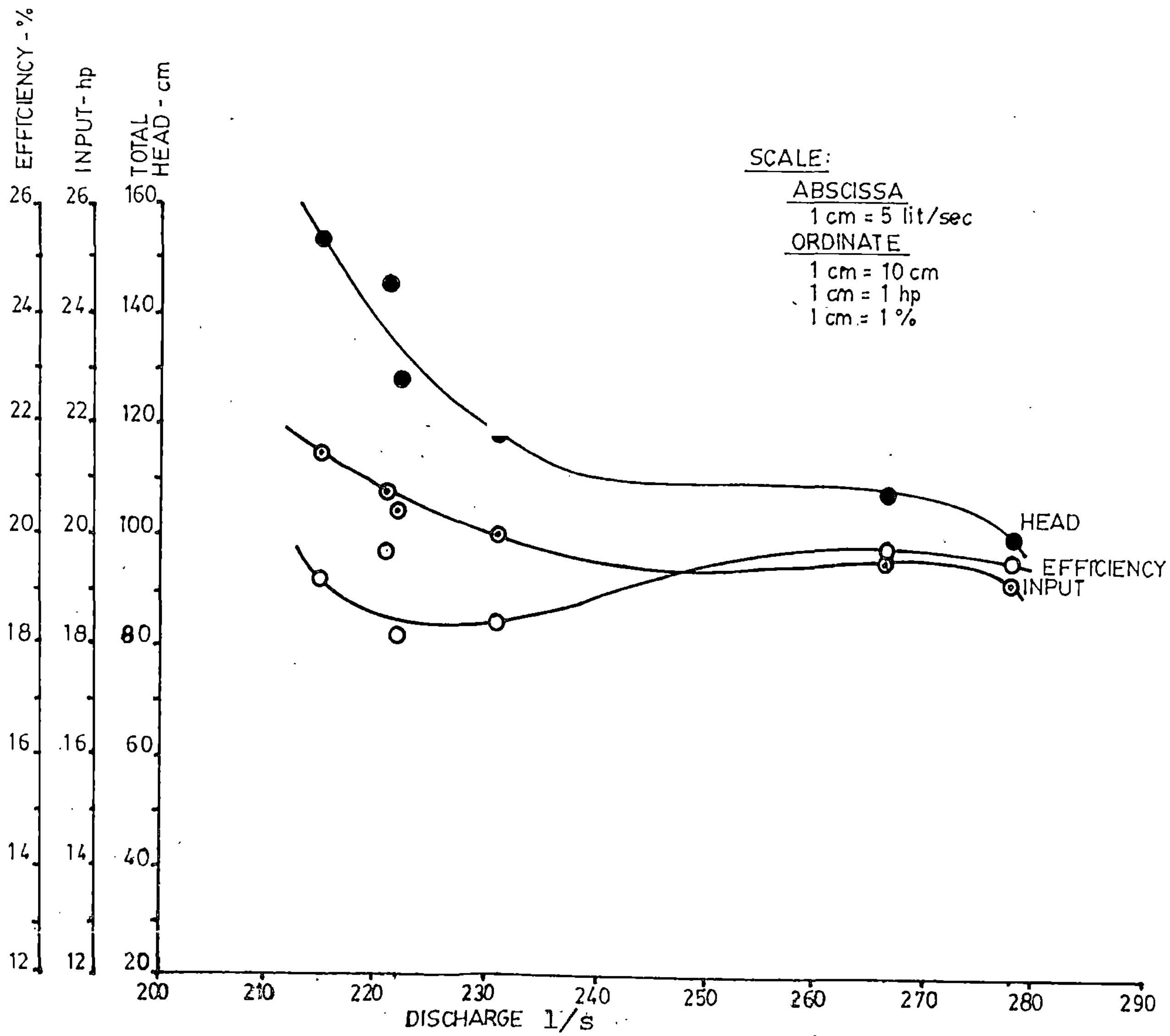


FIG.14 CHARACTERISTICS OF 20 HP 'PETTI AND PARA'

Table 13. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time for 5 revolutions of energy disc(s)	Input power (hp)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)						
1	15.60	19.57	37.00	32.80	87.80	278.30	80.00	16.00	12.70	108.70	4.03	20.56
2	15.50	19.72	36.00	32.50	90.30	260.50	88.00	16.00	11.17	115.20	4.00	20.28
3	15.30	19.98	35.80	32.50	90.80	245.70	93.00	14.00	9.93	116.93	3.83	19.17
4	14.90	20.52	37.50	35.00	93.30	241.10	101.50	13.00	9.56	124.10	3.99	19.45
5	14.70	20.79	37.20	35.00	94.10	221.80	114.00	12.00	8.10	132.10	3.91	18.80
6	14.00	21.84	38.50	37.00	96.00	208.60	127.00	12.00	7.16	146.16	4.07	18.64

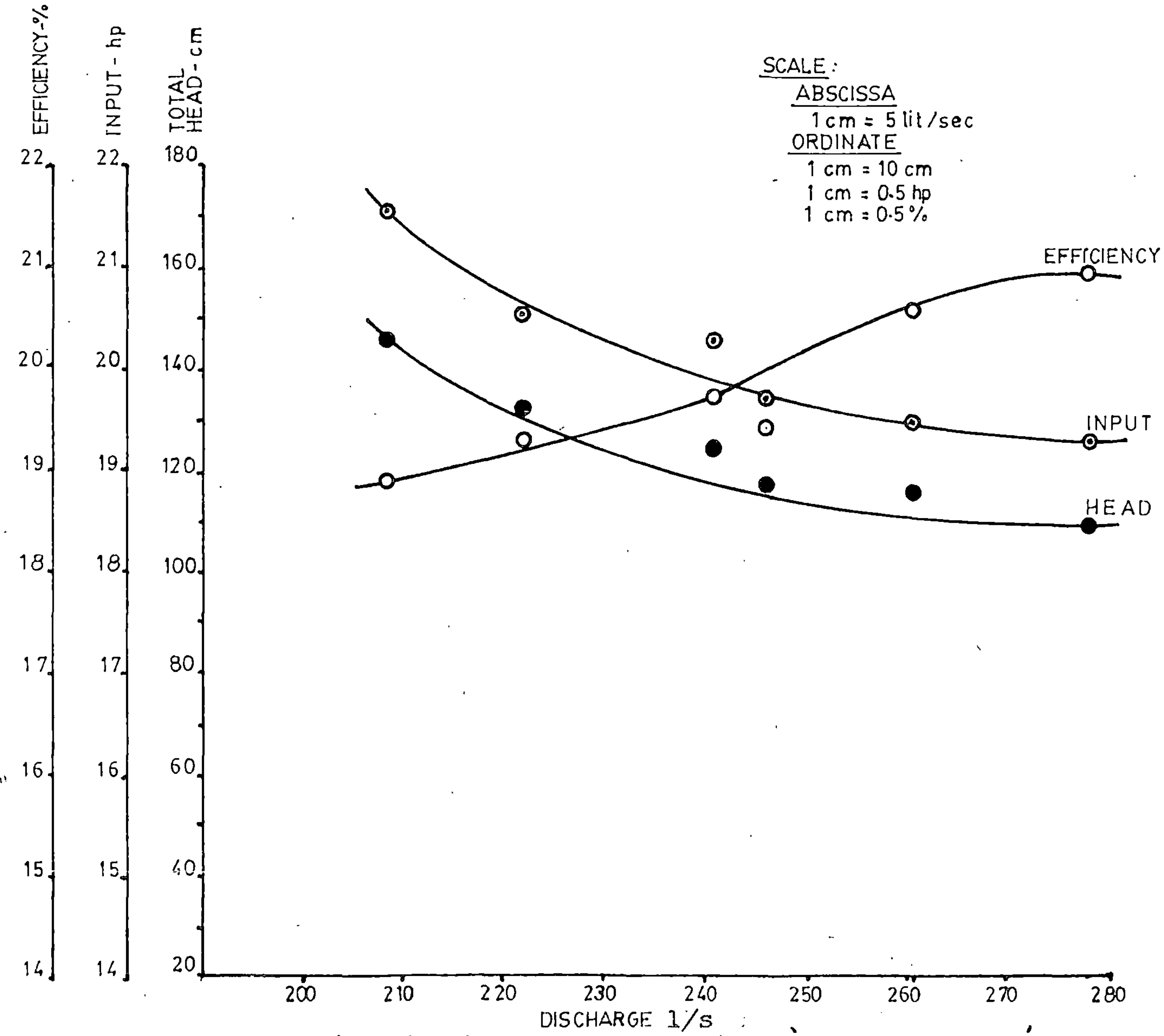


FIG.15 CHARACTERISTICS OF 20 HP 'PETTI AND PARA'

clearance was reduced to 5 mm at one side by welding additional piece of MS sheet. The old 'Para' was replaced by a new one.

When the static head varied from 36 to 103 cm without utilising a bottom 'Para', the discharge varied from 369.5 to 281.2 l/s with a variation in total head from 73.2 to 132.01 cm, and the minimum power consumption was 16.14 hp and the maximum was 19.76 hp, which was erratic in nature, and the efficiency varied from about 21% to about 26% with increase in head (Table 14, Fig. 16).

When the experiment was conducted with bottom 'Para' in position after conducting repairs to the unit and replacing the upper 'Para' with a new one, the discharge varied from 244.1 to 189.2 l/s with a static head variation from 102 to 124.5 cm and the corresponding total head varied from 126.8 to 140.4 cm consuming a power of 18.85 to 20.64 hp and the overall efficiency droped from 21.91 to 17.20% (Table 15, Fig. 17).

An experiment was conducted to assess the effect of outlet flap valve on pump performance. There was a marked increase in discharge of water when the flap was in fully open position (Table 16).

Table 14. Performance of 20 hp 'Petti and Para'

Sl. No.	Time for 2 revolution of energy disc(s)	Input power (hp) (K=30)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha cm	Hb cm	$\frac{Hb}{Ha}\times100$ (%)	Q (1/s)						
1	20.20	16.14	39.50	30.00	76.00	369.50	38.00	15.00	22.50	75.50	3.71	22.18
2	19.50	16.72	38.00	21.50	57.00	367.50	36.00	15.00	22.20	73.20	3.59	21.47
3	19.50	16.72	40.80	34.00	83.30	362.96	41.50	15.00	21.70	78.20	3.78	22.60
4	16.80	19.41	40.00	33.00	82.50	354.50	46.00	15.00	20.68	81.68	3.86	19.88
5	17.00	19.18	36.00	19.00	52.70	337.80	62.00	15.00	18.78	95.78	4.31	22.46
6	17.00	19.18	35.10	15.00	42.70	324.80	68.50	17.00	17.36	102.86	4.45	23.19
7	17.90	18.22	34.00	12.00	35.30	309.00	82.00	16.00	15.70	113.20	4.66	25.58
8	17.60	18.52	33.60	13.50	40.00	303.00	85.50	18.00	15.10	118.60	4.79	25.85
9	17.10	19.06	33.00	12.00	36.40	295.10	94.00	17.00	14.33	125.33	4.93	25.85
10	17.00	19.18	32.50	13.00	40.00	288.15	100.00	16.00	13.66	129.66	4.98	25.96
11	16.50	19.76	32.00	12.00	37.50	281.20	103.00	16.00	13.01	132.01	4.94	24.99

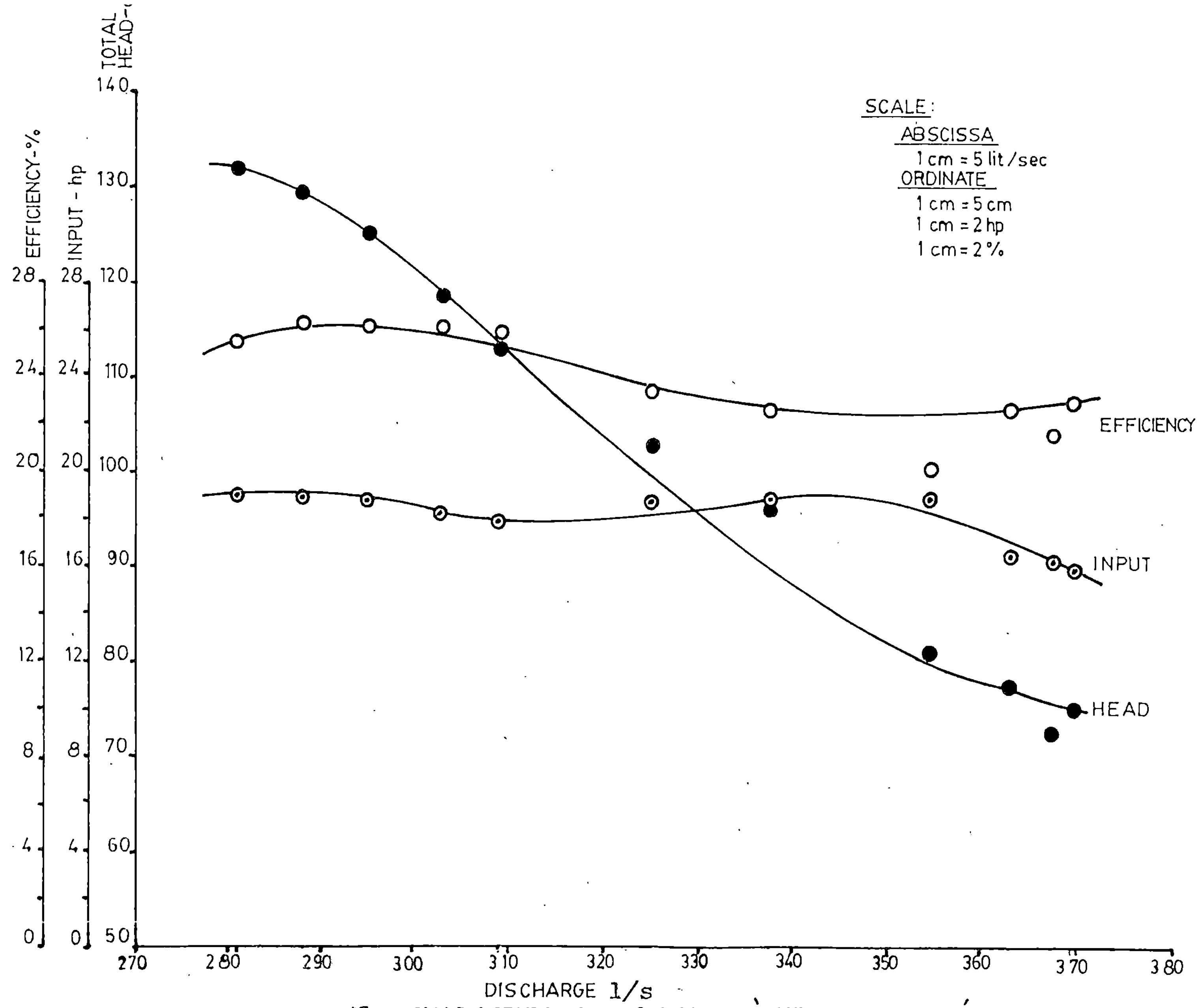


FIG.16 CHARACTERISTICS OF 20 HP 'PETTI AND PARA'

Table 15. Performance of a 20 hp 'Petti and Para'

Sl. No.	Time for 2 revolution of energy disc(s)	Input power (hp) (K=30)	Discharge				Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
			Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (1/s)						
1	17.30	18.85	37.50	35.30	96.00	244.10	102.00	15.00	9.80	126.80	4.13	21.91
2	17.00	19.18	37.50	35.50	94.60	226.20	109.00	15.00	8.41	132.41	3.99	20.80
3	16.40	19.88	37.00	35.00	94.50	221.70	112.00	13.00	8.08	133.10	3.93	19.76
4	16.30	20.00	37.00	35.30	95.40	203.90	117.00	13.00	6.84	136.10	3.64	18.19
5	16.00	20.38	37.00	35.50	96.00	192.00	120.00	12.00	6.07	138.10	3.54	17.36
6	15.80	20.64	36.80	35.40	96.00	189.20	124.50	10.00	5.89	140.39	3.55	17.20

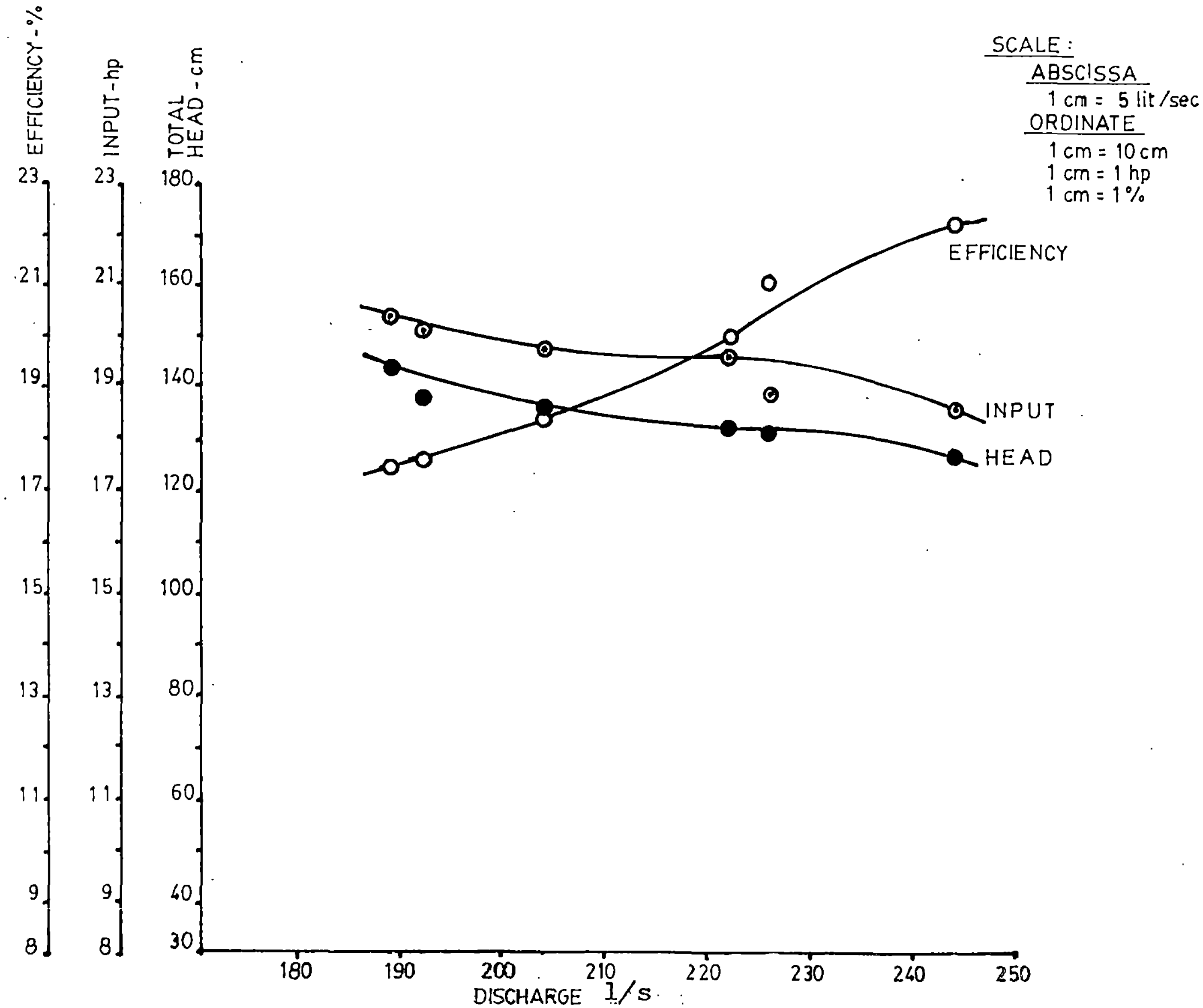


FIG. 17 CHARACTERISTICS OF 20 HP 'PETTI AND PARA'

Table 16. Table showing the effect of cutlet flap value on pump performance

Time for 2 revolution of energy disc(s)	Input power (hp)	Discharge Ha (cm)	Hb (cm)	$\frac{Hb}{Ha}$x100 (%)	Q (l/s)	Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)	
21.0	15.52	39.50	30.00	30.00	369.50	36.00	12.00	22.50	70.50	3.42	22.02	
17.0	19.18	41.00	34.00	83.00	366.18	44.00	15.00	22.06	81.06	3.95	20.59	Flap open
16.8	19.41	36.20	18.00	49.70	340.76	63.00	12.00	19.12	94.12	4.28	22.05	
17.0	19.48	36.50	13.00	35.60	345.20	65.00	14.00	19.61	98.61	4.54	23.67	
17.1	19.07	35.00	20.00	57.00	323.40	69.50	12.00	17.21	98.71	4.26	22.34	
19.5	16.72	38.00	21.50	57.00	367.50	36.00	15.00	22.20	73.20	3.59	21.46	
16.8	19.40	40.90	35.00	85.60	352.60	44.00	16.00	20.45	80.46	3.79	19.53	Flap on Hinged position
16.6	19.64	36.00	19.00	52.70	337.80	62.00	15.00	18.78	95.78	4.31	21.94	
16.5	19.76	35.10	15.00	42.00	324.48	66.00	16.00	17.21	99.21	4.29	21.71	
17.0	19.18	35.00	22.00	62.85	323.40	68.50	17.00	17.36	102.86	4.44	23.15	

All the above results show that the efficiency of 'Petti and Para' type of pumping unit was less in comparison with scientifically designed propeller pumps. Even by giving an allowance for 5% belt slip loss (Appendix-1) and 10 to 15% of motor loss, the efficiency of the pump will be of the order of 25 to 35%.

The experiments proved that even though the efficiency of the 'Petti and Para' was lower than the most scientifically designed units, it is on par with the propeller pumps developed by the Department of Agricultural Engineering, College of Technology, Pantnagar (1985) and Department of Agricultural Engineering, Kerala Agricultural University (1984).

The hub ratio of the pump tested was 0.52. This was in good agreement with values suggested by Addison (1956) and Stepanoff (1957). The number of impeller vanes was 5. As per values suggested by Stepanoff (1957) five vene impellers were utilised for pumps having specific speed below 9000 rpm. The chord spacing ratio of the pump at hub was 1.3 and that at outer periphery was 1.2. This was against the recommendations made by Stepanoff (1957).

4.2.3. Causes of low efficiency

A 'Petti and Para' impeller is having long vanes. The vanes are having different rotative speed along its radius.

The head produced by the vanes will be varing along the radius. This phenomenon makes liquid particles shift in a radial direction in the flow passage of the impellers and down stream of the vanes and decreases the efficiency of the pump. This condition can be avoided by keeping the term 'r Cu' a constant, thus making the liquid particle to flow over a cylinderical surface of appropriate radius.

The 'Para and Petti' joint was a 90° joint, and the transformation takes place from a circular cross section having an area of 2827.4 cm^2 to a rectangular cross section having an area 1760 cm^2. As per N A C A data, the loss at a 90° elbow was $V^2/2g$. In the 'Para and Petti' joint, loss due to sudden contraction also occurs. These losses are significant.

In a 'Petti and Para' installation the sump level was always varying. With reduction in submergence the water could not approach the suction endwith uniform velocity with out disturbance. In short in this type of pumping plant no attention was given for the prevention of vortices in the sump and thus causes the entry of air, causing drop in efficiency. Vortices also occurs due to different vane curvature and vane twist, along the diameter of the impeller. But addition of bottom para helps to improve the flow conditions in the sump.

The size of the sump of a 'Petti and Para' type of pumping plant was determined primarily considering the easyness of installation. It does not satisfy the recommendations on minimum submergence, floor clearance and wall clearance. The usual recommendation for floor clearance for best performance was D/2, where D was the diameter of the impeller. Any deviation from the standard values reduces the efficiency. Efficiency of this pumping unit reduces also due to the absence of inlet guide, outlet diffuser vanes, and also due to the absence of 8° divergence of diffuser casing.

The velocity head in the Petti was greater than 5 per cent of the total head of the pump and this was against the recommendations of Medici (1943).

The impeller of the 'Petti and Para' was fixed in its position by a stand tube, a thrust bearing and two other bearings. A slight misalignment will cause rubbing of the outer tip of the impeller with mild steel sheet protection in the para. This reduces the diameter of the impeller, which results in the reduction of discharge capacity.

The impeller blades of the 'Petti and Para' was made of 4 mm thick mild steel. The efficiency drop in such impellers may be in the order as suggested by Eckert (1944). Also the pumps utilised in Kuttanad conditions were prone to corrosion. Greater relative roughness causes reduction in efficiency.

The results revealed that the input power curve was not too steep. The factors contributing to this characteristics are (1) Number of vanes (2) Impeller vane length and (3) Pre-rotation.

For the same impeller profile an increase in the number of vanes reduces the brake horse power at partial clossure. But increasing the number of vanes has the effect of reducing pump efficiency.

Long impeller vanes or deeper impeller profile, result in a lower brake horse power ratio, because longer vanes causes liquid to rotate with the impeller more than short ones.

The effect of pre-rotation was unloading of impeller and reduction of shut off head and brake horse power, which occurs at partial capacities, because the liquic following a path of least resistance acquires pre-rotation in directio of impeller rotation.

4.3. Testing of 153 mm propeller pump

The propeller pump developed was tested at a constant static head of 120 cm. During testing the discharge varied between 39.64 and 13.34 l/s against a total head of 183.1 and 283.02 cm respectively. The efficiency variation was from 23.72 to 9.60. The power consumption was from 4.076 hp to a maximum of 5.21 hp (Table 17, Fig. 18).

In the original design, the discharge was fixed as 40 l/s, against a head of 1.5 m for 2 hp, with a 3 vane impeller. The maximum discharge obtained was 39.6 l/s against a head of 183.1 cm with 24.02 per cent efficiency for an input of 4.076 hp.

Even though the pump discharged the designed capacity, the power consumption was greater. The testing was conducted with a 10 hp motor. The efficiency of an electric motor at part-load will be very low. Also the power transmission loss, in this unit may be significant. Assuming a motor efficiency of 75% and transmission efficiency of 90%, the maximum efficiency obtained is 35.14%.

The efficiency of the 'Petti and Para' unit tested was about 25% (neglecting the motor loss, and transmission loss). The efficiency of the new axial flow pump developed was also of the same order. Again comparing with the three vane impeller developed by the Department of Agricultural Engineering, College of Technology, Pantnagar (1984), the newly developed pump was on par with them, even by neglecting motor loss and transmission loss.

Table 17. Performance of 153 mm Propeller Pump

Sl. No.	Time for 2 revolutions of energy disc(s)	Input power MHP (K=60)	Time taken to fill 1050 litre tank(s)	Discharge l/s	Hs (cm)	Hd (cm)	$\frac{Vd^2}{2g}$ (cm)	H (cm)	$\frac{QH}{75}$ (hp)	(%)
1	40.00	4.076	26.50	39.60	120.00	10.00	53.10	183.10	0.9670	23.72
2	36.70	4.44	29.70	35.30	120.00	47.00	42.17	209.17	0.9840	22.15
3	36.50	4.46	32.30	32.60	120.00	61.00	35.47	216.47	0.9410	21.06
4	36.00	4.53	36.50	28.78	120.00	90.00	28.03	238.03	0.9134	20.16
5	34.60	4.71	44.00	23.87	120.00	93.00	25.60	238.60	0.8749	18.56
6	33.90	4.81	42.80	24.52	120.00	117.00	20.35	257.35	0.8414	17.49
7	31.70	5.14	67.20	15.64	120.00	143.00	8.28	271.28	0.5657	10.99
8	31.10	5.24	76.40	13.75	120.00	159.00	6.40	285.40	0.5232	9.97
9	31.30	5.21	78.10	13.34	120.00	157.00	6.02	283.02	0.5033	9.6

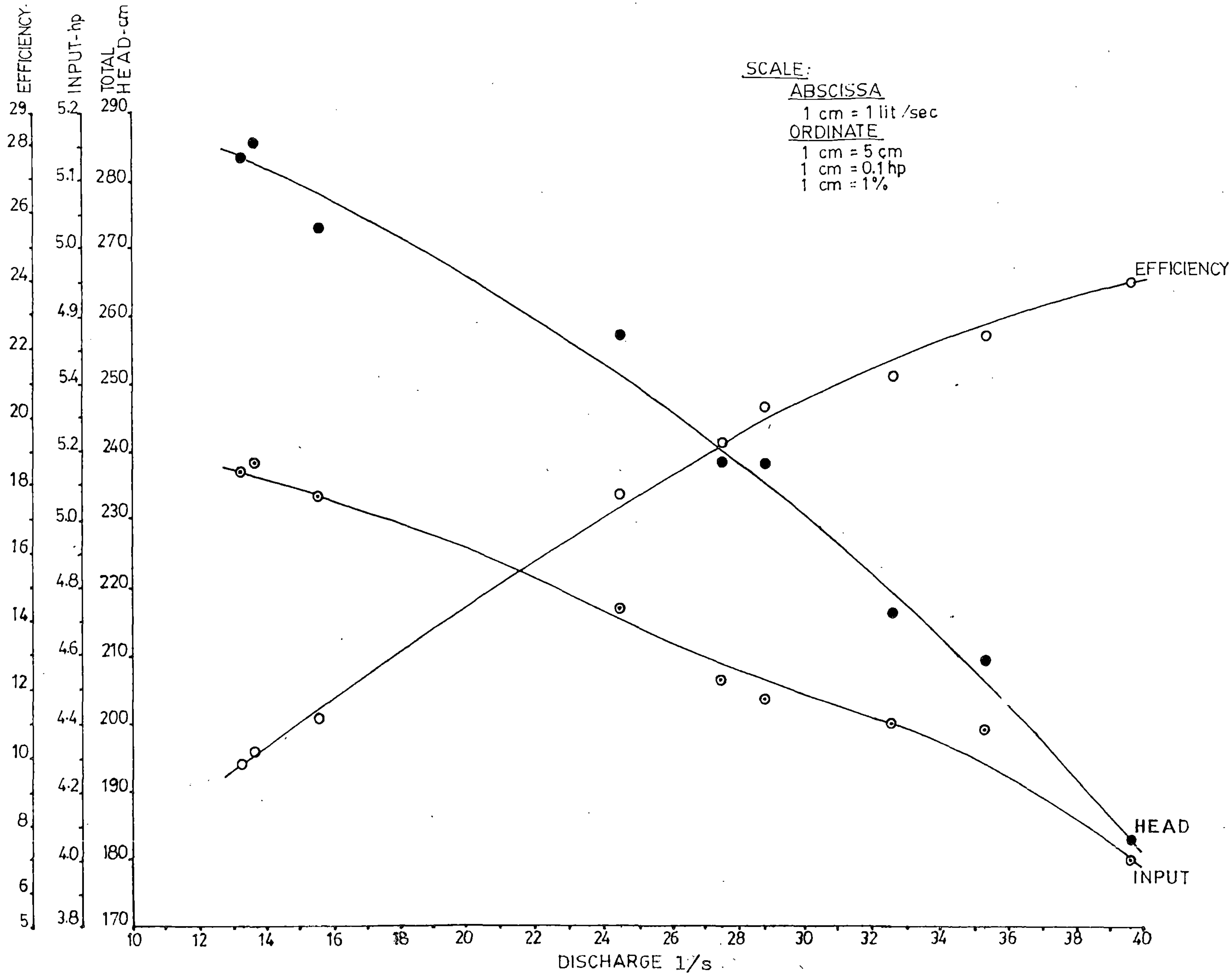


FIG 18 CHARACTERISTICS OF 15·3cm PROPELLER PUMP WITH 3 VANE IMPELLER

The hydraulic loss of the newly developed pump was not fully measurable. The reason for this was that there are so many factors contributing to the hydraulic losses. 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. Losses also occur at the impeller entrance and exit, and are usually called shock losses. Liquid flow in a pump tends to avoid shock by acquiring pre-rotation at the impeller inlet and by establishing a velocity gradient in the casing at the impeller discharge, there by cushioning the shock. The nature of the hydraulic loss at the impeller entrance, when liquid approaches at a high entrance angle, was that caused by a sudden expansion after separation. At the impeller discharge the loss was mostly caused by a high rate of shear due to low average velocity in the casing and high velocity at the impeller discharge.

In the newly developed 3 vane propeller pump, the power variation was between 4.076 hp (Q = 39.6 l/s, H = 183.1 cm) and 5.21 (Q = 13.75 l/s, H = 285.4 cm) ie. the variation in input hp was not high. This was a desirable feature of the pump, because attention of designers has been 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 was minimum.

As the blades were made from MS 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. The efficiency can be increased to a noticeable extent also by a direct coupled electric motor to the pump.

During the testing of the pump the quantity of flow obtained was almost same as the designed quantity, even by assuming 90% volumetric efficiency.

After redesigning and testing the pump at least for three times, necessary suggestions can be made to improve the performance of the existing 'Petti and Para'.

Summary

SUMMARY

'Petti and Para' is the popular pumping equipment used in Kuttanad and Kole lands of Kerala for dewatering agricultural fields. 'Petti and Para' is the local adaptation of the class of pumps called axial flow or propeller pumps. The objective of the project was to evaluate 'Petti and Para' type of pumping unit and to suggest necessary modifications.

A field survey was undertaken to collect information on the general characteristics of 'Petti and Para' and energy consumption for dewatering of the fields in different seasons.

A field pumping test was conducted on a 15 hp and 20 hp 'Petti and Para' using standard methods.

The field survey showed that 'Petti and Para' type of low lift pumping devices are running at rotational speed of about 300 rpm. The impeller diameter of 10 hp pump was 44 cm and that of a 50 hp pump was 79 cm.

For paddy fields having an area less than 20 ha, the mean energy consumption per hectare was 254.8 kWh for punja crop and 454.9 kWh for additional crop. For paddy fields having an area ranging from 20 to 100 ha, the energy consumption was 276.2 kWh/ha and 427 kWh/ha for punja and additional crop respectively. The mean energy consumption

for punja and additional crop were 398.3 kWh and 452.5 kWh per hectare for fields with an area greater than 100 ha.

The experiments on a 15 hp 'Petti and Para' showed that the unit was capable of discharging 217.75 to 143.60 l/s against a head of 65.44 to 100.11 cm with an efficiency from 21.19 to 18.16%.

The experiments on well maintained 20 hp 'Petti and Para' showed that the unit could deliver water at a rate of 369.5 to 281.2 l/s against a total head of 73.2 cm (static head 36 cm) to 132.01 cm (static head 103 cm) with an efficiency of about 21 to 26%.

These experiments proved that even after taking into consideration the transmission efficiency (95%) and motor efficiency (85%), the efficiency of the pump was of the order of 25 to 35% which was lower than the most scientifically designed units.

In order to suggest necessary improvements to the existing pump design a propeller pump was designed and fabricated considering the specific requirements of Kuttanad. The pump was designed as a high specific speed (280 rpm) pump operating at high rotational speed (1900 rpm), so that its physical dimensions were small. A small portable unit having an impeller diameter of 145 mm and 3 vanes was developed initially, so that the work could be extended for the development of bigger units, and to modify existing 'Petti and Para' in corporating the positive results from the initial design, applying principles of similitude. When tested at a constant static head of 120 cm, it delivered 39.64 to 13.34 l/s against a total head of 183.1 cm to 283.02 cm at an efficiency of 23.72 to 9.6 % (without considering motor efficiency and transmission efficiency). The power consumption was from 4.025 to 5.17 hp. The power unit utilised for testing was a 10 hp induction motor. The pump had to be redesigned at least three times. Due to lack of time, further work could not be undertaken.

Comparing with the most scientifically designed axial flow pumps, the efficiency of 'Petti and Para' is very low. Axial flow pumps having an efficiency greater than 80% is very common. At present the stationary elements of the pumps are made up off 'Anjili' wood (<u>Artocarpus hirsuta</u>), which is very costly and needs regular maintenance. So attempt should be made to find out a alternate material for construction preferably ferro cement, in addition to an effort, to modify the pump.

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Appendices

APPENDIX - 1

Transmission loss in 20 hp 'Petti and Para'

Estimated transmission loss (Pump rpm = 960)

rpm of pump	305	310	300	295
% slip loss considering belt thickness	6.10	4.52	7.6	9.14
% slip loss with out considering belt thickness	4.23	2.71	5.85	7.42

APPENDIX - 2

Discharge measurement of 15 hp 'Petti and Para' by velocity area method

The discharge measurement was made in a straight channel leading to the 'Petti and Para'. The channel had a width of about 2.7 m. A longitudinal length of 60 m was taken for discharge measurement. The 60 m longitudinal length was divided into 10 equal divisions, 6 m apart. Cross sections were plotted by dividing the channel cross section (width) into sections of 30 cm, by mid ordinate method. The depth in cm was recorded as the nearest whole number. The sections were plotted at each sump levels, because the channel water level varies with variation in pump sump level and also there exits the chance of erosion and accretion in channel bed. Details are given in following tables.

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in cm at 30 cm interval corresponding to a static head of 43 cm									Area of cross section cm^2
Section 1-1	46	59	67	80	83	72	72	67	57	18090
Section 2-2	56	60	73	76	85	82	76	63	54	18750
Section 3-3	44	63	77	73	80	82	78	75	63	19050
Section 4-4	47	56	72	79	76	82	76	72	63	18690
Section 5-5	50	56	71	83	80	79	78	78	71	19380
Section 6-6	47	60	67	78	90	81	85	83	73	19920
Section 7-7	51	67	81	83	82	80	80	68	58	19500
Section 8-8	56	64	79	86	89	77	75	61	60	19410
Section 9-9	67	77	84	86	90	88	76	70	58	20880
Section 10-10	52	63	72	86	84	89	86	74	62	20040
Section 11-11	53	64	77	82	88	83	76	70	59	19560
									Average area of cross section	19388.2 cm^2

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in cm at 30 cm interval corresponding to a static head of 56 cm									Area of cross section cm^2
Section 1-1	35	49	55	70	72	62	61	56	48	15240
Section 2-2	46	49	62	65	75	70	65	52	45	15870
Section 3-3	35	51	65	64	68	73	67	66	50	16170
Section 4-4	35	47	60	69	65	70	66	61	52	15750
Section 5-5	39	46	59	71	70	71	69	69	59	16590
Section 6-6	36	49	58	66	78	70	73	73	61	16920
Section 7-7	41	55	70	72	71	69	69	58	50	16650
Section 8-8	45	54	67	76	77	66	63	51	49	16440
Section 9-9	57	65	73	74	79	77	66	58	48	17910
Section 10-10	41	52	62	75	72	78	74	62	52	17040
Section 11-11	45	50	66	72	76	72	65	59	47	16560
					Average area of cross section					16467.3 cm^2

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in Cm at 30 cm interval corresponding to a static head of 67 cm									Area of cross section cm^2
Section 1-1	23	35	42	59	57	49	51	43	36	11850
Section 2-2	30	35	51	53	56	52	64	39	32	12360
Section 3-3	23	37	52	52	54	63	53	51	36	12630
Section 4-4	22	33	46	55	53	57	52	47	37	12060
Section 5-5	27	32	46	57	58	57	58	53	46	13020
Section 6-6	24	35	44	55	65	57	59	58	48	13350
Section 7-7	29	41	56	60	60	57	54	46	36	13170
Section 8-8	30	43	55	61	64	54	50	37	36	12900
Section 9-9	45	51	60	62	64	63	56	46	34	14430
Section 10-10	28	40	48	62	61	64	61	49	40	13590
Section 11-11	32	48	52	60	62	59	52	45	35	13350
					Average area of cross section					12973.6 cm^2

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in Cm at 30 cm interval corresponding to a static head of 73 cm									Area of cross section cm^2
Section 1-1	18	36	39	53	56	45	47	39	34	11010
Section 2-2	25	39	51	49	53	49	36	30	28	10800
Section 3-3	20	32	48	48	50	60	51	45	30	11520
Section 4-4	16	30	43	52	50	52	48	45	34	11100
Section 5-5	23	28	43	56	50	52	53	48	41	11790
Section 6-6	20	30	38	50	62	53	54	56	41	12090
Section 7-7	23	34	55	56	52	49	45	31	28	12060
Section 8-8	25	38	51	56	61	51	45	35	30	11760
Section 9-9	42	46	56	60	58	59	51	42	28	13260
Section 10-10	23	35	46	58	57	60	55	46	36	12480
Section 11-11	30	42	50	55	57	56	48	42	29	12270
					Average area of cross section					11830.9 cm^2

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in Cm at 30 cm interval corresponding to static head of 80 cm									Area of cross section cm^2
Section 1-1	16	28	37	49	48	42	46	32	29	9810
Section 2-2	25	33	45	46	46	46	43	28	22	10020
Section 3-3	18	26	44	44	47	55	46	42	28	10500
Section 4-4	15	26	36	47	46	50	43	38	34	10050
Section 5-5	18	23	39	50	51	48	48	46	36	10770
Section 6-6	16	26	35	46	58	49	52	48	39	11070
Section 7-7	24	30	46	53	50	50	47	38	25	10890
Section 8-8	25	32	48	51	56	48	40	32	25	10710
Section 9-9	30	42	50	54	55	55	49	35	26	11880
Section 10-10	19	25	38	53	56	54	52	43	30	11100
Section 11-11	25	39	44	55	52	50	46	39	23	11190
					Average area of cross section					10726.4 cm^2

Contd.

Appendix - 2. Continued

Details of channel section	Channel depth in Cm at 30 cm interval corresponding to a static head of 85 cm									Area of cross section cm^2
Section 1-1	10	24	29	43	44	34	38	28	23	8040
Section 2-2	17	29	39	38	39	40	39	23	18	8460
Section 3-3	10	22	36	40	40	49	41	37	21	8880
Section 4-4	7	20	32	41	40	42	39	33	27	8430
Section 5-5	12	19	33	44	42	42	42	40	31	9150
Section 6-6	10	19	30	41	51	43	44	44	33	9450
Section 7-7	16	26	41	46	45	43	40	33	21	9330
Section 8-8	17	28	41	46	51	41	35	25	20	9120
Section 9-9	23	36	45	48	49	49	41	32	18	10230
Section 10-10	13	27	34	47	48	50	46	36	25	9780
Section 11-11	20	32	38	47	47	45	39	30	20	9540

Average area of cross section 9141.2 cm^2

APPENDIX - 3

Free flow discharge values for 75 cm Parshall measuring flume

Ha (m)	Discharge (m^3/s)	Ha (m)	Discharge (m^3/s)	Ha (m)	Discharge (m^3/s)	Ha (m)	Discharge (m^3/s)
0.025	0.00532	0.140	0.07770	0.310	0.2678	0.540	0.6349
0.030	0.00707	0.145	0.08207	0.320	0.2812	0.550	0.6533
0.035	0.00899	0.150	0.08651	0.330	0.2951	0.560	0.6718
0.040	0.01106	0.155	0.09101	0.340	0.3090	0.570	0.6906
0.045	0.01329	0.160	0.09565	0.350	0.3234	0.580	0.7094
0.050	0.01566	0.165	0.1003	0.360	0.3378	0.590	0.7286
0.055	0.01815	0.170	0.1051	0.370	0.3526	0.600	0.7480
0.060	0.02078	0.175	0.1100	0.380	0.3675	0.610	0.7674
0.065	0.02355	0.180	0.1149	0.390	0.3827	0.620	0.7871
0.070	0.02643	0.185	0.1199	0.400	0.3979	0.630	0.8070
0.075	0.02943	0.190	0.1250	0.410	0.4137	0.640	0.8270
0.080	0.03252	0.195	0.1301	0.420	0.4294	0.650	0.8470
0.085	0.03575	0.200	0.1353	0.430	0.4455	0.660	0.8674
0.090	0.03907	0.210	0.1460	0.440	0.4615	0.670	0.8879
0.095	0.04249	0.220	0.1570	0.450	0.4781	0.680	0.9088
0.100	0.04604	0.230	0.1682	0.460	0.4946	0.690	0.9297
0.105	0.04966	0.240	0.1798	0.470	0.5115	0.700	0.9507
0.110	0.05339	0.250	0.1916	0.480	0.5286	0.710	0.9719
0.115	0.05721	0.260	0.2035	0.490	0.5458	0.720	0.9933
0.120	0.06114	0.270	0.2159	0.500	0.5632	0.730	1.0148
0.125	0.06515	0.280	0.2285	0.510	0.5808	0.740	1.0365
0.130	0.06924	0.290	0.2414	0.520	0.5986	0.750	1.0585
0.135	0.07343	0.300	0.2544	0.530	0.6167		

APPENDIX - 4

SAMPLE CALCULATION

Data taken is Ist set reading in Table-11

Upstream gauge reading in parshall flume Ha	=	36.00 cm (1.18 ft)
Downstream gauge reading Hb	=	32.50 cm
Percentage of submergence	=	$\frac{32.50}{36.00} \times 100 = 90\ \%$
Free flow discharge (Appendix - 3)	=	337.8 l/s
	=	11.932 ft/s
Correction value for submergence (Fig. 7)	=	1.35
Correction factor for 75 cm parshall flume	=	2.1
Actual discharge	=	11.932 - 1.35 x 2.1
	=	9.097 ft/s
	=	257.5 l/s
Hs	=	73.00 cm
Hd	=	13.00 cm
$\frac{Vd^2}{2g}$	=	10.90 cm
Total head $H = Hs + Hd + \frac{Vd^2}{2g}$	=	96.90 cm
Water horse power	=	$\frac{QH}{75} = \frac{257.5 \times 0.969}{75}$
	=	3.33 hp
The input power to the motor	=	$\frac{n}{t} \times \frac{3600}{K} \times \frac{1000}{746}$
Number of revolutions of energy meter disc(n)	=	5
Time taken for 'n' revolutions	=	16.3 s
Energy meter constant	=	80
Input power	=	$\frac{5}{16.3} \times \frac{3600}{80} \times \frac{1000}{736} = 18.76$ h
Efficiency of the unit	=	$\frac{3.33}{18.6} = 17.76$

APPENDIX - 5

Axial hydraulic thrust on 20 hp 'Petti and Para'

Maximum power input to the pump = $P_s = 20$ hp

Corresponding energy per second = $20 \times 75 = 1500$ kg-m/s

Mean blade radius = 0.4475 m

Mean blade velocity = $\dfrac{D_m N}{60} = \dfrac{\times 0.4475 \times 317}{60}$

= 7.43 m/s

Let Pt be the gross tangential thrust on all blades, and Pa the total axial thrust.

Then energy per second = $1500 = Pt \times 7.43$

Pt = 201.88 kg

Mean blade angle = Mean of inlet blade angle and outlet blade angle

= $23° = \tan^{-1} \dfrac{Pt}{Pa}$

Total axial thrust Pa = $\dfrac{201.88}{\tan 23} = 475.6$ kg

0.5 ton

Since the pump was set vertically, the gross load to be carried by the thrust bearing was the sum of axial thrust and the weight of the rotating element.

EVALUATION OF THE CHARACTERISTICS OF PETTI AND PARA' (AXIAL FLOW PUMP)

By

JOSE ABRAHAM

ABSTRACT OF THE THESIS

Submitted in partial fulfilment of the requirement for the degree

Master of Science in Agricultural Engineering

Faculty of Agricultural Engineering

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1988

ABSTRACT

'Petti and Para' is the most popular pumping equipment used in Kuttanad and Kole lands of Kerala for dewatering agricultural fields. It is a local adaptation of the class of pumps called axial flow or propeller pumps. It is fabricated by local blacksmith using their practical experience and is very popular in low head high discharge requirements especially for drainage purposes.

The project was undertaken to evaluate scientifically the 'Petti and Para' and to suggest improvement to its design.

Field survey had been conducted to study the general characteristics of the existing 'Petti and Para' and the energy requirements for punja and additional crop. Field pumping tests were conducted on 15 hp and 20 hp 'Petti and Para' using standard methods, to evaluate its characteristics

A small propeller pump was developed taking into consideration the specific requirements of Kuttanad, so that the results could be utilised in bigger units through the principles of similitude. The pump was designed as a high specific speed unit (280 rpm) operating at high rotational speed (1900 rpm), so that its physical dimensions were small.

The field survey showed that 'Petti and Para' are operating at low speed of 300 rpm. The most common type of 'Petti and Para' in use are 10 hp, 15 hp, 20 hp, 30 hp and 50 hp units. A 10 hp unit has an impeller diameter of 44 cm and 50 hp unit has an impeller diameter of 79 cm.

The average energy consumption for dewatering during punja crop was 309.8 kWh per hectare and that for additional crop was 444.5 kWh/ha.

The efficiency of a 15 hp 'Petti and Para' having an impeller diameter 52 cm was about 20%. It could pump water at a rate of 217.75 to 143.60 l/s against a total head of 65 cm to 100 cm.

A perfectly maintained 20 hp 'Petti and Para' could pump water at a rate of 369.5 to 281.2 l/s under total head of 73.2 to 132.01 cm. The efficiency of the unit varied between 21.47 to 25.96 %.

The newly developed propeller pump having an impeller diameter of 145 mm was tested at constant static head of 120 cm. It could pump water at a rate 39.64 to 13.34 l/s against a total head of 183.1 to 283.02 cm. The efficiency of the unit varied between 23.72 to 9.6. Input power varied between 4.076 to 5.21 hp, while utilising a 10 hp induction motor as power unit. Due to lack of time this work is inconclusive. Further elaborate studies are necessary for making specific recommendation for the improvement in the efficiency of 'Petti and Para'.