PERFORMANCE EVALUATION OF HIGH DISCHARGE LOW HEAD PUMPS

By

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THESIS

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DECLARATION

I hereby declare that this thesis entitled "**PERFORMANCE EVALUATION OF HIGH DISCHARGE LOW HEAD PUMPS**" 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 "**PERFORMANCE EVALUATION OF HIGH DISCHARGE LOW HEAD PUMPS**" is a record of research work done independently by **Kum. S. Rini Rani**, under my guidance and supervision and that it has not previously formed the basis for the award of any degree, fellowship or associateship to her.

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SYMBOLS AND ABBREVIATIONS

Agrl. Engg. - Agricultural Engineering
Agric. - Agriculture
Am. - American
Bull. - Bulletin
cm - centimetre(s)
Co. - Company
Dept. - Department
E - Efficiency
Ed. - Edition
Engg. - Engineering
et al. - and others
Fig. - Figure(s)
ha - hectare
hp - horsepower
ICAR - Indian Council of Agricultural Research
i.e. - that is
IRRI - International Rice Research Institute
J. - Journal of
KCAET - Kelappaji College of Agricultural Engineering and Technology
km - Kilometre(s)
lps (L/s) - litre(s) per second
Ltd. - limited
m - metre(s)
mech. - mechanical
mm - millimetre(s)
M.S. - mild steel
MSL - Mean sea level
m^3/h - cubic metre per hour
m/s - metre(s) per second
M.Tech - Master of Technology
rev/min - revolutions per minute
s - second(s)
Soc. - Society
° - degree

Introduction

INTRODUCTION

Adequate supply of water is one of the basic inputs to agriculture for stable and sustained production. Due to poor drainage, rice production in the coastal regions of India and other countries in South and South-East Asia is seriously affected. Paddy cultivation in Kuttanad and Kole lands of Kerala is made possible only after large scale dewatering operations. The most peculiar feature of paddy cultivation in these regions is the system of drainage of water from the bunded rice fields during the crop season. Both Kuttanad and Kole lands are lying below mean sea level.

The Kuttanad region represents low-lying lands measuring about 25 km east-west and 60 km north-south of the west coast of Kerala State and spread over Alappuzha, Kottayam and Pathanamthitta districts. The region can be divided into Karapadam, Kayal land, Kari land and uplands. The area of Karapadam region is estimated to be about 41,000 ha. The fields lie about 1 to 2 metres below the MSL. Kayal land occupies an area of about 8000 ha. The land is situated about 2 to 3 m below the MSL. Karilands are found in large isolated patches in Ambalapuzha, Shertallai, Vaikom and Kunnathunad taluks, covering an area of 20,000 ha. Usually in Kuttanad region only one crop of paddy (punja) is taken. So these rice tracts are often called `punja' lands.

The Kole region lies in Trichur and Malappuram districts extending partly along Trichur, Chavakkad, Mukundapuram and Ponnani taluks. The Kole areas are reclaimed lake beds below sea level and are lying parallel to sea coast. The area is subjected to inundation which keeps these lands under submerged condition for nearly seven months in an year. At present paddy is the only cultivation in these lands. The Kole region forms a substantial portion of the area under paddy cultivation in Trichur and Malappuram districts. The cultivation in Kole lands is done after pumping the water out of the area after the cessation of heavy rains.

In Kuttanad and Kole lands, soils are very fertile and have high production potential and are suitable for paddy cultivation during the intervening period, namely between the withdrawal of the north-east monsoon and the onset of south-west monsoon. Non-monsoon period paddy crop in these region have very specific requirements in relation to drainage and other water management practices to obtain a high level of sustained production. These water logged lands have to be drained in the shortest possible time to initiate seed bed preparation. The short time available for seed bed preparation need adoption of high discharge low head pumps.

Centrifugal and propeller pumps are two of the most popular mechanically operated pumping devices used in rice production at present. An inherent limitation of centrifugal pumping device is that its efficiency is considerably affected wherever the operating conditions differ from the optimum. A centrifugal pump designed for relatively higher pressure is incapable of giving corresponding increase in delivery rate even if the lift required is reduced to a minimum.

Finding the optimum flow condition is the objective of every pump designer. The design of an axial flow pump as cited by many authors is highly dependant on experimental data. Hence production of characteristic of the axial flow pump and impeller would require detailed information of the different parts of the pump.

Highly efficient factory made propeller pumps used in the Western countries are expensive requiring elaborate procedures in installation, operation and maintenance. The first attempt for large scale pumping in Kuttanad has been the introduction of an axial flow propeller pump called `Petti and Para' in the year 1918 AD by George Brendan, a British engineer. Brendan introduced the device utilizing locally available fabrication facility which confined to the blacksmithy and carpentry prevailing in the region. The device grew in popularity and spread over the entire region during 1920's. The introduction of this locally fabricated propeller pump nearly revolutionized drainage pumping in Kuttanad and Kole region of Kerala. The device, however is inefficient and not easily portable. Presently, the device is the major consumer of electricity in the agricultural sector of the region.

Irrigation and drainage operations on farms often require the use of portable, high-capacity low-lift pumps powered by small engines or electric motors. Axial and Mixed flow pumps used in South Asian countries especially Thailand are suitable for such low lift applications giving moderate to high efficiencies. The axial-flow pump has been popular in Thailand for several decades because (a) there are extensive areas where low-lift pumps are suitable, (b) technology is basically the same as that of outboard motor with its propeller reversed and placed inside a tube which serves as the casing of the pump.

The elaborate study on the performance characteristics of axial and mixed flow pumps imported from Thailand will help to identify the most suitable pump for meeting the specific requirements of irrigation or drainage in Kerala and similar situation elsewhere.

The specific objectives of the study are:

1. To study the design parameters and efficiency characteristics of high discharge low head pumps, namely, axial and mixed flow pumps acquired from Thailand.

2. To conduct laboratory testing of the above pumps at the specially designed testing tanks for the performance evaluation.

3. To conduct field testing of these pumps for performance evaluation.

4. Comparison with `petti and para'.

Review of Literature

REVIEW OF LITERATURE

2.1 Works done on low head high discharge pumps

Hathorn Davey and Company Limited (1926) conducted experiments on screw pumps and studied their advantage over centrifugal pumps. It had stated that though centrifugal pumps were excellent for high lifts they were at disadvantage for service conditions of low and variable heads. At low head or no head condition there would be serious danger of the driving agent being overloaded which would render the centrifugal pump unsuitable for uses like irrigation and drainage because of negligible heads.

Schmidt (1928) conducted tests for two vane impellers of the same inlet and outlet angles but of different projected areas. Two vane impeller with a projected vane area of about 63 per cent was the most efficient with an efficiency of about 79 per cent.

Kaplan and Lechner (1931) observed that for a given wetted area of the vane, the number of vanes should be minimum. With heavy vanes and low chord angle, the maximum number of vanes is almost fixed since adding vanes will restrict the free area of flow.

According to Naysken (1933) the efficiency of Archimedian spiral pump used in drainage of low level areas exceeded the theoretical value by 15-20 per cent although the overall mechanical efficiency of the combined set was not very much.

Sulzer Brothers Limited (1933) compared different impellers of screw and propeller type on the basis of specific speed. The efficiency of screw pump was reported to be higher than the low lift propeller pump. As per literature the following characteristics of propeller pump were noticed.

a. The guiding for water was very simple in propeller pump.

b. Losses were small and risk of chocking was very light.

c. If the head against pumping was too great for a single stage propeller two impellers could be fitted one behind the other on same shaft.

d. Non-return valve was necessary in the delivery side.

e. Pump should be started with delivery gate wide open.

Nillington (1935) in his article gave a good account on the measurement of flow and head in pump testing. He suggested that necessary corrections were to be applied to the pump central axis while calculating the head measured through pressure gauge. He had further recommended that flow measurement through weir was simple and economical but care should be taken in the measurement of readings.

Schlimbach (1935) studied the performance of a four vane impeller, with different vane settings. The vane curvature, remained the same, the discharge angle and inlet angle being changed by the same amount. The same head produced for all vane settings and thus is a function of vane curvature alone. Thus although the tangential component at the impeller discharge was higher at higher values of discharge angle, the tangential component at inlet increased by approximately the same amount. The peripheral velocity being the same at inlet and outlet no change in head resulted. Efficiency was good over a wide range of capacities.

According to Sulzer Brothers Limited (1935) the starting torque of propeller pump showed a parabolic curve with the branch rising more steeply than that of other types of pumps. It was reported that starting of such pumps with the gate valve closed would be dangerous due to excessive starting torque.

Eckert (1944) tested two axial flow impellers, one with airfoil vanes well streamed and polished, the other of the same solidity and camber line but made of stamped steel sheet welded to the hub. The performance of the two impellers proved identical. Later he observed that another impeller of the same airfoil pattern, but made of cast iron with the trailing edge about one-third thick, was lower in efficiency. The efficiency reduction was caused by the greater relative roughness of the cast iron vanes as compared to the polished alloy vane.

According to Stepanoff (1957), for a given impeller profile and impeller discharge angle, there is an optimum number of vanes above which pump efficiency drops appreciably. He conducted tests on a mixed flow pump of specific speed 8000. The impellers used have same impeller profile and discharge angle but different number of vanes. The test results showed that the impeller with three vanes obtained maximum efficiency of about 86.5 per cent.

Stepanoff (1957) designed a 61 cm adjustable vane axial flow pump in which the adjusting mechanism was arranged external to the pump. Except for the impeller the rest of the pump parts were the same as those of pumps, which have one piece impeller. The pump tests showed that, a 25 per cent reduction in capacity resulted in over 50 per cent saving in power. The pump can be started with the discharge valve closed and, when the impeller vanes are flat, the pump requires about 25 per cent of the maximum rated brake horse power.

According to Stepanoff (1957), Ingersoll-Rand Company developed 35 cm elbow type axial flow pump running at 690 rev/min. When the pump was tested without vaned diffuser the discharge varied from 150 to 375 L/s under a head of 3.8 to

8.0 m, with a variation in efficiency from 48 to 77 per cent and the power requirement ranging from 6 to 9.1 hp. When the pump was tested with a vaned diffuser, the discharge varied from 150 to 375 L/s under a head of 4.0 to 10 m with an efficiency of 50.2 to 80.5 per cent and the power requirement varied from 6.2 to 9.8 hp. Thus, the performance of the device improved substantially with the provision of the diffuser and its design. A reduction of head and brake horse power was noted for both the pumps at partial capacities caused by the fact that in an elbow-type axial flow pump liquid rotation is not prevented by the pump casing. In this respect the shape of the inlet pipe has a marked effect on the pump performance at partial capacity. The Ingersoll-Rand Company has also built a 40-cm mixed flow pump of specific speed 7000 with adjustable diffusion casing vanes to be used with a standard impeller. This has shown marked improvement in efficiency at reduced capacities. However the head increased at the same time and there was little or no reduction in brake horse power.

Addison (1976) proposed the design criteria for propeller pumps. According to him, a good axial flow pump should have the value of speed constant (K) between 2.0 and 2.7. K is defined as the ratio of outlet peripheral velocity to the free jet velocity. He conducted a series of trials using K values within the specified range. The results revealed that when K was 2, the efficiency was 56 per cent at a discharge of 86 L/s and head of 2.6 m. A slightly higher efficiency of 57.2 per cent was obtained for a speed constant of 2.7 with 84 L/s discharge and against 2.6 m head.

The International Rice Research Institute, Los Banos, Philippines (1979) designed and developed a portable and low cost axial flow propeller pump which could be coupled to a 5 hp engine or on a power tiller. The pump consists of an axial flow propeller rotating inside a 15 cm diameter casing and is capable of pumping water at a rate of 25 to 50 L/s at heads ranging from 1 to 4 m. The pump inlet was flared to 30° to reduce suction losses. Diffusion vanes were provided to increase the efficiency of the pumping unit. The pumping tests gave a maximum efficiency of 69.1 per cent at a discharge rate of 45 L/s against a head of 2.5 m at 2890 rev/min.

Calilung *et al.* (1982) measured the capacities of an axial flow pump 15 cm diameter and a centrifugal pump 10 cm diameter, for conditions of low lift (1 to 3 m) to compare their relative performance. Each pump was driven by the same 5 hp gasoline engine at approximately the same load. The test results showed that for lifts between 2.8 and 1.07 m the capacity of the axial flow pump was between 2 to 3 times greater than that of centrifugal pump. The two pumps have approximately the same capacity of 4 m lift. Economic considerations of the results illustrated that substantial fuel saving could be gained by using an axial flow pump in place of a centrifugal pump for low lift applications.

The Department of Irrigation and Drainage Engineering, College of Technology, Pantnagar (1982) developed a propeller pump for a discharge range of 30 to 65 L/s at a head of 1 to 3 m. The pump consists of 22 cm diameter impeller and 45 cm long diffuser tapering from 22 to 30 cm diameter. The efficiency of the pump varied from 50 to 65 per cent at a discharge rate of 65 to 45 L/s against a head of 1 to 2 m at 1440 rev/min. The power source was a 5 hp electric motor. Later, the work was further extended to develop a pump of higher discharge. The outer diameter of the propeller was 30 cm. A 7 vaned diffuser 45 cm long with a tapered diameter of 30 cm on the lower end to 37.5 cm on the other end was fabricated from aluminium alloy. The pump was tested at 1440 rev/min 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 3 m. The pump efficiency, however, was low and varied from 29 to 32 per cent.

Chetty (1983) designed a low lift high volume pump taking into account pre-rotation and vortex free or forced condition by considering the Euler's head. The four imepllers developed were the impeller with or without pre-rotation in the free vortex conditions of aerodynamic nature or with or without pre-rotation in the forced vortex condition. The design features included hub ratio of 0.4, speed constant of 1.98, capacity constant of 0.432, head coefficient of 0.128 and capacity coefficient of 0.218. The flow indices of speed constant, head coefficient, capacity constant and capacity coefficients were compared for different speeds, for one-fourth, half and full delivery openings and the pump with the optional provision of diffuser. The impact of delivery opening was distinctly noticed and the full open position alone was found to suit the design conditions. The free vortex impeller with pre-rotation and with the diffuser was identified as the unique one which could meet the design speed constant, capacity constant, head coefficient and capacity coefficients. The discharge obtained with this pump under a total head of 4.3 m was 193 L/s under full open condition.

Sasi (1984) designed and developed a propeller pump with three bladed impeller which was tested at two levels of water, namely, 20 cm above the impeller and 10 cm above the impeller. For the above two conditions at designed head of 1.5 m, the efficiency obtained were 33 per cent and 29.5 per cent, respectively at discharges of 121 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 and 1.54 m at discharge rates of 124.88 and 114.1 L/s respectively. The maximum discharge obtained was 169.19 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 Russian Industry manufactured axial flow pumps of types OI and 011 (Cherhasshy, 1985). In type OI pumps the vanes are rigidly fixed on the hub, while the type 011 pumps were equipped with adjustable vanes. For impeller diameter ranging from 295 to 1850 mm, the discharge of these pumps varied from 4.450 m^3/h to 54,700 m^3/h at a head of 1.9 to 20.9 m and with an efficiency of 81 to 86 per cent respectively.

Taneja *et al.* (1986) designed and fabricated two types of propeller pumps, inclined propeller pumps and vertical propeller pump. The inclined pump was able to lift water at the rate of 44 to 35 L/s against total head ranging from 2 to 2.9 m and correspondingly the efficiency varied from 50 per cent to 80 per cent at 2600 rev/min. A discharge rate of 56.3 to 43 L/s for heads of 1.67 to 3.68 m was obtained at 3000 rev/min and the efficiency varied from 57 to 83 per cent. For vertical propeller pumps at 2800 rev/min, the discharge varied from 32 to 19 L/s at static heads ranging from 84 to 250 cm and the efficiency varied from 65 per cent to 31 per cent.

Anilkumar (1987) designed and fabricated an axial flow pump for a discharge of 250 L/s and head 2 m. Cast iron and mild steel replaced the wooden parts of the conventional model of "petti and para". The petti was made of angle iron and mild steel sheets and para was made from cast iron sheet metal. The pump was tested at 960 and 730 rev/min. The discharge of the pump varied from 161 to 210 L/s against total heads of 98.5 to 160 cm and the efficiency obtained varied from 28.64 to 24.78 per cent at 730 rev/min. The maximum efficiency obtained was 28.72 per cent, at a discharge of 176.5 L/s and against a head of 152.5 cm. At 960 rev/min the maximum efficiency was 23.89 per cent at a discharge of 258.5 L/s and the total head was 152.5 cm.

A small and compact pump was developed by the Agricultural Engineering Division of the Philippines (Anonymous, 1987) and is designed specifically for low lift applications. The Sipa pump, as it is nicknamed, is a low cost propeller or axial flow pump ideal for raising water 1-2 m high. It is easily installed, portable and can be fabricated from locally available materials. A 15 cm diameter pump powered by a 7 hp engine has an output of about 40 L/s for a lift of 1.5 m. This capacity is 2-3 times higher than that of either a centrifugal pump or an axial flow pump using a boat propeller having the same lift and engine power. The difference is due to the specially designed propeller which provides high efficiency for low lift pumping.

Abraham (1988) conducted a field survey and collected information on the general characteristics of `petti and para'. Field pumping tests were conducted on 15-hp and 20-hp `petti and para' using standard methods. The 15-hp `petti and para' was capable of discharging 217.75 to 143.63 L/s against a head variation of 65.44 to 100.11 cm with efficiency varying from 21.19 to 18.16 per cent. The experiments on 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 to 132 cm with efficiency varying from 21 to 26 per cent. Later a propeller pump was designed and fabricated considering the specific requirements of Kuttanad. 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 to 283.02 cm and corresponding efficiency ranged from 23.72 to 9.6 per cent. The power unit was a 10 hp induction motor.

Saji (1994) conducted a study to evaluate the performance of `petti and para' for various designs of the impeller to suggest the optimum number of vanes on the impeller and the optimum working speed for better efficiency. The tests were conducted for the 15-hp `petti and para' using four-bladed, five-bladed and six-bladed impellers for various speeds of operation of the pump. The hub ratio was same for all the three impellers. Analysis of the efficiency-discharge curves indicated that, for all the speeds compared, the efficiency of the five-bladed impeller was maximum and that of the six-bladed impeller was minimum. The detailed analysis of the performance curves of the pump clearly established that the optimum number of vanes on the impeller of 15-hp `petti and para' is five and the optimum working speed is 330 rev/min. A maximum efficiency of 30.09 per cent was obtained at 330 rev/min for five-bladed impeller and the discharge rate was 369.88 L/s against a static head of 46 cm.

Chaiyaphol (1996) studied the performance of 15-cm Thai made axial flow pump and mixed flow pump. Both the pumps were tested under a total static head of 1 m for different speeds. When the axial flow pump was tested over a speed range of 1800 rev/min to 2600 rev/min the discharge rate varied from 14.4 to 30.5 L/s. The mixed flow pump was tested over a speed range of 575 to 1000 rev/min. The efficiency varied from 25.9 per cent to 56.0 per cent and discharge varied from 11.75 to 51.09 L/s. The maximum efficiency of 56.0 per cent was obtained at 1000 rev/min and the discharge rate was 51.09 L/s.

2.2 Theoretical considerations

2.2.1 Basic theory

An axial flow pump sometimes called a propeller pump, develop most of its head by propelling or lifting action of the vanes on the liquid. In axial flow pumps no centrifugal head is impressed on water by centrifugal force. It has a single impeller with the flow entering axially and discharging nearly axially.

The mixed flow pump, on the other hand, employs a chamber or impeller or vane design to impart both an axial and radial motion to water travelling down stream. This powerful motion is created by lifting the water with an impeller while simultaneously giving an outward thrust to the water. Above the impeller, internal curved vane arrangement known as an upper diffuser section guide the water to the column and straighten the flow. By combining axial and centrifugal motion, the mixed flow pump generates greater head than is possible with a single motion axial pump.

2.2.2 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.2.1). The shape of such vector diagram is triangular and they are called velocity triangles. In the design of a pump attention is focussed on the entrance and discharge parts of the impeller vanes and the velocity triangles are called entrance and discharge triangles. The relative velocity of flow (w) is considered relative to the impeller. The absolute velocity of flow (c) is taken with respect to the pump casing and is always equal to the vectorial sum of the relative velocity and the peripheral

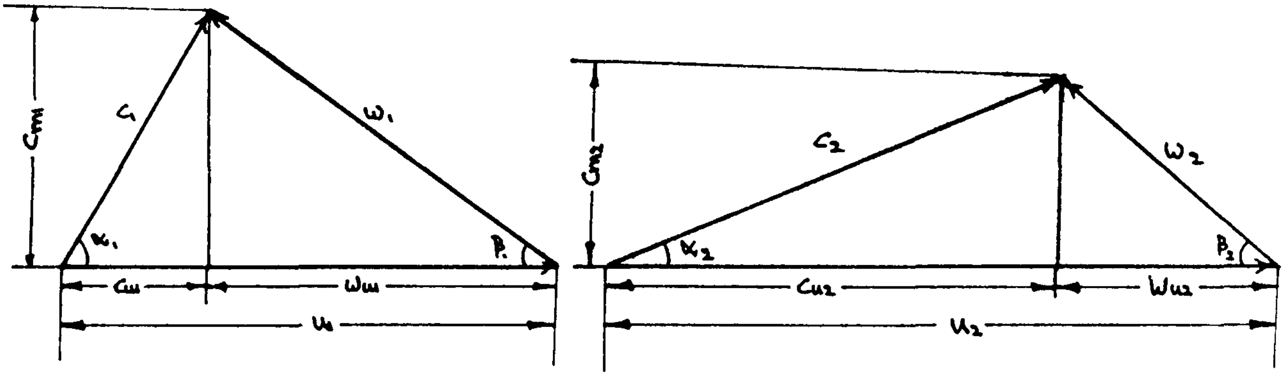


Fig.2.1 a Entrance velocity triangle b Discharge velocity triangle

velocity (u) of the impeller. Subscripts 1 and 2 refer to the entrance and discharge respectively. Tangential components of relative and absolute velocities are given another subscript, u. Components of the absolute velocity normal to the peripheral velocity are designated as c_{m1} and c_{m2} for entrance and discharge diagrams.

2.2.3 Theoretical head

An expression for the theoretical head of a pump is obtained by applying the 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 axis.

Torque exerted by the impeller on the fluid = Angular momentum at discharge - (Angular momentum at entrance)

Let T denote the torque exerted by the impeller on the fluid, c_1 absolute velocity at entrance and c_2 absolute velocity at discharge, dm mass of a small element of liquid, dt the time interval, then

$$T = \frac{dm}{dt} (r_2 c_2 \cos\alpha_2 - r_1 c_1 \cos\alpha_1)$$

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

where,

Γ = Specific weight of liquid, kg/m^2s^2;

Q = Mass of liquid entering/sec, m^3/s and

g = Acceleration due to gravity, m/s^2

Therefore,

$$T = \frac{Q\Gamma}{g} (r_2c_2\cos\alpha_2 - r_1c_1\cos\alpha_1)$$

and power transferred from impeller to water, $P = T\omega$

where,

ω = Angular velocity

Hence,

$$P = \frac{Q\Gamma}{g} (\omega r_2c_2\cos\alpha_2 - \omega r_1c_1\cos\alpha_1)$$

Also from the velocity trangles

$u_2 = \omega r_2$ $c_{u2} = c_2\cos\alpha_2$ $u_1 = \omega r_1$ and $c_{u1} = c_1\cos\alpha_1$

$$\therefore P = \frac{Q\Gamma}{g} (u_2c_{u2} - u_1c_{u1})$$

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.

The theoretical head imparted to the fluid $H_i = \frac{P}{\Gamma Q}$

$$\therefore \quad H_i = \frac{u_2 C_{u2} - u_1 C_{u1}}{g}$$

Since all hydraulic losses between the points where the actual total dynamic head of a pump is measured have been disregarded, the head H_i is a theoretical head and the equation is known as Euler's equation. 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

$$H_i = \frac{u_2 C_{u2}}{g}$$

By geometric substitution from the velocity triangle

$$w_2^2 = c_2^2 + u_2^2 - 2u_2c_2\cos\alpha_2$$

$$w_1^2 = c_1^2 + u_1^2 - 2u_1c_1\cos\alpha_1$$

$$\therefore \quad Hi = \frac{c_2^2 - c_1^2}{2g} + \frac{u_2^2 - u_1^2}{2g} + \frac{w_1^2 - w_2^2}{2g}$$

2.2.4 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 caused by the energy gradient drop. The circular component of flow forms a vortex motion. The type of vortex depends on the velocity and pressure distribution.

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 $u_2 = u_1$ and $c_{m1} = c_{m2}$, the Euler's equation reduces to

$$Hc = \frac{c_{u2}^2 - c_{u1}^2}{2g} + \frac{w_{u1}^2 - w_{u2}^2}{2g}$$

Again assuming first that the liquid approaches the impeller without prerotation ($c_{u1} = 0$ and $w_{u1} = u_1$), the above equation reduces to

$$Hc = \frac{u_2^2}{2g} + \frac{c_{u2}^2}{2g} - \frac{w_{u2}^2}{2g}, \qquad \text{But} \quad c_{u2} = u_2 - w_{u2}$$

$$\therefore Hc = \frac{u_2^2}{g} - \frac{u_2 w_{u2}}{g}$$

This equation indicates that head is generated through the vortex motion and the flow through the impeller is caused by the energy gradient drop $\frac{u_2 w_{u2}}{g}$

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 action of the impellers is the same in a straight radial flow and a straight axial flow pump. This action consists in producing a forced vortex which is superimposed on a radial outward flow in the first case and upon a uniform axial flow in the latter case.

The mixed flow impellers occupy an intermediate position between the radial flow and axial flow types. Therefore all the deductions for radial and axial flow pumps apply as well to the mixed flow pumps. In axial flow pumps the velocity component c_{m1} is not equal to c_{m2}. Normally c_{m1} is greater than c_{m2}. In drawing discharge angles the same procedure is followed as for axial flow pumps. It should be noted that with mixed flow impellers a vane having a prescribed impelling ratio, may have a discharge angle at the hub lower than the entrance angle. This depends on the relative values of c_{m2} and c_{m1} and the impeller profile.

Materials and Methods

MATERIALS AND METHODS

Materials used for the study and methodology adopted are discussed in this chapter.

3.1 Brief description of mixed and axial flow pumps

The mixed flow pump is made up of three sections a bottom part, a middle part and a head part (Fig.3.1). The bottom part consists of a rotating mixed flow impeller housed inside a cylindrical casing (Fig.3.2). A set of diffuser vanes are also provided in the bottom part just above the impeller. As the impeller rotates an outward thrust is given to the water, in addition to imparting an upward velocity. The diffuser provided above the impeller guides the water to the discharge column and straightens the flow. The impeller is suspended on a driving shaft which extends above the bottom part. Rubber bushes are provided in the bottom part to support the driving shaft. Flanges are provided on the bottom part to connect it to the middle part.

The mixed flow impeller consists of 6 blades which are designed in such a way that when it rotates axial and radial motions are imparted to the water (Fig.3.3).

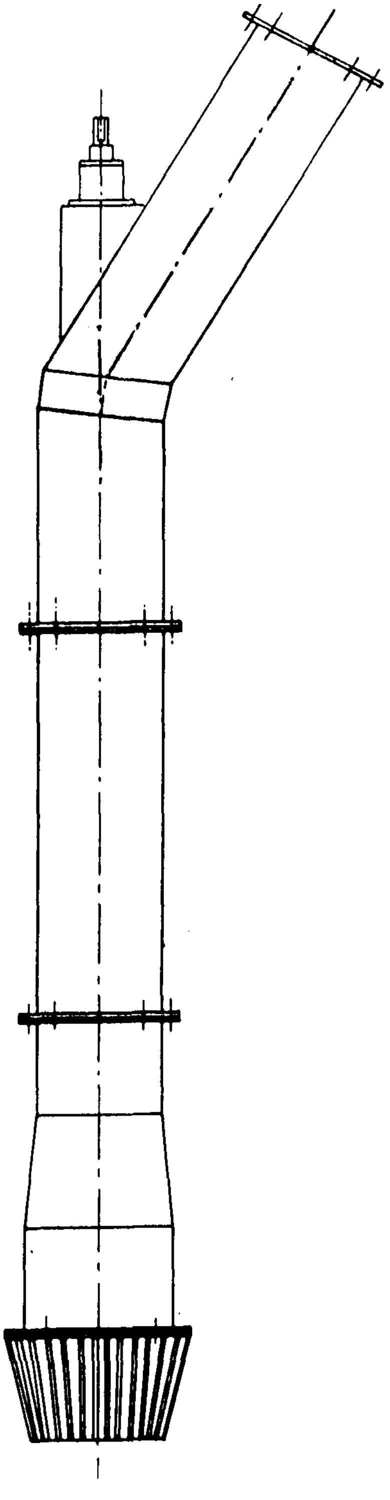
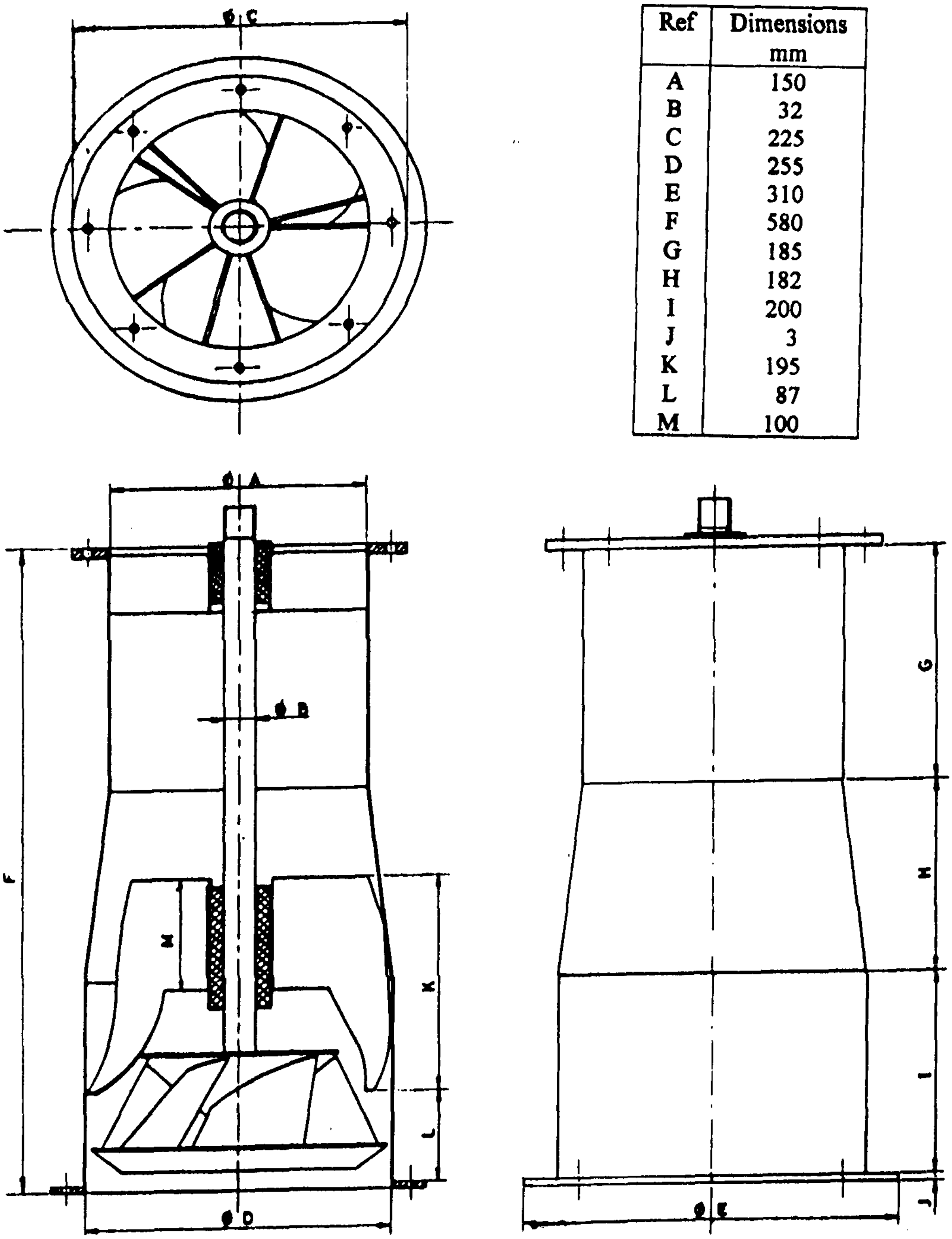
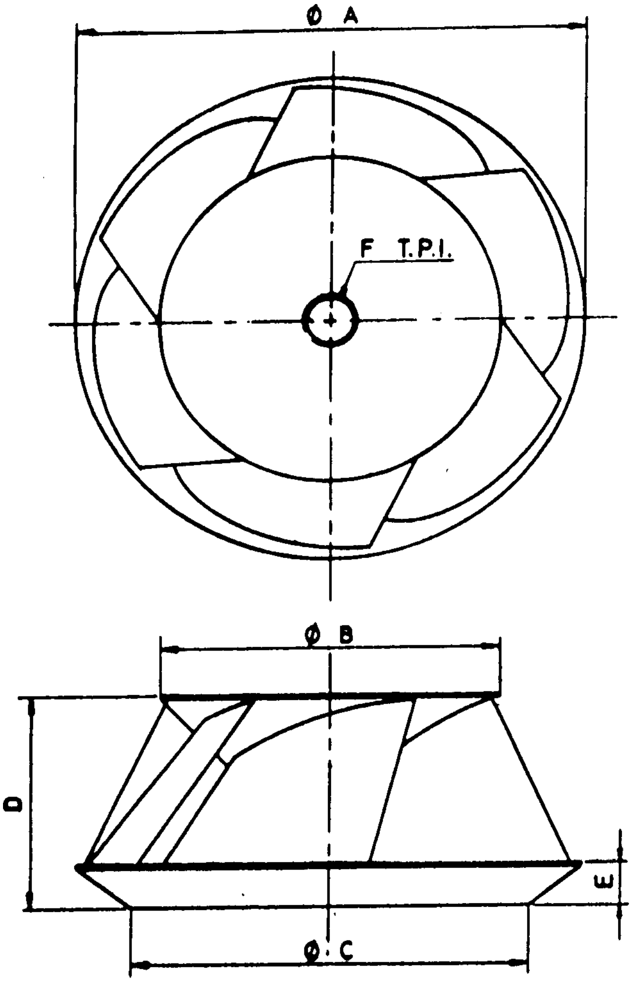


Fig.3.1 View of mixed flow pump



Ref	Dimensions mm
A	150
B	32
C	225
D	255
E	310
F	580
G	185
H	182
I	200
J	3
K	195
L	87
M	100

Fig. 3.2 Views of bottom part of mixed flow pump



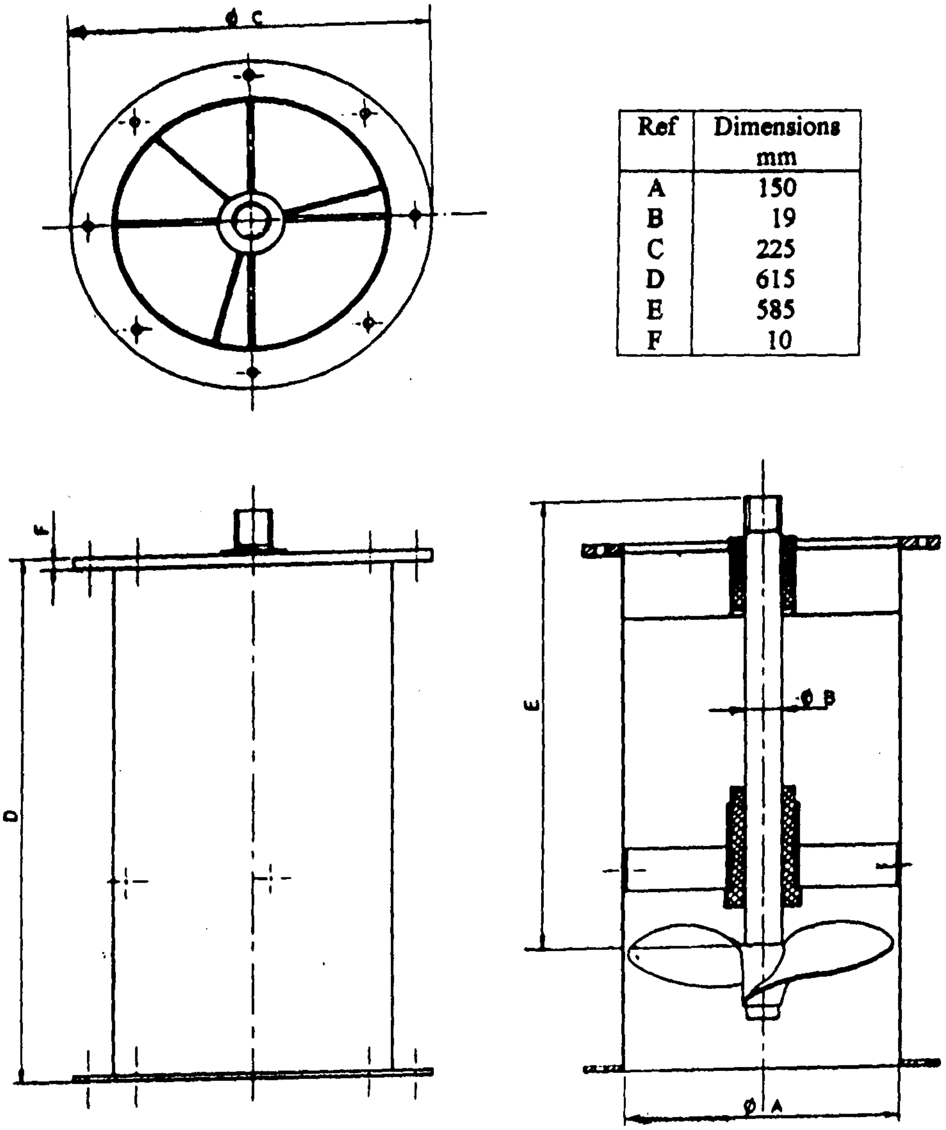
Ref	Dimensions mm
A	235
B	150
C	195
D	118
E	25
F	12

Fig.3.3 Views of mixed flow impeller

The axial flow pump is also made up of three sections. The bottom part consists of an impeller and a driving shaft (Fig.3.4). Rubber bushes are provided on the bottom part to support the driving shaft. The axial flow impeller consists of two blades (Fig.3.5). The impeller blades are designed so as to produce a lifting or propelling action on the water as the impeller rotates.

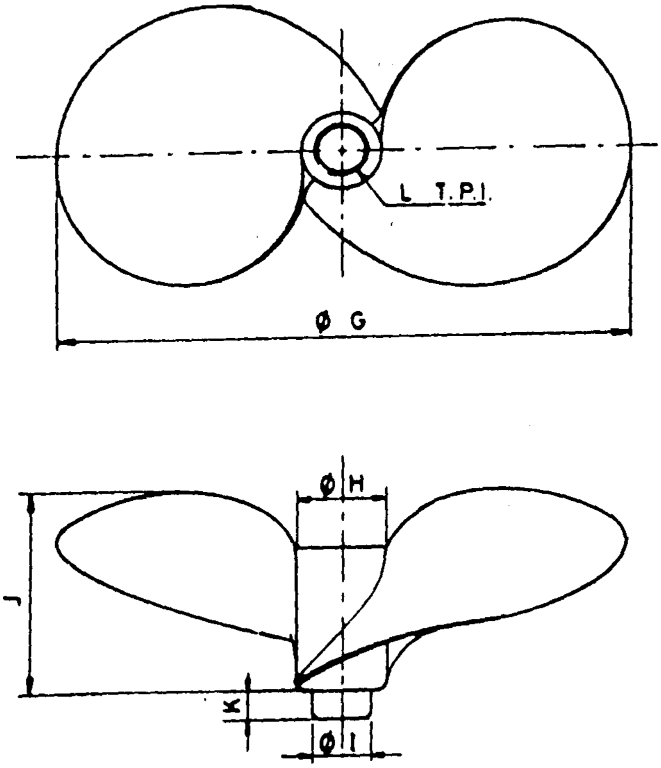
The middle part is a cylindrical section of 1 m length made of galvanized iron sheet (Fig.3.6). Middle part is similar for axial and mixed flow pumps. Flanges are provided on both ends of the middle part which connects the bottom part and head part. Middle part carries a shaft which can be coupled to the shafts in the bottom part and head part. Rubber bushes are provided in the middle part to support the shaft. Middle part forms a major portion of the discharge column.

The head part is also similar for mixed and axial flow pumps (Fig.3.7). This part consists of a portion of discharge column, discharge elbow and discharge pipe. The vertical driving shaft extends outwards at the portion of the discharge elbow to receive the driving force to rotate the impeller. The driving shaft extends from the head down the centre of the column to drive the impeller. The head part consists of only a portion of the driving shaft. One end extends outwards and the other end is coupled to the driving shaft on the middle



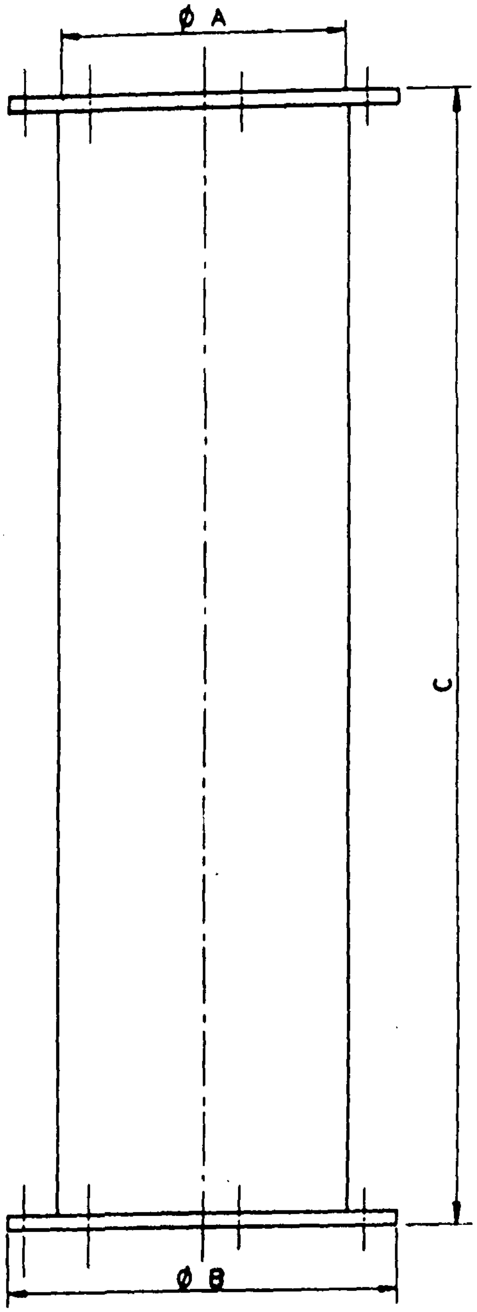
Ref	Dimensions mm
A	150
B	19
C	225
D	615
E	585
F	10

Fig.3.4 Views of bottom part of axial flow pump



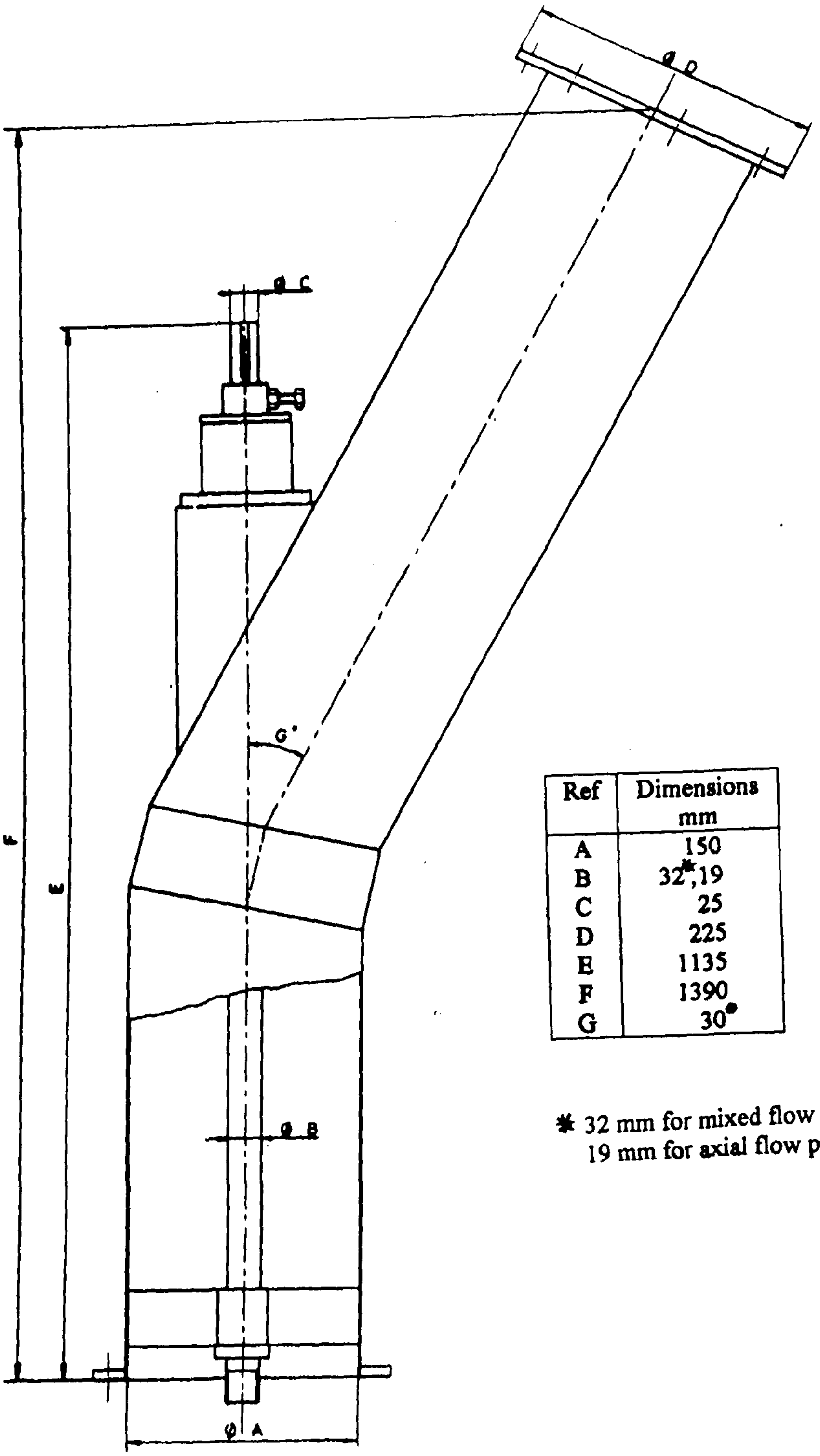
Ref	Dimensions mm
G	140
H	30
I	26
J	45
K	5
L	10

Fig.3.5 Views of axial flow impeller



Ref	Dimensions mm
A	150
B	225
C	1000

Fig.3.6 View of middle part



Ref	Dimensions mm
A	150
B	32*,19
C	25
D	225
E	1135
F	1390
G	30°

* 32 mm for mixed flow pump
19 mm for axial flow pump

Fig.3.7 View of head part

part. Rubber bushes are provided on the head part to support the driving shaft. Flanges are provided on this part to connect this part to the middle part. Head part is also made of galvanized iron sheet.

The whole pump is supported on a tower like structure made of M.S. angles and M.S. flats. The structure has provisions for lowering and raising the pump and for fitting the motor.

The power transmission is achieved by means of V-belts. A pulley is provided on the top of the pump driving shaft. Another pulley is fitted on the motor shaft. V-belt connects the two pulleys and transmits the power from the motor to the pump shaft for driving the impeller. To vary the speed of rotation of the impeller different pulleys with different diameters are used. The pulleys on the motor and pump are made of cast iron and are fixed to the motor shaft and pump shaft using keys. The shaft vibration is reduced by means of ball bearing, provided for the purpose, which is fixed to the supporting structure.

The power for driving the pump was taken from a three phase, 10 hp, induction motor through V-belt drive.

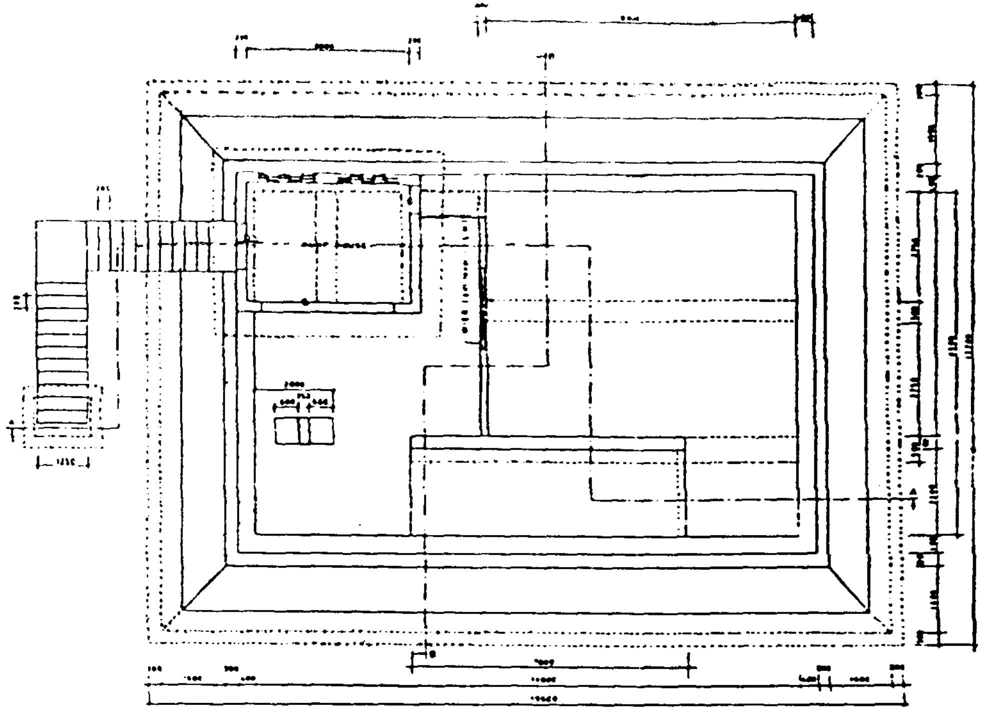
The bottom part and head part of the pump was fabricated in Thailand. The middle part and the supporting structure was fabricated in the research workshop, Mannuthy.

3.2 Details of test bed

The test bed was designed in such a manner that the discharge of the pump could be recirculated back to the pumping sump (Fig.3.8). Thus constant head pump testing could be done. The pump was installed in such a manner that the discharge end of the pump rests on the wall of the concrete channel provided in the testing tank so that the discharge from the pump falls on the discharge channel. The water was pumped from the pumping sump to the discharge channel. The water flowed from the discharge channel to the tank, where the trapezoidal weir was fixed. The water then flowed back to the pumping sump over the weir crest. The depth of the channel section was lesser than that of the weir tank.

A separate level indicator of M.S. flat was provided on the side wall of the pumping sump to indicate the water level in the pumping sump. The zero reading of the level indicator coincided with the weir crest level. Four gate valves were provided at the bottom of the testing tank, for draining water during pump testing.

The power supply was taken from the main line at the pump house. A separate starter, main switch, ammeter, voltmeter and energy meter were fixed to the new meterboard. A starter of semi-automatic type was provided which was set for tripping at overload.



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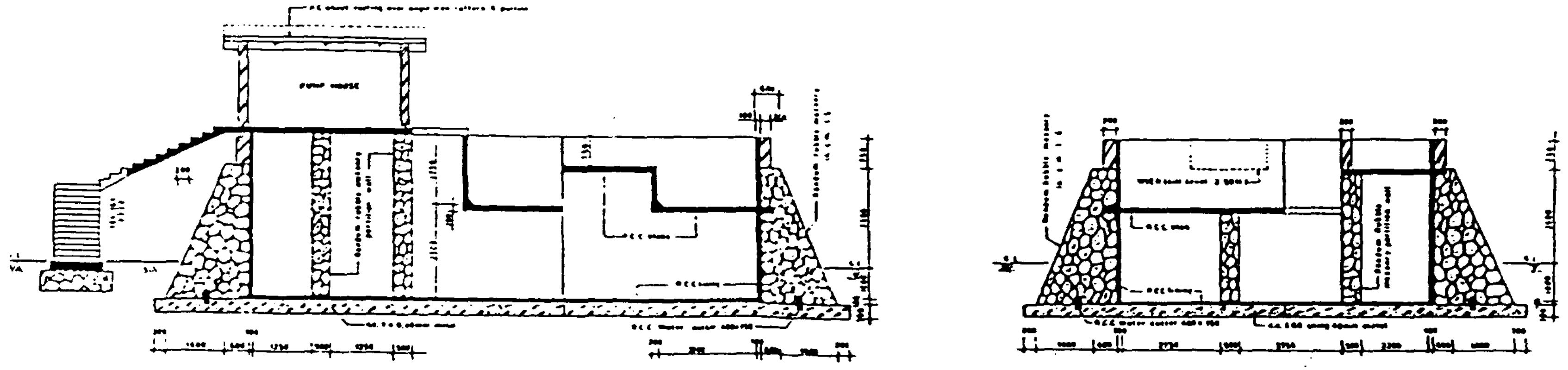

SECTION on A-A

SECTION on B-B

Fig.3.8 Views of test bed

The test bed was constructed with inner dimensions of 14x9x4 m which has the provisions for maintaining constant head.

3.3 Performance evaluation of 15-cm mixed and axial flow pumps

The testing of 15-cm mixed and axial flow pumps were conducted at the specially designed test bed for different speeds. For both the pumps readings were taken at various speeds by changing the diameter of pulleys. For each speed readings were taken at different water levels in the pumping sump by draining water through the gate valves provided at the bottom of the testing tank. The experimental setup is shown in Plate 3.1

The 15-cm mixed flow pump testing was done first. The pump was started when the water level in the pumping sump coincided with the 19 cm reading of the level indicator. The distance from the zero reading of the level indicator to the discharge end of the pump is 71 cm. So the pump was started when the total static head was 90 cm. Since the discharge of the pump was recirculated back to the pumping sump, the pumping water level remained constant. The readings were taken when a steady state condition was reached. The readings of hook gauge, ammeter, voltmeter, kilowattmeter and also the time taken for five revolutions of energy meter disc were

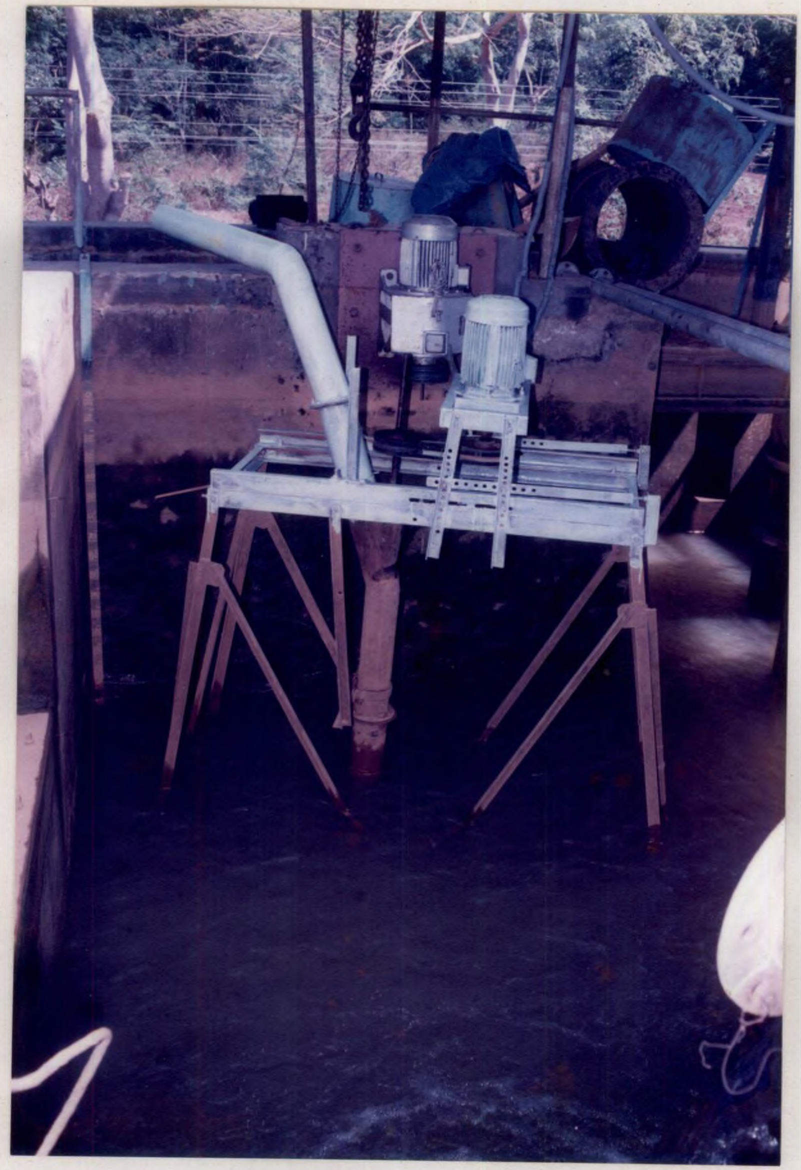

Plate 3.1 View of experimental setup

H_s = total static head or lift, m;

V = mean velocity of the liquid passing through the sectional area at the discharge end, m/s;

g = acceleration due to gravity, m/s^2; and

H_L = hydraulic head losses, m.

3.3.1.1 Total static head measurement

The total static head is the vertical distance between the suction liquid level and the centre of the discharge opening of the pump. During testing the water level below the crest level of the weir, i.e. zero level of water level indicator attached to the wall of the tank. The height above the zero level to the centre of the discharge end was constant and was measured as 0.71 m. Thus the reading on the water level indicator corresponding to the water level in the tank was added to 0.71 m to get the total static head.

3.3.1.2 Hydraulic power losses

The hydraulic power losses of a pump consists of all losses of head which took place between the pump inlet and the discharge end. These included friction losses and all losses due to sudden change in area or direction of flow and eddy losses.

$$H_L = h_f + h_k$$

where,

H_L = hydraulic power losses, m;

h_f = hydraulic power loss due to friction, m; and

h_k = eddy and separation losses, m.

Hydraulic power loss due to friction in the discharge pipe was calculated using the Darcy's equation.

$$h_f = f \frac{L}{D} \frac{V^2}{2g}$$

where,

h_f = friction loss, m;

V = average fluid velocity, m/s;

L = path length; for this experimental set up $L = 5.6$ m;

D = pipe diameter, m; and

f = empirical coefficient dependent upon the Reynold's number, m.

Friction factor can be found out from Moody's Diagram (Fig.3.9). In order to get friction factor from Moody's Diagram relative surface roughness and Reynold's number were calculated. Reynold's number was calculated using the formula

$$R = \frac{VD}{\Gamma}$$

where,

R = Reynold's number;

V = average fluid velocity, m/s;

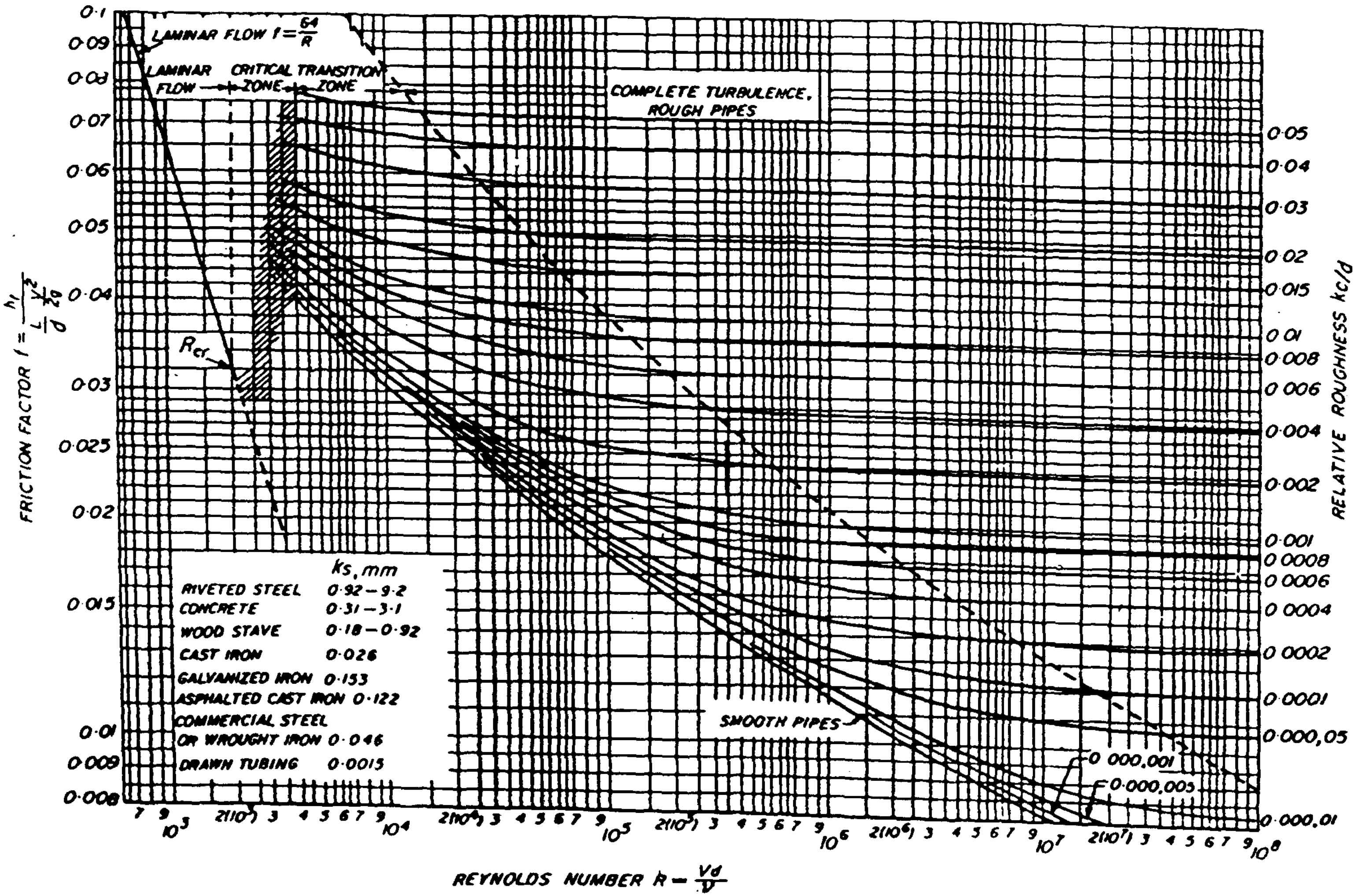

Fig.3.9 Moody's diagram

D = pipe diameter, m; and

Γ = kinemaic viscosity, m^2/s;
for water - 0.804×10^{-6} m^2/s.

All head losses due to sudden change in area or direction of flow and eddy losses that exist between the inlet and discharge were calculated based on the shape of fittings and condition of flow of each pump. The general formula of total head loss due to the eddy and separation losses for the axial and mixed for pumps are

$$h_k = \Sigma k \frac{V^2}{2g}$$

where,

h_k = head loss due to eddy and separation losses, m;

V = average fluid velocity, m/s; and

Σk = summation of the coefficients of all head losses due to sudden change in area or direction of flow and eddy losses exist between the inlet and discharge of tested pumps,

for axial flow pump $\Sigma k = 1.45$;

for mixed flow pump $\Sigma k = 1.65$.

3.3.2 Discharge measurement

The discharge was measured by means of a Cipolletti weir fixed in the tank. A Cipolletti weir is a particular type of tropezoidal weir, the sloping sides of which have an inclination of 1 horizontal to 4 vertical. This weir was invented by an Italian engineer, Cipolletti in 1887, and named after him.

The weir was fabricated from 6 mm M.S. sheet. A coat of anticorrosive paint was applied to it. 3 mm thick rubber packing was provided at the bottom and sides of the tank where the weir was fixed, with the help of stainless steel bolts of diameter 16 mm.

The edge of the weir over which the water flows is known as the sill or crest and its height above the bottom of the tank is known as the crest height. The edge of the weir is bevelled on the down stream side so as to have sharp edged crest resulting in minimum contact with the liquid flowing. The slope of 1 in 4 provided for the sides of this weir results in making the decrease in the discharge over a rectangular weir due to two end contractions just equal to the increase in discharge through the two triangular portions. So that the discharge over a Cipolletti weir may be computed by using the formula for a rectangular weir without end contraction.

On the basis of his own experiments and those of Francis, Cipolletti proposed the following equation for the discharge over a Cipolletti weir.

$$Q = 1.86\ Lh^{3/2}$$

where,

Q = discharge, L/s;

L = length of crest, m; and

h = height of water surface above the crest which is known as the head causing the flow over the weir, cm.

As water flows over the weir, the surface of water over the crest and immediately upstream of it becomes curved. The head h above the crest is therefore measured at a certain distance upstream of the weir, where the water surface may be assumed to be unaffected by the curvature effect. The scale or gauge used for measuring the head should be located at a distance of about four times the approximate head. It should be far enough to one side so that it will be in comparatively still water.

The head h above the crest level was measured with a point gauge. The point gauge consists of a thin vertical rod, pointed at its lower end and it was attached to a rack and pinion arrangement. Main scale and vernier scale were attached to the rod. The rod with the main scale and vernier scale was fixed on a mild steel stand and the stand was fixed to the top of the tank wall. The liquid surface elevation can be read from the main scale and vernier scale. A stilling well of 29 cm diameter and 75 cm depth was provided to get more accurate readings.

3.3.3 Speed measurement

The speed of the pump as well as the motor was noted with a tachometer.

3.3.4 Power measurement

The input energy to the motor was found out using an energy meter, fixed to the switch board. The time taken by the energy meter disc for five revolutions was noted three times at each reading, using a stop watch. From these readings, the average time taken for five revolutions was calculated. Power input to the pump can be calculated using the formula given below,

$$\text{Power input} = \frac{n}{t} \times \frac{3600}{k} \times \frac{1000}{746}$$

where,

n = number of revolutions;

t = time taken for n revolutions, s; and

k = energy meter constant.

Results and Discussion

RESULTS AND DISCUSSION

Experiments were conducted at a specially designed test bed to determine the effect of various parameters on the performance of 15-cm mixed flow and axial flow pumps. The analysis of the observed data and the results obtained from the experiments are presented in this chapter.

4.1 Performance evaluation of the mixed flow pump

The performance characteristics of 15-cm mixed flow pump was evaluated from the tests conducted at the test bed, for various speeds, viz. 600, 650, 700, 750, 800, 850, 900, 950, 1000 and 1050 rev/min and are described below.

Detailed studies were carried out corresponding to all the above speeds. The performance of the pump at various speeds are presented (Appendix I to X). The maximum efficiency of 42.16 per cent was obtained at a speed of 1000 rev/min, against a total static head of 90 cm and a discharge rate of 49.47 L/s (Appendix II). The input power corresponding to this efficiency was 3.4 hp. Efficiency was always less than this for any other combination of head, speed and input power.

From the observation of discharge at each speed for various heads, it was clear that as total head increases discharge decreases. It was also observed that for same total static head, discharge increases with increase in speed. A maximum discharge of 51.67 L/s was obtained at a speed of 1050 rev/min against a total static head of 90 cm. But the efficiency was only 39.41 per cent which less than the maximum efficiency of 42.16 per cent was obtained at 1000 rev/min. For 1050 rev/min discharge varied from 51.67 to 10.37 L/s as total static head varied from 90 to 300 cm. For 1000 rev/min discharge varied from 49.37 to 7.89 L/s as the total static head vary from 90 to 300 cm. Thus for each speed there was considerable decrease in discharge with increase in total static head. As speed decreases total static head against which pump can be operated also decreases. For lower speeds discharge was very less for high total static head. At 600 rev/min discharge was obtained only upto a total static head of 150 cm and the discharge obtained was 2.0 L/s which was very less.

The input power increases with increase in speed. For each speed input power increases as total static head increases or, when discharge decreases. At 1000 rev/min input power varies from 3.4 to 4.0 hp with increase in total head.

Efficiency decreases with increase in total head at various speeds. As speed increases efficiency also increases and reaches a maximum at 1000 rev/min speed and then it decreases. Maximum efficiency of 42.16 per cent was obtained at 1000 rev/min. For each total static head maximum efficiency was obtained at 1000 rev/min speed. Efficiency at 1000 rev/min varied from 42.16 per cent to 7.98 per cent against a total static head varying from 90 to 300 cm and against discharge varying from 49.47 to 7.89 L/s. As speed increases beyond 1000 rev/min there was a slight increase in discharge but there was considerable decrease in efficiency. Hence, 1000 rev/min is perhaps the limit above which the efficiency falls drastically.

Using the data obtained from laboratory tests, performance characteristics curves were fitted for 10 different speeds, namely 600, 650, 700, 750, 800, 850, 900, 950, 1000 and 1050 rev/min. These curves give the relationship of discharge with efficiency, input power and total head at each speed (Fig.4.1 to 4.10).

The plotted curves of discharge-efficiency relationships are linear in nature. The curves are of the form,

$$E = K_1 + K_2Q$$

where,

E = efficiency, %;
Q = discharge, L/s; and

K_1, K_2 = constants.

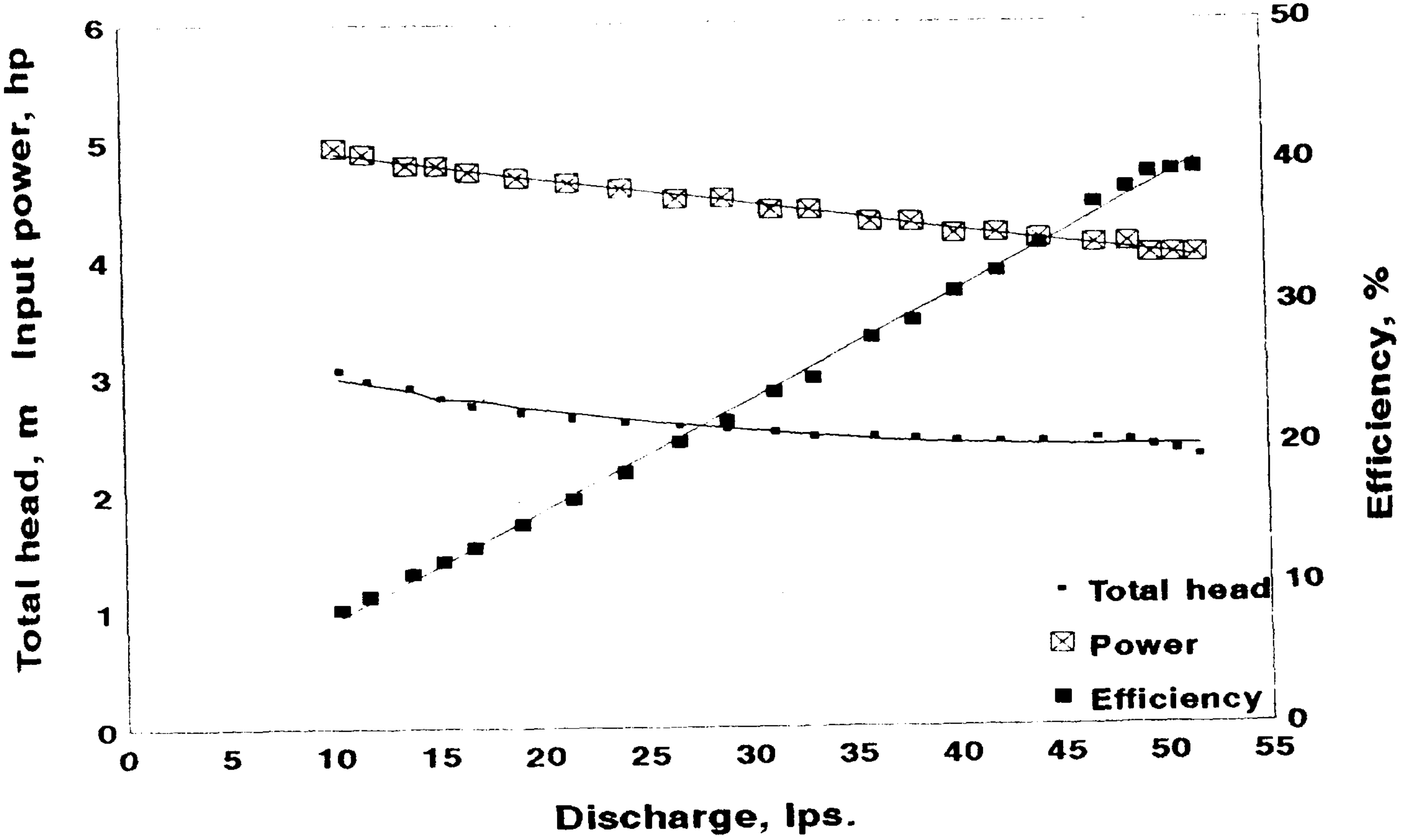

Fig.4.1 Performance characteristic curves of mixed flow pump at 1050 rev/min

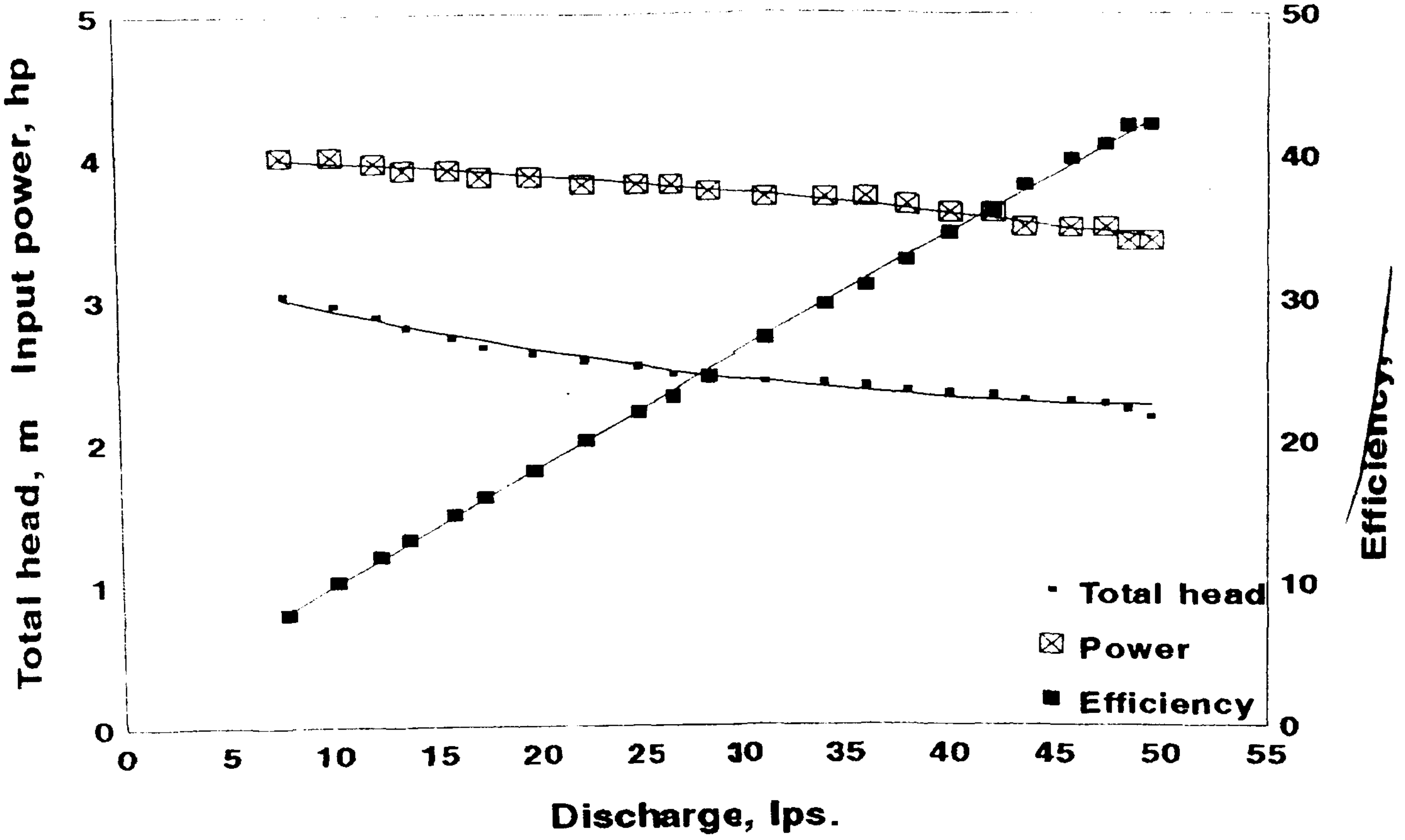

Fig.4.2 Performance characteristic curves of mixed flow pump at 1000 rev/min

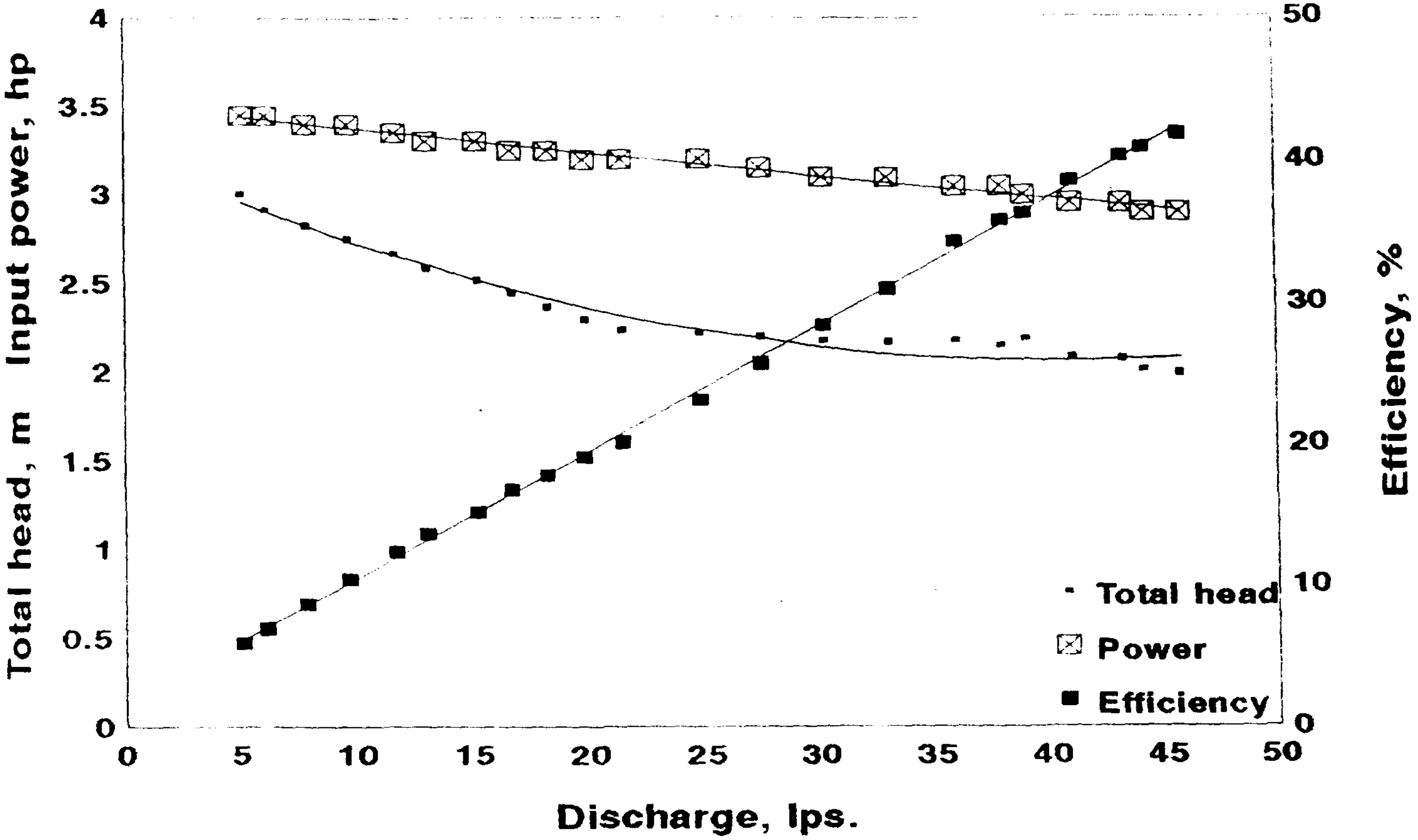

Fig.4.3 Performance characteristic curves of mixed flow pump at 950 rev/min

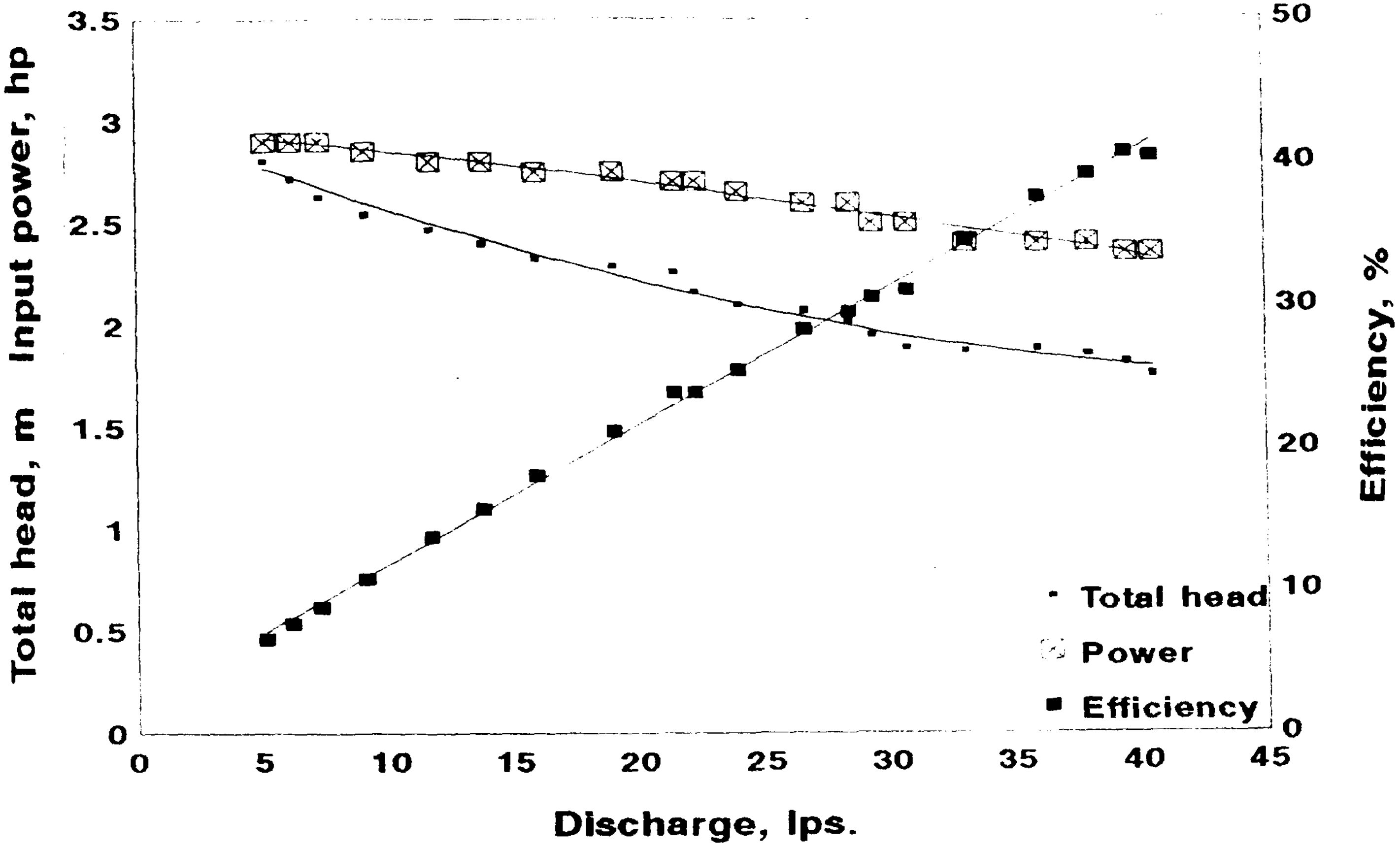

Fig 4.4 Performance characteristic curves of mixed flow pump at 900 rev/min

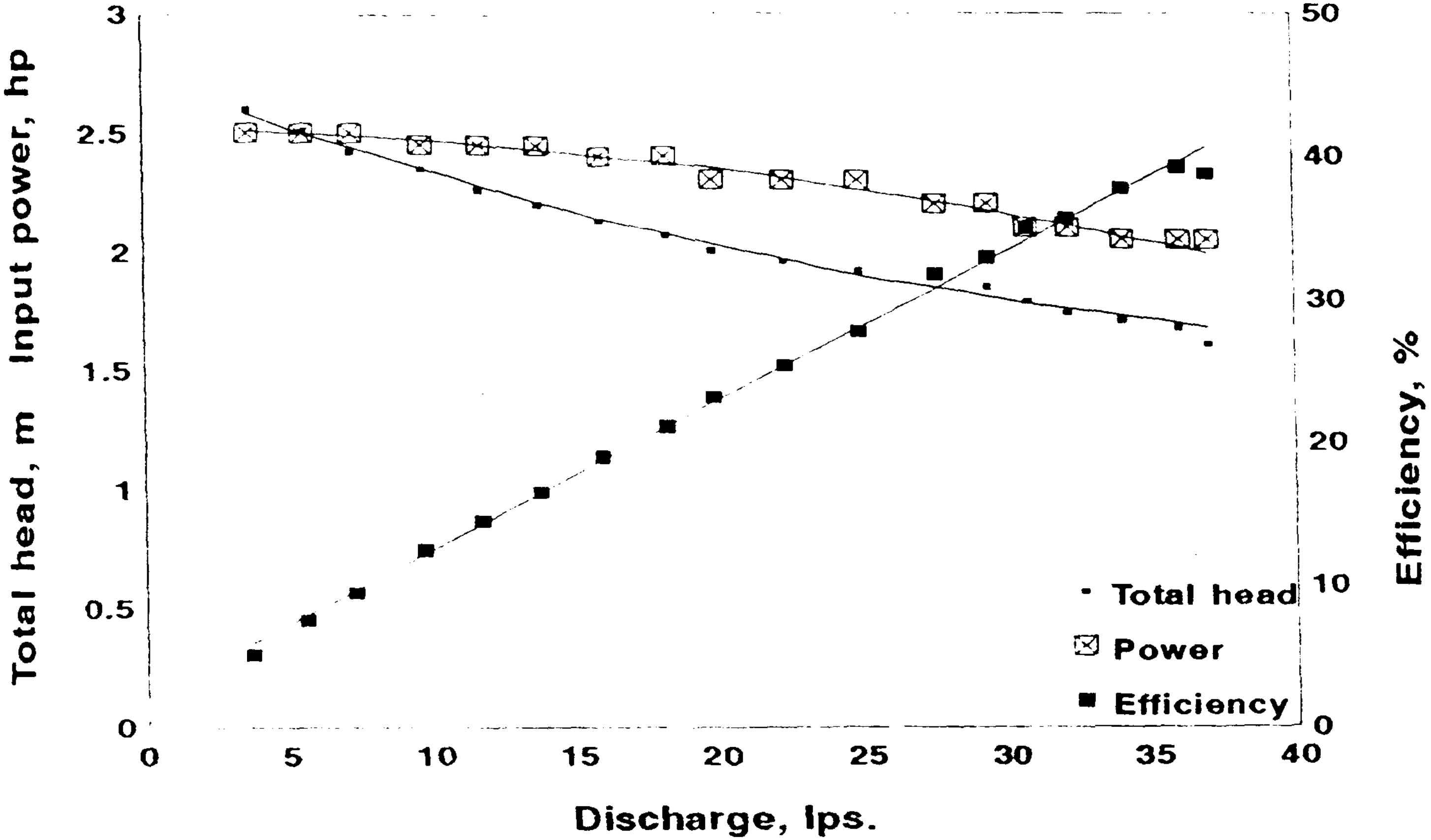

Fig.4.5 Performance characteristic curves of mixed flow pump at 850 rev/min

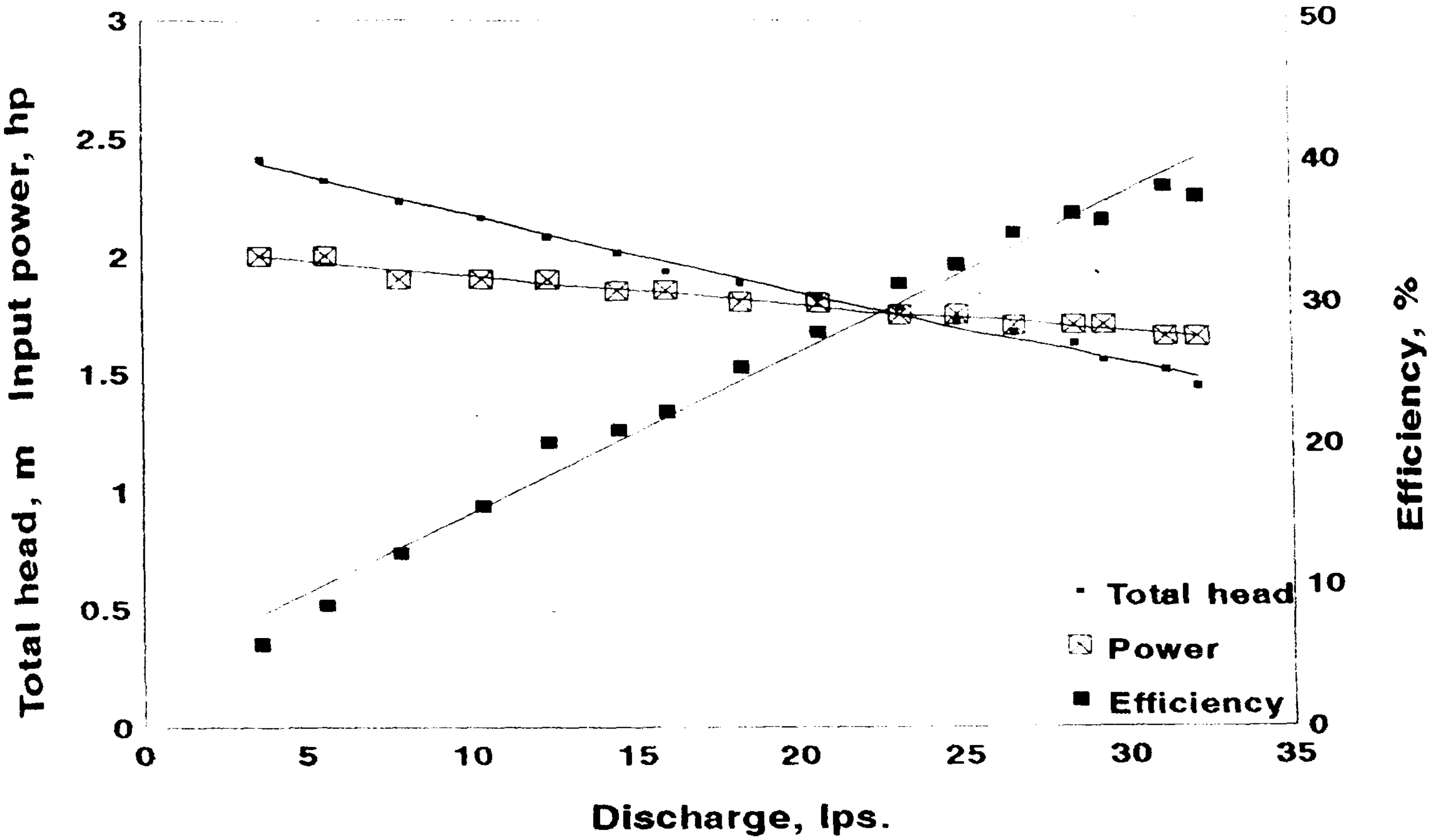

Fig.4.6 Performance characteristic curves of mixed flow pump at 800 rev/min

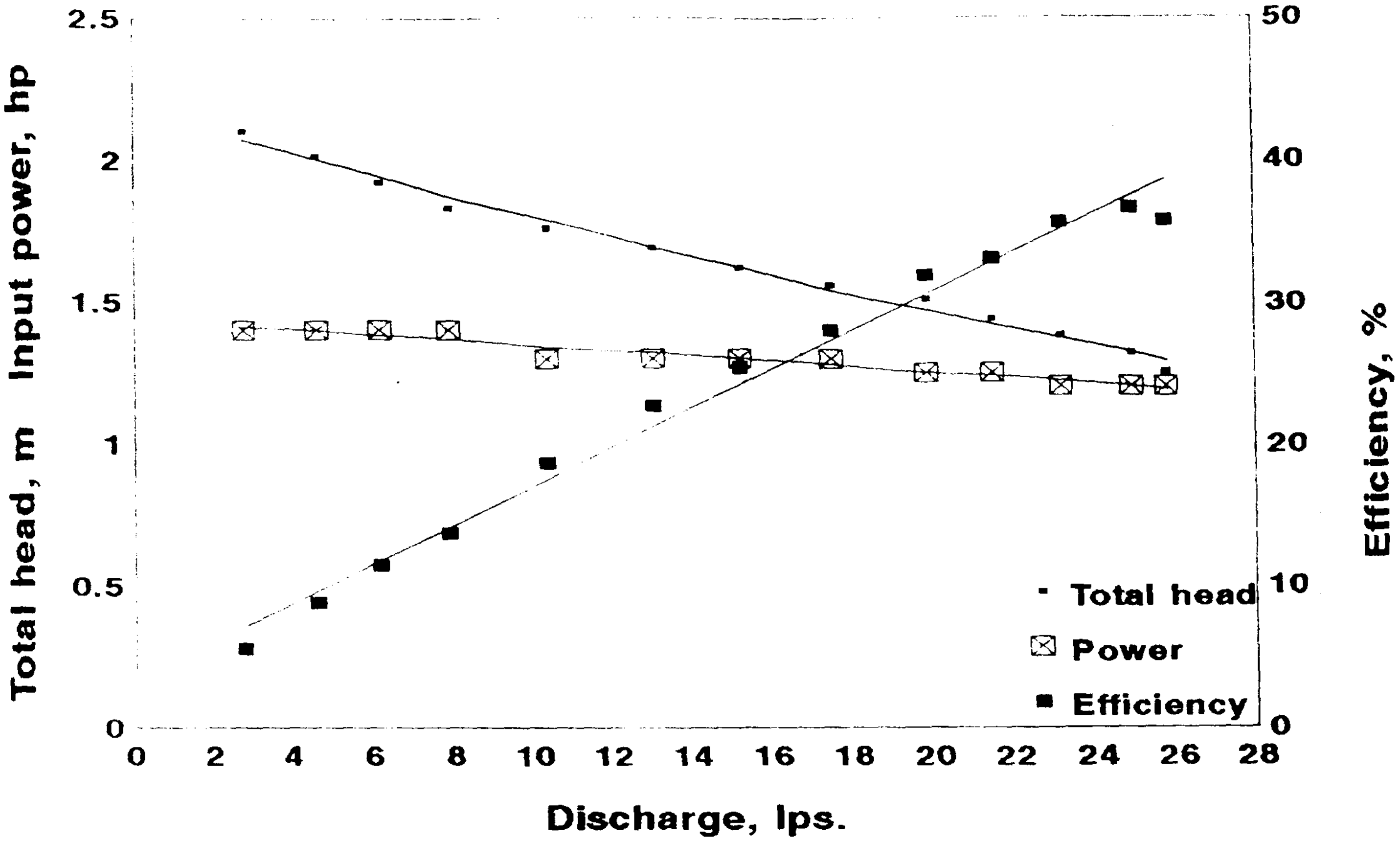

Fig.4.7 Performance characteristic curves of mixed flow pump at 750 rev/min

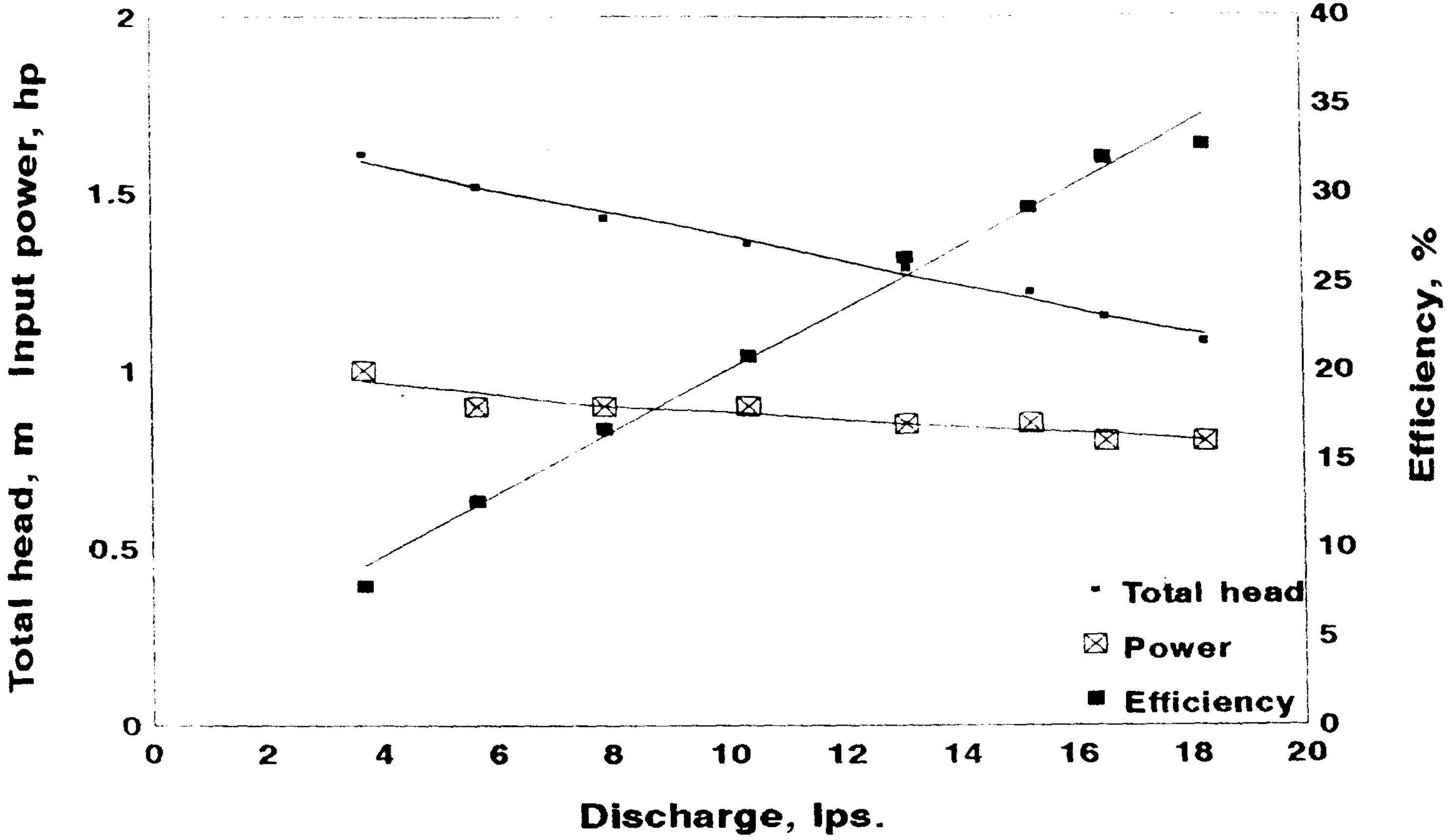

Fig.4.8 Performance characteristic curves of mixed flow pump at 700 rev/min

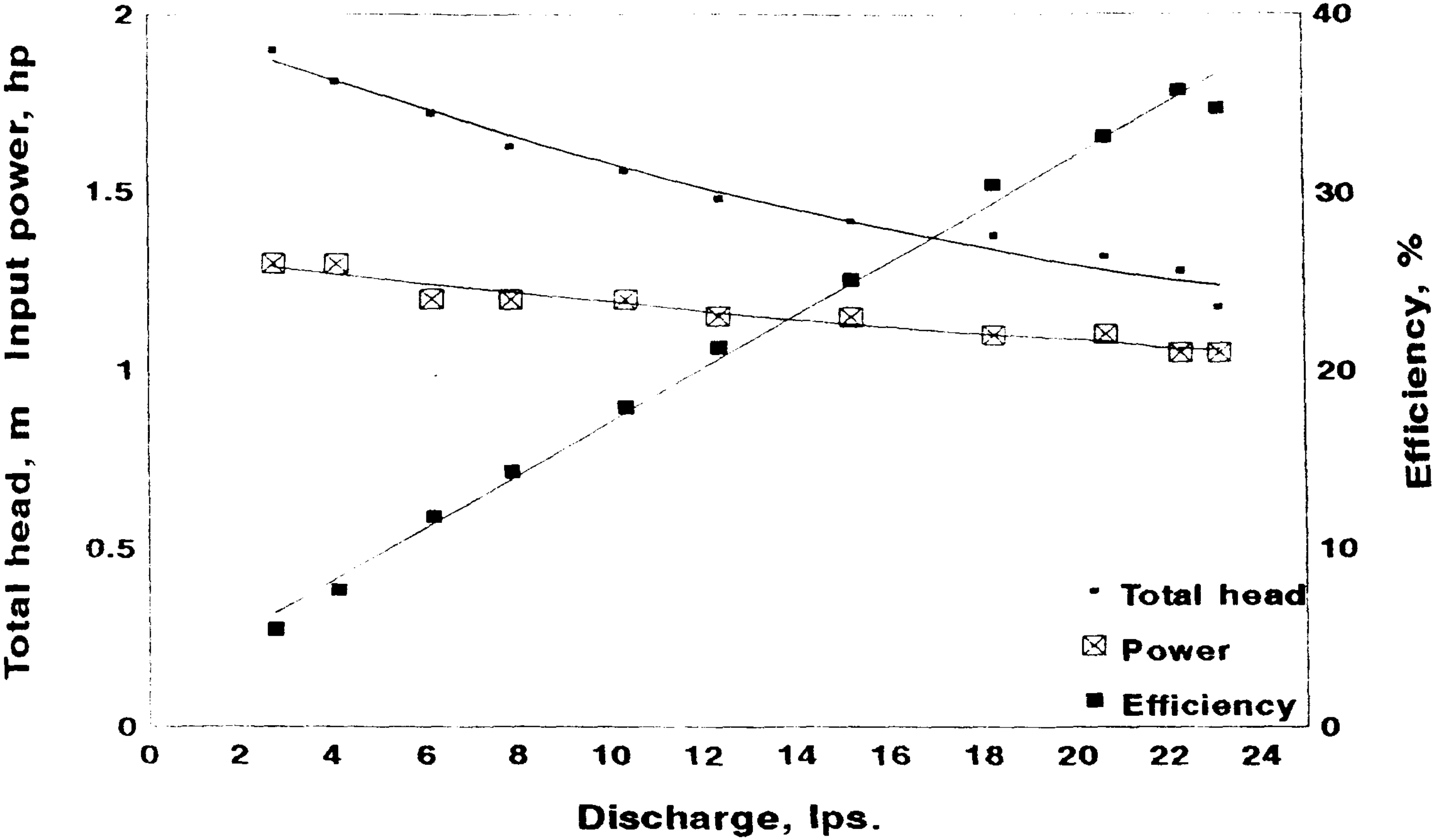

Fig.4.9 Performance characteristic curves of mixed flow pump at 650 rev/min

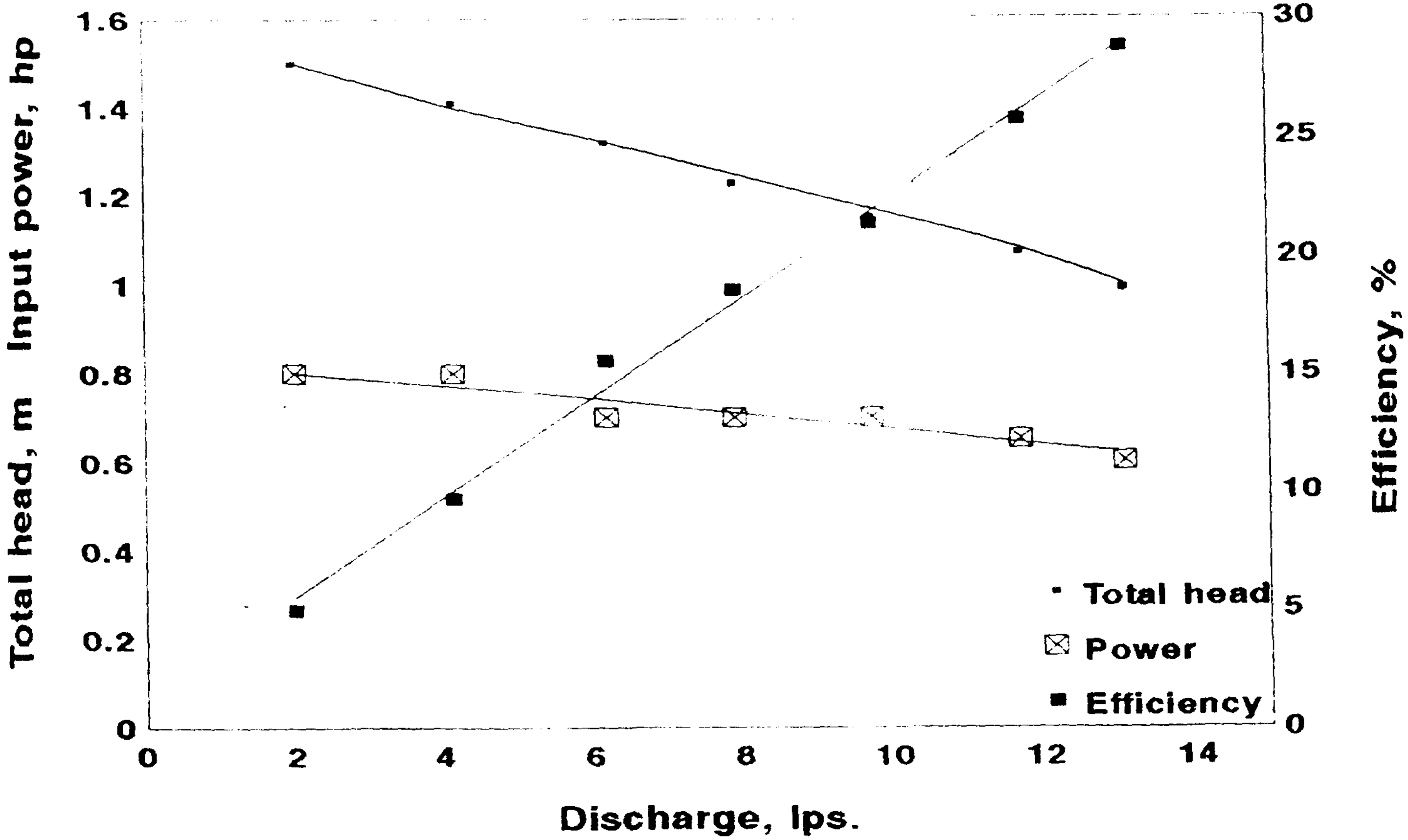

Fig.4.10 Performance characteristic curves of mixed flow pump at 600 rev/min

The fitted equations for various speeds are presented (Table 4.1).

In the same manner the relationships between head and discharge are plotted and the trend is found to be of second degree nature. The second degree curves are of the form,

$$H = K_1 + K_2Q + K_3Q^2$$

where,

H = Total pump head, m;
Q = discharge, L/s; and
K_1, K_2, K_3 = constants.

The fitted equations for the 10 speeds selected are given (Table 4.2).

The fitted curves of input power and discharge were found second degree in nature and is of the form,

$$P = K_1 + K_2Q + K_3Q^2$$

where,

P = input power, hp;
Q = discharge, L/s; and
K_1, K_2, K_3 = constants.

The fitted equations for different speeds, are presented (Table 4.3).

Table 4.1 Discharge - efficiency relationship of 15-cm mixed flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	1050	$E = -0.0928 + 0.7753\ Q$	0.998
2.	1000	$E = 1.5175 + 0.8272\ Q$	0.999
3.	950	$E = 1.8733 + 0.8649\ Q$	0.999
4.	900	$E = 1.9964 + 0.9742\ Q$	0.998
5.	850	$E = 1.7516 + 1.0642\ Q$	0.997
6.	800	$E = 3.8972 + 1.1257\ Q$	0.983
7.	750	$E = 3.2708 + 1.3721\ Q$	0.986
8.	700	$E = 2.2943 + 1.4772\ Q$	0.994
9.	650	$E = 2.6750 + 1.7291\ Q$	0.991
10.	600	$E = 1.3264 + 2.1078\ Q$	0.994

Table 4.2 Discharge - head relationship of 15-cm mixed flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	1050	$H = 3.356 - 0.04\ Q + 0.00041\ Q^2$	0.96
2.	1000	$H = 3.302 - 0.041\ Q + 0.004\ Q^2$	0.98
3.	950	$H = 0.231 - 0.057\ Q + 0.0007\ Q^2$	0.97
4.	900	$H = 3.014 - 0.0491\ Q + 0.00046\ Q^2$	0.99
5.	850	$H = 2.738 - 0.0437\ Q + 0.00041\ Q^2$	0.99
6.	800	$H = 2.521 - 0.0366\ Q + 0.00014\ Q^2$	0.99
7.	750	$H = 2.175 - 0.0398\ Q + 0.00022\ Q^2$	0.99
8.	700	$H = 2.004 - 0.0493\ Q + 0.0007\ Q^2$	0.98
9.	650	$H = 1.719 - 0.0337\ Q + 0.000019\ Q^2$	0.99
10.	600	$H = 1.585 - 0.0452\ Q + 0.00628\ Q^2$	0.95

Table 4.3 Discharge - input power relationship of 15-cm mixed flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	1050	$P = 5.19904 - 0.02744\ Q + 8.036\text{x}10^{-5}\ Q^2$	0.96
2.	1000	$P = 4.03686 - 0.00624\ Q + -0.000123\ Q^2$	0.98
3.	950	$P = 3.51464 - 0.01446\ Q + 2.856\text{x}10^{-5}\ Q^2$	0.98
4.	900	$P = 2.9754 - 0.011104\ Q + -0.000123\ Q^2$	0.98
5.	850	$P = 2.6268 - 0.015295\ Q - 0.0003165\ Q^2$	0.97
6.	800	$P = 2.0448 - 0.013384\ Q + 3.7536\text{x}10^{-5}\ Q^2$	0.98
7.	750	$P = 1.4394 - 0.0090715Q - 1.7632\text{x}10^{-5}\ Q^2$	0.93
8.	700	$P = 1.3372 - 0.016715\ Q + 0.00002025\ Q^2$	0.94
9.	650	$P = 1.02991 - 0.017183\ Q + 0.0002615\ Q^2$	0.83
10.	600	$P = 0.83521 - 0.014805\ Q + 0.000165\ Q^2$	0.90

Performance curves were also plotted for 15-cm mixed flow pump at various constant total static heads. These curves give the relationship between the pump speed and discharge, total head, input power and efficiency (Fig.4.11 to 4.17). From the curves it was clear that as speed increases, discharge, total head and input power also increase. Efficiency increases as speed increases and it reaches maximum at 1000 rev/min and then it decreases. For lower speeds discharge become very less as total static head increases. So curves were drawn for only upto 150 cm constant total static head starting from 90 cm.

Further testing with decrease in total static head was not possible because of the limitation imposed by the test bed.

The plotted speed - efficiency curves at different constant total static heads are second degree in nature. The relationship is of the form,

$$E = K_1 + K_2N + K_3N^2$$

where,

E = efficiency, %;
N = pump speed, rev/min; and
K_1, K_2, K_3 = constants.

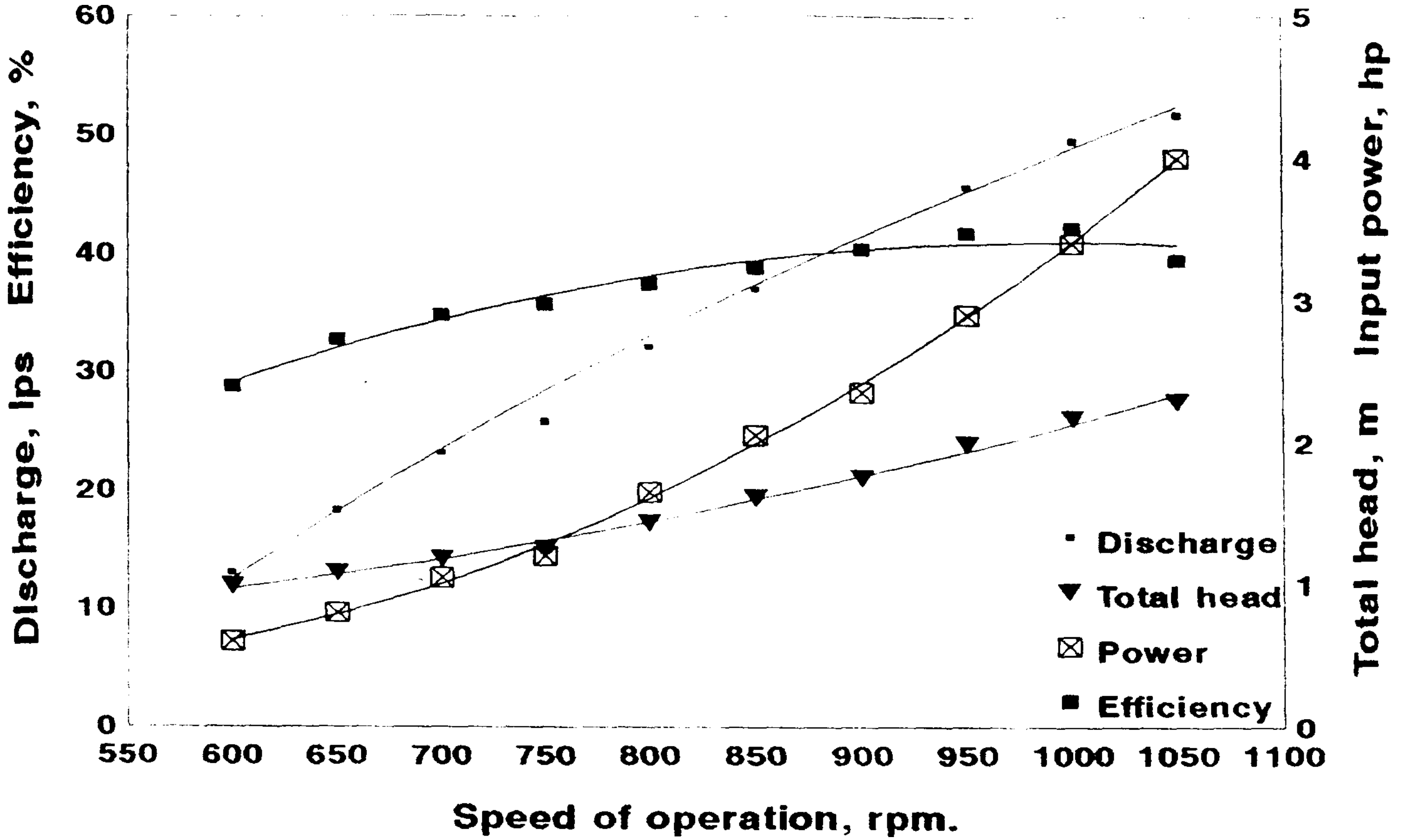

Fig 4.11 Performance characteristics of mixed flow pump at total static head of 0.9m

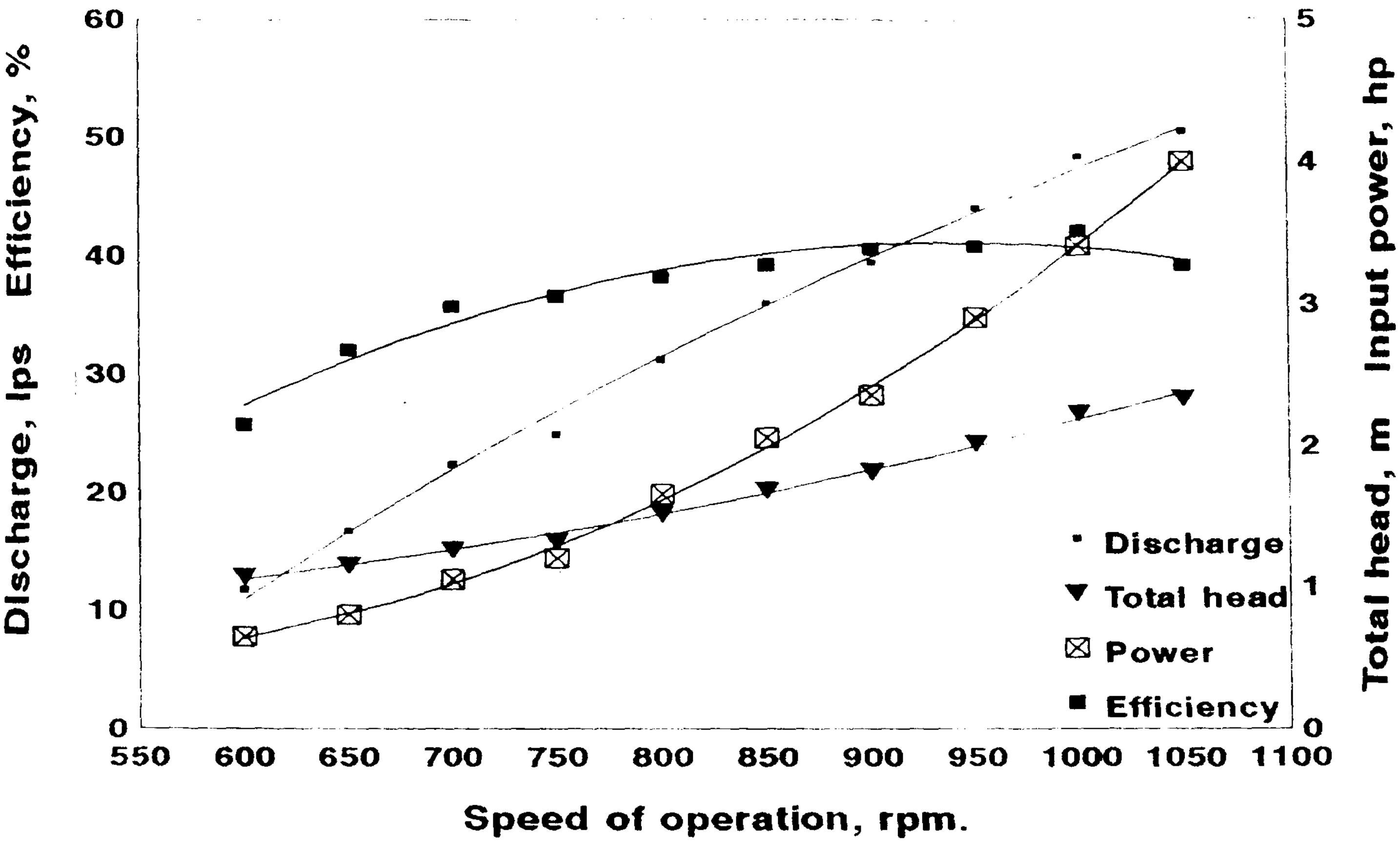

Fig 4.12 Performance characteristics of mixed flow pump at total static head of 1m

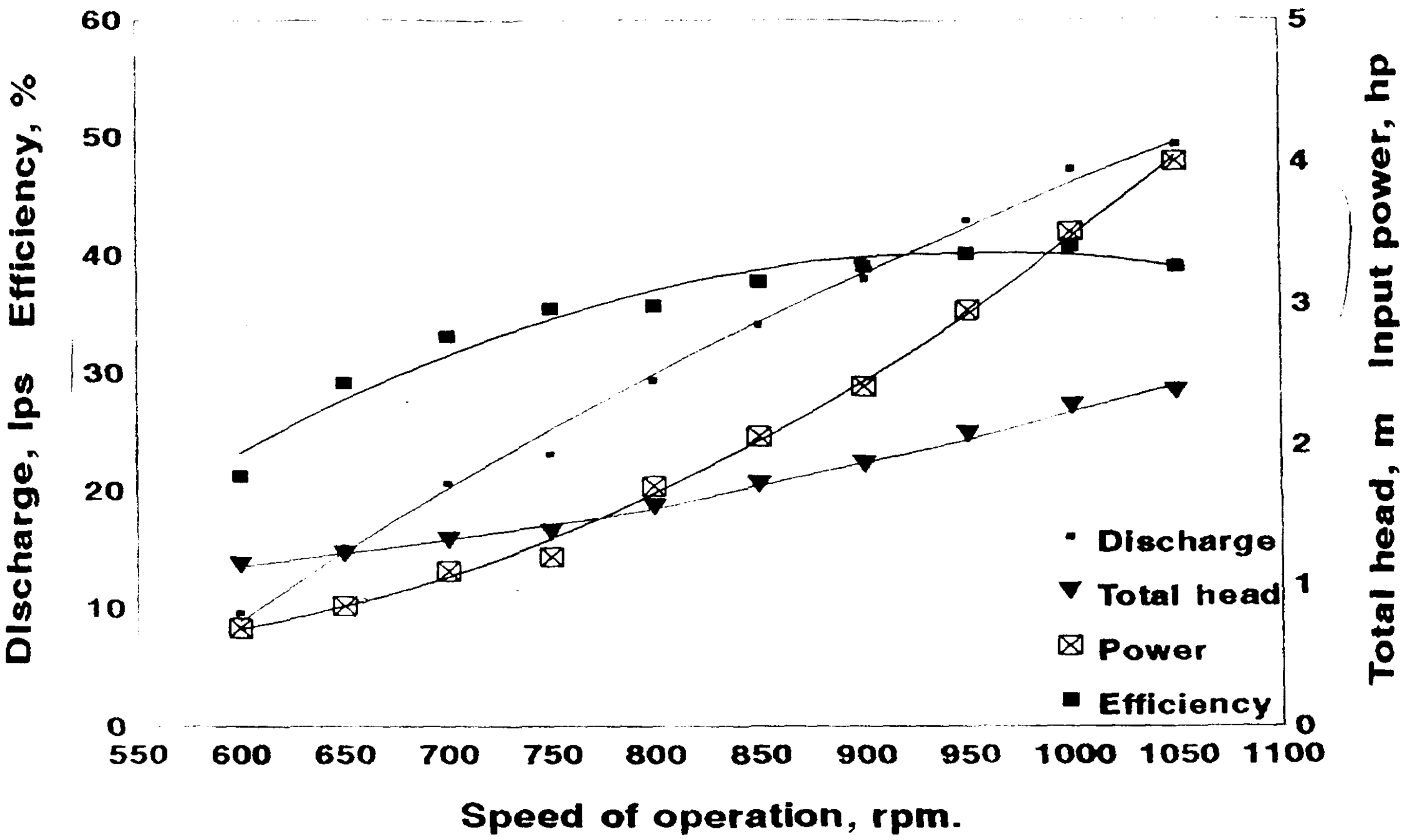

Fig 4.13 Performance characteristics of mixed flow pump at total static head of 1.1m

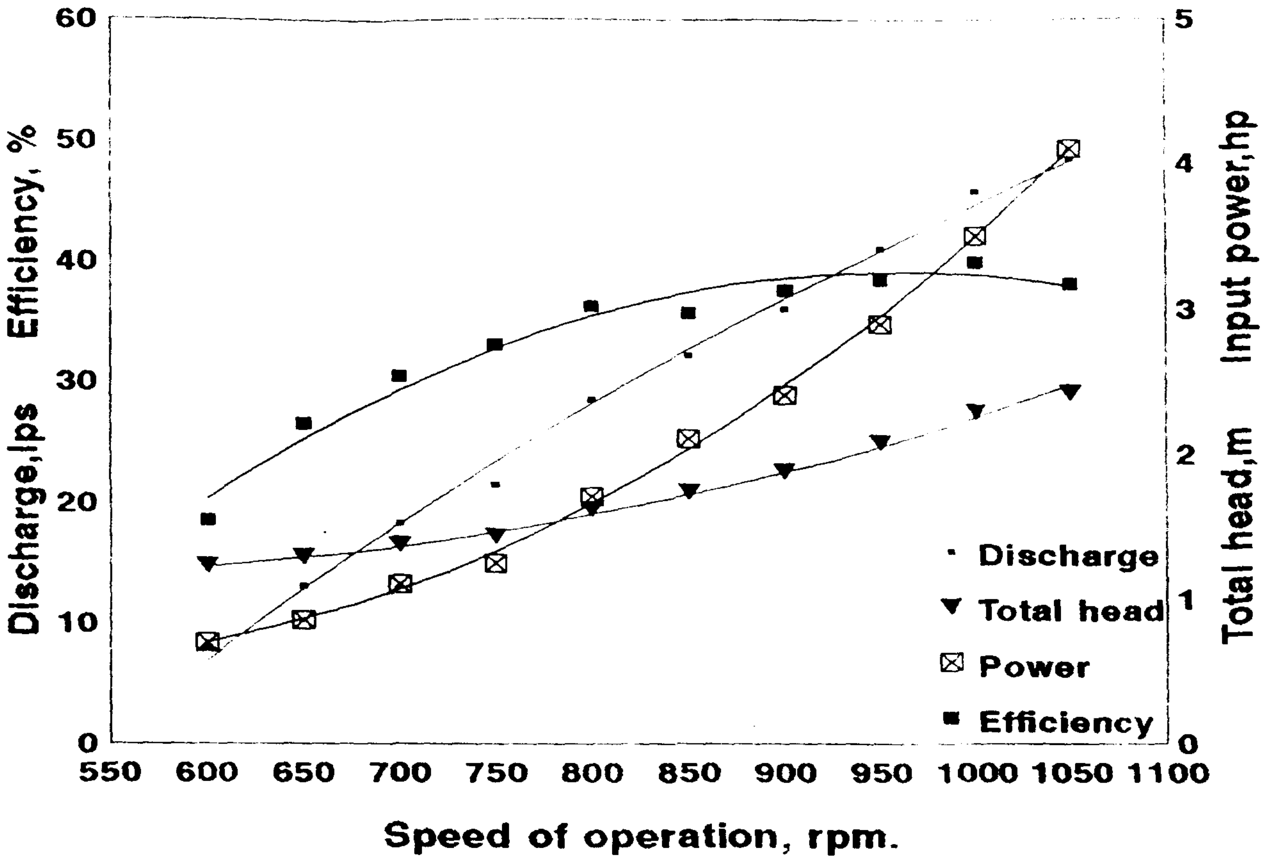

Fig 4.14 Performance characteristics of mixed flow pump at total static head of 1.2 m

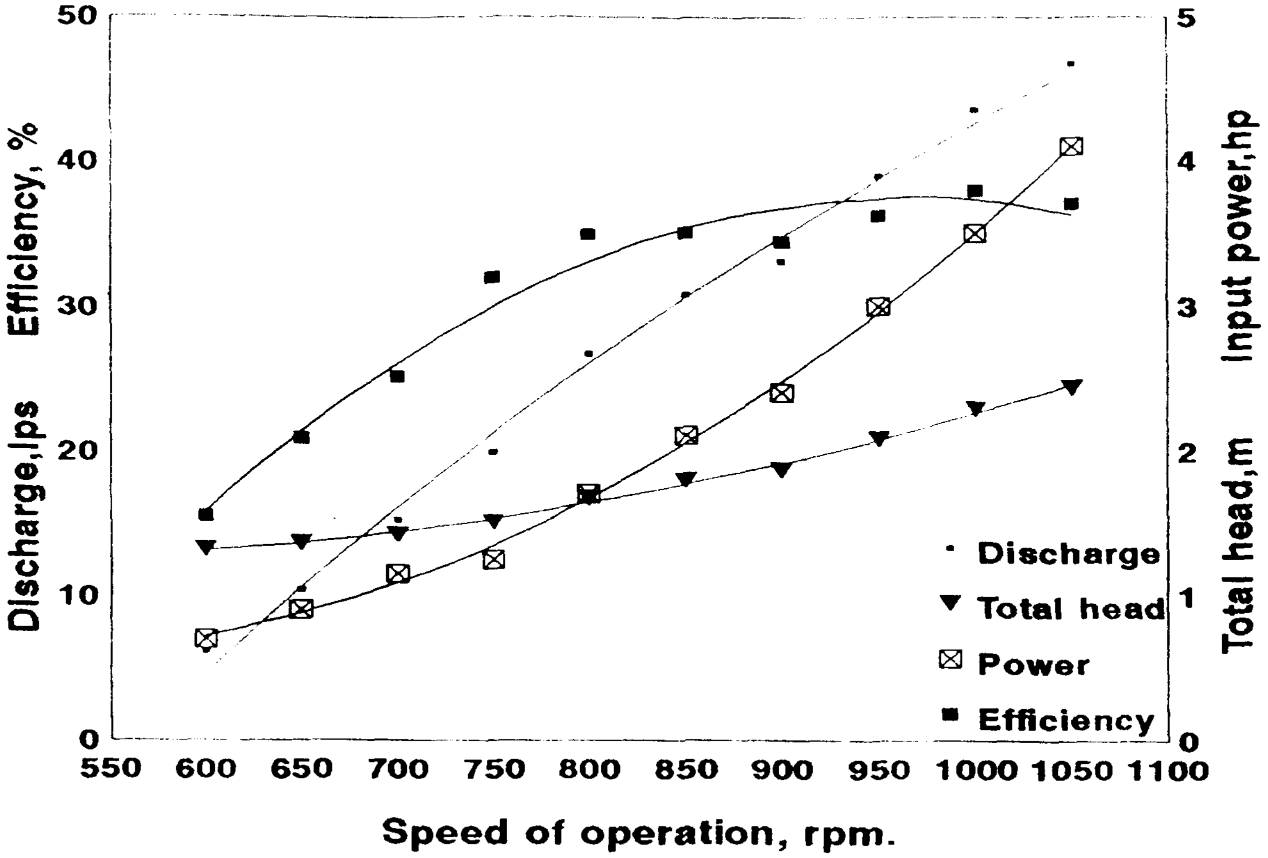

Fig 4.15 Performance characteristics of mixed flow pump at total static head of 1.3m

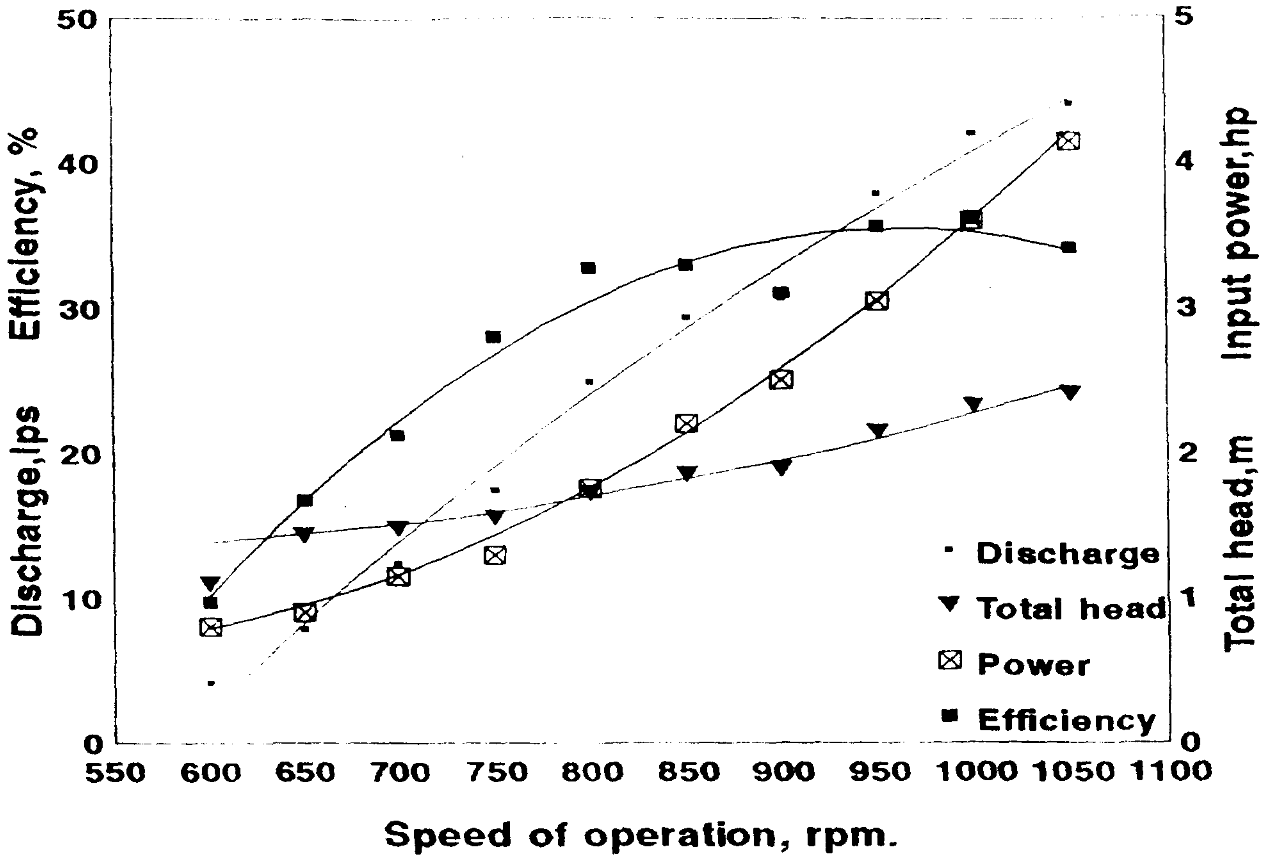

Fig 4.16 Performance characteristics of mixed flow pump at total static head of 1.4m

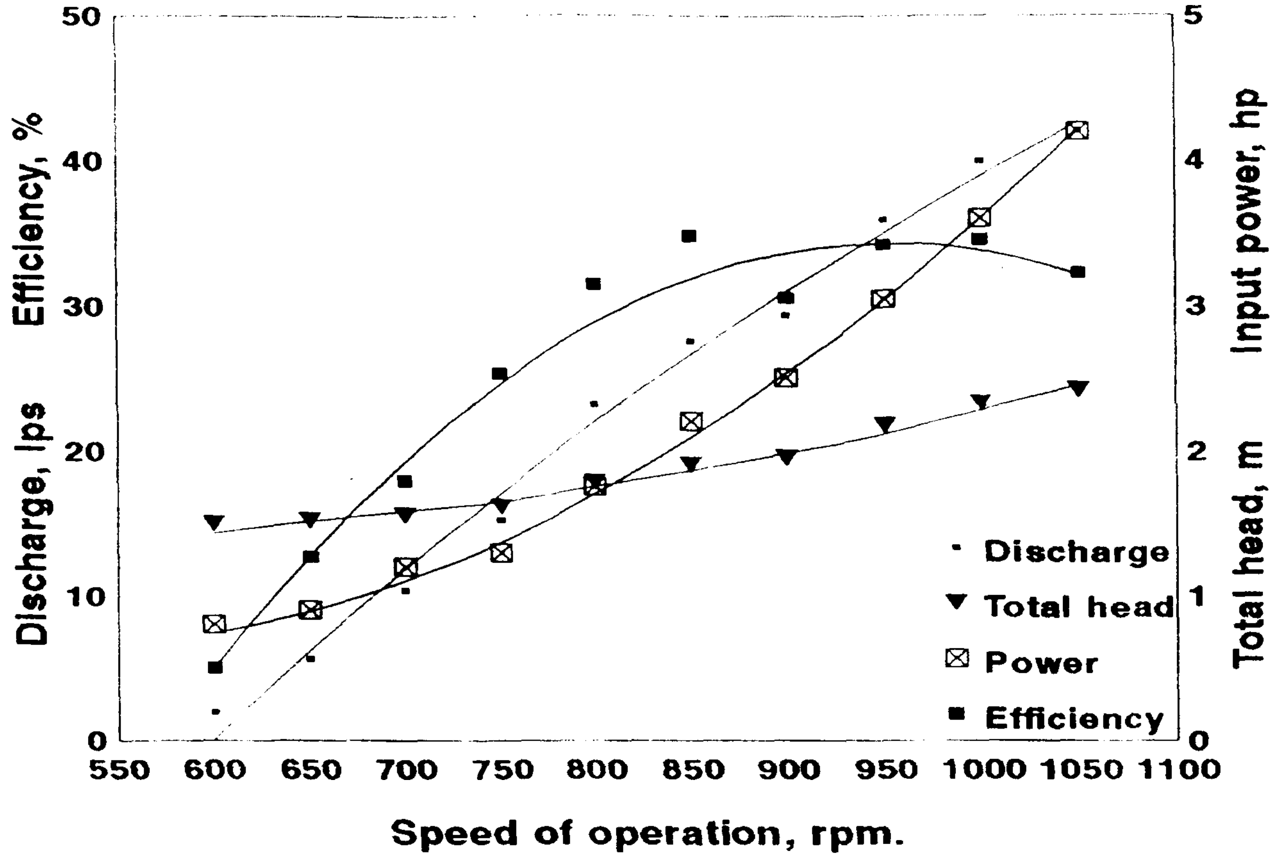

Fig 4.17 Performance characteristics of mixed flow pump at total static head of 1.5m

The fitted equations for various total static head are presented (Table 4.4).

The curves showing the speed - discharge relationship of mixed flow pump are semi-logarithmic in nature and the equations are of the form,

$$Q = K_1 \, Ln \, N - K_2$$

where,

Q = discharge, L/s;

N = pump speed, rev/min; and

K_1, K_2 = constants.

The equations fitted for various total static heads are presented (Table 4.5).

In the same manner the relationship between pump speed and total head are plotted for various total static head and the trend is found to be of second degree nature. The second degree curves are of the form,

$$H = K_1 + K_2N + K_3N^2$$

where,

H = total head, m;

N = pump speed, rev/min; and

K_1, K_2, K_3 = constants.

Table 4.4 Speed - efficiency relationship of 15-cm mixed flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$E = -34.382 + 0.15156\ N - 7.6273\text{x}10^{-5}\ N^2$	0.96
2.	100	$E = -63.452 + 0.22231\ N - 0.0001182\ N^2$	0.95
3.	110	$E = -81.924 + 0.25547\ N - 0.0001336\ N^2$	0.96
4.	120	$E = -94.6 + 0.27858\ N - 0.0001452\ N^2$	0.97
5.	130	$E = -113.43 + 0.31231\ N - 0.00016161\ N^2$	0.97
6.	140	$E = -143.248 + 0.37135\ N - 0.00019291\ N^2$	0.97
7.	150	$E = -178.372 + 0.4597\ N - 0.00023383\ N^2$	0.98

Table 4.5 Speed - discharge efficiency relationship of 15-cm mixed flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$Q = 71.278 \ \mathrm{Ln}\ N - 443.478$	0.995
2.	100	$Q = 71.4097 \ \mathrm{Ln}\ N - 445.897$	0.996
3.	110	$Q = 72.7614 \ \mathrm{Ln}\ N - 456.489$	0.995
4.	120	$Q = 73.753 \ \mathrm{Ln}\ N - 464.896$	0.996
5.	130	$Q = 74.3127 \ \mathrm{Ln}\ N - 470.75$	0.995
6.	140	$Q = 75.4598 \ \mathrm{Ln}\ N - 480.47$	0.990
7.	150	$Q = 75.959 \ \mathrm{Ln}\ N - 485.783$	0.990

The second degree equations fitted for pump speed-total head curves are presented (Table 4.6).

Similarly, the curves showing the relationship between pump speed and input power are plotted for various total static heads. The plotted curves are second degree in nature and are of the form,

$$P = K_1 + K_2N + K_3N^2$$

where,

P = input power, hp;

N = pump speed, rev/min; and

K_1, K_2, K_3 = constants.

The fitted equations for different total static heads are presented (Table 4.7).

4.2 Performance evaluation of the axial flow pump

For the 15-cm axial flow pump tests were conducted at 1600, 1700, 1800, 1900, 2000, 2100, 2200, 2300, 2400, 2500 and 2600 rev/min. The calculated values of discharge, total head, input power and efficiency are presented (Appendix XI to XXI). The maximum efficiency obtained was 18.05 per cent at 2500 rev/min at a total static head of 130 cm and a discharge of 24.88 L/s. The corresponding total head was 160.55 cm.

Table 4.6 Speed - total head relationship of 15-cm mixed flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$H = 0.88 - 0.001489\ N + 2.7424\times10^{-6}\ N^2$	0.990
2.	100	$H = 0.934 - 0.001364\ N + 2.6061\times10^{-6}\ N^2$	0.996
3.	110	$H = 1.29 - 0.0020418\ N + 2.9697\times10^{-6}\ N^2$	0.997
4.	120	$H = 2.052 - 0.0037424\ N + 3.9394\times10^{-6}\ N^2$	0.995
5.	130	$H = 2.032 - 0.0033636\ N + 3.5909\times10^{-6}\ N^2$	0.997
6.	140	$H = 2.012 - 0.0030121\ N + 3.2727\times10^{-6}\ N^2$	0.988
7.	150	$H = 2.11 - 0.002924\ N + 3.1061\times10^{-6}\ N^2$	0.990

Table 4.7 Speed - input power relationship of 15-cm mixed flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$P = 2.41 - 0.008997\ N + 1\times10^5\ N^2$	0.997
2.	100	$P = 2.76 - 0.009802\ N + 1.0455\times10^5\ N^2$	0.997
3.	110	$P = 2.95 - 0.010177\ N + 1.06818\times10^5\ N^2$	0.997
4.	120	$P = 3.14 - 0.01072\ N + 1.10606\times10^5\ N^2$	0.998
5.	130	$P = 3.19 - 0.01075\ N + 1.10606\times10^5\ N^2$	0.998
6.	140	$P = 3.18 - 0.10662\ N + 1.10606\times10^5\ N^2$	0.998
7.	150	$P = 3.50 - 0.011642\ N + 1.17424\times10^5\ N^2$	0.996

It was observed from the analysis that as total head increases discharge decreases. The maximum discharge obtained was 28.43 L/s at 2600 rev/min at a total static head of 90 cm and the efficiency was 15.96 per cent. At 2600 rev/min discharge varied from 28.42 to 13.07 L/s when the total static head varied from 90 to 300 cm. At 2500 rev/min discharge varied from 27.53 to 12.38 L/s against a total static head variation of 90 to 300 cm. It was also clear that for same total static head discharge increases with increase in pump speed. For example at 90 cm total static head discharge varied from 27.53 to 8.49 L/s as pump speed varied from 2600 to 1600 rev/min. Discharge obtained at maximum efficiency was 24.88 L/s which was less than the maximum discharge.

For each speed input power increase as discharge decreases, i.e., as total static head increases. For same constant total static head, input power increases with increase in pump speed. Input power varied from 0.94 to 3.08 hp as pump speed varied from 1600 to 2600 rev/min against a constant total static head of 90 cm. Input power for maximum discharge of 28.43 L/s was 3.08 hp and for maximum efficiency of 18.05 per cent was 2.95 hp.

As discharge increases, efficiency also increases and reaches a maximum value and then starts decreasing at constant pump speed. Maximum efficiency of 18.05 per cent was obtained at 2500 rev/min against a total static head of 130 cm. At

2500 rev/min, efficiency at 300 cm total static head and 12.38 L/s discharge was 12.63 per cent and as discharge increases, efficiency also increases and reaches a maximum of 18.05 per cent, at a discharge of 24.88 L/s and total static head of 130 cm. Then efficiency starts decreasing and reaches 12.63 per cent, when discharge increases to 27.53 L/s against a total static head of 90 cm. Efficiency at different combinations of speed, discharge and total head was less than 18.05 per cent. As speed increases, efficiency at first increases and then starts decreasing for constant total static head. For most total static head conditions, maximum efficiency was obtained at 2500 rev/min. For all constant total static head conditions, efficiency at 2600 rev/min was less than that at 2500 rev/min. So 2500 rev/min is the maximum pump speed limit beyond which efficiency decreases although discharge increases.

Performance characteristic curves were plotted for different speeds, 1600, 1700, 1800, 1900, 2000, 2100, 2200, 2300, 2400, 2500 and 2600 rev/min (Fig.4.18 to 4.28). These curves give the relationship of discharge with efficiency, input power and total head at each speed of 15-cm axial flow pump.

The plotted curves of discharge - efficiency relationships are second degree in nature. The curves are of the form,

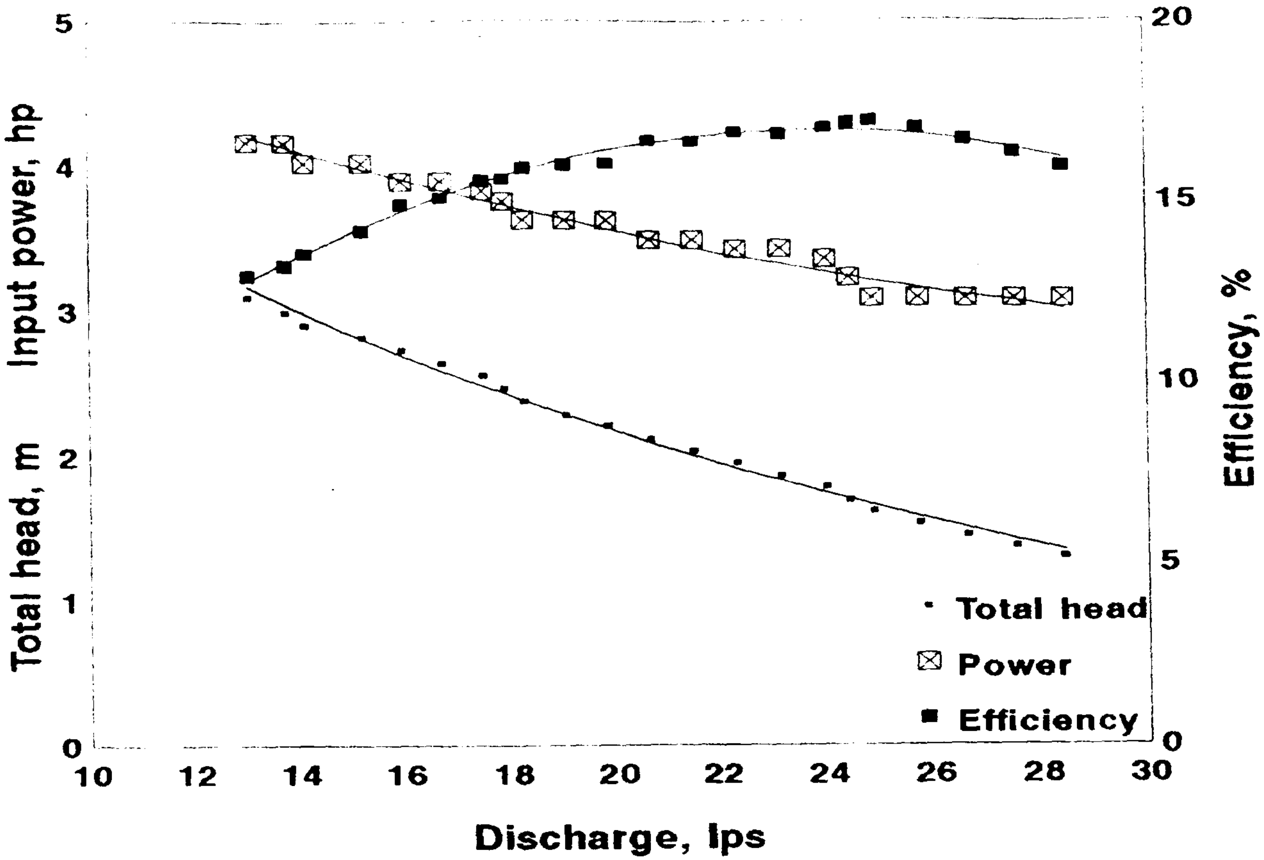

Fig.4.18 Performance characteristic curves of axial flow pump at 2600 rev/min

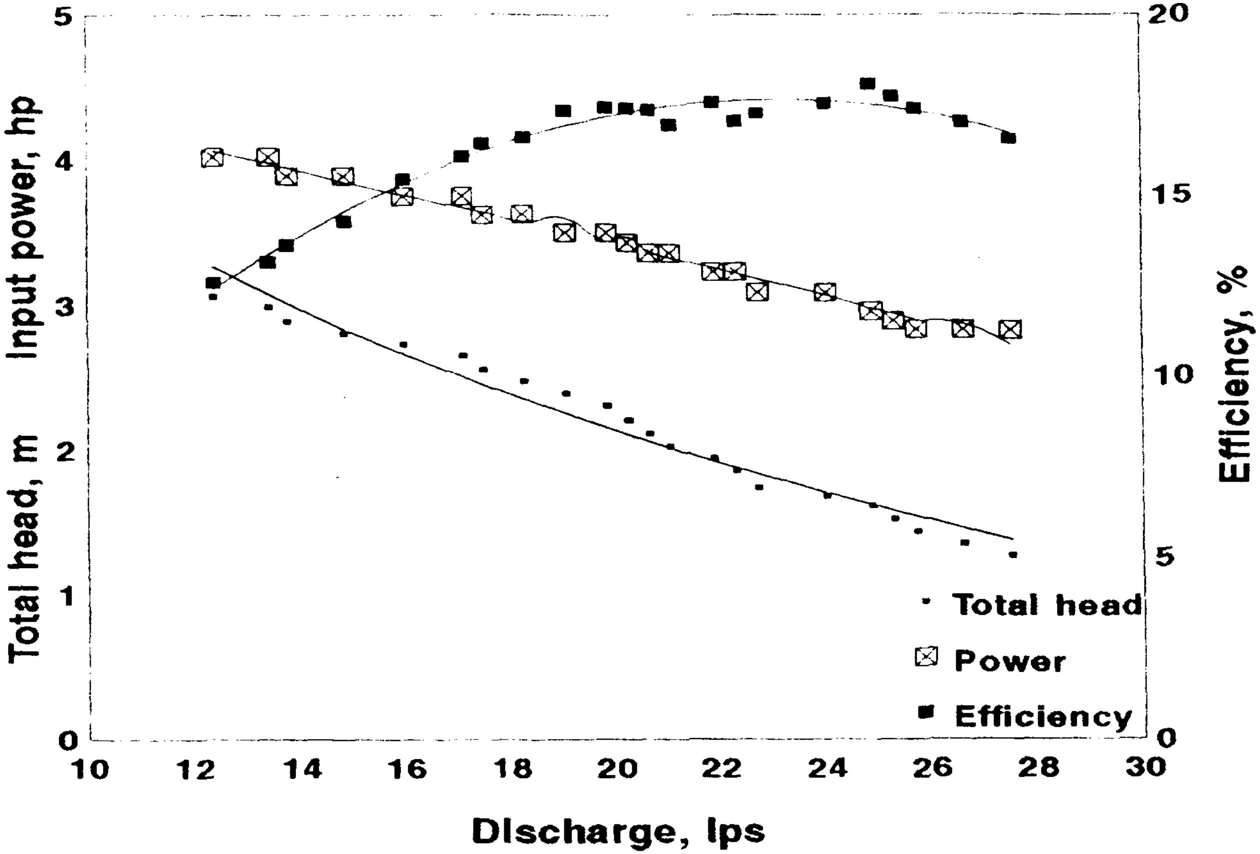

Fig.4.19 Performance characteristic curves of axial flow pump at 2500 rev/min

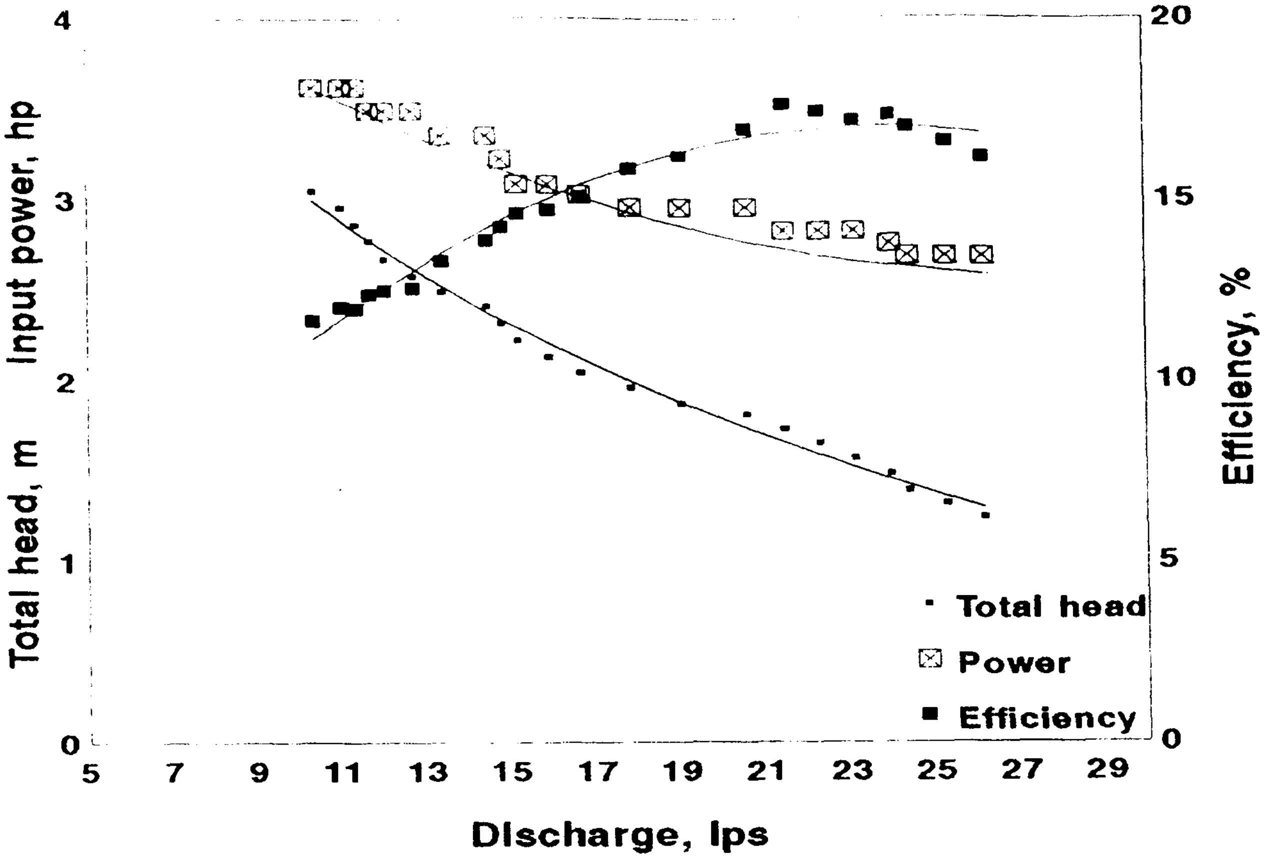

Fig.4.20 Performance characteristic curves of axial flow pump at 2400 rev/min

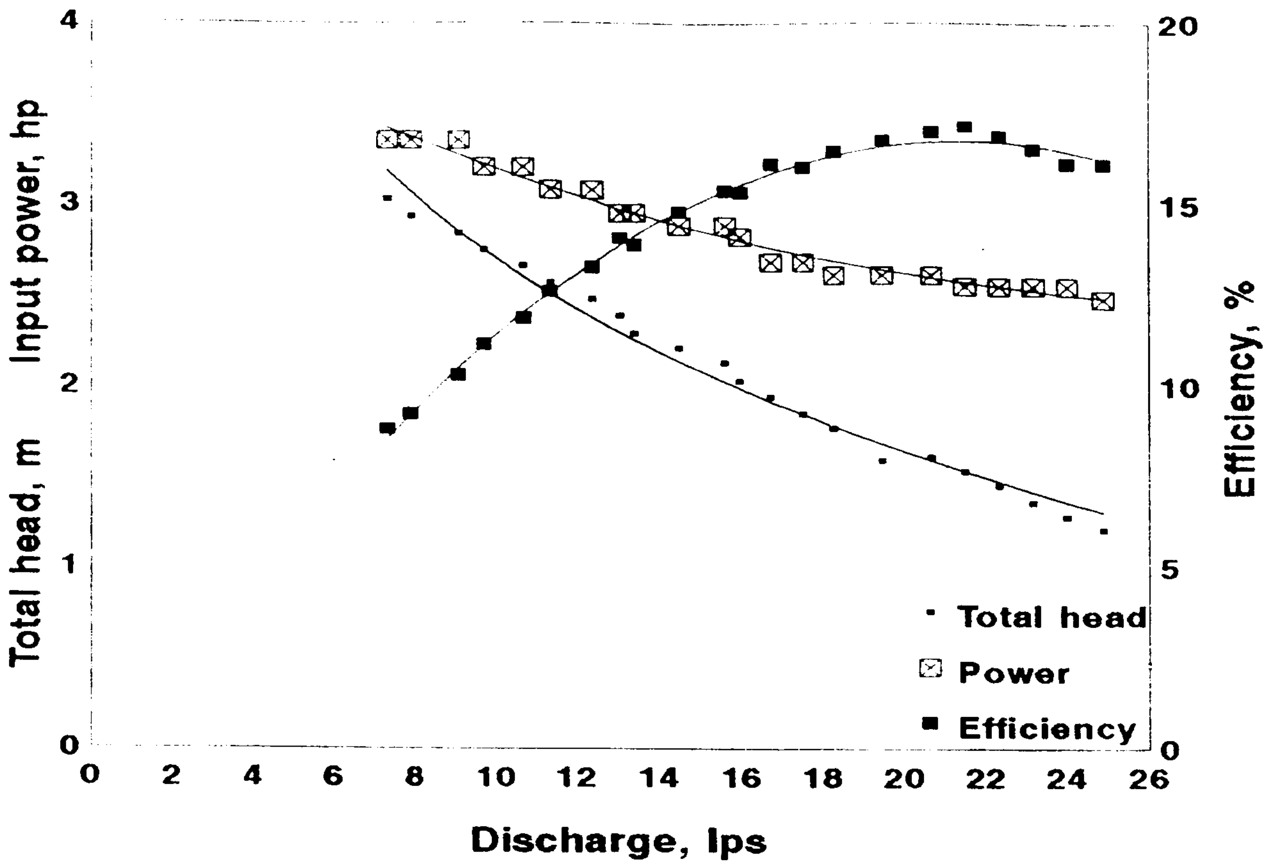

Fig.4.21 Performance characteristic curves of axial flow pump at 2300 rev/min

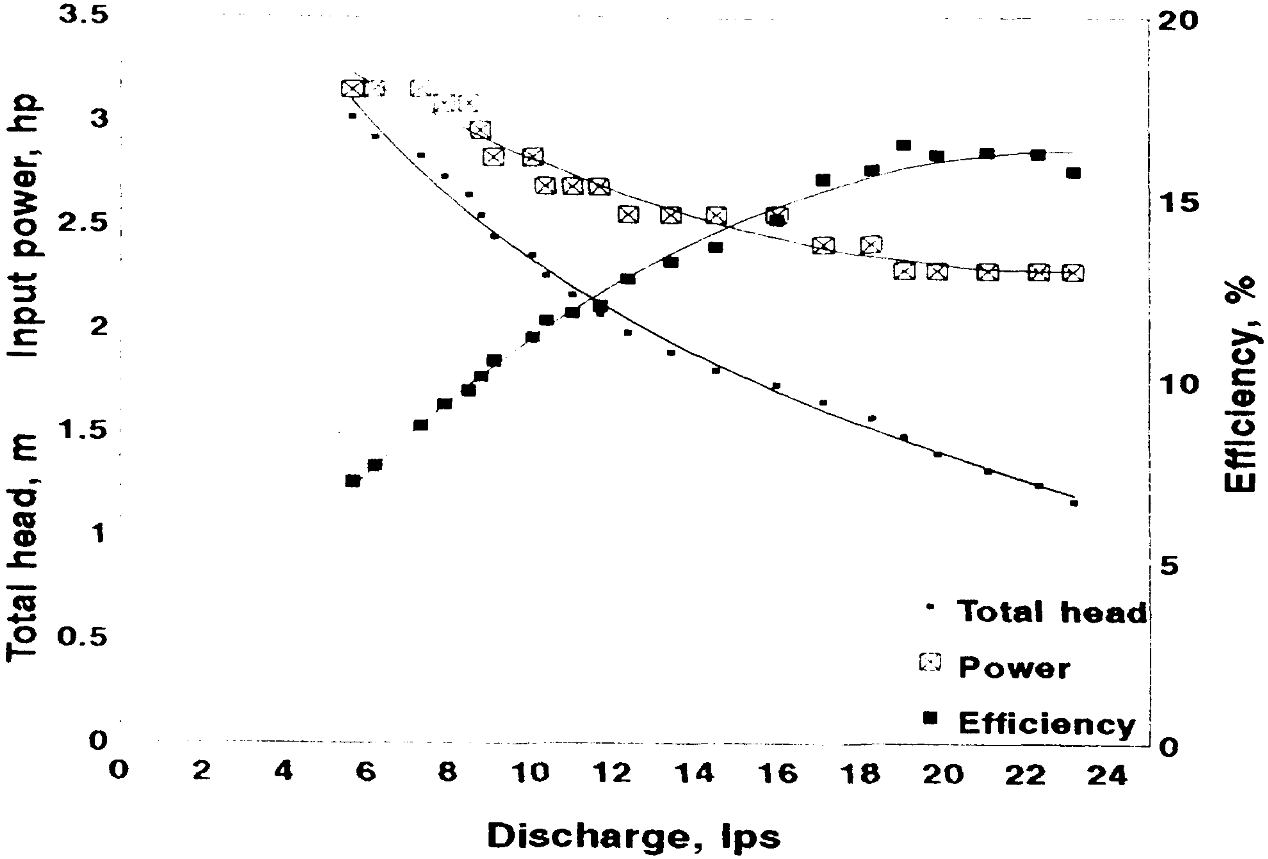

Fig.4.22 Performance characteristic curves of axial flow pump at 2200 rev/min

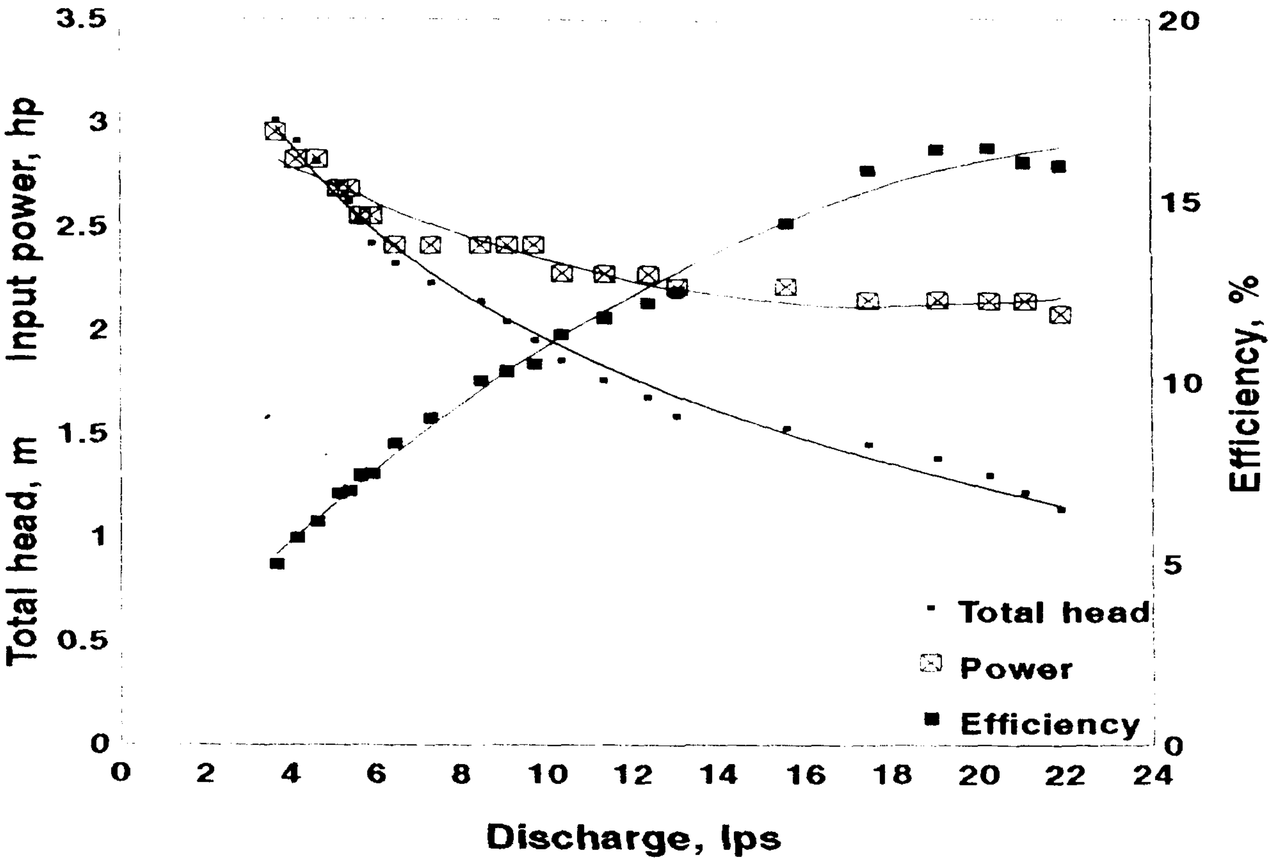

Fig.4.23 Performance characteristic curves of axial flow pump at 2100 rev/min

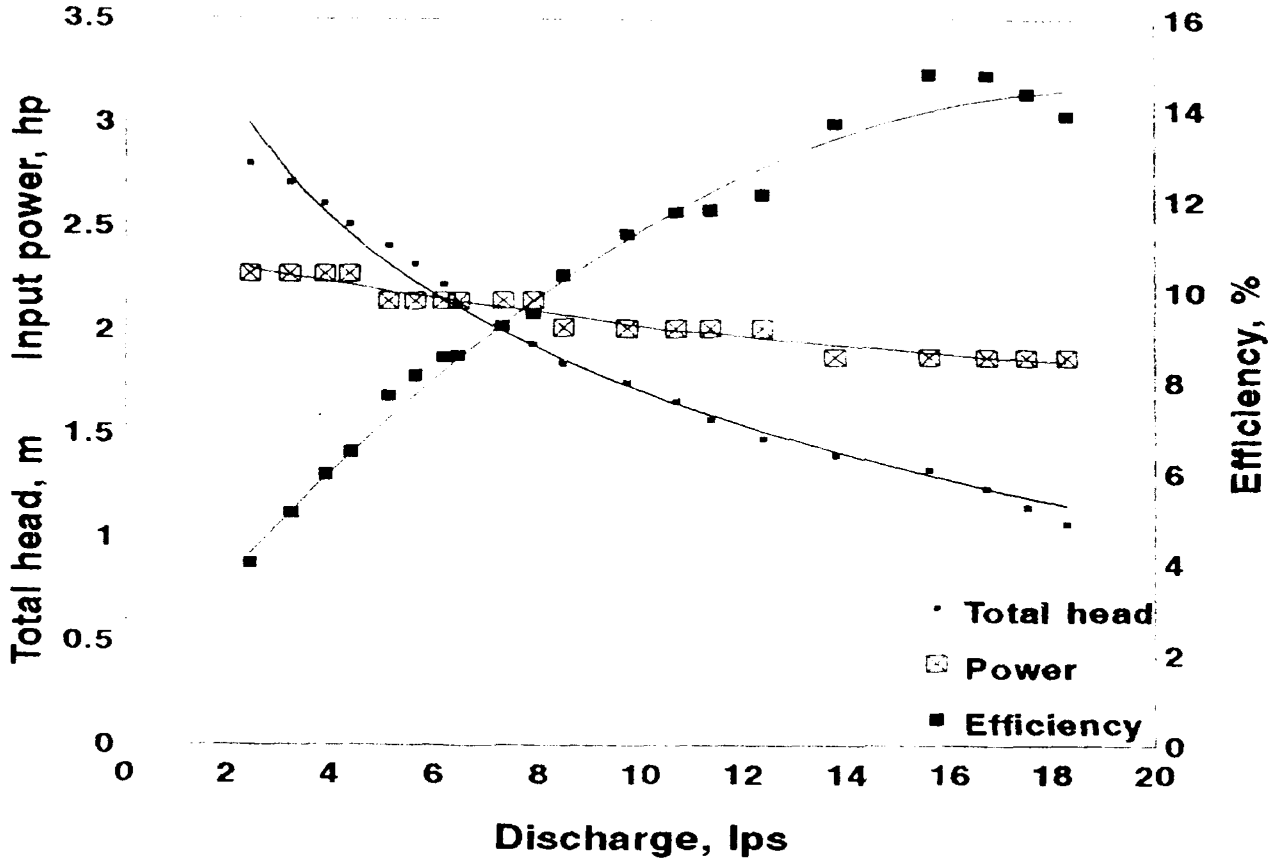

Fig.4.24 Performance characteristic curves of axial flow pump at 2000 rev/min

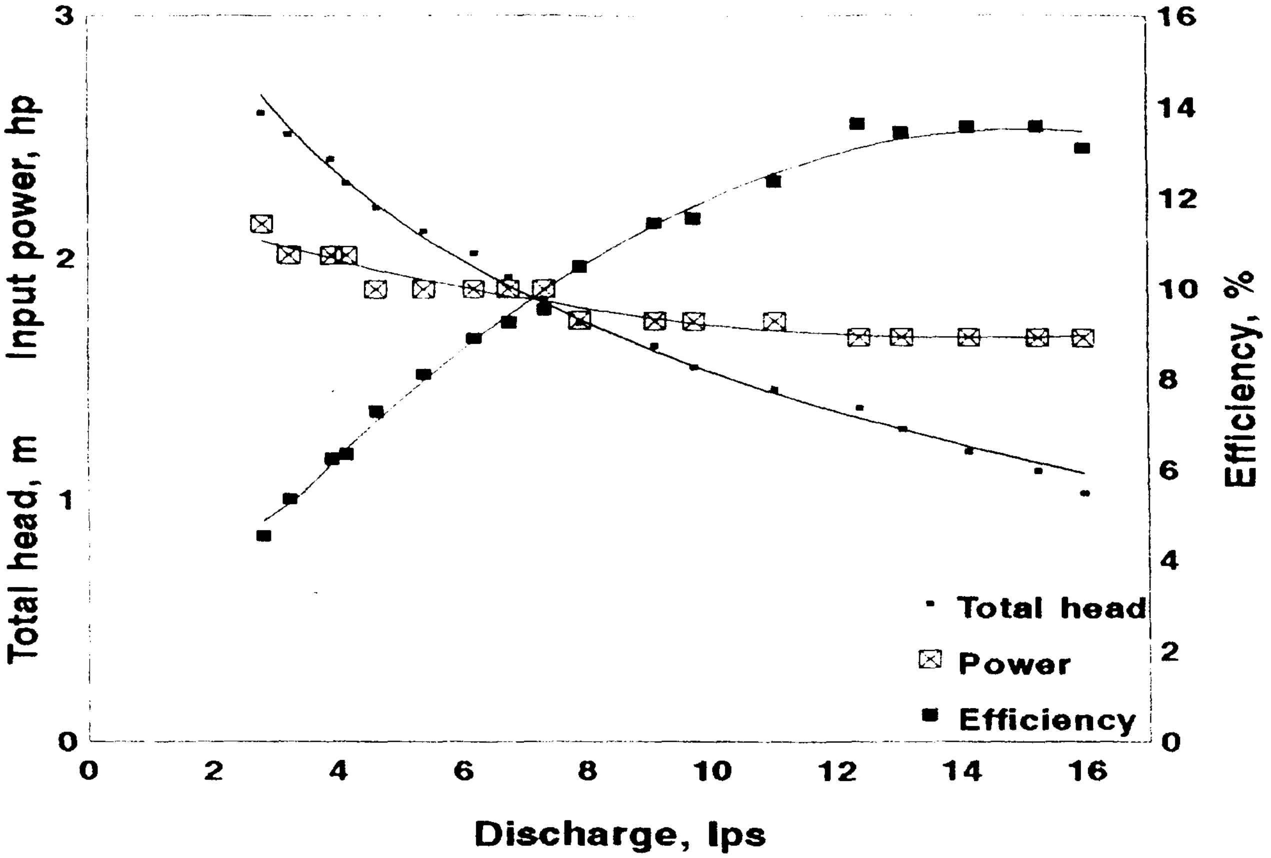

Fig.4.25 Performance characteristic curves of axial flow pump at 1900 rev/min

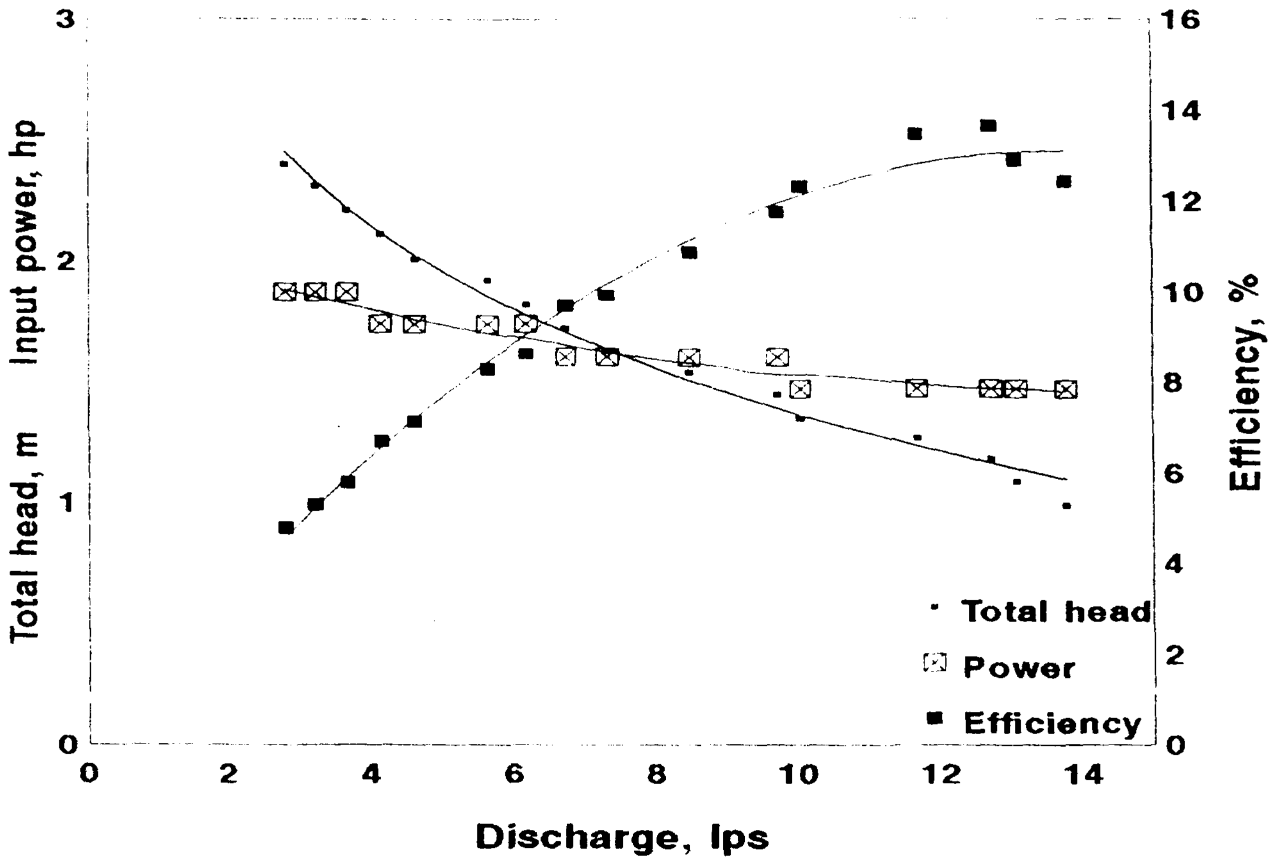

Fig.4.26 Performance characteristic curves of axial flow pump at 1800 rev/min

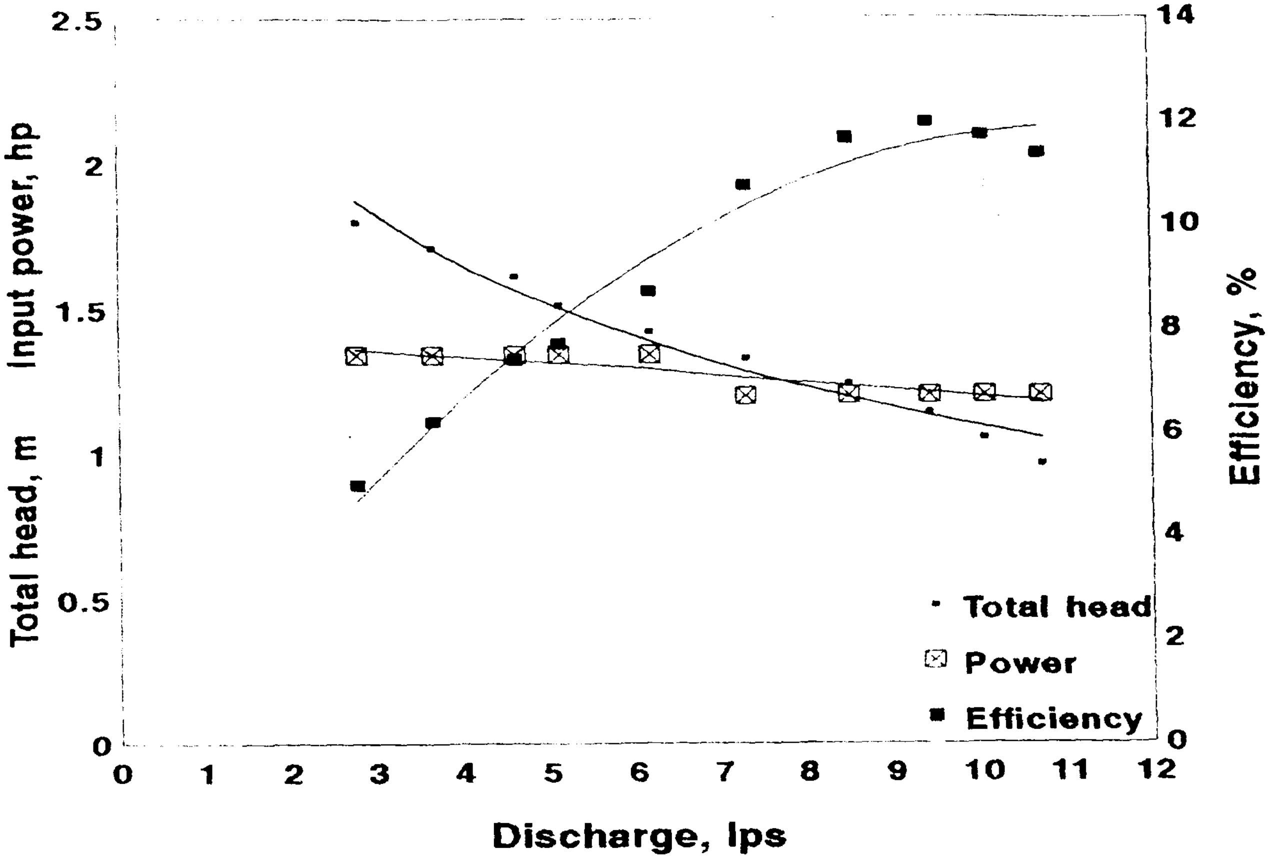

Fig.4.27 Performance characteristic curves of axial flow pump at 1700 rev/min

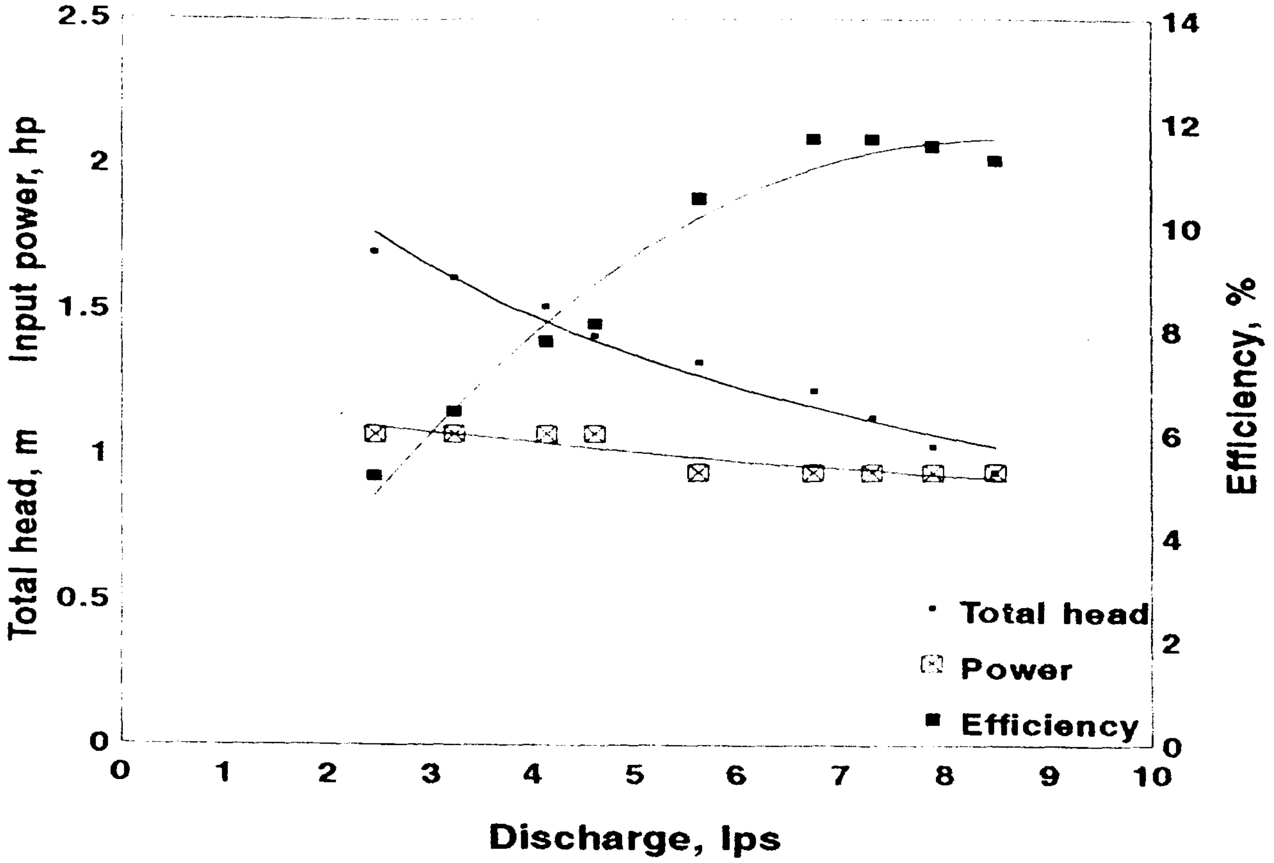

Fig.4.28 Performance characteristic curves of axial flow pump at 1600 rev/min

$$E = K_1 + K_2Q + K_3Q^2$$

where,

E = efficiency, per cent;

Q = discharge, L/s; and

K_1, K_2, K_3 = constants.

The fitted equations for various speeds are presented (Table 4.8).

The relationships between head and discharge are plotted and the trend is found to be of semi-logarithmic nature. The semi-logarithmic curves are of the form,

$$H = K_1 \mathrm{Ln}\ Q + K_2$$

where,

H = Total pump head, m;

Q = discharge, L/s; and

K_1, K_2, K_3 = constants.

The fitted equations for the 11 speeds selected are given (Table 4.9).

In the same manner curves were fitted for input power and discharge. These curves were found to be of second degree in nature and is of the form,

Table 4.8 Discharge - efficiency relationship of 15-cm axial flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	2600	$E = -3.5997 + 1.7289\ Q - 0.03633\ Q^2$	0.99
2.	2500	$E = -6.5467 + 2.0971\ Q - 0.04546\ Q^2$	0.97
3.	2400	$E = -1.4412 + 1.5533\ Q - 0.03274\ Q^2$	0.97
4.	2300	$E = -2.4277 + 1.8038\ Q - 0.04235\ Q^2$	0.99
5.	2200	$E = 0.07376 + 1.4104\ Q - 0.03063\ Q^2$	0.99
6.	2100	$E = 1.05858 + 1.2326\ Q - 0.02414\ Q^2$	0.99
7.	2000	$E = 1.10332 + 1.3878\ Q - 0.03586\ Q^2$	0.99
8.	1900	$E = 0.15338 + 1.7662\ Q - 0.05747\ Q^2$	0.99
9.	1800	$E = -0.27805 + 1.9411\ Q - 0.07048\ Q^2$	0.99
10.	1700	$E = -0.86657 + 2.2864\ Q - 0.10247\ Q^2$	0.98
11.	1600	$E = -1.89585 + 3.2083\ Q - 0.18976\ Q^2$	0.97

Table 4.9 Discharge - head relationship of 15-cm axial flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	2600	$H = -2.34359 \ \text{Ln}\ Q + 9.1846$	0.990
2.	2500	$H = -2.37116 \ \text{Ln}\ Q + 9.2356$	0.970
3.	2400	$H = -1.83713 \ \text{Ln}\ Q + 7.2899$	0.993
4.	2300	$H = -1.52505 \ \text{Ln}\ Q + 6.2148$	0.983
5.	2200	$H = -1.34402 \ \text{Ln}\ Q + 5.4235$	0.990
6.	2100	$H = -1.01784 \ \text{Ln}\ Q + 4.2976$	0.991
7.	2000	$H = -0.91648 \ \text{Ln}\ Q + 3.8171$	0.986
8.	1900	$H = -0.89687 \ \text{Ln}\ Q + 3.5917$	0.994
9.	1800	$H = -0.84798 \ \text{Ln}\ Q + 3.31773$	0.990
10.	1700	$H = -0.61197 \ \text{Ln}\ Q + 2.50068$	0.971
11.	1600	$H = -0.59804 \ \text{Ln}\ Q + 2.30554$	0.960

$$P = K_1 + K_2Q + K_3Q^2$$

where,

P = input power, hp;

Q = discharge, L/s; and

K_1, K_2, K_3 = constants.

The equations fitted for input power - discharge curves for different speeds are presented (Table 4.10).

Performance curves were also plotted for 15-cm axial flow pump at various constant total static heads. These curves give the relationship between pump speed and discharge, efficiency, total pump head, and input power. Efficiency increases as pump speed increases and for almost all total static heads efficiency becomes maximum at 2500 rev/min and then it decreases. Discharge, total head and input power increases as pump speed increases for each total static head. As speed decreases total static head against which pump could be operated also decreases. At 1600 rev/min discharge obtained was very low when total static head reaches 170 cm. The discharge obtained was only 2.45 L/s with an efficiency of 5.2 per cent. Performance curves at constant total static head for 15-cm axial flow pump were plotted only upto a total static head of 160 cm. Curves were plotted for 90, 100, 110, 120, 130, 140, 150 and 160 cm total static heads (Fig.4.29 to Fig.4.36).

Table 4.10 Discharge - input power relationship of 15-cm axial flow pump at various speeds

Sl. No.	Speed (rev/min)	Regression equations	R^2
1.	2600	$P = 5.9412 - 0.1588\ Q + 0.001961\ Q^2$	0.96
2.	2500	$P = 4.9325 - 0.0622\ Q + -0.000664\ Q^2$	0.98
3.	2400	$P = 5.1973 - 0.1862\ Q + 0.003295\ Q^2$	0.97
4.	2300	$P = 4.2246 - 0.1254\ Q + 0.002245\ Q^2$	0.98
5.	2200	$P = 3.9360 - 0.1424\ Q + 0.003052\ Q^2$	0.96
6.	2100	$P = 3.2258 - 0.1230\ Q + 0.003396\ Q^2$	0.93
7.	2000	$P = 2.4152 - 0.0494\ Q + 0.001019\ Q^2$	0.94
8.	1900	$P = 2.2977 - 0.0889\ Q + 0.003138\ Q^2$	0.93
9.	1800	$P = 2.1037 - 0.0871\ Q + 0.002945\ Q^2$	0.93
10.	1700	$P = 1.4258 - 0.0215\ Q + -0.000164\ Q^2$	0.80
11.	1600	$P = 1.1947 - 0.0439\ Q + 0.001399\ Q^2$	0.80

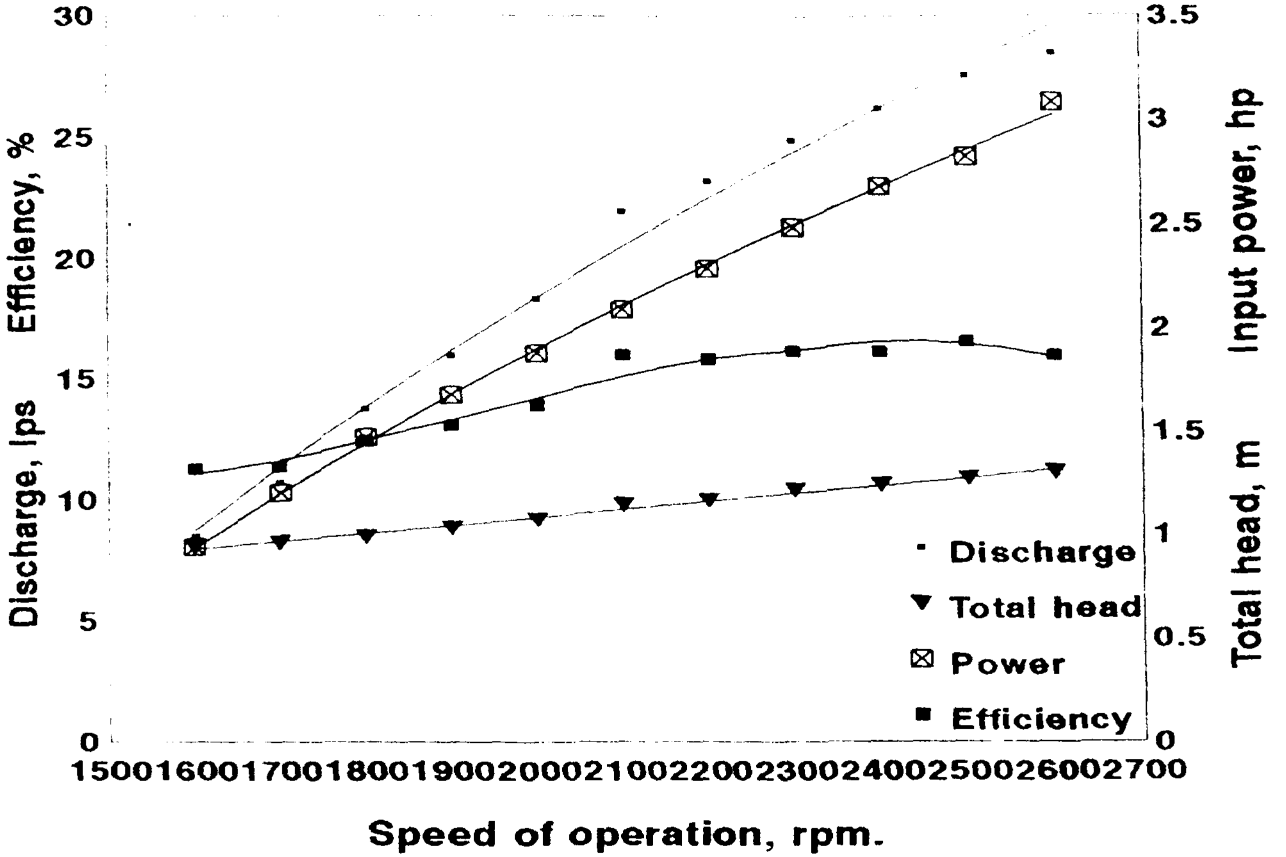

Fig 4.29 Performance characteristics of axial flow pump at total static head of 0.9m

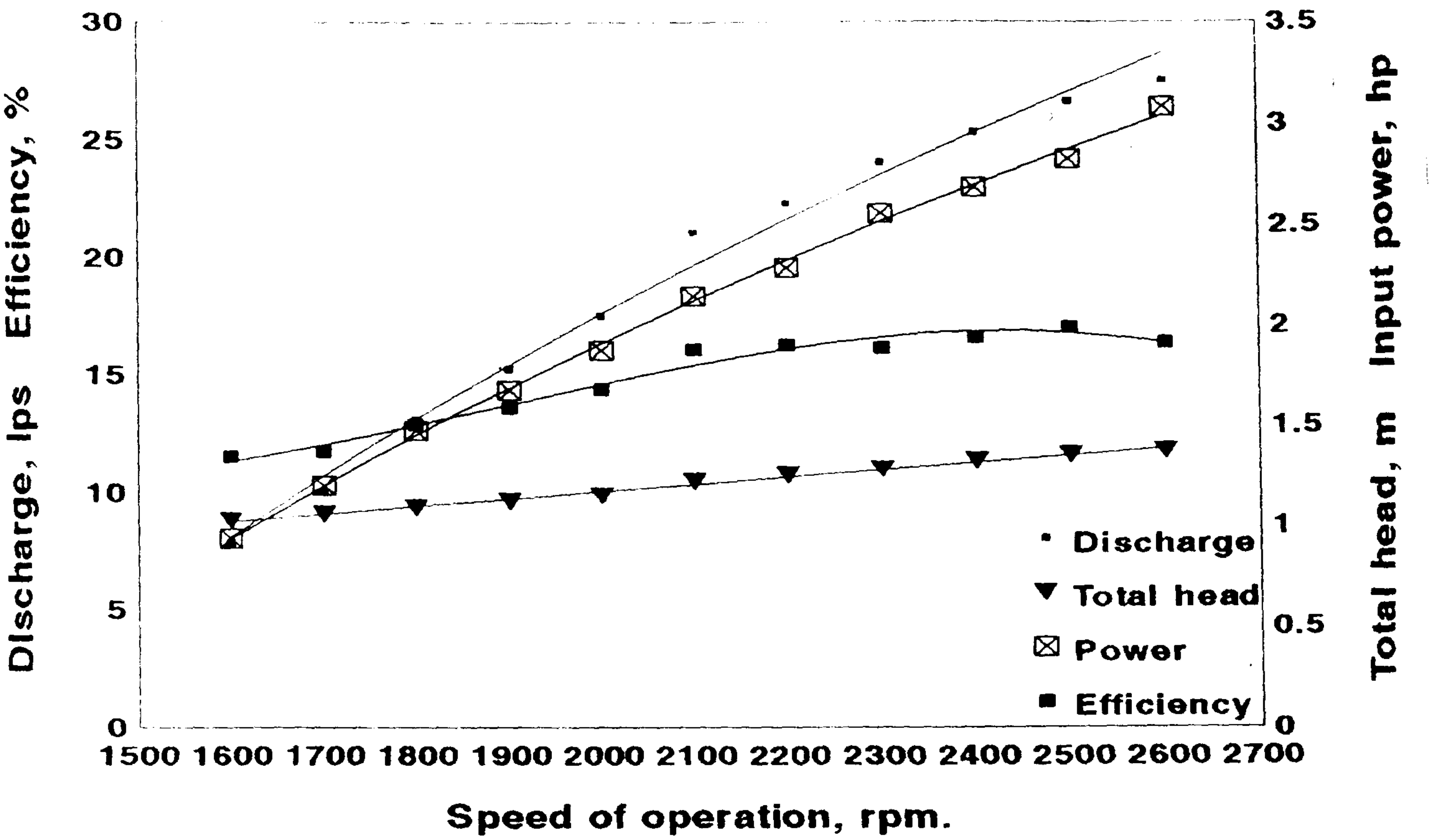

Fig 4.30 Performance characteristics of axial flow pump at total static head of 1m

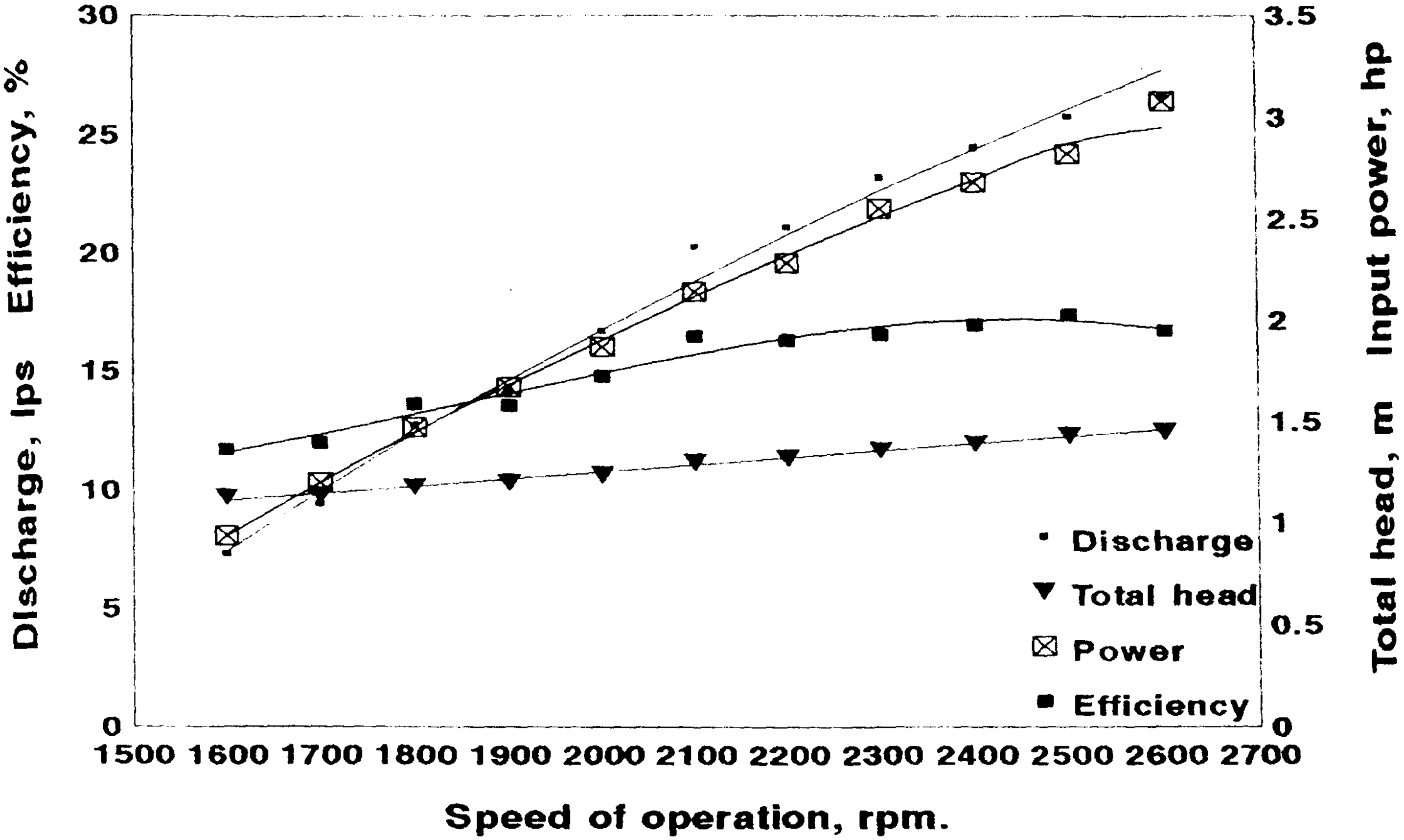

Fig 4.31 Performance characteristics of axial flow pump at total static head of 1.1m

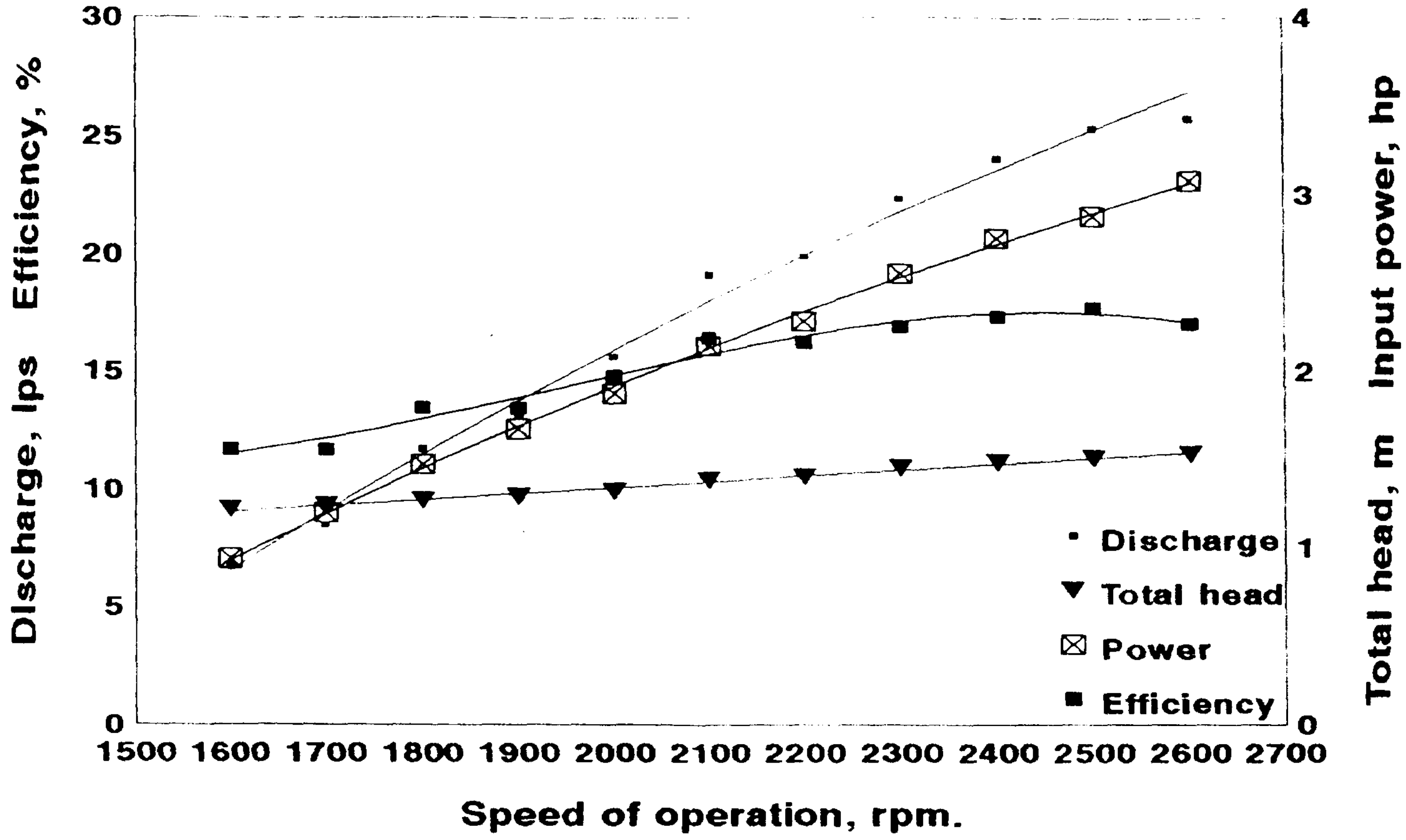

Fig 4.32 Performance characteristics of axial flow pump at total static head of 1.2m

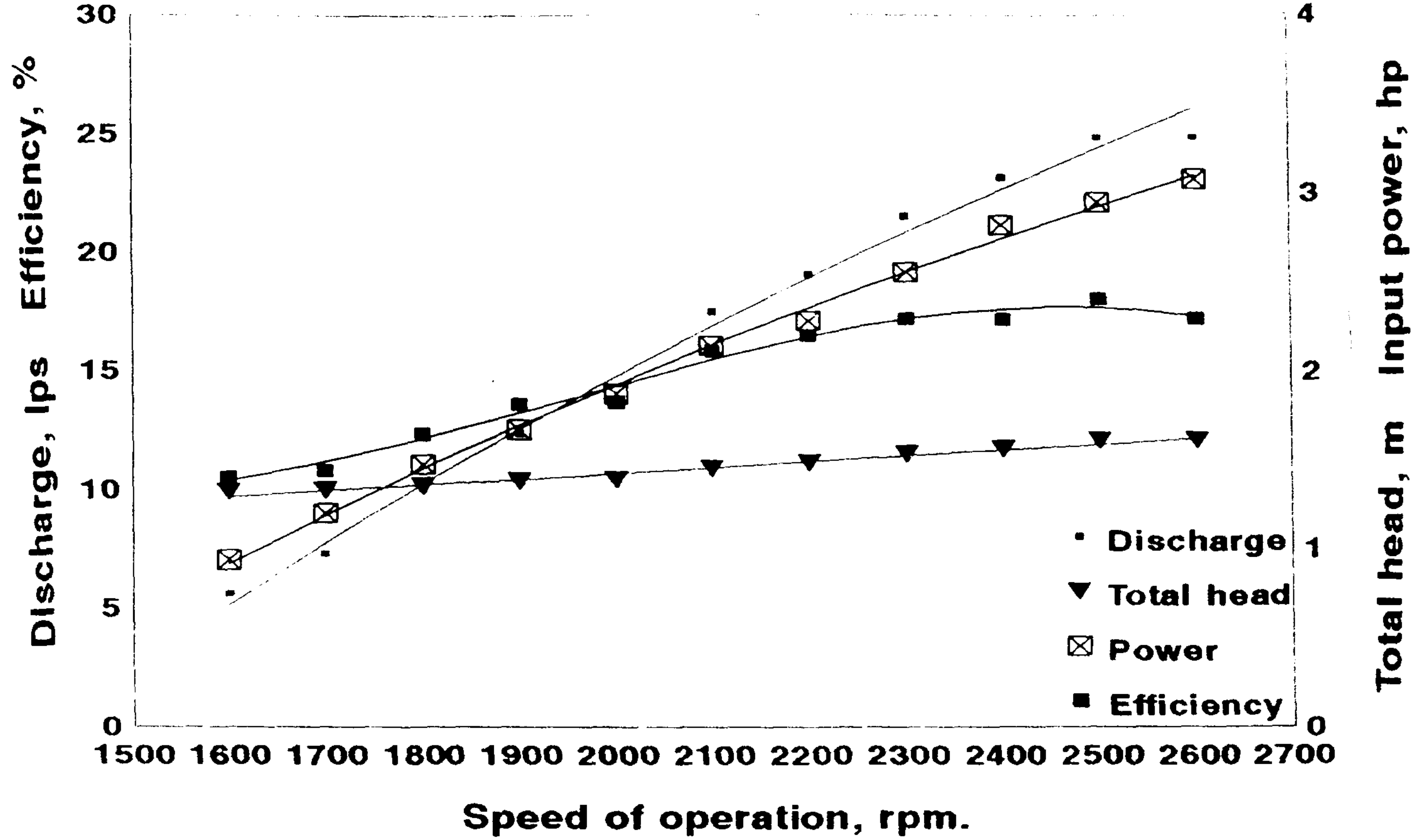

Fig 4.33 Performance characteristics of axial flow pump at total static head of 1.3m

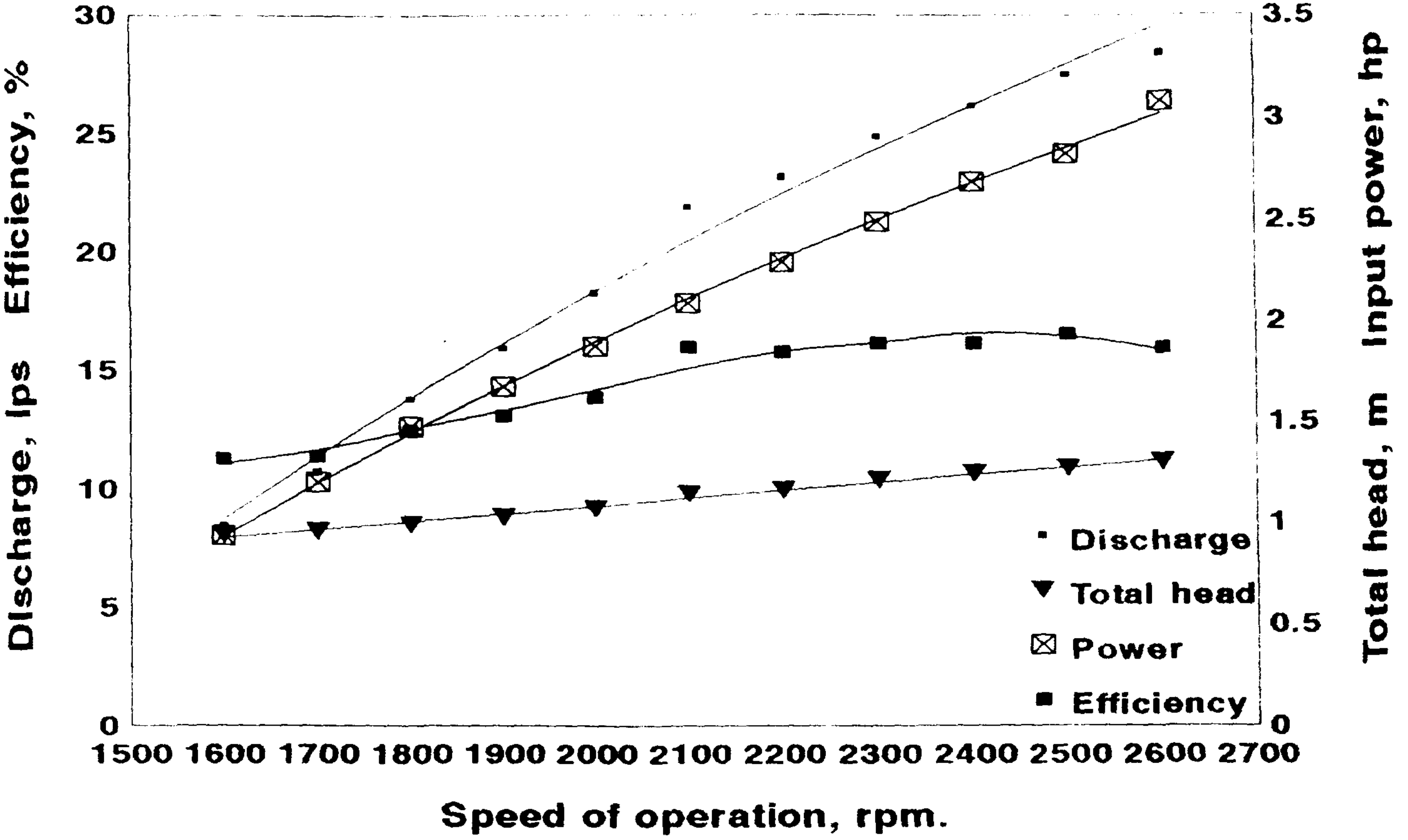

Fig 4.34 Performance characteristics of axial flow pump at total static head of 1.4m

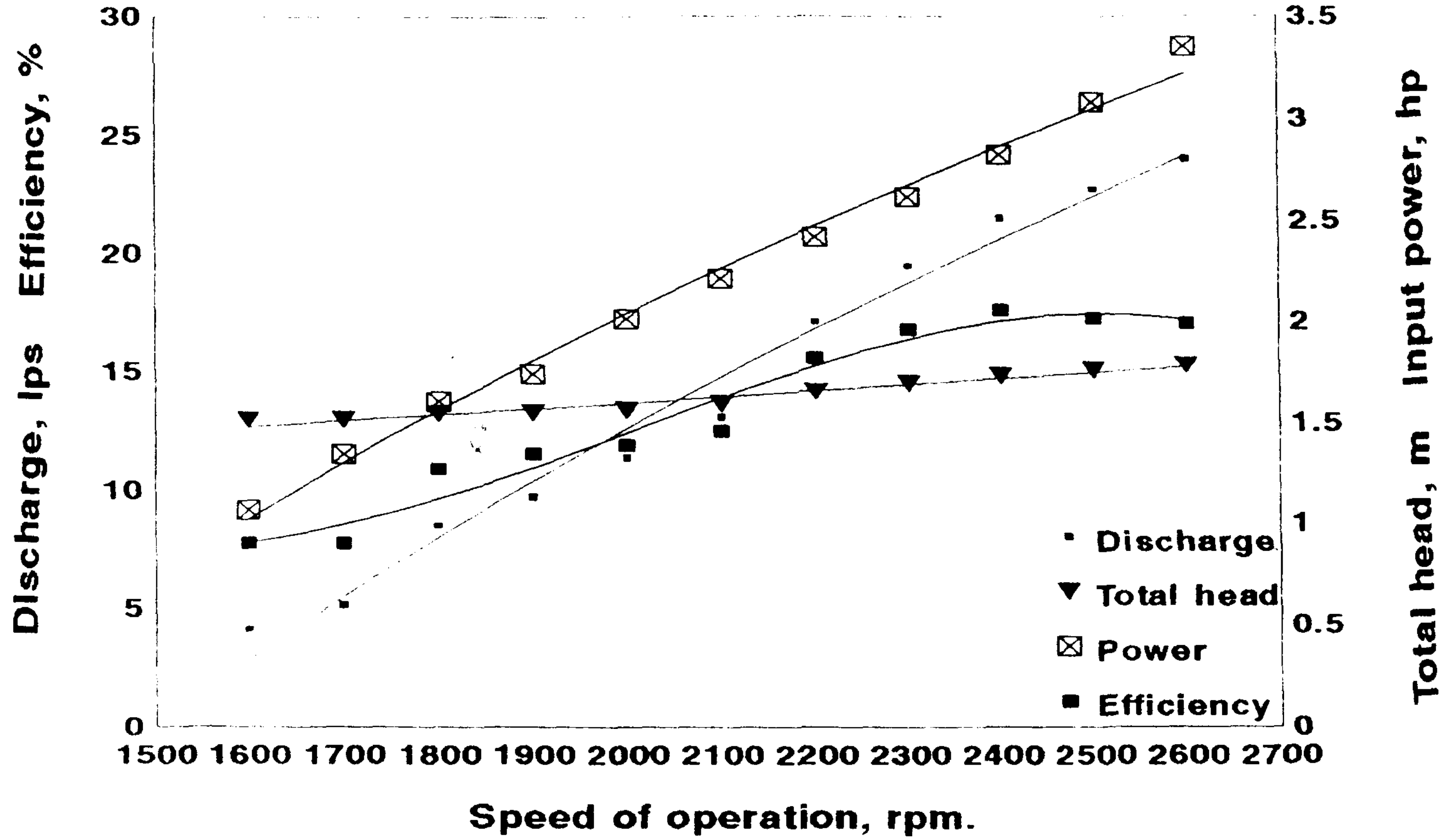

Fig 4.35 Performance characteristics of axial flow pump at total static head of 1.5m

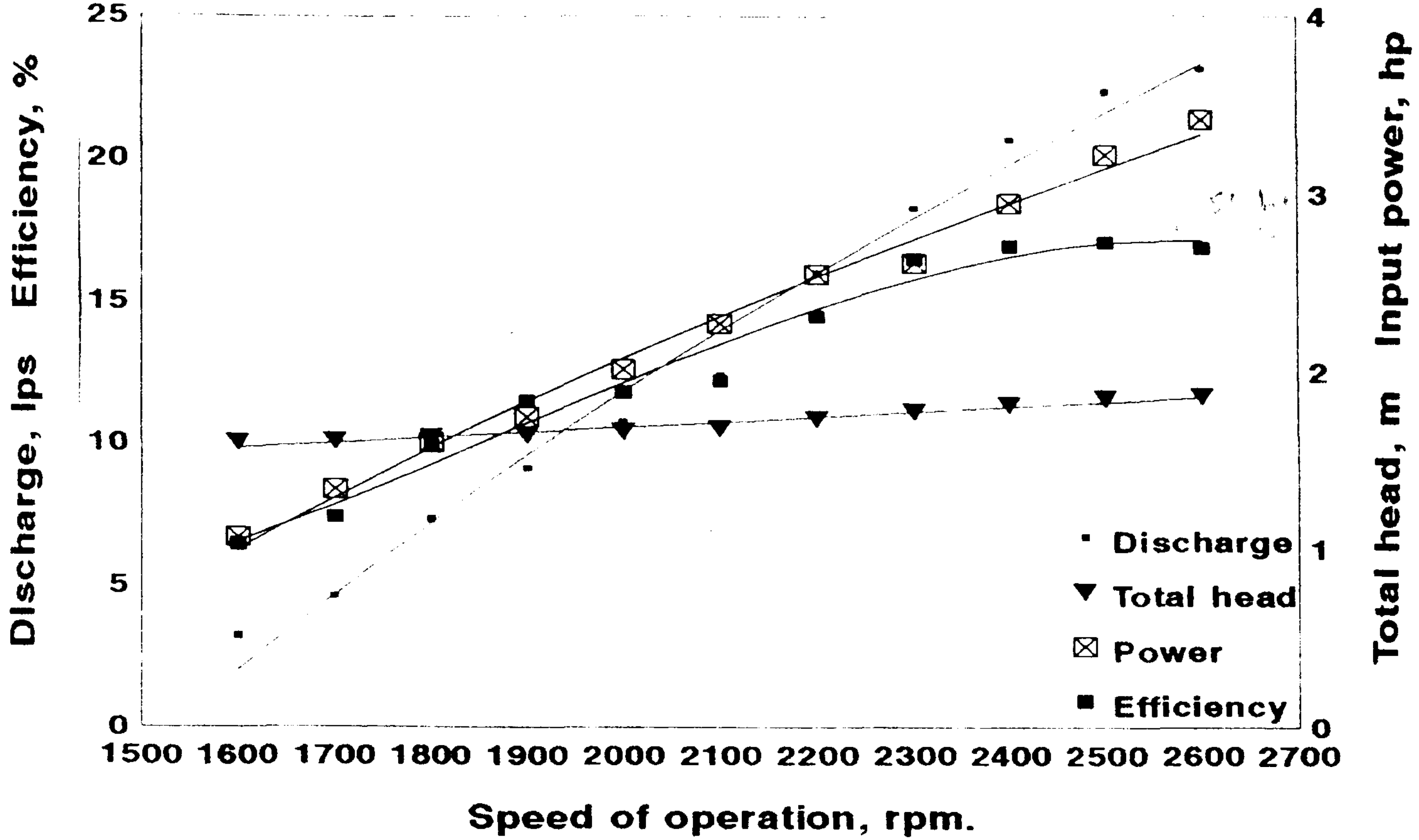

Fig 4.36 Performance characteristics of axial flow pump at total static head of 1.6m

The plotted speed-efficiency curves for different constant total static heads are third degree in nature. The third order equations are of the form,

$$E = K_1 + K_2N + K_3N^2 + K_4N^3$$

where,

E = efficiency, per cent;

N = pump speed, rev/min; and

K_1, K_2, K_3, K_4 = constants.

The fitted equations for various total static heads are presented (Table 4.11).

The curves showing the speed - discharge relationship of 15-cm axial flow pump at various total static heads are semi-logarithmic in nature. The equations are of the form,

$$Q = K_1 \mathrm{Ln}\, N - K_2$$

where,

Q = discharge, L/s;

N = pump speed, rev/min; and

K_1, K_2 = constants.

The equations fitted for various total static heads are presented (Table 4.12).

Table 4.11 Speed - efficiency relationship of 15-cm axial flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$E = 92.972 - 0.1408\ N + 7.725\text{x}10^{-5}\ N^2 - 1.327\text{x}10^{-8}\ N^3$	0.97
2.	100	$E = 70.063 - 0.1063\ N + 6.044\text{x}10^{-5}\ N^2 - 1.057\text{x}10^{-8}\ N^3$	0.98
3.	110	$E = 55.227 - 0.0843\ N + 4.989\text{x}10^{-5}\ N^2 - 8.902\text{x}10^{-9}\ N^3$	0.97
4.	120	$E = 83.071 - 0.1252\ N + 6.936\text{x}10^{-5}\ N^2 - 1.191\text{x}10^{-8}\ N^3$	0.97
5.	130	$E = 93.570 - 0.1458\ N + 8.076\text{x}10^{-5}\ N^2 - 1.383\text{x}10^{-8}\ N^3$	0.98
6.	140	$E = 77.294 - 0.1302\ N + 7.547\text{x}10^{-5}\ N^2 - 1.319\text{x}10^{-8}\ N^3$	0.97
7.	150	$E = 140.43 - 0.2210\ N + 0.000118\ N^2 - 1.953\text{x}10^{-8}\ N^3$	0.96
8.	140	$E = 52.141 - 0.0952\ N + 5.810\text{x}10^{-5}\ N^2 - 1.026\text{x}10^{-8}\ N^3$	0.97

Table 4.12 Speed - discharge relationship of 15-cm axial flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$Q = 42.9626 \ \text{Ln}\ N - 308.218$	0.990
2.	100	$Q = 42.3228 \ \text{Ln}\ N - 304.120$	0.990
3.	110	$Q = 41.7980 \ \text{Ln}\ N - 300.975$	0.990
4.	120	$Q = 41.9145 \ \text{Ln}\ N - 302.703$	0.991
5.	130	$Q = 43.2197 \ \text{Ln}\ N - 313.743$	0.991
6.	140	$Q = 43.9006 \ \text{Ln}\ N - 319.931$	0.990
7.	150	$Q = 43.8561 \ \text{Ln}\ N - 320.730$	0.983
8.	160	$Q = 43.8765 \ \text{Ln}\ N - 321.705$	0.987

The relationship between pump speed and total head are plotted for various total static heads and the trend is found to be of first degree nature. The first degree equations are of the form,

$$H = K_1 + K_2N$$

where,

H = Total head, m;

N = pump speed, rev/min; and

K_1, K_2 = constants.

The first degree equations fitted at various total static heads are presented (Table 4.13).

Similarly the curves showing the relationship between pump speed and input power are plotted for various total static heads. The plotted curves are semi-logarithmic in nature and are of the form,

$$P = K_1 \text{ Ln } N - K_2$$

where,

P = input power, hp;

N = pump speed, rev/min; and

K_1, K_2 = constants.

The fitted equations for different total static heads are presented (Table 4.14).

Table 4.13 Speed - total head relationship of 15-cm axial flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	$H = 0.3096 + 0.0003855 \; N$	0.991
2.	100	$H = 0.4457 + 0.0003609 \; N$	0.990
3.	110	$H = 0.5619 + 0.0003445 \; N$	0.990
4.	120	$H = 0.6731 + 0.0003327 \; N$	0.992
5.	130	$H = 0.7701 + 0.0003264 \; N$	0.981
6.	140	$H = 0.8861 + 0.00031182 \; N$	0.979
7.	150	$H = 0.9965 + 0.0002991 \; N$	0.963
8.	160	$H = 0.5799 + 0.0002846 \; N$	0.960

Table 4.14 Speed - input power relationship of 15-cm axial flow pump at various total static heads

Sl. No.	Total static head (cm)	Regression equations	R^2
1.	90	P = 4.2943 Ln N - 30.7488	0.990
2.	100	P = 4.32345 Ln N - 30.9705	0.998
3.	110	P = 4.32345 Ln N - 30.9705	0.998
4.	120	P = 4.4077 Ln N - 31.5907	0.998
5.	130	P = 4.4994 Ln N - 32.2783	0.997
6.	140	P = 4.4140 Ln N - 31.5226	0.997
7.	150	P = 4.5276 Ln N - 32.3787	0.993
8.	160	P = 4.7974 Ln N - 34.3901	0.991

The above detailed evaluation of the performance of 15-cm axial flow and mixed flow pump establishes that efficiency and discharge of mixed flow pump are more than that of axial flow pump at low total static heads. The drop in efficiency as total head increases is more for mixed flow pump than axial flow pump. Maximum efficiency of axial flow pump was 18.05 per cent and the discharge obtained was 24.88 L/s at this efficeincy, whereas maximum efficiency of mixed flow pump was 42.16 per cent and the discharge obtained at this efficiency was 49.47 L/s. Also analysis of the head-discharge, efficiency discharge and input power discharge relationships at various speeds indicate that the optimum working speed of 15-cm axial flow pump is 2500 rev/min and that of 15-cm mixed flow pump is 1000 rev/min.

Limitations of the study

Since the present study was conducted as a part of the post-graduate research programme, the study had the inherent limitations in terms of time and coverage. Since the third objective would require more time, involvement of lot of man power and the necessary physical conditions were not feasible, it was decided to drop the third objective.

Besides, the `petti and para' could not be tested at heads higher than 90 cm and at the same time the axial flow pump and mixed flow pump could not be tested at heads less than 90 cm due to the limitations in the test bed. Therefore the fourth objective could not be pursued.

Further research work

The present study being the first investigation in India in the performance of the Thai axial flow and mixed flow pumps has limitations. The studies on the pump performance were limited to testing only two units imported from Thailand namely 15-cm axial flow and 15-cm mixed flow pumps. However, there are various other sizes of pumps which need to be studied. The theoretical investigations were limited to the analysis of basic hydraulics and functional relationships.

Summary

SUMMARY

Irrigation and drainage requirements on farms often require the use of portable, high capacity low lift pumps powered by small engines or electric motors. At present centrifugal pump is the most popular mechanically operated pumping device used. But the efficiency of centrifugal pump is considerably affected whenever the operating condition differ from the optimum. Highly efficient factory made propeller pumps used in the western countries are expensive and require elaborate procedures in installation, operation and maintenance. At the same time axial flow and mixed flow pumps are in extensive use in Thailand mainly for dewatering.

Considering the above aspects it was decided to study the performance characteristics of a 15-cm axial and a 15-cm mixed flow pumps imported from Thailand. The objective of the project was to study the design parameters and efficiency characteristics of these pumps and to conduct laboratory and field testing of the above pumps for performance evaluation.

The laboratory tests were conducted at a specially designed and constructed test bench at the head quarters of the Kerala Agricultural University, Vellanikkara, Trichur. The test bench has the facility for testing pumps with discharges upto 3000 L/s and heads upto 3.5 m. The tests were

conducted for 15-cm mixed flow and axial flow pumps for different speeds. These speeds were obtained by using pulleys of different diameters. For each speed, tests were done at different water levels in the pumping sump. The readings of level indicator and hook gauge and the time taken for five revolutions of energy meter disc were taken at each water level, when a steady state condition was reached. For each test, from the observed readings discharge, input power and efficiency were calculated.

The mixed flow pump was tested at the speeds of 600, 650, 700, 750, 800, 850, 900, 950, 1000 and 1050 rev/min. The maximum efficiency of 42.16 per cent was obtained at a speed of 1000 rev/min, against a total head of 217.33 cm and a discharge rate of 49.47 L/s. The input power corresponding to this efficiency was 3.4 hp. The efficiency was always less than this for any other combination of head, speed and input power. It was observed that as total head increases discharge decreases. It was also observed that for same total static head discharge increases with increase in speed. The input power increases with increase in speed and for each speed input power increases as total head increases. Efficiency decreases with increase in total head at various speeds. As speed increases, efficiency also increases and reaches a maximum at 1000 rev/min speed and then it decreases. For each total static head, maximum efficiency was obtained at 1000 rev/min.

Characteristic curves of 15-cm mixed flow pump were fitted for 10 different speeds. The plotted curves of discharge - efficiency relationships are linear in nature. The curves are of the form,

$$E = K_1 + K_2Q$$

Where,

E = efficiency, %;

Q = discharge, L/s; and

K_1, K_2 = constants.

The fitted curves of head and discharge are second degree in nature and is of the form,

$$H = K_1 + K_2Q + K_3Q^2$$

Where,

H = total pump head, m;

Q = discharge, L/s; and

K_1, K_2, K_3 = constants.

The curves of input power and discharge relationships are also second degree in nature.

Performance curves were also plotted for the mixed flow pump keeping the total static head constant. The speed - discharge relationship at various constant total static heads are of semi-logarithmic nature. Curves of speed - efficiency, speed - total head and speed - input power relationships are of second degree in nature.

Tests for axial flow pumps were conducted at 1600, 1700, 1800, 1900, 2000, 2100, 2200, 2300, 2400, 2500 and 2600 rev/min. The maximum efficiency obtained was 18.05 per cent at 2500 rev/min at a total head of 160.55 cm and a discharge of 24.88 L/s. The corresponding input power was 2.95 hp. It was observed that as total head decreases discharge increases and for same total static head discharge increases with increase in pump speed. For each speed input power increases as total head increases and for constant total static head as speed increases input power increases. For each speed,efficiency at first increases and reaches a maximum value and then decreases as total head increases. For most total static head conditions, maximum efficiency was obtained at 2500 rev/min.

Performance characteristic curves were plotted for different speeds. The plotted curves of discharge - efficiency and discharge - input power relationships are second degree in nature. The head - discharge relationship curves are semi-logarithmic in nature and are of the form,

$$H = K_1 \operatorname{Ln} Q + K_2$$

Where,

H = total pump head, m;

Q = discharge, L/s; and

K_1, K_2 = constants.

Performance curves were also plotted for 15-cm axial flow pump keeping total static head constant. The curves showing the speed - discharge and speed - input power relationships are semi-logarithmic in nature. Speed -efficiency curves for different constant total static heads are third degree in nature. The curves showing the speed - total head relationships are first degree curves.

Detailed evaluation of the performance of 15-cm axial flow and mixed flow pumps indicate that the mixed flow has greater discharge than the axial flow pump at all total heads. The best performance of mixed flow pump was noted at a working speed of 1000 rev/min and that of axial flow pump was noted at a working speed of 2500 rev/min.

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Appendices

Appendix I

Performance of 15-cm mixed flow pump at 1050 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	40.90	97.90	228.80	7.00	51.67	1.58	4.00	39.41
2	100	39.17	93.72	232.89	6.90	50.57	1.57	4.00	39.26
3	110	37.50	83.89	237.33	6.80	49.47	1.57	4.00	39.14
4	120	35.87	85.93	241.80	6.70	48.38	1.56	4.10	38.04
5	130	33.51	80.28	243.80	6.55	46.76	1.52	4.10	37.07
6	140	29.82	71.46	241.26	6.30	44.12	1.42	4.15	34.20
7	150	27.07	64.96	242.03	6.10	42.03	1.36	4.20	32.29
8	160	24.49	58.86	243.30	5.90	39.98	1.30	4.20	30.89
9	170	22.08	53.06	245.40	5.70	37.97	1.24	4.30	28.86
10	180	19.84	47.83	247.67	5.50	35.99	1.19	4.30	27.63
11	190	16.76	40.40	247.16	5.20	33.08	1.07	4.40	24.77
12	200	14.91	36.38	251.29	5.00	31.19	1.05	4.40	23.75
13	210	12.78	30.90	253.68	4.75	28.88	0.98	4.50	21.70
14	220	10.87	26.28	257.15	4.50	26.63	0.91	4.50	20.29
15	230	8.84	21.41	260.24	4.20	24.02	0.83	4.60	18.11
16	240	7.07	17.17	264.24	3.90	21.49	0.76	4.65	16.28
17	250	5.56	13.53	269.09	3.60	19.06	0.68	4.70	14.55
18	260	4.29	10.47	274.76	3.30	16.73	0.61	4.75	12.90
19	270	3.55	8.70	282.25	3.10	15.23	0.57	4.80	11.94
20	280	2.91	7.13	290.04	2.90	13.77	0.53	4.80	11.06
21	290	2.01	4.90	296.95	2.60	11.70	0.44	4.90	9.45
22	300	1.64	4.06	305.70	2.40	10.37	0.42	4.95	8.54

Appendix II

Performance of 15-cm mixed flow pump at 1000 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	37.50	89.83	217.33	6.80	49.47	1.43	3.40	42.16
2	100	35.87	85.93	221.80	6.70	48.38	1.43	3.40	42.08
3	110	34.28	82.13	226.41	6.60	47.31	1.43	3.50	40.81
4	120	32.00	76.79	228.78	6.45	45.70	1.39	3.50	39.83
5	130	29.11	69.85	228.96	6.25	43.51	1.33	3.50	37.95
6	140	27.07	64.96	232.03	6.10	42.03	1.30	3.60	36.12
7	150	24.49	58.86	233.35	5.90	39.98	1.24	3.60	34.55
8	160	22.08	53.06	235.14	5.70	37.97	1.19	3.65	32.61
9	170	19.84	47.83	237.67	5.50	35.99	1.14	3.70	30.82
10	180	17.94	42.79	240.54	5.30	34.04	1.09	3.70	29.51
11	190	14.91	36.38	241.29	5.00	31.19	1.00	3.70	27.12
12	200	12.38	29.94	242.32	4.70	28.43	0.92	3.75	24.50
13	210	10.87	26.28	247.15	4.50	26.63	0.88	3.80	23.09
14	220	9.48	22.96	252.44	4.30	24.88	0.84	3.80	22.04
15	230	7.63	18.48	256.11	4.00	22.32	0.76	3.80	20.06
16	240	6.04	14.63	260.67	3.70	19.86	0.69	3.85	17.93
17	250	4.69	11.45	266.14	3.40	17.49	0.62	3.85	16.12
18	260	3.91	9.54	273.45	3.20	15.97	0.58	3.90	14.90
19	270	2.91	7.13	280.04	2.90	13.77	0.51	3.90	13.18
20	280	2.35	5.78	288.13	2.70	12.38	0.48	3.95	12.04
21	290	1.64	4.06	295.70	2.40	10.37	0.41	4.00	10.22
22	300	0.95	2.38	303.33	2.00	7.89	0.32	4.00	7.98

Appendix III

Performance of 15-cm mixed flow pump at 950 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	32.00	76.79	198.78	6.45	45.70	1.21	2.90	41.76
2	100	29.82	71.44	201.26	6.30	44.12	1.18	2.90	40.82
3	110	28.42	68.20	206.62	6.20	43.07	1.19	2.95	40.22
4	120	25.76	61.81	207.57	6.00	41.00	1.13	2.95	38.46
5	130	23.27	55.92	209.19	5.80	38.97	1.09	3.00	36.26
6	140	22.08	53.06	215.14	5.70	37.97	1.09	3.05	35.71
7	150	19.84	47.83	217.67	5.50	35.99	1.04	3.05	34.25
8	160	16.76	40.40	217.16	5.20	33.08	0.96	3.10	30.89
9	170	14.03	33.87	217.90	4.90	30.26	0.88	3.10	28.36
10	180	11.61	28.07	219.68	4.60	27.53	0.81	3.15	25.60
11	190	9.41	22.96	222.44	4.30	24.88	0.74	3.20	23.06
12	200	7.07	17.17	224.25	3.90	21.49	0.64	3.20	20.08
13	210	6.05	14.63	230.67	3.70	19.86	0.61	3.20	19.09
14	220	5.11	12.45	237.56	3.50	18.26	0.58	3.25	17.80
15	230	4.28	10.45	244.73	3.30	16.72	0.55	3.25	16.79
16	240	3.55	8.66	252.21	3.10	15.23	0.51	3.30	15.52
17	250	2.62	6.42	259.04	2.80	13.07	0.45	3.30	13.68
18	260	2.01	4.95	266.95	2.60	11.70	0.42	3.35	12.43
19	270	1.45	3.59	275.04	2.30	9.73	0.36	3.40	10.49
20	280	0.95	2.38	283.33	2.00	7.89	0.30	3.40	8.77
21	290	0.59	1.48	292.07	1.70	6.18	0.24	3.45	6.98
22	300	0.40	1.03	301.43	1.50	5.13	0.21	3.45	5.98

Appendix IV

Performance of 15-cm mixed flow pump at 900 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	25.12	60.28	175.40	5.95	40.49	0.95	2.35	40.30
2	100	23.87	57.38	181.25	5.85	39.47	0.95	2.35	40.58
3	110	22.08	53.06	185.14	5.70	37.97	0.94	2.40	39.05
4	120	19.84	47.83	187.67	5.50	35.99	0.90	2.40	37.50
5	130	16.76	40.40	187.16	5.20	33.08	0.83	2.40	34.40
6	140	14.46	34.92	189.38	4.95	30.72	0.78	2.50	31.03
7	150	13.19	31.89	195.08	4.80	29.34	0.76	2.50	30.53
8	160	12.38	29.94	202.32	4.70	28.43	0.77	2.60	29.50
9	170	10.87	26.28	207.15	4.50	26.63	0.74	2.60	28.29
10	180	8.84	21.41	210.24	4.20	24.03	0.67	2.65	25.41
11	190	7.63	18.48	216.11	4.00	22.32	0.64	2.70	23.82
12	200	7.07	17.17	224.24	3.90	21.49	0.64	2.70	23.80
13	210	5.56	13.53	229.09	3.60	19.06	0.58	2.75	21.17
14	220	3.91	9.54	233.45	3.20	15.97	0.50	2.75	18.08
15	230	2.91	7.53	240.04	2.90	13.77	0.44	2.80	15.73
16	240	2.01	4.95	246.95	2.60	11.70	0.39	2.80	13.76
17	250	1.27	3.15	254.42	2.20	9.10	0.31	2.85	10.83
18	260	0.82	2.06	262.88	1.90	7.31	0.26	2.90	8.84
19	270	0.59	1.48	272.07	1.70	6.18	0.22	2.90	7.73
20	280	0.40	1.03	281.43	1.50	5.13	0.19	2.90	6.64

Appendix V

Performance of 15-cm mixed flow pump at 850 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	20.94	50.33	161.27	5.60	36.97	0.80	2.05	38.77
2	100	19.84	47.83	167.67	5.50	35.99	0.80	2.05	39.24
3	110	17.94	42.79	170.54	5.30	34.04	0.77	2.05	37.76
4	120	15.82	38.19	174.01	5.10	32.13	0.75	2.10	35.61
5	130	14.46	34.92	179.38	4.95	30.72	0.73	2.10	34.99
6	140	13.19	31.89	185.08	4.80	29.34	0.72	2.20	32.91
7	150	11.61	28.07	189.68	4.60	27.53	0.70	2.20	31.76
8	160	9.48	22.96	192.44	4.30	24.88	0.64	2.30	27.76
9	170	7.63	18.48	196.11	4.00	22.32	0.58	2.30	25.37
10	180	6.04	14.63	200.67	3.70	19.86	0.53	2.30	23.10
11	190	5.11	12.45	207.56	3.50	18.26	0.51	2.40	21.06
12	200	3.91	9.54	213.45	3.20	15.97	0.45	2.40	18.93
13	210	2.91	7.13	220.04	2.90	13.77	0.40	2.45	16.49
14	220	2.01	4.95	226.96	2.60	11.70	0.35	2.45	14.45
15	230	1.45	3.59	235.04	2.30	9.73	0.30	2.45	12.44
16	240	0.82	2.06	242.88	1.90	7.31	0.24	2.50	9.47
17	250	0.49	1.24	251.73	1.60	5.64	0.19	2.50	7.57
18	260	0.21	0.54	260.75	1.20	3.67	0.13	2.50	5.10

Appendix VI

Performance of 15-cm mixed flow pump at 800 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	15.82	38.19	144.01	5.10	32.13	0.62	1.65	37.39
2	100	14.91	36.38	151.29	5.00	31.19	0.63	1.65	38.13
3	110	13.19	31.89	155.08	4.80	29.34	0.61	1.70	35.69
4	120	12.38	29.94	162.32	4.70	28.43	0.62	1.70	36.19
5	130	10.87	26.28	167.15	4.50	26.63	0.59	1.70	34.91
6	140	9.48	22.96	172.44	4.30	24.88	0.57	1.75	32.69
7	150	8.22	19.90	178.12	4.10	23.16	0.55	1.75	31.43
8	160	6.54	15.84	182.38	3.80	20.67	0.50	1.80	27.92
9	170	5.11	12.45	187.56	3.50	18.27	0.46	1.80	25.38
10	180	3.91	9.54	193.45	3.20	15.97	0.41	1.85	22.26
11	190	3.26	7.98	201.24	3.00	14.50	0.38	1.85	21.03
12	200	2.35	5.78	208.13	2.70	12.38	0.38	1.90	20.15
13	210	1.64	4.06	215.70	2.40	10.37	0.30	1.90	15.70
14	220	0.95	2.37	223.32	2.00	7.89	0.23	1.90	12.36
15	230	0.49	1.26	231.73	1.60	5.64	0.17	2.00	8.71
16	240	0.21	0.54	240.75	1.20	3.67	0.12	2.00	5.89

Appendix VII

Performance of 15-cm mixed flow pump at 750 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	10.16	24.61	124.77	4.4	25.75	0.43	1.20	35.70
2	100	9.48	22.96	132.44	4.3	24.88	0.44	1.20	36.61
3	110	8.22	19.91	138.13	4.1	23.16	0.43	1.20	35.54
4	120	7.07	17.17	144.24	3.9	21.49	0.41	1.25	33.06
5	130	6.04	14.63	150.67	3.7	19.86	0.40	1.25	31.92
6	140	4.69	11.45	156.14	3.4	17.49	0.36	1.30	28.00
7	150	3.55	8.66	162.21	3.1	15.23	0.33	1.30	25.33
8	160	2.62	6.42	169.04	2.8	13.07	0.29	1.30	22.66
9	170	1.64	4.06	175.70	2.4	10.37	0.24	1.30	18.69
10	180	0.95	2.38	183.33	2.0	7.89	0.19	1.40	13.78
11	190	0.59	1.48	192.07	1.7	6.18	0.15	1.40	11.30
12	200	0.33	0.84	201.17	1.4	4.62	0.12	1.40	8.85
13	210	0.12	0.31	210.43	1.0	2.79	0.08	1.40	5.59

Appendix VIII

Performance of 15-cm mixed flow pump at 700 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	8.22	19.90	118.12	4.1	23.16	0.36	1.05	34.74
2	100	7.63	18.48	126.10	4.0	22.32	0.38	1.05	35.74
3	110	6.54	15.83	132.38	3.8	20.67	0.36	1.10	33.13
4	120	5.11	12.45	137.56	3.5	18.27	0.34	1.10	30.46
5	130	3.55	8.70	142.25	3.1	15.23	0.29	1.15	25.11
6	140	2.35	5.78	148.13	2.7	12.38	0.24	1.15	21.26
7	150	1.64	4.06	155.70	2.4	10.37	0.22	1.20	17.94
8	160	0.95	2.37	163.32	2.0	7.89	0.17	1.20	14.32
9	170	0.59	1.48	172.07	1.7	6.18	0.14	1.20	11.82
10	180	0.26	0.73	180.99	1.3	4.14	0.10	1.30	7.69
11	190	0.12	0.31	190.43	1.2	2.79	0.07	1.30	5.45

Appendix IX

Performance of 15-cm mixed flow pump at 650 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	5.11	12.45	107.56	3.5	18.26	0.26	0.80	32.73
2	100	4.28	10.45	114.73	3.3	16.72	0.26	0.80	31.97
3	110	3.55	8.66	122.21	3.1	15.23	0.25	0.85	29.20
4	120	2.62	6.42	129.04	2.8	13.07	0.23	0.85	26.45
5	130	1.64	4.06	135.70	2.4	10.37	0.19	0.90	20.84
6	140	0.95	2.37	143.32	2.0	7.89	0.15	0.90	16.75
7	150	0.49	1.24	151.73	1.6	5.64	0.11	0.90	12.68
8	160	0.21	0.54	160.75	1.2	3.67	0.08	1.00	7.87

Appendix X

Performance of 15-cm mixed flow pump at 600 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	2.62	6.42	99.03	2.8	13.07	0.17	0.60	28.76
2	100	2.10	5.15	107.25	2.6	11.70	0.17	0.65	25.74
3	110	1.45	3.59	115.04	2.3	9.73	0.15	0.70	21.32
4	120	0.95	2.38	123.33	2.0	7.89	0.13	0.70	18.54
5	130	0.59	1.48	132.07	1.7	6.18	0.11	0.70	15.54
6	140	0.26	0.73	140.99	1.3	4.14	0.08	0.80	9.70
7	150	0.06	0.16	150.22	0.8	2.00	0.04	0.80	5.00

Appendix XI

Performance of 15-cm axial flow pump at 2600 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	12.38	27.46	129.84	4.70	28.43	0.49	3.08	15.96
2	100	11.61	25.75	137.36	4.60	27.53	0.50	3.08	16.35
3	110	10.87	24.11	144.98	4.50	26.63	0.51	3.08	16.70
4	120	10.16	22.57	152.73	4.40	25.75	0.52	3.08	17.00
5	130	9.48	21.07	160.55	4.30	24.88	0.53	3.08	17.21
6	140	9.15	20.33	169.48	4.25	24.44	0.55	3.22	17.16
7	150	8.84	19.63	178.48	4.20	24.01	0.57	3.35	17.04
8	160	8.22	18.26	186.48	4.10	23.16	0.58	3.42	16.85
9	170	7.63	16.95	194.58	4.00	22.32	0.58	3.42	16.93
10	180	7.07	15.76	202.83	3.90	21.49	0.58	3.49	16.67
11	190	6.54	14.53	211.07	3.80	20.67	0.58	3.49	16.69
12	200	6.04	13.49	219.53	3.70	19.86	0.58	3.62	16.06
13	210	5.56	12.41	227.97	3.60	19.06	0.58	3.62	16.00
14	220	5.11	11.45	236.56	3.50	18.27	0.58	3.62	15.92
15	230	4.90	10.97	245.89	3.45	17.88	0.59	3.75	15.62
16	240	4.69	10.50	255.19	3.40	17.49	0.60	3.82	15.57
17	250	4.29	9.60	263.89	3.30	16.72	0.59	3.89	15.12
18	260	3.91	8.75	272.66	3.20	15.97	0.58	3.89	14.94
19	270	3.55	7.97	281.52	3.10	15.23	0.57	4.02	14.21
20	280	3.06	6.89	289.95	2.95	14.14	0.55	4.02	13.59
21	290	2.91	6.55	299.45	2.90	13.77	0.55	4.16	13.23
22	300	2.62	5.91	308.53	2.80	13.07	0.54	4.16	12.94

Appendix XII

Performance of 15-cm axial flow pump at 2500 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	11.61	25.75	127.36	4.60	27.53	0.47	2.82	16.56
2	100	10.87	24.11	134.98	4.50	26.63	0.48	2.82	17.00
3	110	10.16	22.57	142.73	4.40	25.75	0.49	2.82	17.38
4	120	9.82	21.82	151.64	4.35	25.31	0.51	2.88	17.71
5	130	9.48	21.07	160.55	4.30	24.88	0.53	2.95	18.05
6	140	8.84	19.63	168.48	4.20	24.02	0.54	3.08	17.53
7	150	7.92	17.60	175.52	4.05	22.74	0.53	3.08	17.28
8	160	7.63	16.95	184.48	4.00	22.32	0.55	3.22	17.06
9	170	7.35	16.36	193.70	3.95	21.90	0.57	3.22	17.57
10	180	6.81	15.38	202.19	3.85	21.08	0.57	3.35	16.95
11	190	6.54	14.53	211.07	3.80	20.67	0.58	3.35	17.36
12	200	6.29	14.04	220.33	3.75	20.26	0.60	3.42	17.40
13	210	6.04	13.49	229.53	3.70	19.86	0.61	3.49	17.43
14	220	5.56	12.41	237.97	3.60	19.06	0.60	3.49	17.32
15	230	5.11	11.45	246.56	3.50	18.27	0.60	3.62	16.49
16	240	4.69	10.50	255.19	3.40	17.49	0.60	3.62	16.44
17	250	4.48	10.04	264.51	3.35	17.11	0.60	3.75	16.09
18	260	3.91	8.75	272.66	3.20	15.97	0.58	3.75	15.48
19	270	3.38	7.61	280.99	3.05	14.89	0.56	3.89	14.32
20	280	2.91	6.55	289.45	2.90	13.77	0.53	3.89	13.66
21	290	2.76	6.21	298.97	2.85	13.42	0.53	4.02	13.18
22	300	2.35	5.31	307.65	2.70	12.38	0.51	4.02	12.63

Appendix XIII
Performance of 15-cm axial flow pump at 2400 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	10.51	23.35	123.84	4.45	26.19	0.43	2.68	16.13
2	100	9.82	21.82	131.64	4.35	25.31	0.44	2.68	16.57
3	110	9.15	20.33	139.48	4.25	24.44	0.45	2.68	16.96
4	120	8.84	19.63	148.48	4.20	24.01	0.48	2.72	17.30
5	130	8.22	18.26	156.48	4.10	23.16	0.48	2.82	17.17
6	140	7.63	16.95	164.58	4.00	22.32	0.49	2.82	17.40
7	150	7.07	15.76	172.83	3.90	21.49	0.50	2.82	17.59
8	160	6.54	14.58	181.07	3.80	20.67	0.50	2.95	16.92
9	170	5.56	12.41	187.97	3.60	19.96	0.48	2.95	16.19
10	180	4.90	10.79	195.87	3.45	17.88	0.47	2.95	15.83
11	190	4.29	9.60	203.09	3.30	16.72	0.45	3.02	15.07
12	200	3.91	8.75	212.66	3.20	15.97	0.45	3.08	14.69
13	210	3.55	7.97	221.52	3.10	15.23	0.45	3.08	14.59
14	220	3.38	7.61	230.99	3.05	14.86	0.46	3.22	14.22
15	230	3.22	7.25	240.46	3.00	14.50	0.46	3.35	13.87
16	240	2.76	6.21	248.97	2.85	13.42	0.45	3.35	13.29
17	250	2.48	5.60	258.08	2.75	12.72	0.44	3.49	12.56
18	260	2.22	5.01	267.23	2.65	12.04	0.43	3.49	12.48
19	270	2.10	4.73	276.83	2.60	11.69	0.43	3.49	12.38
20	280	1.98	4.47	286.47	2.55	11.36	0.43	3.62	11.98
21	290	1.86	4.21	396.07	2.50	11.03	0.44	3.62	12.03
22	300	1.64	3.73	305.37	2.40	10.37	0.42	3.62	11.66

Appendix XIV

Performance of 15-cm axial flow pump at 2300 rev/min

Sl No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	9.48	21.07	120.55	4.30	24.88	0.40	2.48	16.12
2	100	8.84	19.63	128.48	4.20	24.01	0.41	2.55	16.13
3	110	8.22	18.26	136.48	4.10	23.16	0.42	2.55	16.53
4	120	7.63	16.95	144.58	4.00	22.32	0.43	2.55	16.87
5	130	7.07	15.76	152.83	3.90	21.49	0.44	2.55	17.17
6	140	6.54	14.53	161.07	3.80	20.67	0.44	2.61	17.01
7	150	5.80	12.95	168.75	3.65	19.46	0.44	2.61	16.78
8	160	5.11	11.45	176.56	3.50	18.27	0.43	2.61	16.47
9	170	4.69	10.50	185.19	3.40	17.49	0.43	2.68	16.03
10	180	4.29	9.60	193.89	3.30	16.72	0.43	2.68	16.12
11	190	3.91	8.75	202.66	3.20	15.97	0.43	2.82	15.32
12	200	3.73	8.36	212.90	3.15	15.60	0.44	2.82	15.37
13	210	3.22	7.25	220.46	3.00	14.50	0.43	2.88	14.79
14	220	2.76	6.21	228.97	2.85	13.42	0.41	2.95	13.89
15	230	2.62	5.91	238.53	2.80	13.07	0.42	2.95	14.09
16	240	2.35	5.31	247.65	2.70	12.38	0.41	3.08	13.27
17	250	1.98	4.47	256.45	2.55	11.36	0.39	3.08	12.61
18	260	1.75	3.99	265.75	2.45	10.70	0.38	3.20	11.85
19	270	1.45	3.43	276.80	2.30	9.73	0.36	3.20	11.14
20	280	1.27	2.89	284.16	2.20	9.10	0.34	3.35	10.27
21	290	0.95	2.18	293.13	2.00	7.89	0.31	3.35	9.21
22	300	0.82	1.88	302.70	1.90	7.31	0.30	3.35	8.81

Appendix XV

Performance of 15-cm axial flow pump at 2200 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	8.22	18.26	116.48	4.10	23.16	0.36	2.28	15.78
2	100	7.63	16.95	124.58	4.00	22.32	0.37	2.28	16.26
3	110	6.81	15.38	132.19	3.85	21.08	0.37	2.28	16.30
4	120	6.04	13.49	139.53	3.70	19.86	0.37	2.28	16.21
5	130	5.56	12.41	147.97	3.60	19.06	0.37	2.28	16.49
6	140	5.11	11.45	156.56	3.50	18.27	0.38	2.41	15.82
7	150	4.48	10.04	164.51	3.35	17.11	0.38	2.41	15.57
8	160	3.91	8.75	172.66	3.20	15.97	0.37	2.55	14.44
9	170	3.22	7.24	180.46	3.00	14.50	0.35	2.55	13.70
10	180	2.76	6.21	188.97	2.85	13.42	0.34	2.55	13.28
11	190	2.35	5.31	197.65	2.70	12.38	0.33	2.55	12.81
12	200	2.10	4.73	206.83	2.60	11.69	0.32	2.68	12.03
13	210	1.86	4.21	216.07	2.50	11.03	0.32	2.68	11.85
14	220	1.64	3.73	225.37	2.40	10.37	0.31	2.68	11.63
15	230	1.55	3.52	235.07	2.35	10.05	0.31	2.82	11.17
16	240	1.27	2.89	244.16	2.20	9.10	0.30	2.82	10.52
17	250	1.19	2.70	253.88	2.15	8.79	0.30	2.95	10.09
18	260	1.10	2.51	263.62	2.10	8.49	0.30	3.08	9.69
19	270	0.95	2.18	273.13	2.00	7.89	0.29	3.08	9.33
20	280	0.82	1.88	282.70	1.90	7.31	0.28	3.15	8.75
21	290	0.59	1.37	291.96	1.70	6.18	0.24	3.15	7.64
22	300	0.49	1.14	301.63	1.60	5.64	0.23	3.15	7.20

Appendix XUI

Performance of 15-cm axial flow pump at 2100 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	7.35	16.36	113.70	3.95	21.90	0.33	2.08	15.96
2	100	6.89	15.38	122.19	3.85	21.08	0.34	2.14	16.05
3	110	6.29	14.04	130.33	3.75	20.26	0.35	2.14	16.45
4	120	5.56	12.41	137.97	3.60	19.06	0.35	2.14	16.38
5	130	4.69	10.50	145.19	3.40	17.49	0.34	2.14	15.62
6	140	3.73	8.36	152.90	3.15	15.60	0.32	2.21	14.37
7	150	2.62	5.91	158.53	2.80	13.07	0.28	2.21	12.50
8	160	2.35	5.31	167.65	2.70	12.38	0.28	2.27	12.19
9	170	1.98	4.47	176.45	2.55	11.36	0.27	2.27	11.77
10	180	1.64	3.73	185.37	2.40	10.37	0.26	2.27	11.29
11	190	1.45	3.43	194.89	2.30	9.73	0.25	2.41	10.49
12	200	1.27	2.89	204.16	2.20	9.10	0.25	2.41	10.28
13	210	1.10	2.51	213.62	2.10	8.49	0.24	2.41	10.03
14	220	0.82	1.88	222.90	1.90	7.31	0.22	2.41	9.00
15	230	0.64	1.48	232.12	1.75	6.46	0.20	2.41	8.30
16	240	0.54	1.26	241.80	1.65	5.91	0.19	2.55	7.47
17	250	0.49	1.14	251.63	1.60	5.64	0.19	2.55	7.42
18	260	0.44	1.04	261.49	1.55	5.38	0.19	2.68	7.00
19	270	0.40	0.95	271.35	1.50	5.13	0.19	2.68	6.93
20	280	0.33	0.77	281.10	1.40	4.62	0.17	2.82	6.15
21	290	0.26	0.62	290.62	1.30	4.14	0.16	2.82	5.70
22	300	0.21	0.49	300.70	1.20	3.67	0.15	2.95	4.98

Appendix XVII

Performance of 15-cm axial flow pump at 2000 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	5.11	11.45	106.56	3.50	18.27	0.26	1.87	13.88
2	100	4.69	10.50	115.19	3.40	17.49	0.27	1.87	14.36
3	110	4.29	9.60	123.89	3.30	16.72	0.28	1.87	14.77
4	120	3.73	8.36	132.90	3.15	15.60	0.28	1.87	14.78
5	130	2.91	6.55	139.45	2.90	13.77	0.26	1.87.	13.69
6	140	2.35	5.31	147.65	2.70	12.38	0.24	2.01	12.12
7	150	1.98	4.47	156.45	2.55	11.36	0.24	2.01	11.79
8	160	1.75	3.99	165.75	2.45	10.70	0.24	2.01	11.76
9	170	1.45	3.43	174.89	2.30	9.73	0.23	2.01	11.28
10	180	1.10	2.51	183.62	2.10	8.49	0.21	2.01	10.34
11	190	0.95	2.18	193.13	2.00	7.89	0.20	2.14	9.49
12	200	0.82	1.88	202.70	1.90	7.31	0.20	2.14	9.23
13	210	0.64	1.48	212.12	1.75	6.46	0.18	2.14	8.58
14	220	0.59	1.37	221.96	1.70	6.18	0.18	2.14	8.55
15	230	0.49	1.14	231.63	1.60	5.64	0.17	2.14	8.14
16	240	0.40	0.95	241.35	1.50	5.13	0.17	2.14	7.71
17	250	0.29	0.69	250.98	1.35	4.38	0.15	2.27	6.46
18	260	0.23	0.55	260.78	1.25	3.90	0.14	2.27	5.97
19	270	0.16	0.38	270.55	1.10	3.22	0.12	2.27	5.12
20	280	0.09	0.24	280.32	0.95	2.45	0.09	2.27	4.03

Appendix XVIII

Performance of 15-cm axial flow pump at 1900 rev/min

Sl. No.	Total static head H_s (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	3.91	8.75	102.66	3.20	15.97	0.22	1.67	13.09
2	100	3.55	7.97	111.52	3.10	15.23	0.23	1.67	13.56
3	110	3.06	6.89	119.95	2.95	14.14	0.23	1.67	13.54
4	120	2.62	5.91	128.53	2.80	13.07	0.23	1.67	13.41
5	130	2.35	5.31	137.66	2.70	12.38	0.23	1.97	13.61
6	140	1.86	4.21	146.07	2.50	11.03	0.21	1.74	12.35
7	150	1.45	3.43	154.89	2.30	9.73	0.20	1.74	11.55
8	160	1.27	2.89	164.16	2.20	9.10	0.20	1.74	11.45
9	170	0.95	2.18	173.13	2.00	7.89	0.18	1.74	10.46
10	180	0.62	1.88	182.70	1.90	7.31	0.18	1.87	9.52
11	190	0.70	1.62	192.32	1.80	6.74	0.17	1.87	9.24
12	200	0.59	1.37	201.96	1.70	6.18	0.17	1.87	8.90
13	210	0.44	1.04	211.49	1.55	5.38	0.15	1.87	8.11
14	220	0.33	0.77	221.10	1.40	4.62	0.14	1.87	7.28
15	230	0.26	0.62	230.88	1.30	4.14	0.13	2.01	6.34
16	240	0.23	0.55	240.78	1.25	3.90	0.13	2.01	6.23
17	250	0.16	0.38	250.55	1.10	3.22	0.11	2.01	5.35
18	260	0.12	0.28	260.41	1.00	2.79	0.10	2.14	4.53

Appendix XXI

Performance of 15-cm axial flow pump at 1600 rev/min

Sl. No.	Total static head Hs (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	1.10	2.51	93.62	2.10	8.49	0.11	0.94	11.27
2	100	0.95	2.18	103.13	2.00	7.89	0.11	0.94	11.54
3	110	0.82	1.88	112.70	1.90	7.31	0.11	0.94	11.69
4	120	0.70	1.62	122.32	1.80	6.74	0.11	0.94	11.69
5	130	0.49	1.14	131.63	1.60	5.64	0.10	0.94	10.53
6	140	0.33	0.77	141.10	1.40	4.62	0.09	1.07	8.12
7	150	0.26	0.62	150.88	1.30	4.14	0.08	1.07	7.78
8	160	0.16	0.38	160.55	1.10	3.22	0.07	1.07	6.44
9	170	0.09	0.24	170.32	0.95	2.45	0.06	1.07	5.20

Appendix XX

Performance of 15-cm axial flow pump at 1700 rev/min

Sl. No.	Total static head H_S (cm)	Velocity head $V^2/2g$ (cm)	Hydraulic head losses H_L (cm)	Total head H (cm)	Flow over weir crest h (cm)	Discharge Q L/s	Output hp	Input hp	Efficiency %
1	90	1.75	3.99	95.75	2.45	10.70	0.14	1.20	11.38
2	100	1.55	3.52	105.07	2.35	10.05	0.14	1.20	11.73
3	110	1.36	3.01	114.37	2.25	9.42	0.14	1.20	11.97
4	120	1.10	2.51	123.62	2.10	8.49	0.14	1.20	11.66
5	130	0.82	1.88	132.70	1.90	7.31	0.13	1.20	10.78
6	140	0.59	1.37	141.96	1.70	6.18	0.12	1.34	8.73
7	150	0.40	0.95	151.35	1.50	5.13	0.10	1.34	7.73
8	160	0.33	0.77	161.10	1.40	4.62	0.10	1.34	7.41
9	170	0.21	0.49	170.70	1.20	3.67	0.08	1.34	6.23
10	180	0.12	0.28	180.41	1.00	2.79	0.07	1.34	5.00

PERFORMANCE EVALUATION OF HIGH DISCHARGE LOW HEAD PUMPS

By

S. RINI RANI

ABSTRACT OF A THESIS

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ABSTRACT

Paddy cultivation in Kuttanad and Kole lands is made possible only after large scale dewatering operations. These water logged lands have to be drained in the shortest possible time to initiate seed bed preparation. The short time available for seed bed preparation need adoption of high discharge low head pumps. The most commonly used centrifugal pump is incapable of giving corresponding increase in discharge rate even if the lift required is reduced to a minimum. So this study was undertaken to help to identify a suitable portable pump for meeting the specific requirement of drainage in Kerala. Mixed flow and axial flow pumps imported from Thailand were used for this purpose.

The study included testing of 15-cm mixed and axial flow pump at the specially designed and constructed test bed at different speeds and at different water levels in the pumping sump. For each test, from the data obtained discharge, input power and efficiency were calculated. Using the calculated values performance characteristic curves were plotted and regression equations were developed for discharge - efficiency, discharge - total head and discharge - input power relationship at each speed. Performance curves were also plotted keeping total static head const ant and regression

equations of speed - discharge, speed - total head, speed - input power and speed - efficiency relationships were developed.

The maximum efficiency of 15-cm mixed flow pump was 42.16 per cent at a speed of 1000 rev/min, against a total head of 217.33 cm and a discharge rate of 49.47 L/s. The input power corresponding to this efficiency was 3.4 hp. The best performance of mixed flow pump was noted at a working speed of 1000 rev/min. The maximum efficiency of 15-cm axial flow pump tested was only 18.05 per cent at a total head of 160.55 cm and a discharge of 24.88 L/s. The corresponding speed was 2500 rev/min and input power was 2.95 hp. The best performance of axial flow pump was noted at a working speed of 2500 rev/min.