DEVELOPMENT OF EJECTOR SYSTEMS FOR INCREASING THE DISCHARGES OF CENTRIFUGAL PUMPS

By

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THESIS

Submitted in partial fulfilment of the
requirement for the degree

Master of Science in Agricultural Engineering

Faculty of Agricultural Engineering
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Tavanur - Malappuram

1988

DECLARATION

I hereby declare that this thesis entitled "Development of ejector systems for increasing the discharges of centrifugal 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.

Vellanikkara,
1988

LATHA.A.KOSHY

CERTIFICATE

Certified that this thesis entitled "Development of ejector systems for increasing the discharges of centrifugal pumps" is a record of research work done independently by Smt.Latha A.Koshy 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 USED

An	Area of nozzle
At	Area of throat
Cd	Coefficient of discharge
Cm	Centimetre (s)
Dn	Diameter of nozzle
Dt	Diameter of throat
F	Multiplication factor
Fig	Figure
g	Acceleration due to gravity, Gram(s)
GI	Galvanised iron
hr	Hour
Hd	Discharge head
Hn	Effective head to the nozzle
Hs	Suction head
HP	Horse power
kg	Kilogram(s)
l	Litres
lpm	Litres per minute
m	Metre(s)
M	Flow ratio
M*	Capacity ratio
M.S	Mild steel
N	Pressure ratio
N*	Head ratio
Pd	Discharge pressure
P_1	Pressure at nozzle

P_2	Suction pressure
$\dot{Q}$	Discharge
Qn	Discharge through nozzle
Qs	Secondary flow through ejector system
R	Area ratio
rpm	Revolutions per minute
Rs	Rupee(s)
s	second (s)
η	Efficiency, MN
η^*	Efficiency, M* N*
%	Per cent
π	Pi (22/7)
°	Degree
/	Per
θ	Angle
ϕ	Diameter

Introduction

INTRODUCTION

The primary problem in agriculture is the timely supply of adequate and assured quantity of water. Canal irrigation systems in addition to their large capital investments are characterised by many soil and water management problems. Since lift irrigation offers the farmer superior water control, it is becoming increasingly popular in Asian countries.

The pump capability requirement in rice production is essentially one of low lift and high capacity. Yet considerable variations in the pumping heads may occur during crop seasons. Generally centrifugal pumpsets are used where medium suction lift condition in the range of 3 to 6 metres exists. But sometimes centrifugal pumps are used in low lift conditions as in the case of Kuttanad region and Trichur kole lands. Here many low lift high capacity centrifugal pumps are used for dewatering as well as pumping water for irrigation, but because of the low lift conditions, the full capacity of the centrifugal pump cannot be used.

The centrifugal pump is unable to give a maximum discharge, exceeding 150 per cent of that obtained at the best efficiency point. This behaviour of the centrifugal pump is a serious limitation in the use for low lift applications. Any device providing a combined discharge which exceeds the actual maximum flow of the centrifugal pump may be technically justified as long as it does not require any additional power. By using a suitably designed ejector system, the centrifugal pump can be brought to work under the best efficiency condition during the low lift condition. With the attachment of the ejector system the discharge capacity can also be multiplied. In many instances this can be a cheaper solution than to go in for a high specific speed pump. The ejector can also be detached for a subsequent dry season pumping operation where the lift is likely to be more.

A simple and novel design concept is utilised in this project to increase the discharge capacity of centrifugal pumps.

The present study is carried out to formulate the relationship between the suction lift and the discharge capacity of the ejector system.

Review of Literature

REVIEW OF LITERATURE

The primary problem in agriculture is the timely supply of an adequate and assured quantity of water (Johnson, 1965). Canal irrigation systems are characterised by many soil and water management problems in addition to their large capital investment (Wickham, 1974). Since pump irrigation offers the farmer superior water control, it is becoming increasingly popular in Asian countries.

Lift irrigation equipments

Molenar (1956) had made a comprehensive review of the various types of water lifting devices used in different parts of the world. The performance data of these devices is summerised in Table 1.

Although all manual and animal powdered devices have limited output capabilities, for centuries they have been effectively utilised for irrigation by operating several units in series to increase lift, or in parallel to increase capacity.

Nowadays power operated pumps have almost repleced indegenous water lifting devices. The most commonly used

Table 1. Comparison of performance of some manual and animal-operated water lifting devices (Molenar, 1956)

Equipment	Height lifted (m)	Range of lift (m)	Capacity (m^3/h)	Men for 8 - hour operation	Animals for 8-hour operation
A. Manual					
1. Scoop (Swing type)	1.0	-	8.0	1	-
2. Swing basket	0.6	-	5.0	4	-
3. Counterpoise lift	2.5	-	3.0	2	-
4. Archemedian screw	0.5	0.25-0.75	22.5	2	-
5. Improved paddle wheel	0.3	0.15-0.45	100.0	2	-
6. Water ladder	0.9	-	8.0	2	-
7. a. Chain pump	1.5	-	30.0	4	-
b. Chain pump	3.0	-	20.0	4	-
B. Animal - operated					
1. Self-emptying bucket	8.0	-	8.0	1	2
2. a. Two-bucket lift	3.0	-	22.0	1	2
b. Two-bucket lift	10.0	-	5.5	1	2
3. Persian wheel	9.0	1.5-9.0	9.0	1	2
4. Sakia	0.3	0.3-1.8	110.0	1	2

devices for irrigation purpose are the centrifugal and propeller or axial flow pumps.

A propeller pump operates at greatly reduced capacity when the operating head is above the design point and may overload the prime mover (Addison, 1966).

For small capacities with lifts upto 40 m a special type of pumping unit has been developed. This consists of a combination of a centrifugal pump and a jet pump or ejector (Stepanoff _et al_., 1964). The first is mounted next to the motor at the ground surface and furnishes the driving head and capacity for this jet pump placed in the well below the water surface. The details of the pump are given in Appendix 1.

Industry employs two types of jet machines – water jet pumps (eduction) and steam jet compressors (Cherhasshy, 1977). In water-jet pumps the motive fluid is water, and in steam jet compressors it is steam. The operation of water jet pumps and steam jet compressors are basically same, the difference being in the motive fluids used. The basic design of a jet compressor built for industrial application is presented in Appendix-2.

Stepanoff (1964) stated that water jet pumps can be used advantageously to solve temporary dewatering problems.

Silvester (1968), proposed the use of a centrifugal jet pump combination for maintaining high and low pressure water supply in hydraulic laboratories.

Samuel (1972) designed an ejector system to improve the discharge capacity of conventional irrigation pumpset. He had studied the influences of area ratios, throat entry profile, nozzle spacing, throat length and diffuser angle on the performance of the ejector systems.

Duff (1973) pointed out that the U.S.Navy had been using for some time an ejector system for rapid drainage of flooded compartments in ships.

Samuel and Jacob (1977) proposed a novel fluidisation technique for harvesting the salvinia weed, based on the principle of a high capacity water jet device originally developed by Samuel in 1972 to improve the discharge capacity of the centrifugal pumpset. A portable pumpset was used as the prime mover to produce the primary flow which will induce the weed to move through an ejector system into the collection tank.

Surya Rao Singamsetti and Vishnu Kumar Goenka (1977) had conducted studies on the performance characteristics of jet pumps. Measurement of gross characteristics of jet pump had shown that the dimensionless performance parameters of the jet pump such as the ratio of head developed to the velocity head of the jet and efficiency depended upon the flow ratio, jet Reynolds number, jet diameter and the geometric parameters of the jet pump such as angle of the suction nozzle, angle of diffuser, ratios of the jet diameter to the mixing chamber diameter, ratio of the nozzle to throat distance to the jet diameter.

Sankaranarayanan (1981) of Kerala Agricultural University had developed one prototype salvinia harvesting machine based on the fluidisation technique proposed by Samuel and Jacob (1977).

Chandramohan (1984), of Kerala Agricultural University had developed a small scale dredging unit for the collection of sand using the above fluidisation technique.

Hajilal (1987), of Kerala Agricultural University had developed a high capacity ejector system for the collection of salvinia using the fluidisation technique.

Principles of operation

The ejector system operates on the principle of the transfer of energy and momentum from primary to secondary fluid through a process of turbulent mixing (Samuel, 1975). A schematic representation of an ejector system is shown in Figure 1. The primary fluid, which has a higher pressure, is accelerated to a high velocity by means of the nozzle. The secondary fluid is entrained by and mixed with the primary fluid in the constant diameter throat section. The mixed fluids then pass through a diffuser in which a portion of the velocity head is converted to static pressure.

The fundamental mechanism of the ejector system is the turbulent mixing. A schematic representation of the mixing velocity profile is shown in Figure 2 (Sanger, 1968). He described "At the nozzle exit, the primary stream is essentially a core of constant velocity fluid which is separated from the secondary fluid by a region of higher shear. Initially the high region is made up of a thin sheet of vortices or eddies which give rise to mixing on the periphery of the high velicity core".

As axial distance from the nozzle exit is increased, the shear or mixing layer increases in thickness. The potential core region progressively grows smaller as its

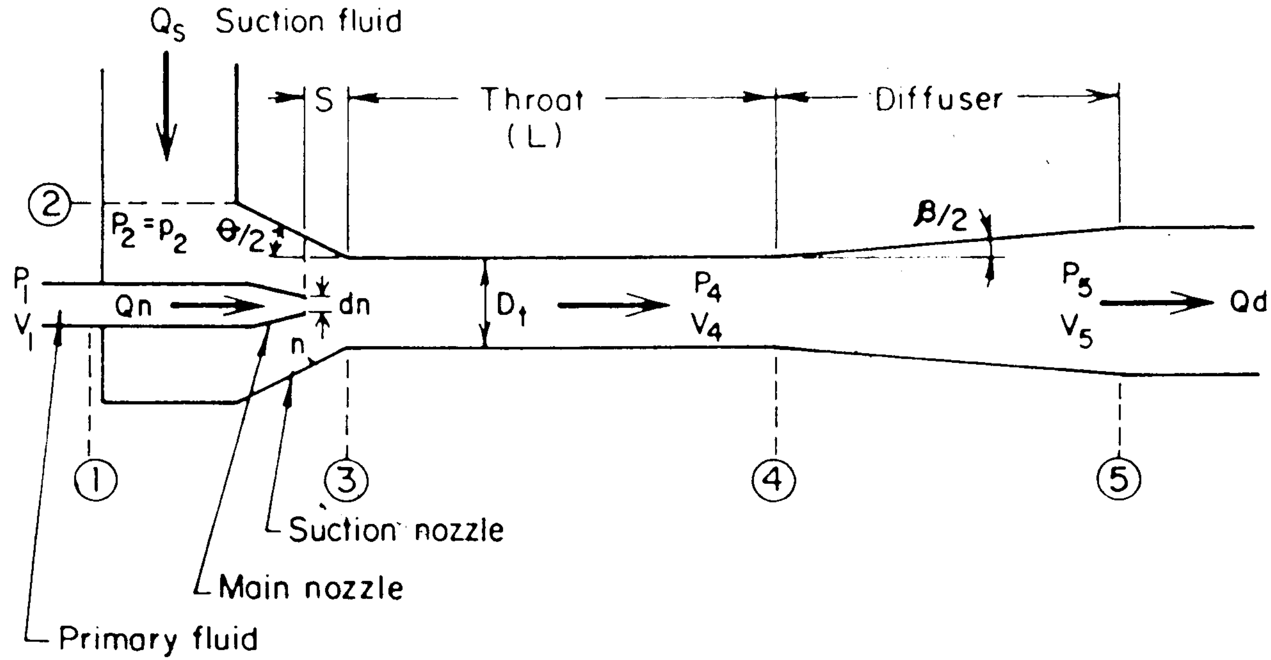


FIG.1. SCHEMATIC REPRESENTATION OF AN EJECTOR SYSTEM

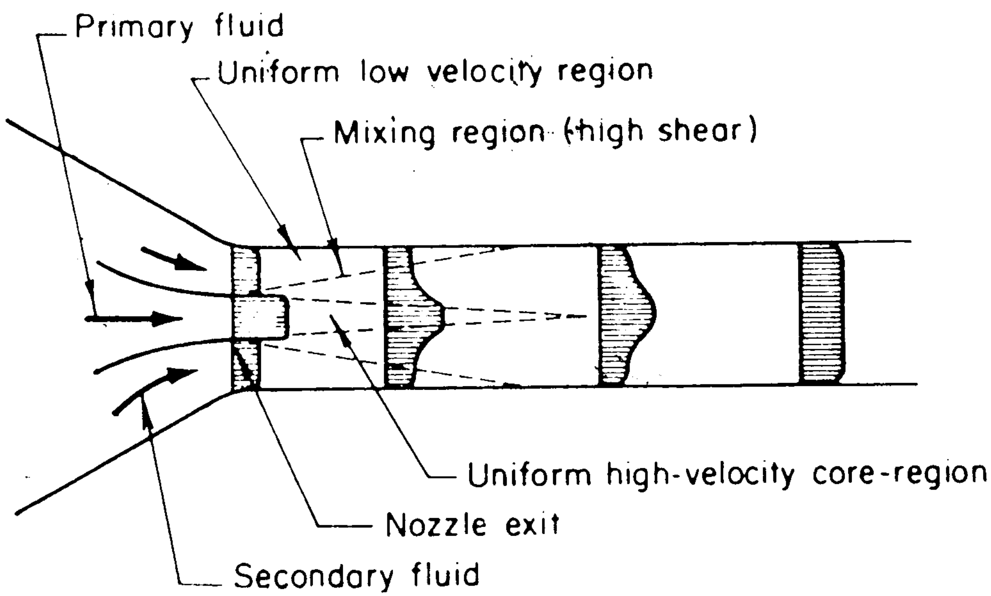


FIG.2. SCHEMATIC REPRESENTATION OF THE MIXING VELOCITY PROFILE IN AN EJECTOR SYSTEM

energy is dissipated to turbulence in the mixing layer. Due to the transport of momentum, the primary stream is decelerated while the secondary stream is accelerated. The lateral growth of the mixing layer continues until it finally meets the jet centre line and the constant velocity core disappears.

The concept of low lift pumping

Application of the water jet pump principle for possible salvaging of the power loss associated with the use of many medium pressure centrifugal pumps for rice irrigation in Kerala, was independently proposed by Samuel (1967). This low lift pumping arrangement which utilizes a low cost water jet pump in combination with a centrifugal pump is shown in Fig.3. The details of the ejector system are shown in Fig.4. In this arrangement the entire delivery of a centrifugal pump is fed to a jet pump which lifts an additional quantity of water over a small vertical distance. This is different from the conventional jet - centrifugal pump combination where the jet pump is used on the suction side to improve the suction lift capability of the centrifugal pump.

Generally, in these applications, an unlimited pressure supply is available and the requirement is only

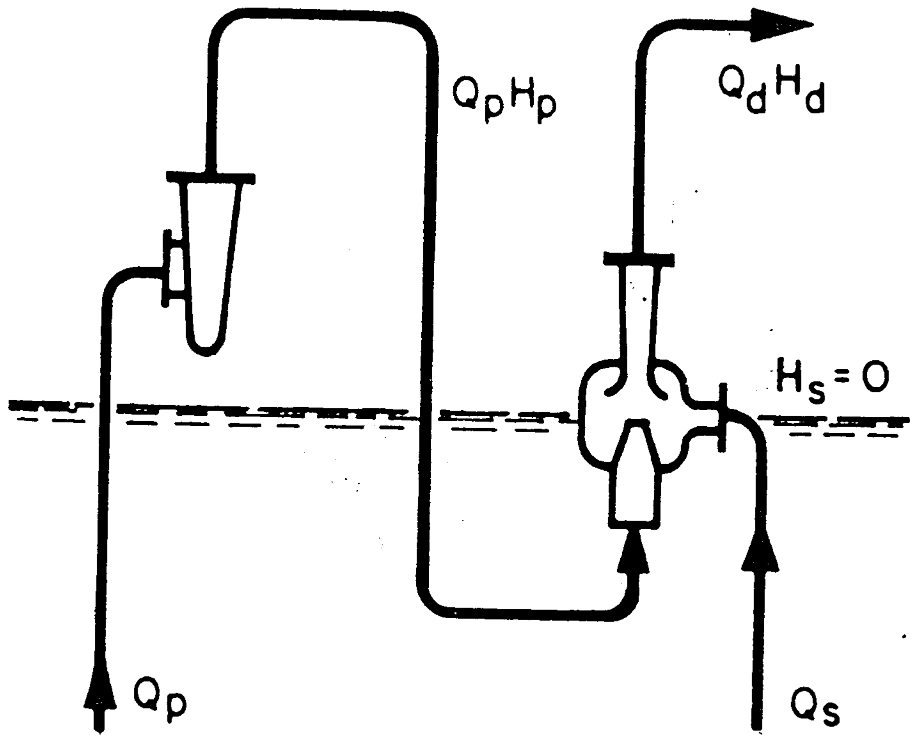


FIG 3. A LOW LIFT PUMPING ARRANGEMENT WHICH UTILISES A CENTRIFUGAL EJECTOR COMBINATION

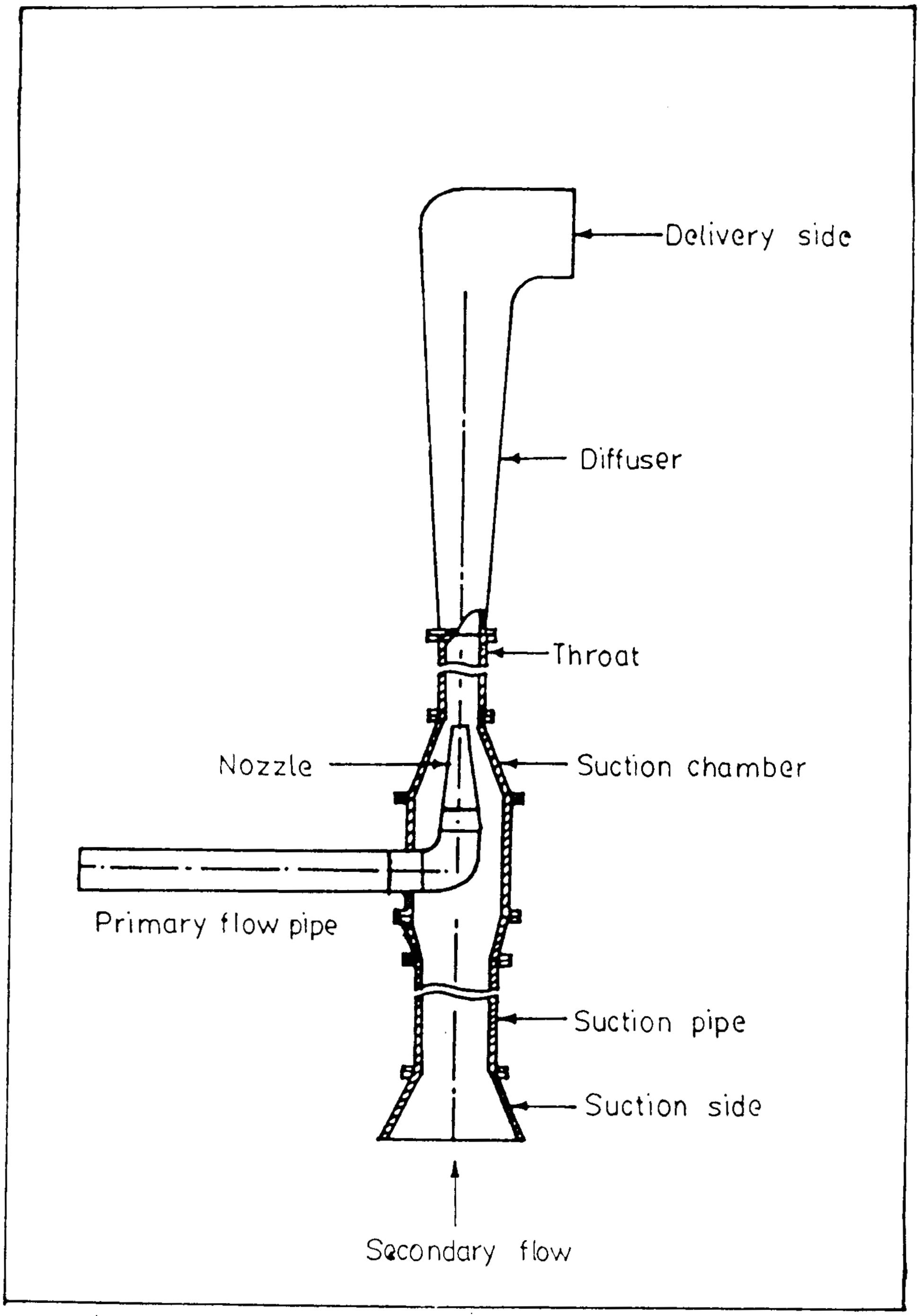


FIG. 4. DETAILS OF EJECTOR SYSTEM (Samuel, 1972)

high flow ratios. This inturn necessitate the use of high capacity (low area ratio) jet pump systems. The performance of low area ratio jet pumps, however, has been studied in detail only recently, by the U.S.National Aeronautics and Space Agency (NASA) in connection with their application in space systems (Sanger, 1970).

Ejector design

Design procedure

In designing a water ejector system for a particular application, the most critical feature to consider is the area ratio. An accepted procedure for the design of water jet pumps for any application is that described by Stepanoff (1964). Here the flow phenomenon is related to physical parameters of the jet pump by equating the momentum of the driving fluid and the discharge fluid which consists of both the driving and driven fluids. Assuming no pressure changes within the throat (mixing chamber) the relation is

$$\rho \quad Qn \quad Vn = \rho \quad (Qs + Qn) \quad Vt \quad - \quad (1)$$

where ρ is the density, Vn is the velocity through the nozzle and Vt the velocity at the throat exit.

Since $R = \dfrac{An}{At}$ by applying continuity relations it can be shown that, the ratio of secondary flow to primary flow through ejector, i.e. Flow ratio,

$$M = \frac{1}{\sqrt{R}} - 1 \qquad (2)$$

where An = Area of nozzle, and

At = Area of throat

This equation gives the particular area ratio to be used for a required flow ratio. From an efficiency envelop curve stepanoff located the corresponding best efficiency value. This value, when divided by the flow ratio specified, provides the best efficiency pressure ratio at which the jet pump can be operated. The component parts of the jet pump are then proportioned on the basis of existing recommendations made from experimental investigations.

Cairns and Na (1969) proposed an empirical N-M envelope equation,

$$N = 0.38 \; M^{-0.81} \qquad - \quad (3)$$

where N = Head ratio of the system = $\frac{\text{Total head of the ejector}}{\text{Net input head}}$

for use in the design of jet pumps. This equation when compared with equations (1) and (2) will describe the relations among R, N and M for optimum conditions.

A good deal of information is available in the literature regarding jet pump proportions. The recommendations given by Samuel (1975) in Table 2 are in agreement with those of Stepanoff (1964). However, the extent to which they are applicable to extremity low area ratio jet pumps is unknown - except for the results reported by Sanger (1970).

Later an attempt to update the efficiency envelope curve developed by Stepanoff was done by Samuel (1975). Fig.5 shows the maximum efficiency values quoted by Stepanoff for area ratio 0.5 and above, and values obtained by recent researchers for area ratios below 0.5. A first approximation efficiency curve drawn through the points was used in calculating the corresponding N values using equation (2). This inturn indicated that the N-R relations could be represented by a straight line. Fig.5 also shows the curves using N-M slope values of 1.5 and 1.6 which formed an upper and lower envelope to the efficiency values indicating that an equation

$$N = 1.5 R \qquad - \qquad (4)$$

would meet the desired criterion satisfactorily. Since $\eta = M \times N$, this in turn provided the emperical

Table 2. Jet pump proportions: Comparison of recommendations by different researchers (Samuel, 1975)

	Reference		S/d	L/Dt	Main nozzle shape	Suction nozzle shape/ angle (θ)	Diffuser angle (θ)
1	Cunningham	(1957)	$\frac{\frac{1}{4}(1-R)}{R}$	4	Quarter ellipse	Shortcone 120	8 - 9.5°
2	Stepanoff	(1964)	1	6	-	-	-
3	Silvester	(1968)	1	7	Conical	Bellmouth or conical 20 - 90°	5°- 0
4	Reddy	(1968)	1-2	$\frac{18d}{Dt}$	Conical 16-20°	20-24°	5° - 0
5	Sanger	(1970)	(1+M)	5.66	Ellipse	Bellmouth	6 - 8°

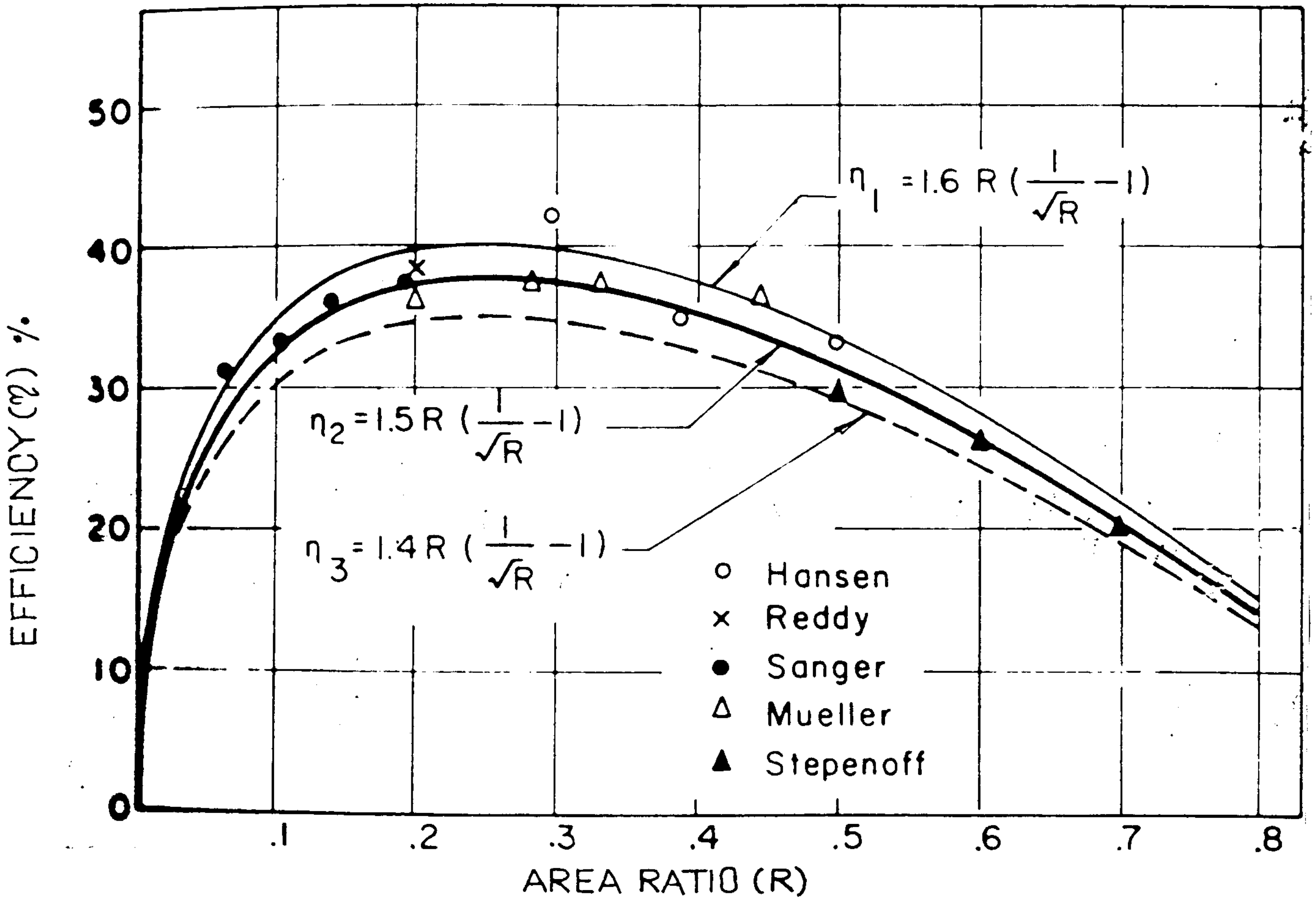


FIG.5. EFFICIENCY ENVELOPES OF JETPUMP PERFORMANCE BASED ON TWO N-R RELATIONSHIPS (Samuel, 1975)

equation for emperical envelope as

$$\eta = 1.5 \quad (R^{\frac{1}{2}} - R) \quad - \quad (5)$$

Thus for conventional applications of the ejector system, the set of equations (2), (4) and (5) could be used as design equations, all of which are related only to R as the independent parameter as shown in Fig.6. They indicate that a maximum efficiency of 37.5 per cent can be expected at a flow ratio of one for an area ratio of 0.25, the corresponding pressure ratio being 0.375. These equations can be rearranged in terms of the flow ratio to give

$$R = (1 + M)^{-2} \qquad (6)$$

$$N = 1.5\ (1 + M)^{-2} \qquad (7)$$

$$\eta = 1.5\ M\ (1 + M)^{-2} \qquad (8)$$

According to Gosline and O'Brien (1934) the characteristics of the water jet pump can be represented by three ratios. These ratios are the nozzle - throat area ratio, R = An/At, secondary-primary flow ratio M = Qs/Qn and pressure ratio $N = (P_5-P_2)/(P_1-P_2)$. The ratios used to compute efficiency can be different and are determined by the mode of use of the pump. The conventional form, based on the concept of useful work done as Qs (Hd-Hs)

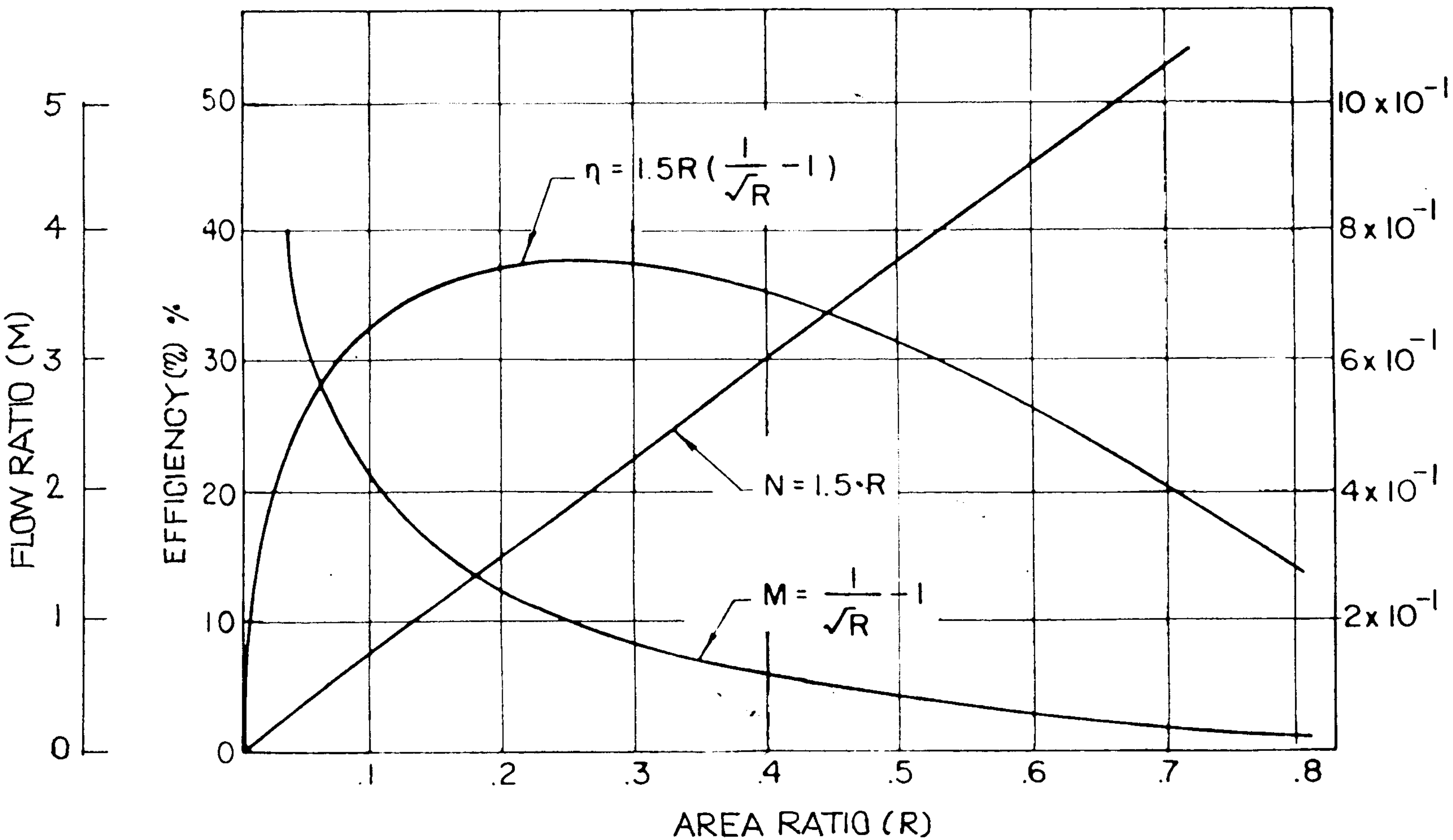


FIG.6. SUGGESTED DESIGN RELATIONSHIPS FOR CONVENTIONAL JET PUMPS APPLICATIONS (Samuel, 1975)

and input energy expended as Qn (Hp-Hd), for expressing efficiency is

$$\eta = \frac{Qs\ (Hd-Hs)}{Qn\ (Hp-Hd)} = \frac{Qs}{Qn} \times \frac{P_5-P_2}{P_1-P_5} = MN \quad - \quad (9)$$

This relation is shown graphically in Fig.7.

Silvester (1960) pointed out that when used in a booster system, and the primary and secondary liquids are drawn from the same source the useful work consists of both Qn and Qs being elevated through a head of Hp-Hs. The centrifugal pump then has only to supply a head of Hp-Hs to Qn which results in the efficiency ratio.

$$\eta^* = \frac{Qs + Qn}{Qn} \quad \frac{Hd - Hs}{Hp - Hs} = \frac{Qd}{Qn} \times \frac{P_5-P_2}{P_1-P_2}$$

$$= (1 + M) \quad \frac{N}{N + 1} \quad = M^* N^* \quad - \quad (10)$$

M* and N* can be named as the capacity ratio and head ratio respectively.

For use in low lift pumping application equations (6), (7), (8) were modified in accofdance with equation (10). This lead to the following set of equations.

$$M^* = \frac{1}{R^{\frac{1}{2}}} \qquad (11)$$

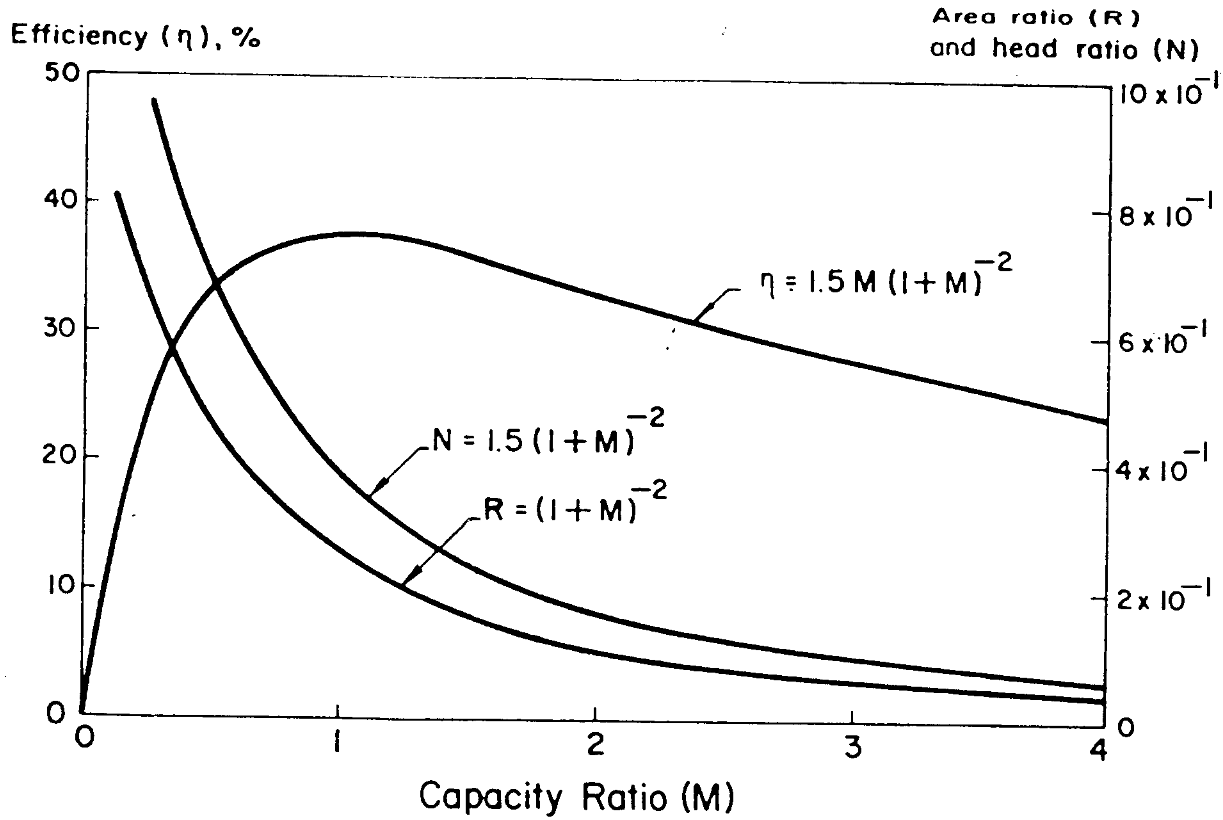


FIG.7. DESIGN RELATIONSHIPS FOR WATER JET PUMPS

(Samuel, 1975)

$$N^* = 1.5\ R\ (1+1.5\ R)^{-1} \qquad (12)$$

$$\eta^* = 1.5\ R^{\frac{1}{2}}\ (1+1.5\ R)^{-1} \qquad (13)$$

$$R = M^{*\ -2} \qquad (14)$$

$$N^* = 1.5\ R\ (1.5 + M^{*2})^{-1} \qquad (15)$$

$$\eta^* = 1.5\ M\ (1.5 + M^{*2})^{-1} \qquad (16)$$

These relations are shown graphically in Fig.8 and Fig.9.

It may be seen from the graphs that when using a centrifugal-ejector combination the efficiency will be higher in low lift applications than the conventional applications.

The design criteria adopted for the work under the study was based on the work done by Sankaranarayanan (1981) and Hajilal (1987).

An experimental ejector system was designed and developed by Sankaranarayanan (1981) to match a 5 HP pumping system. Here the test run speed of the pump was 3300 rpm. In the present study the specific speed of 3600 rpm was maintained throughout. Another ejector system to match 10 HP pumpset, which was obtained by series connection of 2 numbers of 5 HP pumpsets also was fabricated.

The detailed design procedure is discussed in the chapter Materials and Methods.

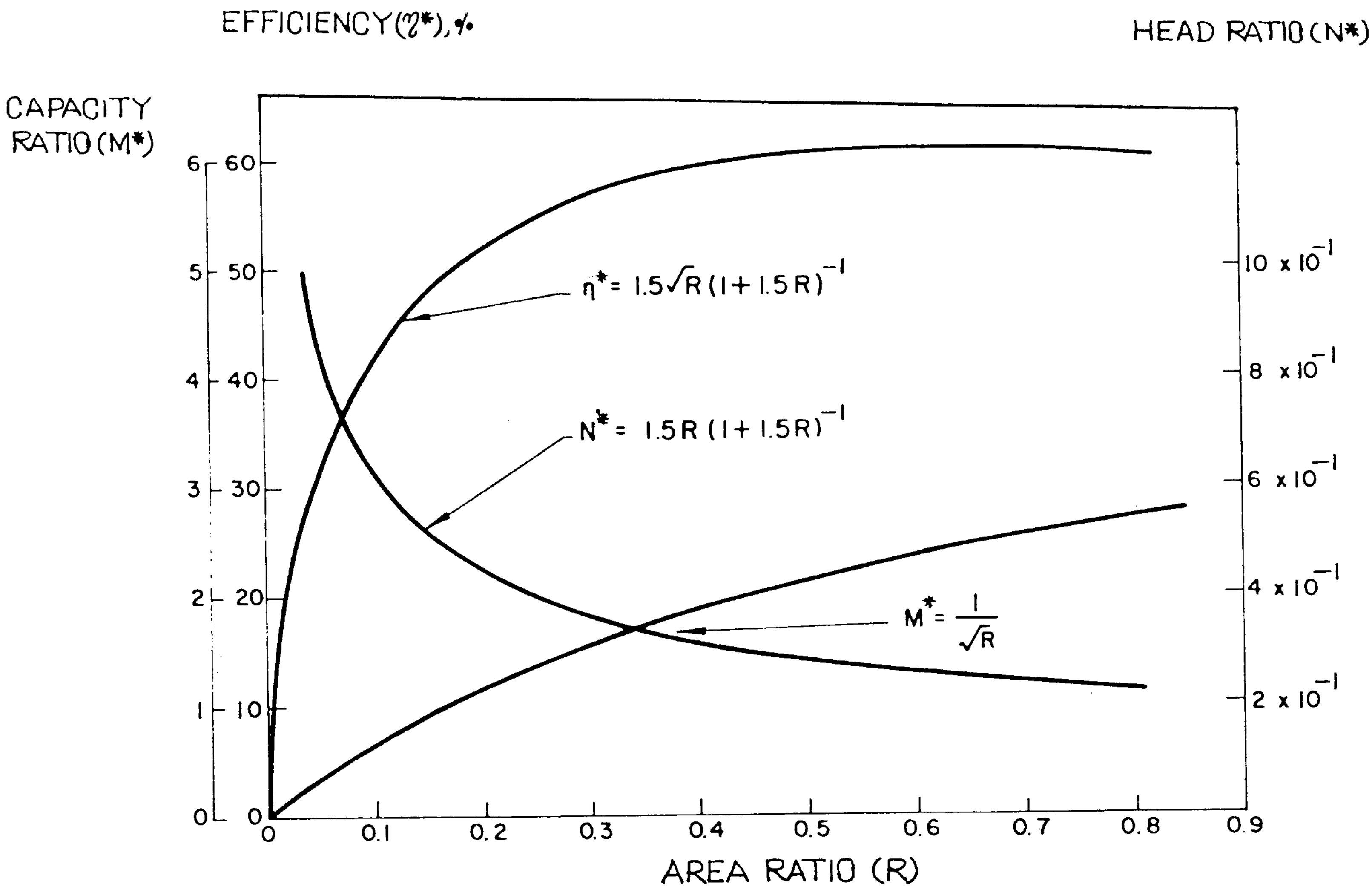


FIG.8. DESIGN RELATIONSHIPS IN CENTRIFUGAL JET PUMP (EJECTOR) COMBINATION FOR VARIOUS AREA RATIOS (Samuel, 1975)

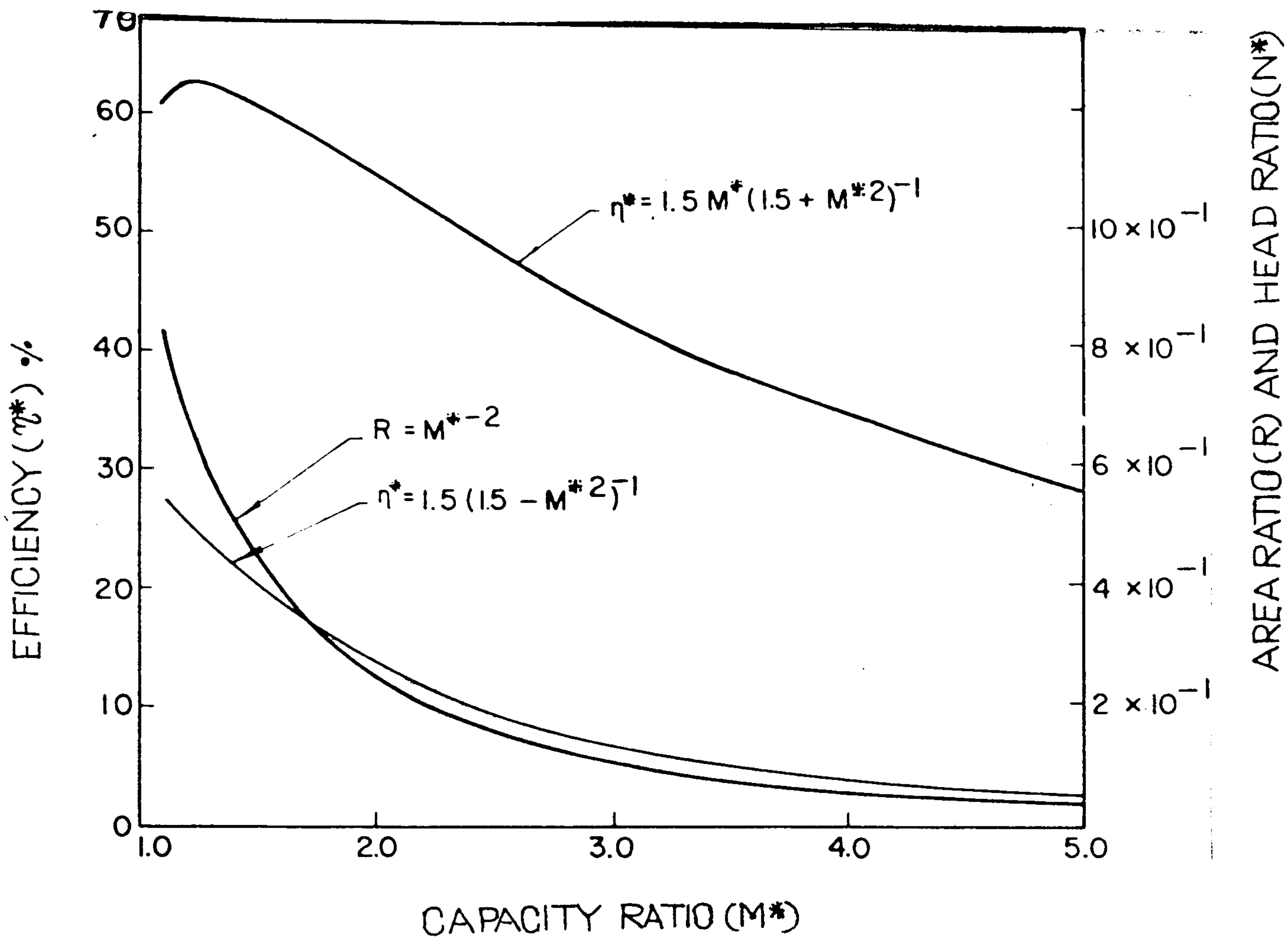


FIG 9. DESIGN RELATIONSHIPS FOR WATER JET PUMPS

(Samuel, 1975)

Materials and Methods

MATERIALS AND METHODS

3.1 Objectives

The ultimate objective of the project was to increase the capacity of the conventional centrifugal pump sets in the low lift condition using an ejector attachment.

The specific objectives were:-

1. To study the design parameter and efficiency characteristics of high capacity ejector system (high volume jet pump) to suit conventional centrifugal pump sets.

2. To develop high volume ejector system suitable for the conventional centrifugal pumpset.

3. To conduct laboratory testing/field testing for different types of ejector systems for different discharge capacities under different suction lifts.

3.2 Concepts utilised

A simple design concept was utilised in this project to increase the discharge capacity of centrifugal pumps.

The equipment consisted of a low-cost, high capacity ejector device which when used in combination with a medium or high head (pressure) centrifugal pump utilises the discharge of the said pump to suck an additional quantity of water through the device.

In this arrangement the entire delivery of a centrifugal pump was fed to a jet pump (ejector) which lifted an additional quantity of water for a small vertical distance. Then it differed from the more conventional jet centrifugal pump combination where the jet pump was used on the suction side to improve the suction lift capability of the centrifugal pump.

The passage of pump discharge in the form of a high velocity jet of water caused the energy to be transferred to the stationary column of water present in the device. The movement of this otherwise stationary water with the jet, further resulted in drawing additional quantities of water through the suction side of the ejector. The quantities of water thus sucked into the ejector system might be several times the discharge capacity depending on the discharge pressure developed by the centrifugal pump. It was found that the total capacity (including the primary flow and the secondary flow) varied with the change in suction head. Studies were conducted to find the optimum suction head. The technique also allows the use of any locally available portable engine, power tiller or a tractor driven pump set as a prime mover, which permits a considerable reduction of the fixed cost of operation justifying the ejector system.

3.3 Design of ejector to match 5 HP pump

Step 1

From the performance characteristics curves of the 5 HP pump supplied by the manufacturer (Fig.10) the best efficiency values were taken. The best efficiency values for the discharge Q and Head H for the pump set were taken to be 480 lpm and 26 metres respectively. An allowance of 20 per cent (5.0 m) of the total head was given for system losses at this point. Hence the effective head, Hn and capacity, Qn available to the ejector system were taken to be 21 metres and 480 lpm respectively for design purpose. From these two parameters, the critical diameter 'Dn' of nozzle of the ejector system was determined as follows:

$$Qn = Cd \times An \times Vn$$

Where Qn is the discharge through the nozzle, Cd the coefficient of discharge, An the area of the nozzle and Vn the velocity of flow through the nozzle.

$$\text{Hence} \quad An = \frac{Qn}{Cd\sqrt{2g\ Hn}}$$

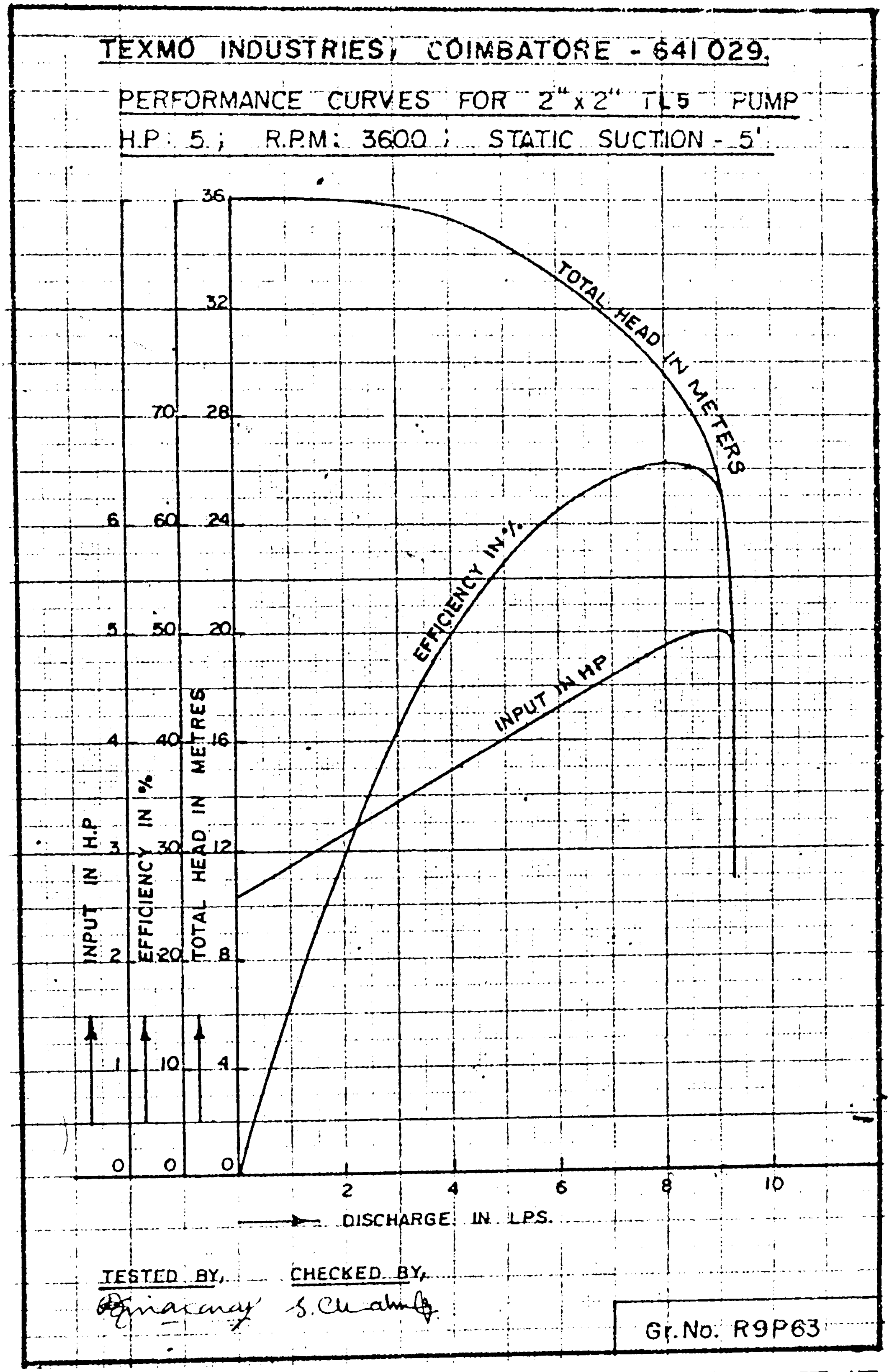


FIG.10. PERFORMANCE CURVES OF PRIME MOVER PUMPSET AT 3600 rpm (MANUFACTURER'S DATA)

Assuming Cd = 0.95

$$An = \frac{480 \times 100 \times 100}{1000 \times 60 \times 0.95 \times \sqrt{2 \times 9.81 \times 21}}$$

$$An = 4.15 \text{ cm}^2$$

But $An = \frac{\pi Dn^2}{4}$

Hence $Dn = \sqrt{\frac{4 An}{\pi}}$

= 2.29 cm

= Say 23 mm

Step 2

The main characteristics of the ejector system can be defined by the following ratios.

1. Flow ratio, $M = \frac{Qs}{Qn} = \frac{\text{Secondary flow}}{\text{Primary flow (through nozzle)}}$

2. Capacity ratio, $M^* = \frac{Q_t}{Q_n} = \frac{\text{Total flow}}{\text{Primary flow (through nozzle)}}$

3. Multiplication factor, $F = \frac{Q_t}{Q_p} = \frac{\text{Total flow}}{\text{Primary flow (without nozzle)}}$

4. Head ratio, $N = \frac{Hd-Hs}{Hn-Hd} = \frac{\text{Total head of ejector system}}{\text{Net input head to the ejector system}}$

where

Hd = the discharge head of the ejector system

Hs = the suction head

Hn = the effective input head to nozzle

The following experical design equations were utilised.

$$M = \frac{1}{\sqrt{R}} - 1 \qquad (a)$$

$$N = 1.5 \; R \qquad (b)$$

$$\eta = M \times N = (1.5\,R) \left(\frac{1}{\sqrt{R}} - 1\right) \qquad (c)$$

where

R = An ÷ At = Nozzle tip area ÷ Throat area

= the efficiency of the ejector system

Hd is the discharge head of the ejector system, Hs the suction head, Hn the effective input head to nozzle and the efficiency of the ejector system.

From eq (10), $M^* = (I+M)$ and $N^* = \frac{N}{N+1}$ $\eta^* = M^*N^*$

Assuming Hs is 1 metre and Hd is 2 metres (to include sudden contraction and friction losses in the suction line, at least one metre of static lift, and friction and velocity head losses in the discharge line, respectively) and substituting the values.

$$N = \frac{2(-1)}{20-2} = \frac{1}{6}$$

$$R = \frac{N}{1.5} = \frac{1}{6 \times 1.5} = \frac{1}{9}$$

$$M = \frac{1}{\sqrt{R}} - 1 = 2$$

The throat diameter 'Dt' can be calculated from the relation

$$Dt = Dn \times R^{-\frac{1}{2}}$$

$$= 23 \times \frac{1}{\sqrt{1/9}} = 69 \text{ mm}$$

$$= \text{say } 70 \text{ mm}$$

Step 3

The remaining dimensions of the ejector system were found out by the relation developed by Samuel (1975).

Throat length = 5 Dt = 5x70 = 350 mm

Nozzle spacing = 0.75 Dt. = 0.75 x 70 = 52.5 mm

Throat entry profile = 90°

Diffuser angle = 7°

The expected efficiency of the ejector system was found from the equation

$$\eta = M \times N = 2 \times \frac{1}{6} \times 100 = 33\%$$

3.4 Fabrication works of ejector system

The nozzle was fabricated with 10 guage sheet metal (Fig.11). The cone angle was taken as 16° and the nozzle

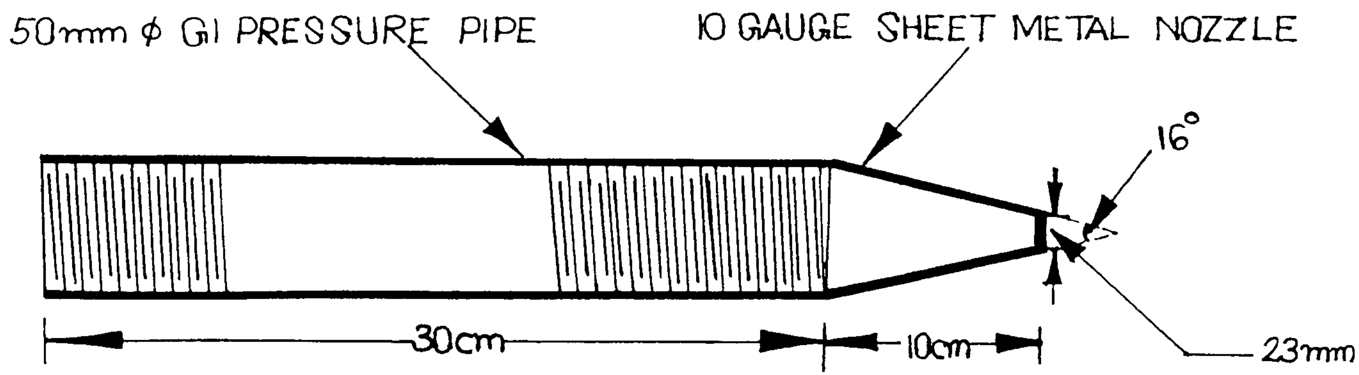


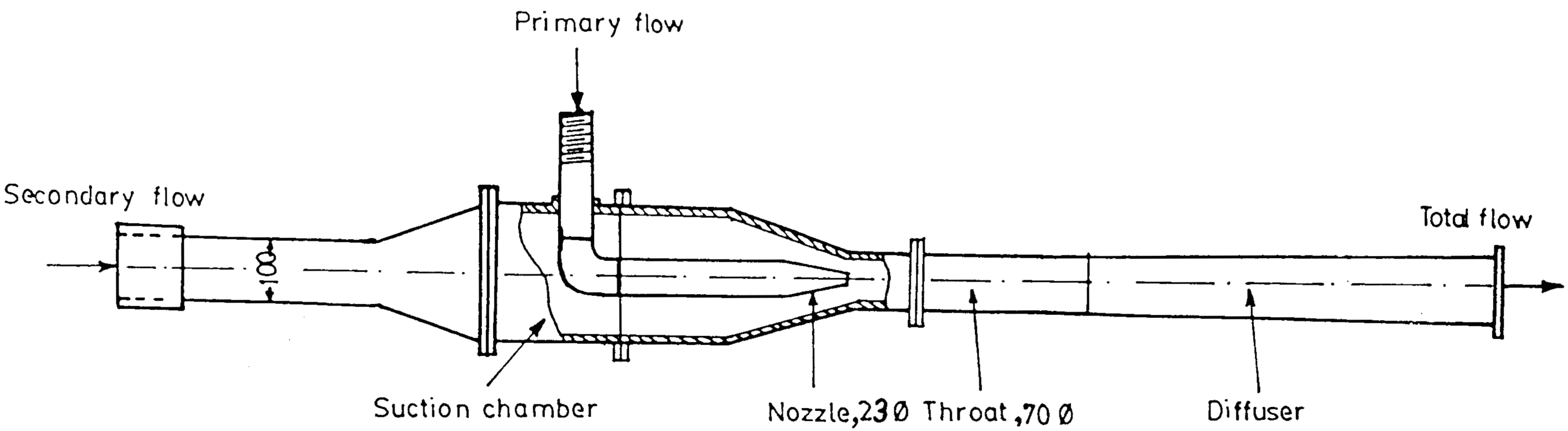
FIG.11. NOZZLE ASSEMBLY

length was calculated as 11 cm in order to weld it to a piece of standard 50 mm GI pipe. A 50 mm GI coupling was provided on the suction chamber to which the nozzle was fitted. The pipe of the nozzle assembly was externally threaded to a length of 50 mm which enabled the adjustment of the nozzle throat spacing. To study the effects of direction changes in primary and secondary flows, four different ejectors were fabricated with the above desinged values. The directions of the primary flow as well as secondary flow were changed in these ejectors as shown in Figures 12 to 15.

In the ejector (E1), the primary flow was made at right angle to the secondary flow and the secondary flow was kept straight (Plate 1).

In the ejector (E_2), the primary flow was made straight and a direction change of 40° was given to the secondary flow by using a portion of a 10 cm diameter 90° elbow as the suction chamber (Plate 2).

Ejector (E_3) was also fabricated with the primary flow straight and a change of direction of 20° was given to the secondary flow with the help of a portion of 10 cm diameter 90° bend (Plate 3).



Scale 1:10

All dimensions in mm

FIG.12. EJECTOR SYSTEM E1

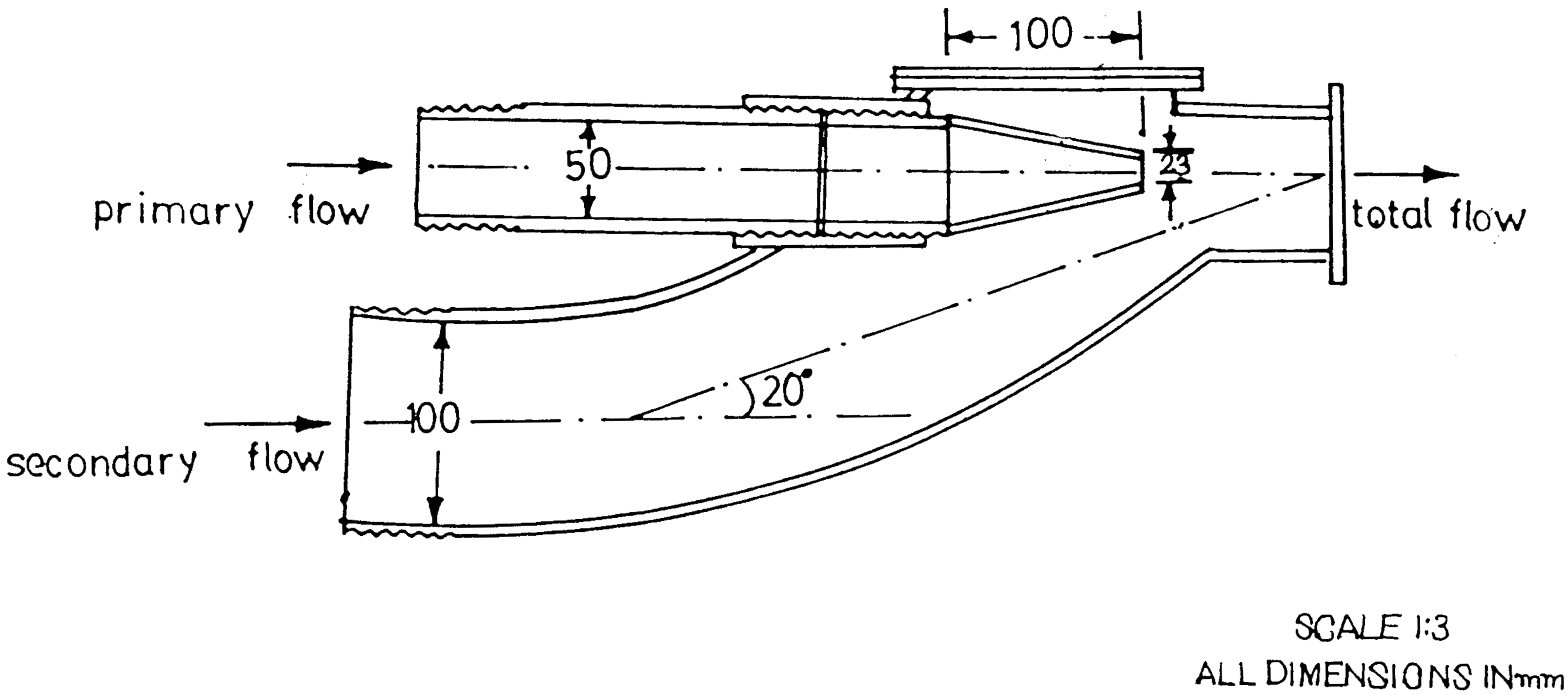
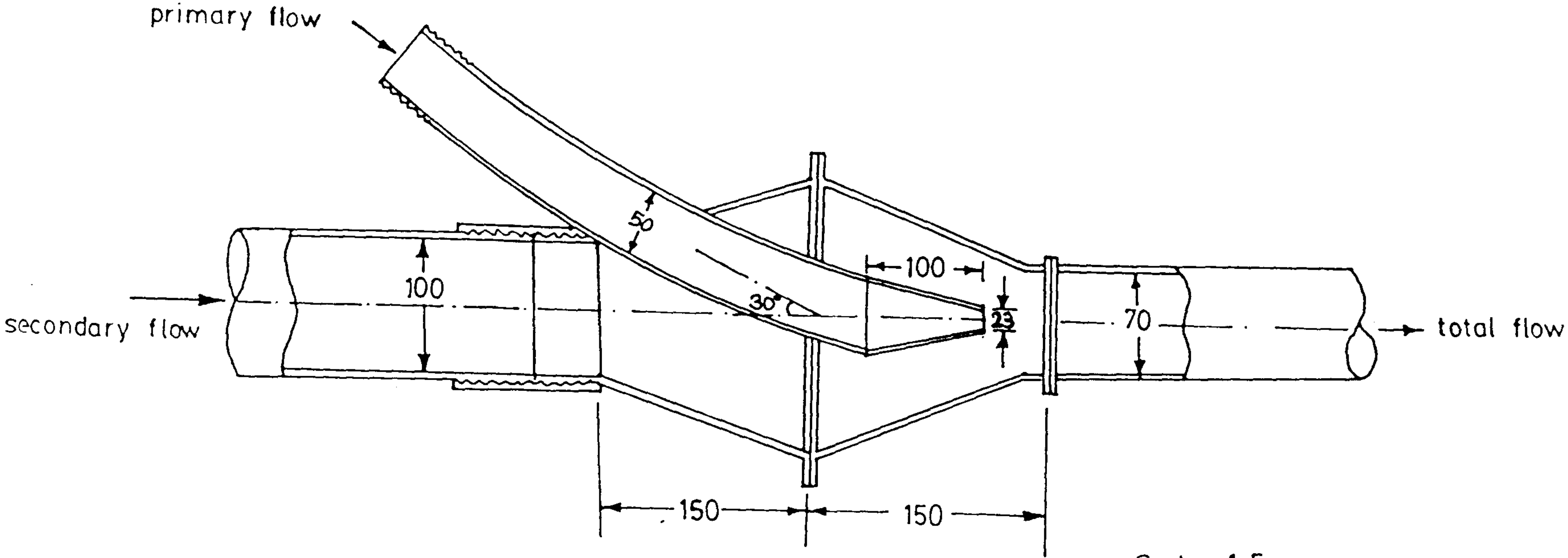


FIG.14. EJECTOR SYSTEM E3



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All dimensions in mm

FIG.15. 30° EJECTOR SYSTEM (E4)

The ejector (E_4) was fabricated with 16 guage M.S.sheet with the secondary flow straight and a 30° change of direction to the primary flow. The ejector suction and discharge pipe size were chosen as 10 cm for convenience in handling and fitting of pipe connections (Plate 4).

3.5 Preliminary experiments

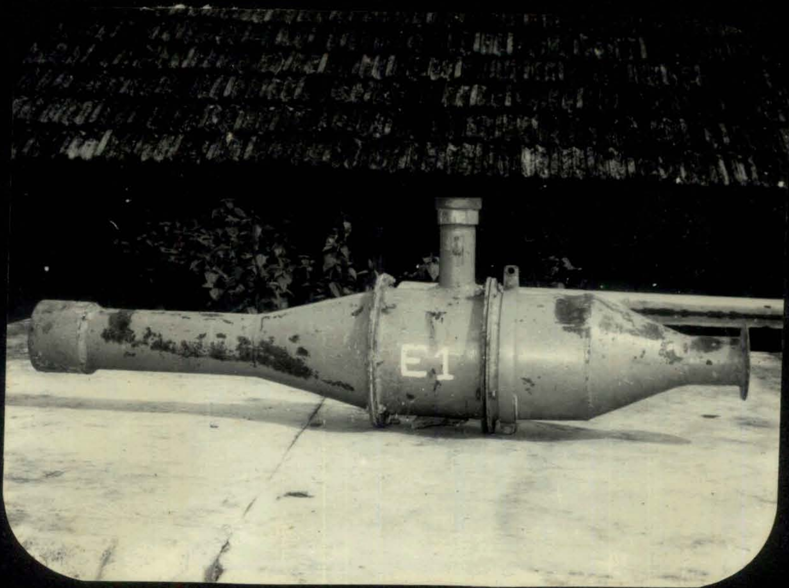
These four ejectors were tested at different suction heads and the corresponding capacities were noted. A light weight 5 HP, 3600 rpm "Greaves Lombardini", Diesel engine coupled to a 10 - 35 m head, 3600 rpm "TEXMO" pump was used as the prime mover (Specification of the 5 HP pump set are given in Appendix III).

The tests were conducted at the test tank of Agricultural Engineering Research Workshop at Mannuthy. There is a twin tank each with a capacity of 12,000 litres and separated in between by a platform. The prime mover pumpset and also the ejector system were kept on the platform as in Plate (5).

The two tanks were of size 4 m x 2 m x 1.5 m. There was a flow path between the two tanks at a depth of 20 cm from the top of the platform. Till this depth water

Plate 1. Ejector E_1

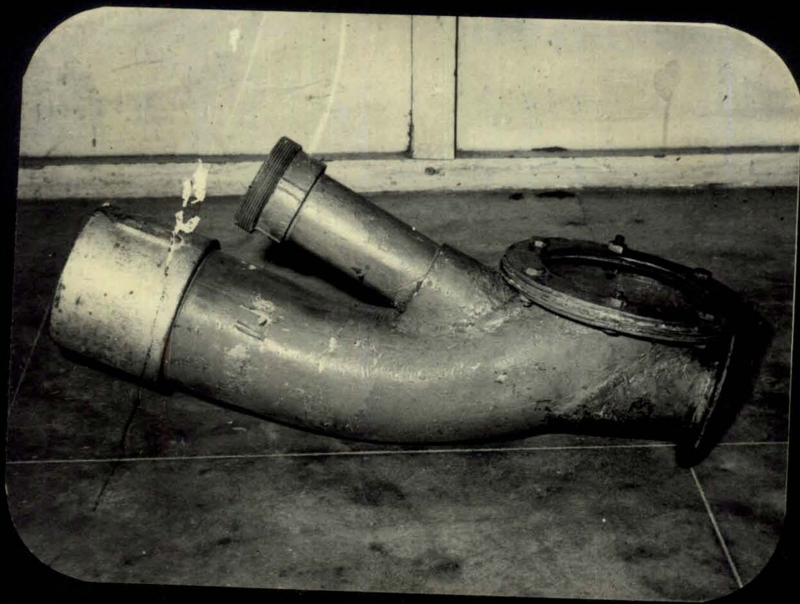
Plate 2. Ejector E_2

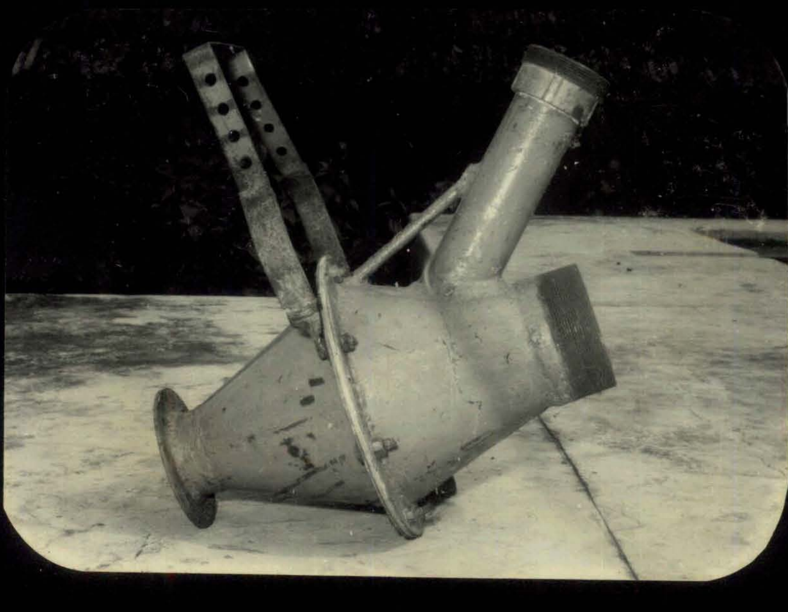

E1


E2

Plate 3. Ejector E_3

Plate 4. Ejector E_4





could flow from one tank to the other. The suction pipe of the prime mover and the suction side of the ejector system were kept in tank B. The suction lift is taken as the distance between the centre of the pump and the water level. The desired suction lift was maintained in the tank B by pumping out water from it. The suction lift was measured by a steel tape. While doing experiments the water discharged was collected in a drum, which was kept in the tank A. This enabled in preventing even very small loss of water. The tank A was always kept full (upto the passage). When additional water came into tank A, this additional water would come back to tank B, thus keeping the head constant during a particular test.

The experiments were done in such a way that the delivery head was made zero. For this the top of the collecting drum was kept in the same horizontal line as the centre of the pump. The frictional loss in the pipe line and fittings were almost same in all the experiments. Hence total head could be replaced by suction lifts in plotting the graphs. Due to the absence of adequate devices in measuring the frictional loss, it could not be measured in the present study.

A drum of size 0.6 m diameter and 0.9 m height was used to collect water. The capacity of the drum was approximately 250 litres.

The pumping capacity was determined as follows. The water was collected in a 250 litre drum, and the time required to fill the drum was noted by an accurate stop watch (Plate 6). Later the capacity was converted into litres per minute.

In operation, when the centrifugal pump was started after priming, water was drawn through its (primary) suction pipe and delivered foreecefully to the nozzle of the ejector. The nozzle converted the high pressure water to high velocity jet of water. This jet of water caused the stationary water in the ejector to move with it. An induced flow of water is simultaneously created at the ejector (secondary) suction and a large volume of water was drawn in. Combined flow was then forced out through the delivery pipe of the ejector system.

The water level in both the tanks were kept at a depth of 2 cms from top of the plantform. This helped in keeping the suction lift at 0.30 metres. The ejector was kept on the platform in such a way that the centre of the ejector system and the centre of the pump were at the same horizontal plane. Flexible pipes were used. The diameter of suction pipes and delivery pipes of the centrifugal pump was 50 mm and those of the ejector was 100 mm.

Plate 5. Preliminary experiments

Plate 6. Collection of water into a 250 litre oil drum

A 250 litre capacity oil drum was used for measuring the discharged water. A stop watch (Racer) was utilised to note the time for filling the drum. The experiments were conducted at five suction lifts of 0.30 m, 0.60 m, 0.90 m, 1.20 m and 1.50 m. All the four ejectors were tested at these suction lifts. Both primary flow and primary plus secondary flow were measured at each suction lift. Primary flow was noted in two ways - Before and after passing through the ejector - This was done to find out the percentage loss of primary flow while passing through the ejector. It helped in finding out the multiplication factor also.

The length of the inter connection between the prime mover and the ejector system was considerably reduced which gave the additional advantage of reduced frictional losses between the centrifugal pump and the ejector nozzle. The suction pressure and delivery pressure were noted using two pressure guages.

Preliminary experiments were conducted only upto 1.5 metre suction lift for convenience. For greater depth, a temporary platform should have been constructed. To avoid this and thus to make the preliminary experiment simple, the suction lift was limited to 1.5 metres. There was a lot of fluctuations observed in the reading of the veccum guage fitted on the suction side of the prime mover pumpset. Hence performance of the test was analysed by measuring the actual suction lift condition of each test criteria by measuring the depth of water level from the centre of the pumpset by a steel tape.

3.6 Main experimental programme

In preliminary experiments E_2 and E_3 gave almost similar results. But E_2 was selected for further experiments due to the following reasons.

1. Easiness of fabrications - E_3 was found to have fabrication difficulties - because of the small angle of inclination of the secondary pipe to the primary pipe. E_2 was constructed with the easily available GI elbow, whereas E_3 was fabricated with a special bent, available only in a few shops of business locality.

2. Easiness of fitting - Due to the small angle of inclination fitting of ejector to the delivery of the prime mover centrifugal pump set was found to be tiresome.

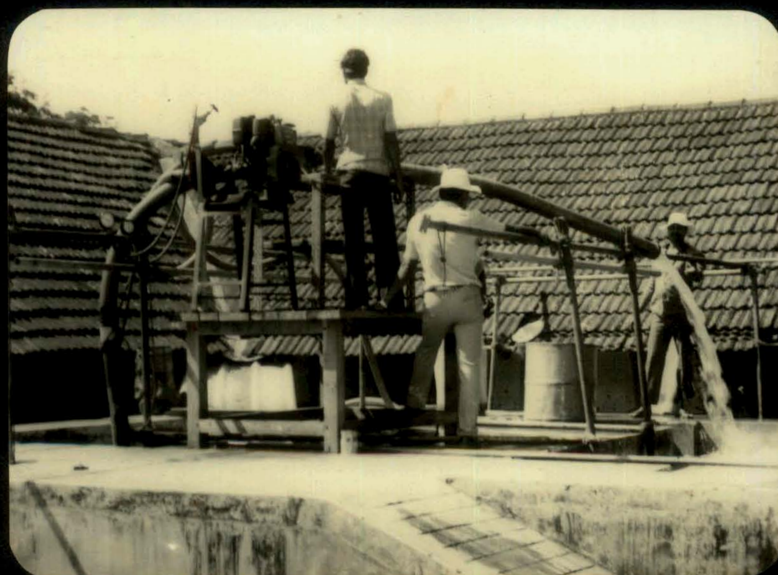
3.6.1 Experiments with a single 5 HP engine pumpset

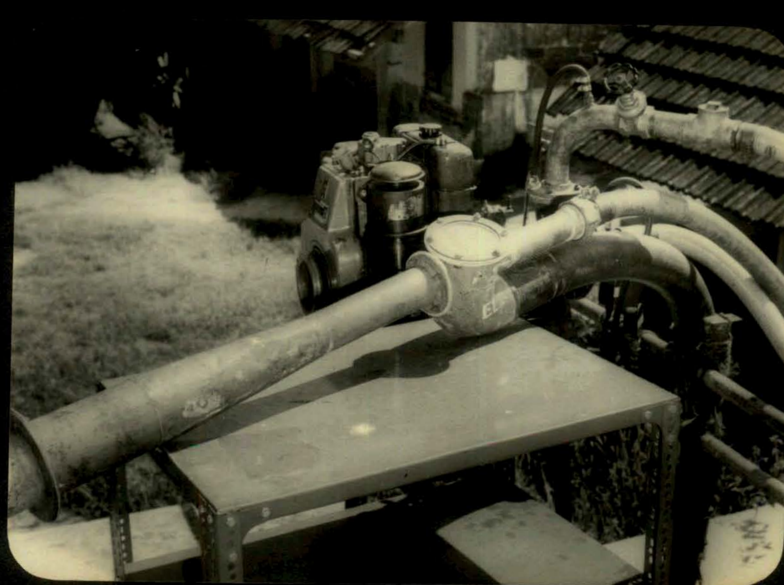
The experiment was conducted using E_2 upto a suction lift of 3.00 metres. For conducting experiments beyond a suction lift of 1.50m a strong wooden table was placed on the platform. The pumpset was fixed on the table. The ejector system also was kept on the table. The base plate was fixed to the top of the table so that the variations could be reduced to a minimum. To conduct tests at further suction lifts the pump and the ejector system had to be raised. For this an iron frame was made as shown in Plate (7). The pumpset was fitted to the frame and the frame was fixed to the table. An iron table of the same height was also made to support the ejector system (Plate 8). Experiments were carried out upto a suction lift of 3 metres. The arrangement of pressure guages is shown in plate 9. The readings were noted at suction lifts of 1.80 m, 2.10 m, 2.40 m, 2.70 m, 3.00 m.

Some modifications were made at the suction side of the secondary tube. As the secondary pipe was kept on the table, along with the pumpset there was considerable reduction in the secondary suction. This may be due to the high air column formed in the secondary tube. Then it was necessary to fit a foot valve to the secondary suction side.

Plate 7. 5 HP pumpset mounted on an iron frame

Plate 8. Ejector system mounted on an iron table in the 5 HP-E_2 combination





As the secondary pipe was of 100 mm diameter no foot valve of such size was easily available in the market. Hence one simple type foot valve was fabricated locally in the Agricultural Engineering Research Workshop with 6 guage MS sheet and leather washer.

3.6.2 Experiments with a 10 HP prime mover pumpset

The same experiments were repeated using the ejector E_2 in a 10 HP arrangement also (shown in plate 10). Here two 5 HP pumpset were connected in series, and a nozzle having tip diameter 20 mm was used in the ejector system.

The size of the nozzle matching the 10 HP pumpsets was designed in the same way as in the case of 5 HP pumpset.

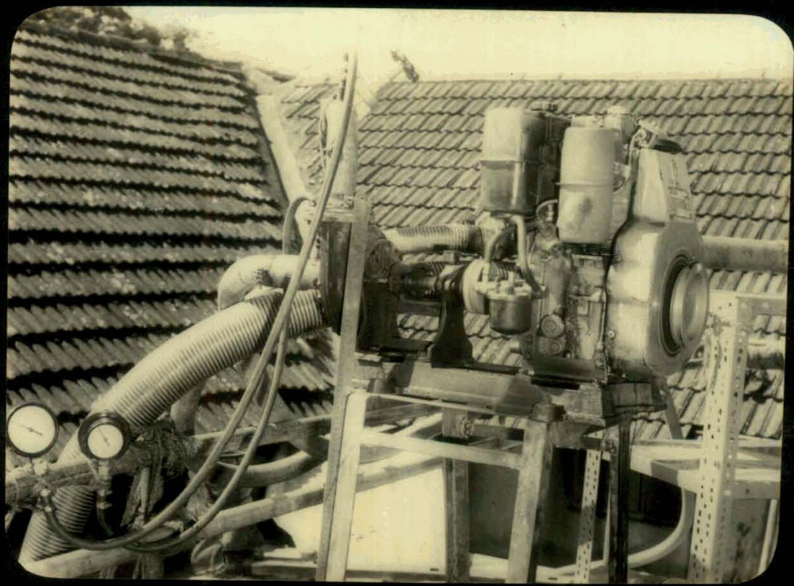
Step 1. In view of the series connection of the prime mover pumpsets the best efficiency head of the combination was taken as 52 metres at 3600 rpm.

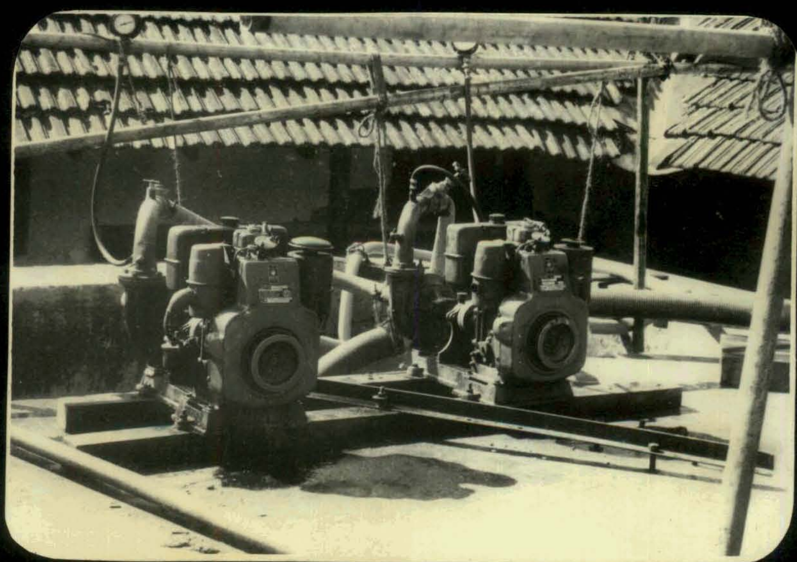
The best efficiency discharge capacity in turn would be 500 lpm at 3600 rpm respectively.

Step 2. Assuming a higher system loss, for the combination as 33.33 per cent on account of the series connection, the effective head, Hn and capacity, Qn available to the

Plate 9. Arrangement of the pressure gauges

Plate 10. Experiments using a 10 HP-E_2 combination





ejector system were taken to be 35 metres and 500 lpm respectively for design purposes

$$An = \frac{Qn}{Cd\sqrt{2g\,Hn}} \quad \text{and } Dn = 2\sqrt{\frac{An}{\pi}}$$

Substituting the values for Qn = 500 lpm, Hn = 35 metres Cd = 0.95 and g = 9.81 m/Sec^2

$$An = \frac{500}{1000 \times 60 \times 0.95 \times \sqrt{2 \times 9.81 \times 35}} \times 100 \times 100 \text{ cm}^2$$

$$An = 3.3 \text{ cm}^2, \qquad Dn = 2\sqrt{\frac{3.3}{\pi}}$$

Dn = 2.04 cm Say 20 mm

Step 3

The nozzle was fabricated with a 16° cone angle and a base diameter of 50 mm was welded to a nipple of the same size. The opposite end of the nipple in turn was fitted to an elbow to introduce the nozzle to the suction chamber in a direction parallel to the secondary flow.

Step 4

Slightly increased values of Hs = 1.5 metre and Hd = 2.5 metres were assumed for ejector system in view of the higher flow rates to be expected.

Accordingly $N = \frac{Hd - Hs}{Hn - Hd}$

$$= \frac{2.5 - (-1.5)}{40 - 2.5} = 0.107$$

$$R = \frac{N}{1.5} = \frac{0.107}{1.5} = 0.0713$$

$$M = \frac{1}{\sqrt{R}} - 1 = 3.75 - 1 = 2.75$$

$Dt = Dn \times R^{-\frac{1}{2}} = 7.49$ cm, say 75 mm

Step 5

Throat length = 5 Dt = 375 mm

Nozzle spacing = 0.75 Dt = 56 mm

Throat entry profile = 90°

Diffusion angle = 7°

The expected efficiency of the ejector system will be M x N = 2.75 x 0.107 = 29.4%

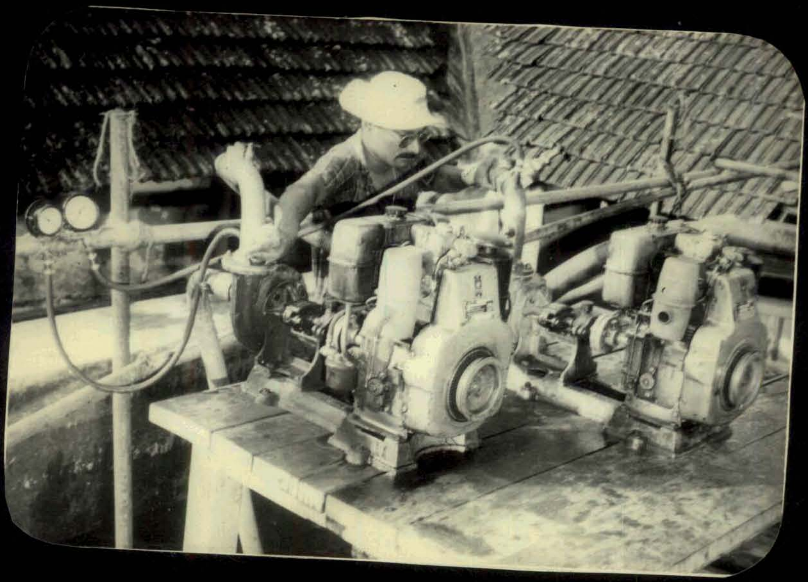
The compoments of the ejector system were made out of 16 guage M.S. sheet and joined by means of flanges at various sections. A size of 46 cm x 20 cm was chosen for the suction chamber. The sizes of suction and discharge pipe of ejector were chosen as 100 mm for convenience in handling and fitting of pipe connections.

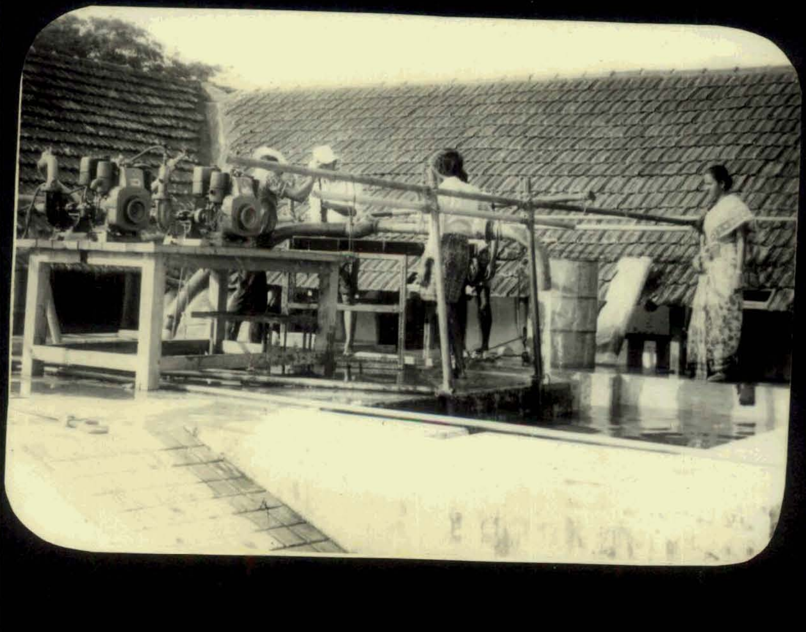
While doing experiments the mixing chamber used for 5 HP pumpset was used by changing the nozzle. The experiment was conducted at different suction heads of 0.30 m, 0.60 m, 0.90 m, 1.20 m, 1.50 m, 2.10 m, 2.40 m etc. The readings were noted as in the case of 5 HP pumpset.

The pumpsets and the ejector were kept on the platform of the test tank upto a suction lift of 1.5 m (plate 11). Then they were mounted on the wooden table for further suction lifts. The ejector system was kept in the same level by placing it on an iron table as shown in plate (12). Experiments upto a suction lifts of 2.4 m were conducted. Experiments at further suction lifts could not be conducted due to practical difficulties.

Plate 11. Two 5 HP pumpsets, mounted on a wooden table and connected in series

Plate 12. The ejector system mounted on an iron table in the 10 HP-E_2 combination





Results and Discussion

RESULTS AND DISCUSSION

4.1 Preliminary experiments

Testing of ejector systems

The results of the preliminary tests using different ejector systems are shown in Tables 3 to 6.

The four ejectors E_1, E_2, E_3 and E_4 were tested upto a suction head of 1.5 metres. The capacity ratio, flow ratio and multiplication factor in each case are calculated and shown in Tables 7 to 10. The variations of capacity ratio, flow ratio and multiplication factor for different suction lifts are shown in Figures 16 to 18.

Capacity ratio

Capacity ratio is the ratio of total flow to the primary flow through the ejector. The graph gives a descending curve from left to right. As head increases capacity ratio decreases. The highest capacity ratios obtained for ejectors E_1, E_2, E_3 and E_4 at a suction head of 0.30 m are 3.67, 4.18, 3.9 and 3.33 respectively.

E_2 and E_3 gave better results than the other two ejectors. Between them E_2 was found to be more advantageous than E_3 due to the reasons already explained in Chapter 3.

In E_1 and E_4 there were changes in the direction of primary flow and due to this, loss of primary head was developed at the entry of the throat of the ejector system. Due to this loss of head, there was a reduction of partial vacuum developed inside the mixing chamber. This was the reason for the decreased capacity ratio in E_1 and E_4 with compared to the capacity ratio of E_2 and E_3.

Flow ratio

Flow ratio is the ratio of secondary flow to the primary flow through the ejector. Thus the capacity ratio = flow ratio + 1. The shape of the curves were same as those of capacity ratio.

Multiplication factor

Multiplication factor is the ratio of total flow to primary flow without ejector (i.e. the discharge of the centrifugal pump). This ratio gives a clear idea of the utility of the ejector system in a set up. It indicates the number of times by which the flow is increased when an ejector system is introduced in the delivery side of a centrifugal pump.

Table 3. Calculation of discharges of 5 HP-E_1 combination

Suction lift (m)	Delivery pressure (kg/Cm^2)	Primary flow			Primary flow through ejector			Total flow (Primary through ejector + secondary flow)		
		Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)
(1)	(2)	(3)			(4)			(5)		
0.30	2.8	250	23.30	644	250	29.60	506	250	8.08	1856
0.60	2.7	250	23.39	641	250	30.90	485	250	9.10	1649
0.90	2.6	250	23.70	633	250	31.33	479	250	10.36	1448
1.20	2.5	250	24.19	620	250	32.08	468	250	11.57	1296
1.50	2.4	250	25.00	600	250	32.70	459	250	12.32	1217

Table 4. Calculation of discharges of 5 HP-E_2 combination

Suction lift (m)	Delivery pressure (kg/Cm^2)	Primary flow			Primary flow through ejector			Total flow (Primary through ejector + Secondary flow)		
		Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)
(1)	(2)	(3)			(4)			(5)		
0.30	2.8	250	23.30	644	250	30.92	485	250	7.40	2026
0.60	2.7	250	23.39	641	250	31.33	479	250	8.47	1771
0.90	2.7	250	23.70	633	250	32.34	464	250	9.28	1616
1.20	2.6	250	24.19	620	250	32.70	459	250	10.28	1459
1.50	2.5	250	25.00	600	250	33.05	454	250	11.13	1348

Table 5. Calculation of discharges of 5 HP-E_3 combination

Suction lift (m)	Delivery pressure (kg/Cm^2)	Primary flow			Primary flow through ejector			Total flow (Primary through ejector + secondary flow)		
		Qty.of water colle-cted (1)	Time taken (s)	Dis-charge (lpm)	Qty.of water colle-cted (1)	Time taken (s)	Dis-charge (lpm)	Qty.of water collected (1)	Time taken (s)	Dis-charge (lpm)
(1)	(2)	(3)			(4)			(5)		
0.30	2.8	250	23.30	644	250	31.0	484	250	7.95	1886
0.60	2.6	250	23.39	641	250	32.00	469	250	9.12	1645
0.90	2.6	250	23.70	633	250	32.25	465	250	9.66	1553
1,20	2.5	250	24.19	620	250	33.70	445	250	11.18	1341
1.50	2.4	250	25.0	600	250	34.48	435	250	11.87	1264

Table 6. Calculation of discharges of 5 HP-E_4 combination

Suction lift (m)	Delivery pressure (kg/Cm^2)	Primary flow			Primary flow through ejector			Total flow (Primary through ejector + secondary flow)		
		Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)
(1)	(2)	(3)			(4)			(5)		
0.30	2.8	250	23.30	644	250	31.58	475	250	9.47	1584
0.60	2.7	250	23.39	641	250	32.25	465	250	10.40	1441
0.90	2.6	250	23.70	633	250	33.43	449	250	11.59	1294
1.20	2.4	250	24.19	620	250	34.38	436	250	13.23	1134
1.50	2.4	250	25.00	600	250	35.08	428	250	14.56	1030

Table 7. Capacity ratio, flow ratio and multiplication factor of 5 HP-E_1 combination

Suction lift (m)	Primary flow (lpm)	Primary flow through ejector (lpm)	Total flow (Primary flow through ejector + secondary flow) (lpm)	Secondary flow (lpm)	Capacity ratio M*	Flow ratio M	Multi-plication factor F
(1)	(2)	(3)	(4)	(5) =(4)-(3)	(6)=(4)	(7)	(8)
0.30	644	506	1856	1350	3.67	2.67	2.88
0.60	641	485	1649	1164	3.40	2.40	2.57
0.90	633	479	1448	969	3.02	2.02	2.29
1.20	620	468	1296	829	2.77	1.77	2.09
1.50	400	459	1217	759	2.65	1.65	2.03

Table 8. Capacity ratio, flow ratio and multiplication factor of 5 HP-E_2 combination

Suction lift (m)	Primary flow (lpm)	Primary flow through ejector (lpm)	Total flow (Primary flow through ejector + secondary flow) (lpm)	Secondary flow (lpm)	Capacity ratio M*	Flow ratio M	Multiplication factor F
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0.30	644	485	2026	1541	4.18	3.18	3.15
0.60	641	479	1771	1293	3.70	2.70	2.76
0.90	633	464	1616	1153	3.49	2.49	2.56
1.20	620	459	1459	1000	3.18	2.18	2.35
1.50	600	454	1348	894	2.97	1.97	2.25

Table 9. Capacity ratio, flow ratio and multiplication factor of 5 HP-E_3 combination

Suction lift (m)	Primary flow (lpm)	Primary flow through ejector (lpm)	Total flow (Primary flow through ejector + secondary flow) (lpm)	Secondary flow (lpm)	Capacity ratio M*	Flow ratio M	Multiplication factor F
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0.30	644	484	1886	1403	3.90	2.90	2.93
0.60	641	469	1645	1176	3.51	2.51	2.57
0.90	633	465	1553	1088	3.34	2.34	2.45
1.20	620	445	1341	896	3.01	2.01	2.16
1.50	600	435	1264	829	2.91	1.91	2.11

Table 10. Capacity ratio, flow ratio and multiplication factor of 5 HP-E_4 combination

Suction lift (cm)	Primary flow (lpm)	Primary flow through ejector (lpm)	Total flow (primary flow through ejector + secondary flow (lpm)	Secondary flow (lpm)	Capacity ratio	Flow ratio	Multiplication factor
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0.30	644	475	1584	1109	3.33	2.33	2.46
0.60	641	465	1441	976	3.10	2.10	2.25
0.90	633	449	1294	845	2.88	1.88	2.05
1.20	620	436	1134	698	2.60	1.60	1.83
1.50	600	428	1030	601	2.41	1.41	1.72

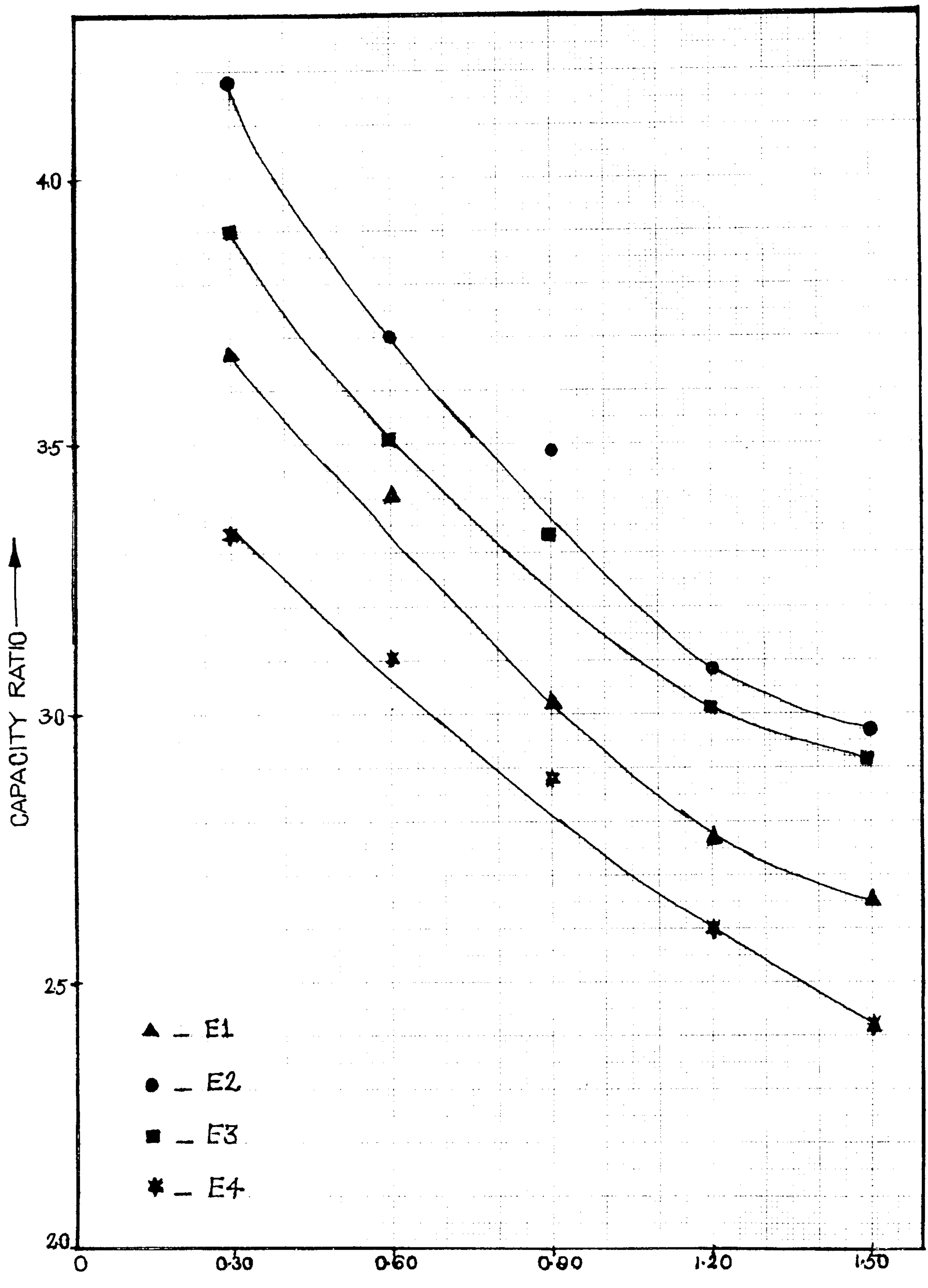


FIG 16. SUCTION LIFT Vs CAPACITY RATIO

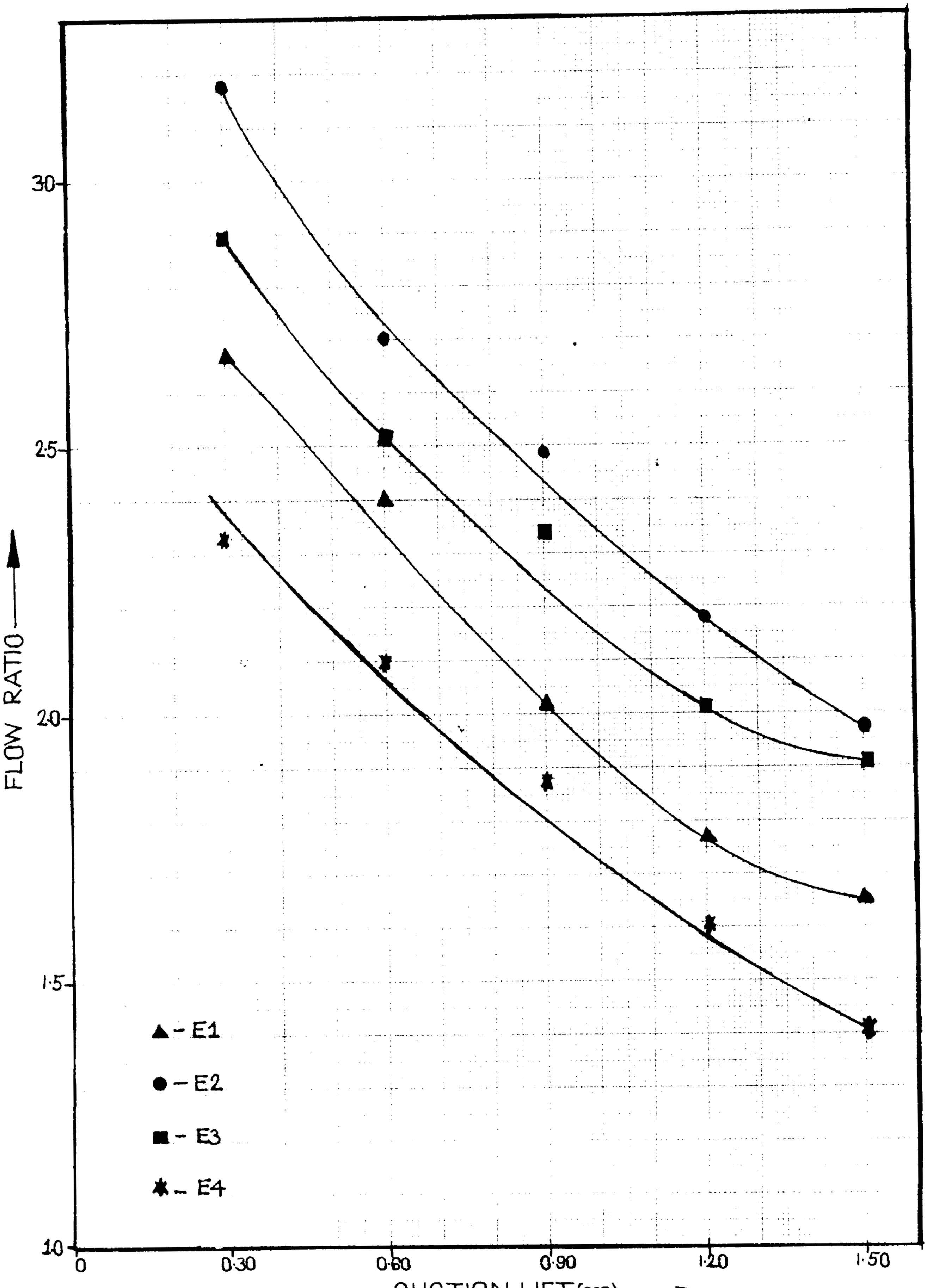


FIG 17. SUCTION LIFT Vs FLOW RATIO

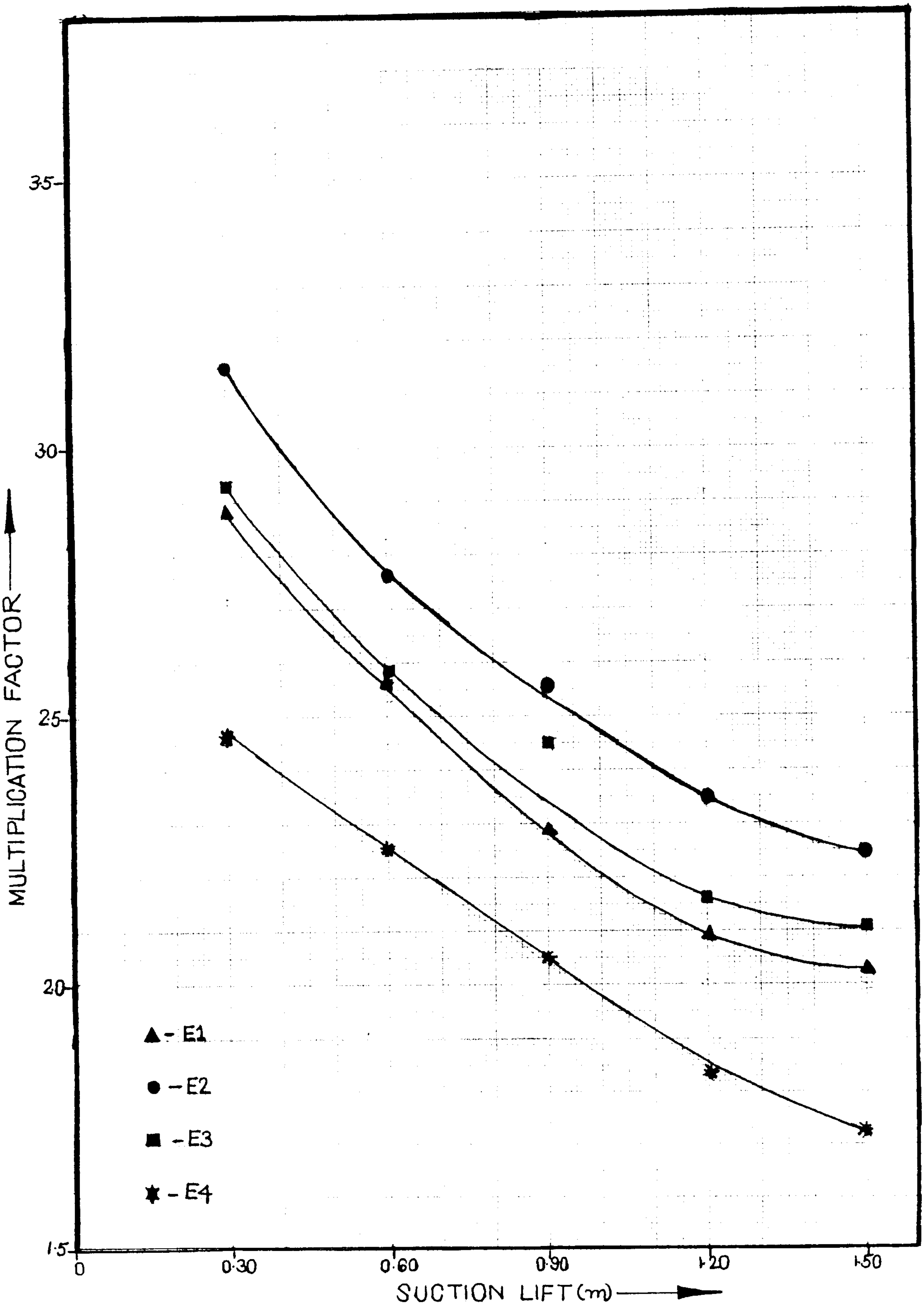


FIG 18. SUCTION LIFT Vs MULTIPLICATION FACTOR

Due to the addition of the ejector system in the delivery side, there was loss in the primary flow of the centrifugal pump. The percentage loss of primary flow is shown in Tables 11, 12, 13 and 14. The range of loss is 20 to 30 per cent.

The curves are similar to that of capacity ratio and flow ratio. The multiplication factor reduced as head increased. For E_1, E_2, E_3 and E_4, the multiplication factor at a suction lift of 0.30 m are 2.88, 3.15, 2.93 and 2.46 respectively. E_2 also gives better multiplication factor than other ejectors. For one metre suction lift the multiplication factor of E_2 is 2.5.

In general the performance values obtained show that a low cost ejector attachment could be utilised to multiply the discharge of a centrifugal pump 2 to 3 times in the suction lift range of 0 to 1.5 metres, without much initial cost.

4.2 Main experiments

4.2.1 Combination of 5 HP prime mover pumpset and ejector E_2

E_2 was selected as the best ejector from the preliminary experiments. The results of the main experiments using E_2 upto a suction lift of 3 metres are shown in

Table 11. Percentage loss of primary flow while passing through ejector E_1

Suction lift (m)	Primary flow without ejector	Primary flow through ejector	Percentage loss at the ejector
0.30	644	506	21.4
0.60	641	485	24.3
0.90	633	479	24.3
1.20	620	468	24.5
1.50	600	459	23.5

Table 12. Percentage loss of primary flow while passing through ejector E_2

Suction lift (m)	Primary flow without ejector	Primary flow through ejector	Percentage loss at the ejector
0.30	644	485	24.7
0.60	641	479	25.3
0.90	633	464	26.7
1.20	620	459	26.0
1.50	600	454	24.3

Table 13. Percentage loss of primary flow while passing through ejector E_3

Suction lift (m)	Primary flow without ejector	Primary flow through ejector	Percentage loss at the ejector
0.30	644	484	24.8
0.60	641	469	26.8
0.90	633	465	26.5
1.20	620	445	28.2
1.50	600	435	27.5

Table 14. Percentage loss of primary flow while passing through ejector E_4

Suction lift (m)	Primary flow without ejector	Primary flow through ejector	Percentage loss at the ejector
0.30	644	475	26.2
0.60	641	465	27.4
0.90	633	449	29.0
1.20	620	436	29.6
1.50	600	428	28.7

Table 15. The capacity ratio, flow ratio and multiplication factor are calculated and shown in Table 16.

For one metre suction lift the multiplication factor is about 2.6. In Kuttanad and Trichur Kole lands during the dewatering period, the range of suction lift is only 0.5 to 1.50 metres. Within this range the multiplication factor is between 2.8 and 2.5. Hence the ejector can very well be used with the centrifugal pump to increase its capacity.

But for a suction lift of 3 metre, the multiplication factor is only 1. Beyond 3 m the discharge capacity will be reduced with the ejector system. Hence the use of ejector system to increase the discharge capacity of the centrifugal pump is restristed to a suction lift less than 3 m. Ejector system is very advantageous to a suction lift upto 1.5 m.

Head capacity characteristics

The head capacity curve is plotted as shown in Fig.19. The graphs are plotted with capacity (total flow) in the abcissa and suction head in the ordinate. Here suction head is taken instead of total head for reasons already explained in Chapter 3.

Table 15. Calculation of discharges of 5 HP-E_2 combination

Suction lift (m)	Delivery pressure (kg/cm^2)	Primary flow			Primary flow through ejector			Total flow (Primary flow through ejector + secondary flow)		
		Qty. of water collected (l)	Time taken (s)	Discharge (lpm)	Qty. of water collected (l)	Time taken (s)	Discharge (lpm)	Qty. of water collected (l)	Time taken (s)	Discharge (lpm)
(1)	(2)	(3)			(4)			(5)		
0.30	7.0	250	23.30	644	250	30.92	485	250	7.40	2026
0.60	5.0	250	23.39	641	250	31.33	479	250	8.47	1771
0.90	3.0	250	23.70	633	250	32.34	464	250	9.28	1616
1.20	2.5	250	24.19	620	250	32.70	459	250	10.28	1459
1.50	2.5	250	25.00	600	250	33.05	454	250	11.13	1348
1.80	2.5	250	25.48	589	250	33.80	444	250	11.65	1287
2.10	2.4	250	25.97	578	250	36.36	413	250	12.70	1181
2.40	2.4	250	27.03	555	250	38.46	390	250	13.86	1083
2.70	2.3	250	28.57	525	250	40.54	370	250	16.15	929
3.00	2.1	250	30.77	488	250	45.28	331	250	30.40	493

Table 16. Capacity ratio, flow ratio and multiplication factor of 5 HP-B_2 combination

Suction lift (m)	Primary flow without ejector (lpm)	Primary flow through ejector (lpm)	Total flow Primary flow through ejector + secondary flow (lpm)	Secondary flow (lpm)	Capacity ratio	Flow ratio	Multiplication Factor
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
0.30	644	485	2026	1541	4.18	3.18	3.15
0.60	641	479	1771	1292	3.70	2.70	2.76
0.90	633	464	1616	1152	3.49	2.49	2.56
1.20	620	459	1459	1000	3.18	2.18	2.35
1.50	600	454	1348	894	2.97	1.97	2.25
1.80	589	444	1287	843	2.90	1.90	2.19
2.10	578	413	1181	768	2.86	1.87	2.05
2.40	555	390	1083	693	2.78	1.78	1.95
2.70	525	370	929	559	2.51	1.51	1.77
3.00	488	331	493	162	1.49	0.49	1.01

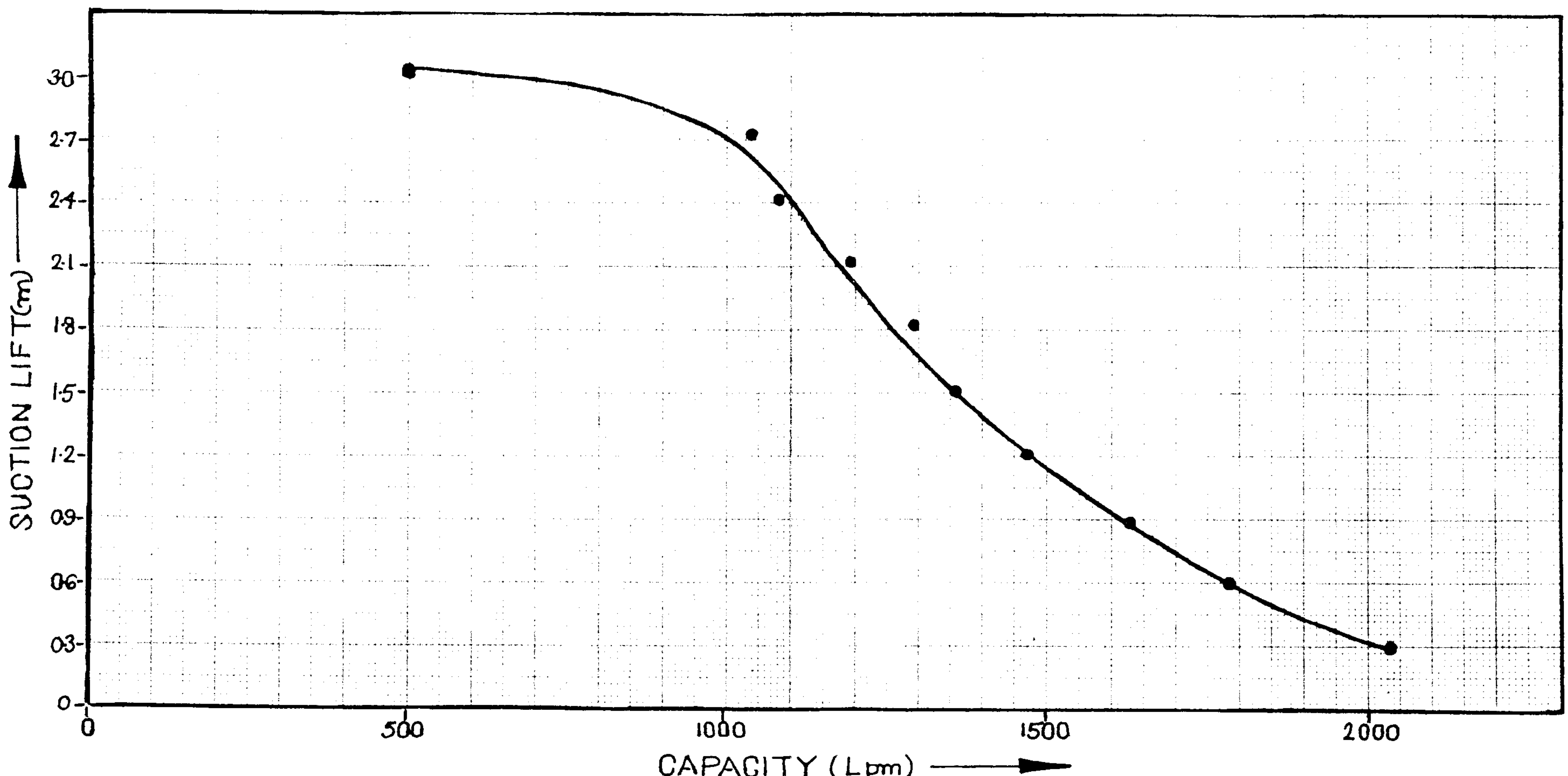


FIG.19. VARIATION OF CAPACITY WITH SUCTION LIFT IN A 5HP-E2 COMBINATION

As suction head increases primary flow decreases, thus causing a reduction in the combined output. At a certain suction head (around 3 m) the suction flow has ceased and the output of the ejector system became the same as the primary flow. The performance of the ejector upto this point may be considered as its useful range as a low lift pumping device. Beyond this point of zero suction flow, as the head increased, the jet pump neither abruptly stopped pumping nor continued to give the constant flow which was derived from the primary pump. Instead the pump kept on discharging at a reduced rate. This meant that the flow from the primary pump divided itself as it entered the ejector system. Part of the flow was delivered at the discharge outlet while the other part flowed down the suction line of the ejector to the tank. The curves resembles the head capacity curve obtained for the experimental jet pump described by Samuel J (1975). It could also be seen that the useful range of the head capacity curve is almost similar to the characteristic curve of a centrifugal pump.

Efficiency characteristics

Efficiency curve is plotted taking capacity ratio in the horizontal axis and efficiency in the vertical axis as shown in Figure 20. The calculation of efficiency using the equation $\eta^{*} = \frac{1.5\ M}{1.5 + M^{*2}}$ is given in the Table 17.

The highest efficiency of 29% is given for a capacity ratio of 2.5 corresponding to a suction lift of 2.7 metres. Upto a suction lift of 3 m i.e. for a capacity ratio of 1.5 the curve shows an abrupt fall. These curves are similar to the efficiency curves obtained for the experimental jet pump described by Samuel (1975).

4.2.2 Experiments using a 10 HP pumpset - E_2 combination.

The ejector E_2 was also tested using a 10 HP prime mover pumpset, by connecting two 5 HP pumpsets in series. The results are tabulated in Table 18. The capacity ratio, flow ratio and multiplication factor are calculated as shown in Table 19. Here a capacity ratio of 4.30 and a multiplication factor of 3.35 are obtained at a suction lift of 0.30 metres. For a suction lift of 1 m, the multiplication factor is about 2.8 and capacity ratio is about 3.6.

The experiment with 10 HP prime mover had to be limited to a suction lift of 2.70 m. As two 5 HP prime movers were connected in series, it was difficult to lift the pumpsets to further heights.

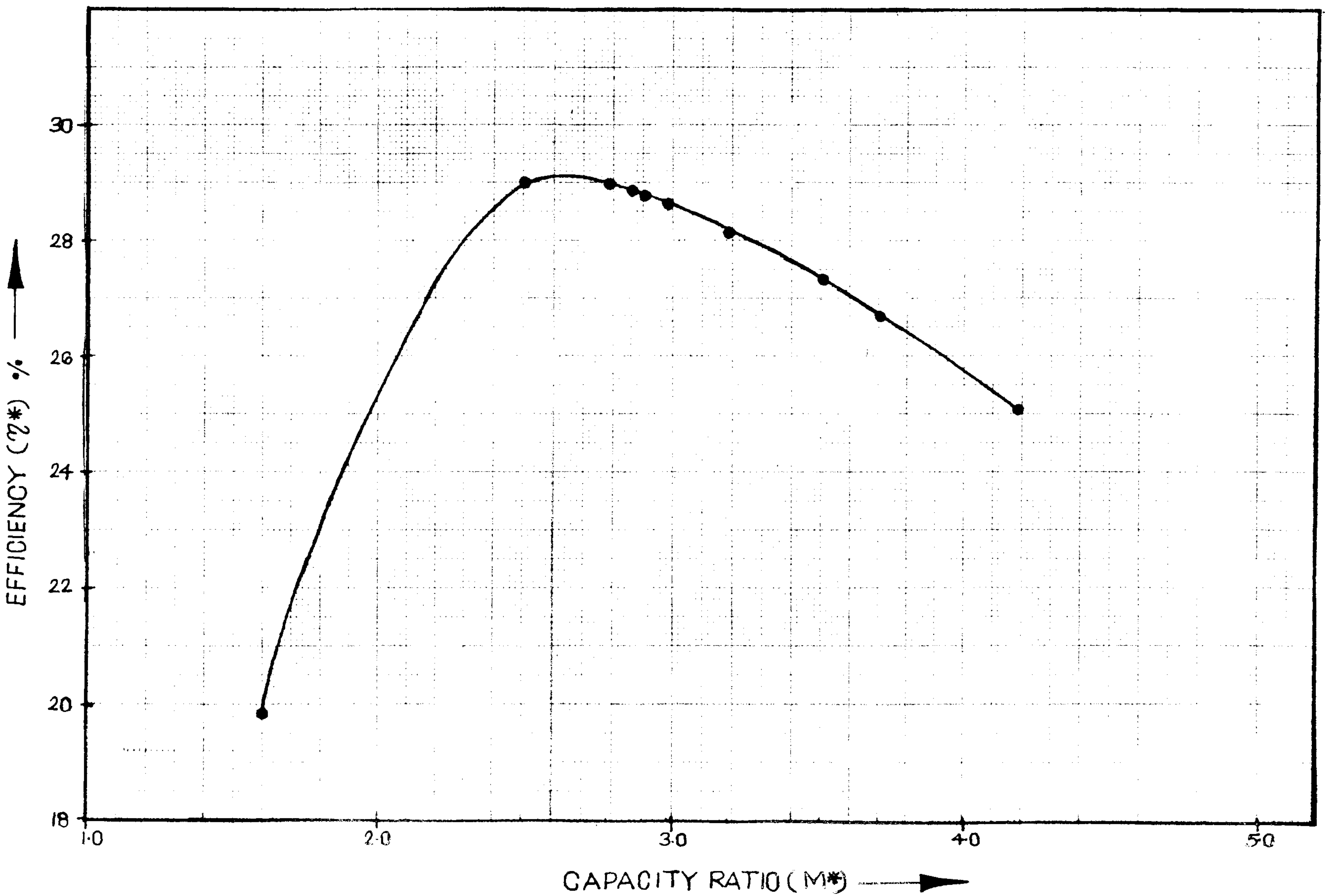


FIG 20. VARIATION OF EFFICIENCY WITH CAPACITY RATIO IN A 5HP. E2 COMBINATION

Table 17. Calculated value of efficiency for 5 HP-E_2 combination

Suction lift (m)	Total flow lpm	Capacity Tatio, M*	Flow ratio M	$\eta = \frac{1.5\ M}{1.5 + M*2}$ %
0.30	2026	4.18	3.18	25.1
0.60	1771	3.70	2.70	26.7
0.90	1616	3.49	2.49	27.3
1.20	1459	3.18	2.18	28.1
1.50	1348	2.97	1.97	28.6
1.80	1287	2.90	1.90	28.7
2.10	1181	2.86	1.87	28.8
2.40	1083	2.78	1.78	28.9
2.70	929	2.51	1.51	29.0
3.00	493	1.49	0.49	19.8

Table 18. Calculation of discharges of 10 HP-E_2 combination

Suction lift (m)	Pressure (kg/cm^2)	Primary flow without ejector			Primary flow through ejector			Total flow (Primary flow through ejector + secondary flow)		
		Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)	Qty.of water collected (l)	Time taken (s)	Discharge (lpm)
(1)	(2)		(3)			(4)			(5)	
0.30	4.7	250	16.0	938	250	20.5	731	250	4.77	3143
0.60	4.5	250	16.4	915	250	21.2	709	250	5.42	2765
0.90	4.5	250	16.8	893	250	21.8	688	250	5.74	2614
1.20	4.4	250	17.4	863	250	22.6	664	250	6.68	2244
1.50	4.2	250	17.6	853	250	23.0	653	250	7.20	2083
1.80	4.2	250	17.9	838	250	23.3	645	250	7.80	1922
2.10	3.8	250	18.4	815	250	24.2	620	250	8.37	1792
2.40	2.4	250	19.4	774	250	26.0	578	250	9.80	1531

Table 19. Capacity ratio, flow ratio and multiplication factor of 10 HP-E_2 combination

Suction lift (m)	Primary flow without ejector (lpm)	Primary flow throgh ejector (lpm)	Total flow (primary flow through ejector + secondary flow)(lpm)	Secondary flow (lpm)	Capa-city ratio, M*	Flow ratio, M	Multiplication factor, F
1	2	3	4	5	6	7	8
0.30	938	731	3143	2412	4.30	3.30	3.35
0.60	915	709	2765	2056	3.90	2.90	3.02
0.90	893	688	2614	1926	3.80	2.80	2.93
1.20	863	664	2244	1580	3.38	2.38	2.60
1.50	853	653	2083	1430	3.19	2.19	2.44
1.80	838	645	1922	1277	2.98	1.98	2.29
2.10	815	620	1792	1172	2.89	1.89	2.20
2.40	774	578	1531	953	2.65	1.65	1.98

The curves showing suction lift Vs capacity ratio and suction lift Vs multiplication factor are shown in Fig.21.

Head - capacity characteristics

The head capacity curves are plotted as shown in Fig.22. It is similar to the head capacity curve obtained for the 5 HP - E_2 combination.

Efficiency characteristics

A curve is plotted with capacity ratio in the horizontal axis and efficiency in the vertical axis as shown in Fig.23. The calculation of efficiency using the equation $\eta^* = \frac{1.5\ M}{1.5 + M*2}$ is given in Table 20.

The efficiency curve is also similar to that of the 5 HP-E_2 combination.

4.3 Advantages of ejector system

The advantages of utilising a centrifugal ejector pump combination can be listed as follows.

1. Attachment of an ejector system increases the discharge capacity of a centrifugal pump at the expense of

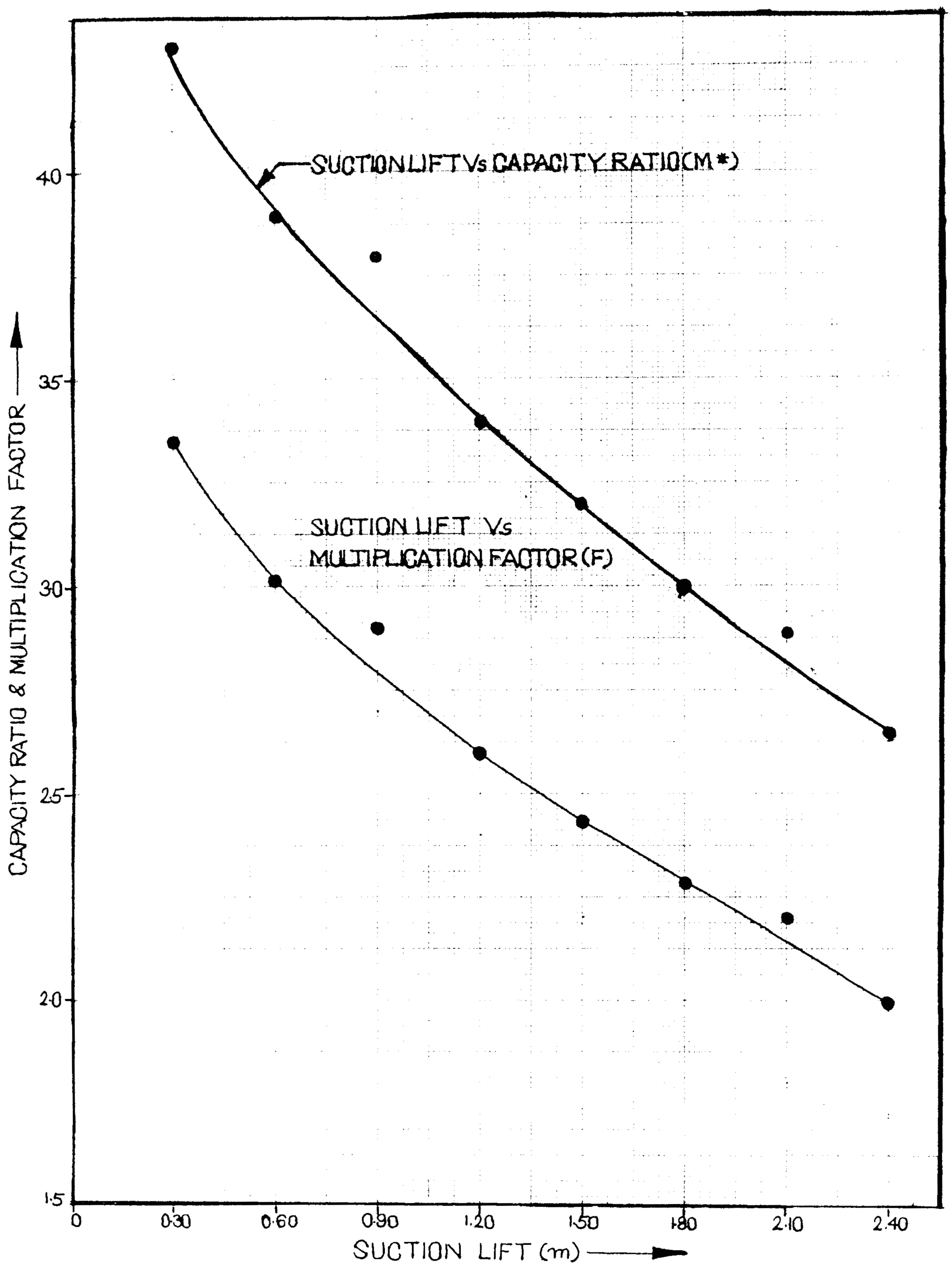


FIG 21. VARIATION OF M* AND F WITH SUCTION LIFT IN A 10 HP-E2 COMBINATION

Table 20. Calculated value of efficiency for 10 HP-E_2 combination

Suction lift (m)	Total flow (lpm)	Capacity ratio M^*	Flow ratio M	$\eta^* = \frac{1.15\ M}{1.5 + M^{*2}}$ %
0.30	3143	4.30	3.30	24.76
0.60	2765	3.90	2.90	26.00
0.90	2614	3.80	2.80	26.30
11.20	2244	3.38	2.38	27.80
1.50	2083	3.19	2.19	28.10
1.80	1922	2.98	1.98	28.60
2.10	1792	2.89	1.89	28.77
2.40	1531	2.65	1.65	29.00

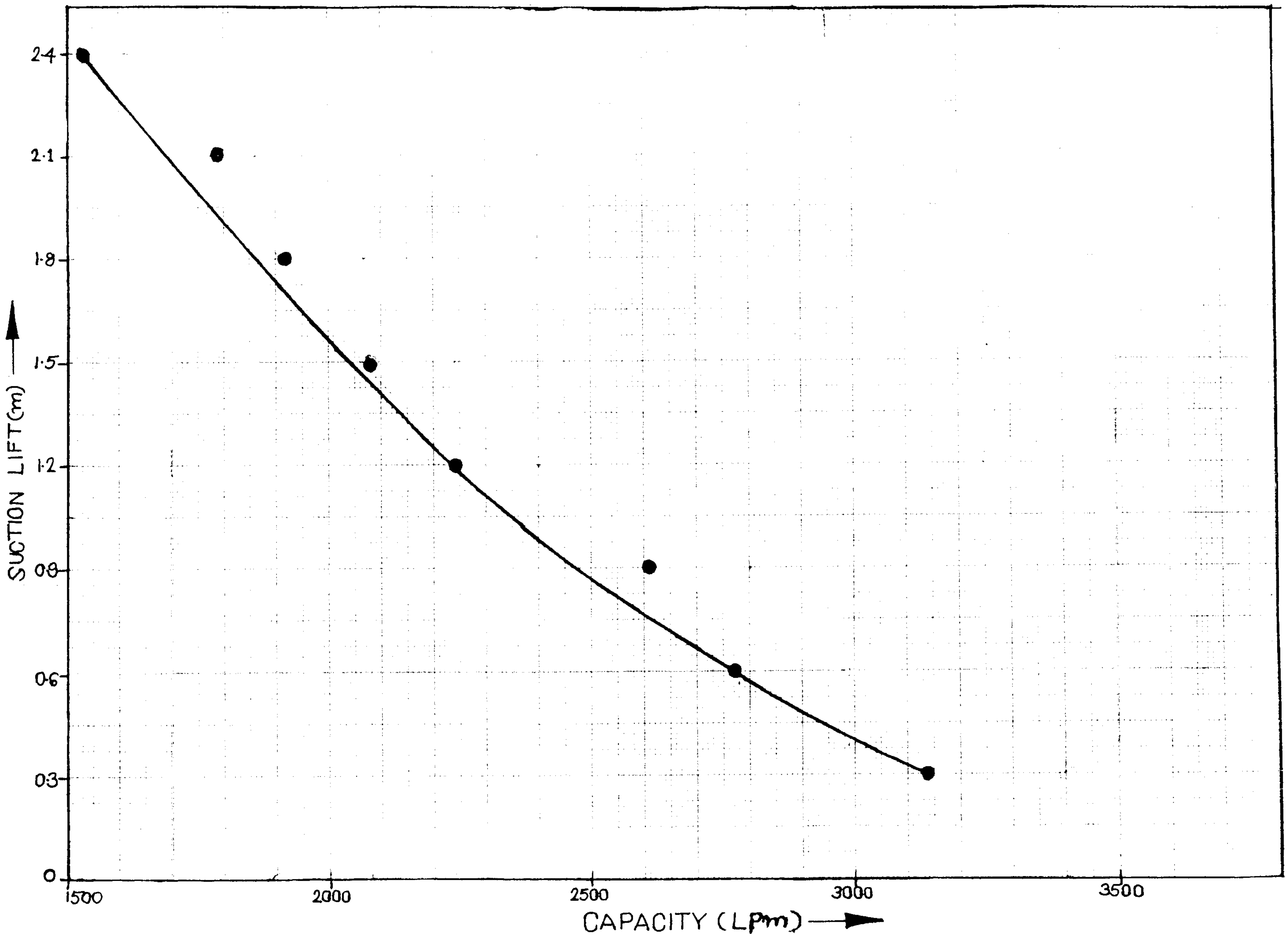


FIG 22. VARIATION OF CAPACITY WITH SUCTION LIFT IN A 10HP-E2 COMBINATION

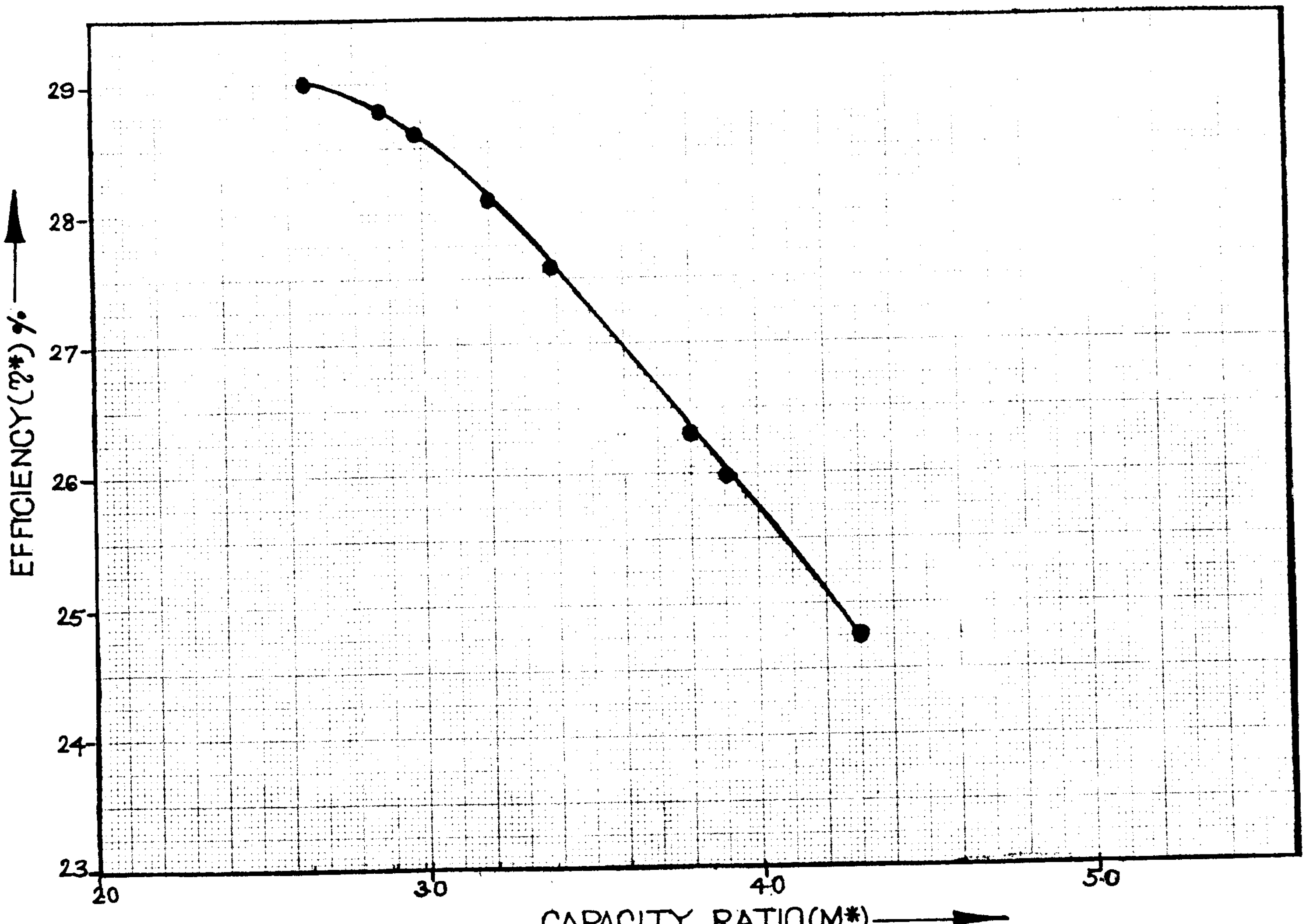


FIG 2.3. VARIATION OF EFFICIENCY WITH CAPACITY RATIO IN A 10 HP-E2 COMBINATION

its pressure head. Thus a jet pump attachment would permit the salvaging of the power loss associated with the use of a vast number of centrifugal pumps now employed in rice production.

2. In situations where there is a large variation in the pumping head requirement, the ejector attachment permits the selection of a high pressure pump as the basic unit. The attachment of an ejector system would enable this pump to work efficiently at low lift conditions also.

3. A centrifugal ejector combination requires much less capital investment than buying two pumps, one for high head conditions and the other for low head conditions.

4. The piping and housing installations required by a centrifugal ejector combination are less than those for a low lift, large volume pump. Also the ejector can be detached when higher heads are required.

5. Since no additional running or maintenance expenditure is involved, the ejector system offers both economic advantages and flexibility for the pumping system.

6. Ejector system has no complicated parts and can be fabricated in an ordinary workshop.

7. Muddy and debris laiden water can be pumped through the ejector system as it has no moving parts.

4.4 Economic analysis

The cost of fabrication of the ejector system and the cost of the accessories are worked out in Appendix IV. The total cost is estimated as Rs. 870/-.

Summary

SUMMARY

Present study was on the development of ejector systems, for increasing the discharges of centrifugal pumps. The pump capability requirement in agriculture, especially in rice production is essentially one of low lift and high capacity. Because of the low lift conditions the full capacity of the centrifugal pump cannot be used. By using an ejector system the centrifugal pump is brought to work under the best efficiency condition during the low lift condition also.

An attempt to increase the capacity of centrifugal pumps by attaching an inexpensive and simple ejector system was done in this project. The main objective was to find out the useful range of suction lifts in which the attachment can be advantageous. A study on the design parameters and efficiency characteristics of the ejector system was also done. Four types of ejector systems were tested to choose the best one out of them based on the capacity ratio and multiplication factor obtained. Capacity ratio was defined as the ratio of total flow to the primary flow through the ejector. Multiplication factor was defined as the ratio of total flow to the primary flow without ejector.

The ejectors were having the following specifications.

E_1 - The secondary flow was straight and the primary flow inclined by 90° to the secondary flow.

E_2 - The primary flow was straight and the secondary flow inclined by 40° to the primary flow.

E_3 - The primary flow was straight and the secondary flow inclined by 20° to the primary flow.

E_4 - The secondary flow was straight and primary flow inclined at 30° to the secondary flow.

The results obtained are summarised below:

1. The preliminary experiments were conducted using a 5 HP prime mover pumpset with all the four ejectors, upto a suction lift namely E_1, E_2, E_3 and E_4 upto a suction lift of 1.5 metre. Out of them ejector E_2 gave the highest capacity ratio of 4.2 and highest multiplication factor of 3.15 at a suction lift of 0.3 metres.

2. For a suction lift of 1 metre E_2 gave a capacity ratio of 3.4 and multiplication factor of 2.6.

3. E_3 was found to yield results nearer to E_2. In

both these cases the primary flow was kept straight and the secondary flow inclined.

4. In E_1 and E_4 there was a change in the direction of primary flow and due to this a loss of primary head was developed. E_4 gave the lowest capacity ratio and multiplication factor.

5. E_2 was found to have many other advantages such as easiness of fabrication and fixing.

6. Due to the addition of ejector system in the delivery side of the centrifugal pump, there was a loss in the primary flow which was about 20 to 30 per cent.

7. For a suction lift of 3 metres, the multiplication factor was 1. Hence the use of the ejector system to increase the discharge capacity of the centrifugal pump was restricted to a suction lift below 3 metres.

8. With a 10 HP prime mover pumpset E_2 gave the highest capacity ratio of 4.3 and multiplication factor of 3.4 at a suction lift of 0.3 metres. For a suction lift of 1 metre the capacity ratio was 3.6 and multiplication factor 2.8.

9. In the head-capacity curves plotted for the 5 HP - E_2 combination, at a certain suction head just above 3 m, the suction flow had ceased and the output of the ejector system became the same as the primary flow.

10. The head-capacity curve resembled those obtained for the experimental jet pump developed by Samuel (1975).

11. The useful range of the head capacity curve is almost similar to the characteristic curve of a centrifugal pump.

12. Capacity ratio Vs efficiency curves in both 5 HP-E_2 combination and 10 HP-E_2 combination were similar to those obtained for the experimental jet pump developed by Samuel (1975).

13. The materials and fabrication cost of the ejector system including the cost of suction and delivery pipes comes to about Rs.870/-.

The advantages of the ejector system can be summarised as follows:

1. The attachment of an ejector system would enable a high head centrifugal pump to work at its best efficiency at low lift conditions also.

2. A centrifugal ejector combination requires less capital investment than buying more than one centrifugal pumps to work at different suction lifts.

3. The ejector system can be detached when higher heads are required.

4. Ejector system has no complicated parts and can be fabricated in ordinary workshops.

5. Muddy and debris laiden water can be pumped through the ejector system as it has no moving parts.

6. Repair and maintenance expenditure are little. Thus ejector system offers both economic advantages and flexibility for the pumping system.

7. The time of pumping can be reduced by the addition of an ejector system, thus saving power either electricity or diesel.

The following are some of the works suggested for further investigation.

1. Further industrial extension and design work should be initiated to locate specific areas in which the device would have immediate application.

2. As the ejector system has no moving parts, the possibility of developing a mud pump based on this principle appears to have good potentials. Further work may be done for exploring this possibility.

3. As the primary flow passes through the ejector system, there is about 20 to 30 per cent loss. Further work may be taken up to reduce this loss.

References

REFERENCES

Addison, H. 1966. Centrifugal and other rotodynamic pumps. Chapman & Hall, London.

*Cairn, J.R. and Na, T.Y. 1969. Optimum design of water jet pumps. Transactions of the ASME. Journal of Engineering for Power, January.

Chandramohan, V. 1984. Development of small scale dredging unit. M.Sc. thesis. Kerala Agric. Univ., Vellanikkara.

Cherhasshy, V.M. 1985. Pumps. Fans. Compressors. Mir Publishers, Moscow. 2nd ed. pp.364

*Cunningham, R.G. 1957. Jet pump theory and performance with fluids of high viscosity. Transactions of the ASME, November (In) Samuel J. 1975. Water jet pumps for low lift applications. Ph.D.thesis, Indian Agric. Res. Inst., New Delhi.

*Gosline, J.A. and O'Brien, M.P. 1934. The water jet pump. Univ. of California publications in Engg. Vol.3 (In) Samuel, J. 1975. Water jet pumps for low lift applications. Ph.D. thesis, Indian Agric. Res.Inst., New Delhi.

Hajilal, M.S. 1987. Design and Development of a high capacity salvinia Harvester. M.Sc.thesis Kerala Agric. Univ., Vellanikkara.

Hansen, H.C. and Kinnavy, R. 1965. The design of water jet pumps - Part I - Experimental determination of optimum design parameters. ASME paper No.65-WA/FE-31 (In) Samuel, J. 1975. Waterjet pumps for low lift applications. Ph.D.thesis, Indian Agric. Res. Inst., New Delhi.

*Kroll, A.E. 1947. The design of jet pumps. Chemical Engineering progress. Vol.1, December (In) Samuel, J. 1975. Waterjet pumps for low lift applications. Ph.D.thesis, Indian Agric. Res. Inst., New Delhi.

Samuel, J. 1967. Studies on water lifting devices. Technical report Agricultural College & Research Institute, Vellayani, August.

Samuel, J., Nichols, F.E. and Roy, S.E. 1974. Design of Contrifugal jet pump combination - A new approach for rice irrigation and drainage. Paper presented at the Annual convention of Philippine Society of Agricultural Engineers, April.

Samuel, J. 1975. Water jet pumps for low lift applications. Ph.D. thesis. Indian Agric. Res. Institute, New Delhi.

Samuel, J. 1980. Development of Salvinia harvesting machine. Paper presented at the seminar held at the College of Agriculture, Vellayani (unpublished).

Sankaranarayanan, M.R. 1981. Mechanical control of floating type aquatic weed. Salvinia molesta (African Payal). M.Sc.thesis, Kerala Agric. Univ., Vellanikkara.

*Silvester, R. 1960. Characteristics and applications of the water jet pump. La Houlle Balance, Aug-Sept.(In) Samuel, J. 1975. Water jet pumps for low lift applications, Ph.D. thesis, Indian Agric. Res. Inst., New Delhi.

Stepanoff, A.J. 1964. Centrifugal and axial flow pumps. Theory, design and application. John Wiley & Sons Inc. 2nd 2d. pp.402-415.

Surya Rao Singamsetti and Vishnukumar Goenka, 1977. Performance characteristics of Jet pumps. Irrigation and power journal. Volume 34. Number 4 pp.487-501.

* Originals not seen

Appendices

Appendix-I

Conventional Centrifugal - Jet pump

For small capacities with lifts up to 40 metres, a special type of pumping unit has been developed which consists of a combination of a centrifugal pump and a jet pump or ejector. The first is mounted next to the motor at the ground surface and furnishes the driving head and capacity for the jet pump placed in the well below the water surface as shown in Fig.24. For shallow wells up to 8 metres the jet pump can be placed on the surface of the ground or built into the centrifugal pump casing. This arrangement provides much mechanical advantage, as there are no moving parts in the well, and the centrifugal pump, with its motor, can be placed at some convenient point. The hydraulic advantages are: steep head-capacity characteristics with operating head about 50 per cent higher than that of the centrifugal pump alone and a brake hores power curve which is non-overloading. The peak efficiency of the combination is equal to or better then that of the jet pump but is lower than that of centrifugal or vertical turbine pumps. However, at the operating capacity the efficiency is equal to or better than that of the centrifugal pump at the same capacity. In small sizes, this type of pumping unit is widely used for the domestic water supply.

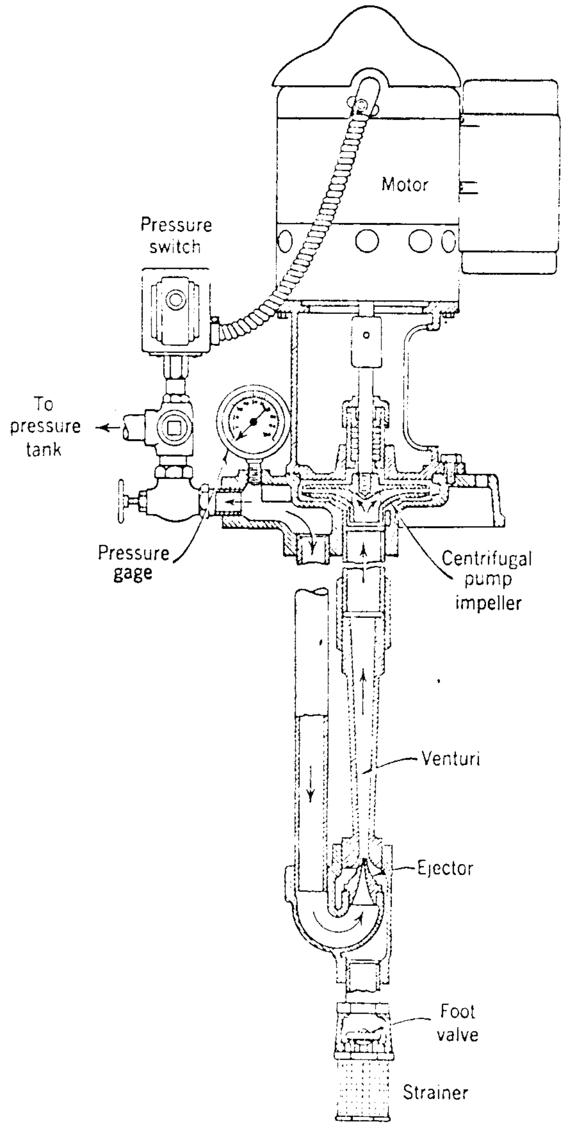


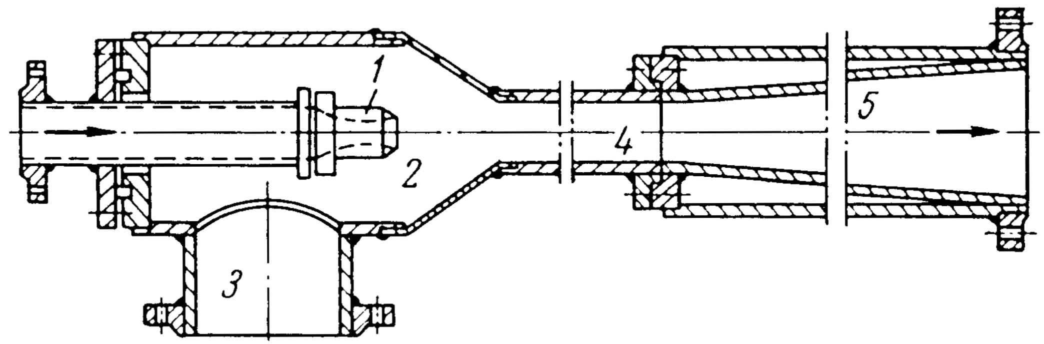
FIG24 CONVENTIONAL CENTRIFUGAL-JET PUMP USED FOR DOMESTIC WATER SUPPLY

Appendix-II

STEAM-JET COMPRESSOR

The basic design of a jet compressor built for industrial applications is presented in Figure 25.

A motive fluid (water, steam, gas) issues at a high velocity from nozzle 1 into suction chamber 2, where it makes contact with a suction liquid coming through pipe 3. Due to friction and exchange of momentum at the surface of the motive fluid jet, the latter entrains the suction liquid and moves to it mixing chamber 4 and then to conical diffuser 5. The mixing chamber is the place where an exchange of momentum between the motive fluid and the suction liquid takes place; the diffuser converts the kinetic into the potential energy. From the diffuser the liquid flows into the discharge line.



1 NOZZLE

2 SUCTION CHAMBER

3 PIPE

4 MIXING CHAMBER

5 CONICAL DIFFUSER

FIG.25. STEAM-JET COMPRESSOR

Appendix-III

Specification of 5 HP pumpset

a. Greaves Lombardini Diesel Engine

Series	-	500
Type	-	520
Bore	-	78 mm
Stroke	-	68 mm
Displacement	-	228 cm^2
Compression ratio	-	18:1
Speed	-	3600 rpm
HP-I. S-1601	-	5
Specific fuel consumption	-	220 g/hp/hr
Lubricating oil consumption	-	13 gm/hr
Capacity of oil sump	-	1 litre
Dry weight	-	38 kg

b. TEXMO centrifugal pumpset

Size	-	2 x 2
Type	-	TL5
Head	-	30 - 110
HP	-	5
Speed	-	3600 rpm
S.No	-	12959-80

Appendix-IV

COST ANALYSIS

Particulars of the item	Quantity or number	Rate Rs.Ps.	Amount Rs. Ps.
110 mm flexible pipe for secondary suction	2.5 m	96.00 per m	240.00
110 cm flexible pipe for secondary delivery	0.5 m	96.00 per m	48.00
110 mm hose connector	2 Nos.	24.00 per No.	48.00
110 mm clamp	2 Nos.	4.00 per No.	8.00
110 mm GI coupling	1 No	36.00 per No.	36.00
Fabrication cost of nozzle, mixing chamber, diffuser and secondary foot valve including cost of materials			450.00
Overhead charges			40.00
Total			Rs. 870.00

DEVELOPMENT OF EJECTOR SYSTEMS FOR INCREASING THE DISCHARGES OF CENTRIFUGAL PUMPS

By

LATHA A. KOSHY

ABSTRACT OF A THESIS

Submitted in partial fulfilment of the requirement for the degree

Master of Science in Agricultural Engineering

Faculty of Agricultural Engineering
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Kelappaji College of Agricultural Engineering and Technology
Tavanur - Malappuram

1988

ABSTRACT

The pump capability requirement in agriculture especially in rice production is essentially one of low lift and high capacity. Because of the low lift conditions the full capacity of the centrifugal pump cannot be used. By attaching an ejector system, the centrifugal pump is brought to work under the best efficiency condition during low lifts also.

Four types of ejector systems were developed and the best one was chosen, based on the capacity ratio and multiplication factor. Experiments were conducted to find out the useful range of suction lifts in which the attachment can be advantageous. Design parameters and efficiency characteristics also were studied.

The study revealed that the ejector E_2 with primary flow straight and secondary flow inclined at 40° gave higher capacity ratio and multiplication factor than the other three ejectors. Hence E_2 was selected for further experiments. For a suction lift of one metre E_2 gave a capacity ratio of 3.4 and multiplication factor of 2.6 for the 5 HP centrifugal pumpset. With a 10 HP centrifugal

pumpset E_2 gave a capacity ratio of 3.6 and multiplication factor 2.8, for a suction lift of one metre. E_2 is also having many other advantages than the other three ejector such as easiness of fabrication and fixing.

A centrifugal ejector combination requires less capital investment than buying several pumps for use at different heads. Muddy and debris laiden water can be pumped through the ejector system as it has no moving parts. Time of pumping can be reduced by the addition of an ejector system, thus saving power, either electricity or diesel. It can be fabricated in a local workshop and transportation is also very easy.

The estimated cost of fabrication of the ejector system is Rs.870/-.