TESTING AND MODIFICATION OF AN AXIAL FLOW IRRIGATION PUMP MANUFACTURED IN VIETNAM

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ABSTRACT

The performance of a commonly used, inclined shaft, axial flow pump manufactured in Vietnam was evaluated. The pump tested had a 37 cm diameter thrust impeller and 40 cm outlet diameter. This pump was initially evaluated to establish the base performance curves for three total static heads of 1.45 m, 1.75 m and 2.15 m at a constant recommended speed of 980 rpm. In the field survey, it was found that the problems of the pump were concerned with the pump shaft, brass sleeve, impeller and lubricating system. These parts of the pump were modified and then it was tested again at the same test conditions used for the original one.

Maximum efficiency of the original pump varied from 56.11% to 53.15%, and that of the modified pump from 57.63% to 54.52% when the total static head varied from 1.45 m to 2.15 m. At these total static heads, the discharge, the total head and the power input varied from 387 to 347 l/s, 4.25 to 4.60 m and 28.72 to 29.38 kW, respectively, for the original pump and from 388 to 346 l/s, 4.29 to 4.63 m and 28.23 to 28.91 kW, respectively, for the modified pump. The efficiency of the pump after modification increased by more than 1.5% and the power input decreased by 1.7%.

INTRODUCTION

There are many types of axial flow pumps, manufactured by both government factories as well as local industries, available in Vietnam. But most of these pumps have not been tested adequately and have very short operating lives and low efficiency. The users are also not aware of the actual performance of these pumps, and so they are unable to choose a pump suitable to their requirements. Therefore, it was necessary to study the performance of such a pump in Vietnam.

According to Fraenkel (1986), the choice of pump for irrigation is based on life cycle costs, unit output costs, status or availability of the technology, operational convenience, skill requirements for installation and maintenance, durability, reliability, useful life and potential of the local manufacturers. Based on this fact a study was carried out to evaluate the performance of the locally-made Vietnamese pump to identify the problems of the present design and to suggest a better design which would be useful for the manufacturers as well as the users. Since inclined shaft axial flow pumps are being widely used in Vietnam, these pumps were selected for the performance evaluation study (Khanh, 1995). A field survey at 12 pumping stations in Nam Dinh and Vinh Phu provinces was carried out to find out the problems of the pump operation. A commonly-used axial flow pump was used for this study.

MATERIAL AND METHODS

Technical specifications of the original pump

During the preliminary field survey, the pump with specifications given in the Table 1 was found to be most widely-used in the Red and Mekong river deltas of Vietnam, where there is rather a low water head. This pump was used for testing (fig. 1).

Setup for testing

The experiments were conducted at the National Research Institute of Water Resources, Hanoi, Vietnam. The impeller was of the thrust axial flow type and the length of the pump shaft was 8 m. The suction source was a pond with depth 2.5 m, length 6.9 m and width 2.7 m. The discharge pond was a channel with the length 40 m and width 2 m. A wave damping wall and a sharp crest cipolleti weir were placed across the channel. The water head in the discharge pond against the elevation of the weir was measured by a meter placed in a hole adjacent to the pond.

Installation and testing of the pump

The pump was driven by a 55 kW electric motor coupled to the pump shaft by a cacdan coupling (Fig. 1). Three piezometric openings of 5 mm diameter were fitted on the discharge pipe for the measurement of discharge head through a pressure gauge. A gate valve was fitted on the discharge pipe (1.4 m after the piezometric openings) to vary the flow rate. At each flow rate, discharge head, flow rate, power input, shaft torque and rotational speed were measured and recorded. The pump was tested at three different static heads of 1.45, 1.75 and 2.15 m with constant rotational speed of 980 rpm as recommended by the manufacturer. Five replications were made for each set of conditions. The pump was installed on an inlined frame with inclining discharge nozzle.

Measurements

The flow was measured in the open channel with the sharp crest cipolleti weir built across it. The water flow was delivered back to the suction pond to facilitate recirculation. The pump was run for a fixed duration (30 minute) at each set of operating conditions before recording the data. The discharge was computed by using the standard equation (JSA, 1978).

For discharge head measurement of the pump, a pressure gauge connected to the three piezometric openings (fitted on the discharge pipe) through a steel tube with four branches was used. During each set of operating conditions, the corresponding variation of pressure indicated on the gauge was recorded along with the total static head. Discharge head was obtained by using the standard formula (Anon, 1972).

The speed of the pump under test was determined by a tachometer attached to the shaft connecting electric motor with the pump.

A voltage meter, ampere meter and cosφ meter were used to measure the power input to the motor. The motor and these meters were calibrated. Four torque sensors attached to the driven shaft were connected to a torque transducer that was fitted on the shaft connecting motor with the pump for measurement of power input to the pump. The torque transducer was connected to a data processor through an amplifier. The results were processed and shown in the form of voltage output.

The efficiency of the pump was calculated using the following formula:

$$E_p = (P_{ou} / P_s) * 100 \quad (1)$$

$$E_m = (P_s / P_{in}) * 100 \quad (2)$$

$$E_t = E_p * E_m \quad (3)$$

where E_p = pump efficiency, %

E_m = motor efficiency, %

E_t = total efficiency, %

P_{ou} = pump output power, kW

P_s = power input to pump, kW

P_{in} = power input to electric motor, kW

RESULTS AND DISCUSSION

Performance of the Original Pump

Initially, the pump was tested for pumping water at three static heads, i.e. 1.45, 1.75 and 2.15 m, to establish the performance base curves. Experimental results obtained at (the maximum efficiency point) different operating conditions are given in Table 2. Performance of this pump showed that for 1.45 m total static head, the pump had maximum efficiency of 56.11% with 387 l/s discharge at a total head of 4.25 m and power input of 28.72 kW.

When compared with design parameters, the actual performance of the pump was too low. a) Testing parameters : E = 56.11%, H = 4.25 m and Q = 387 l/s
b) Design parameters : E = 72%, H = 5 m and Q = 416 l/s
The actual efficiency of the pump was 16% less, at 0.75 m less head and 29 l/s less discharge.

Experimental results showed that when total static head increased from 1.45 m to 2.15 m, discharge decreased by 10.4% (from 387 l/s to 347 l/s), total head increased by 8.2% (from 4.25 m to 4.59 m), efficiency decreased by 5.3% (from 56.11% to 53.15%), and power input increased by 2.3% (from 28.72 to 29.38 kW), when all other operating conditions were the same. These results showed that the performance of the pump was significantly affected by change in the static head.

General observations on the original pump during and after testing

The pump used for testing was quite a new one and was tested for about 120 hours of working with different operating conditions. During and after testing the pump, the following observations were made:

a) At the end of the testing period (about 25 days from the beginning), there were some leakages at the stuffing box and at the pump casing joint.
b) Some smoke was observed at the upper bearing three hours after the start of the testing for the lowest level of discharge.
c) The components such as brass sleeve, sleeve cup, pump shaft were worn out severely leading to misalignment of the shaft. Pump shaft diameter measured at the point contacting the brass sleeve was worn out by 0.5 mm (measured - 54.5 mm, designed - 55 mm).
d) Thickness of the impeller vanes at their edge was much higher (2 mm) than its design which decreased the pump efficiency. The gap between the impeller and the seal was too large (4 mm) when compared with the design (0.5 mm). A flaw of 10 cm^2 was observed at the end of one impeller vane.
e) The bottom end of the pump shaft was blocked by a flinger fitted to the guide vanes by 4 bolts of 8 mm diameter. Under centripetal forces these bolt joints loosened and as a result it made the impeller to strike the seal. This resulted in high noise and vibration at the end of testing period (the gap between impeller and water seal at one side measured only 0.2 mm. The edge of impeller was worn out by 3 mm (at maximum point).
f) Water ring seal components functioned very poorly.
g) The 8 bolts used to connect electric motor and driven shaft were not tight enough, the coupling plate was damaged.
h) Suction belt was highly pock - marked.

These indicated both the poor mechanical and technical facilities of the local industries as well as the quality of the pump components.

Modifications of original pump

Based on the field survey results and observations during the testing, the original pump was modified with the aims of i) increasing the efficiency, ii) increasing the service life and iii) making the operational requirement simple. Under these guidelines the following parts of the pump were modified : pump shaft, brass sleeve, lubricating system and the impeller (Fig. 2).

The *shaft* of the original pump was replaced by a shaft of three segments connected together by cacdan couplings. The first segment was placed within guide vanes and diffuser vanes. The impeller was attached to one end of this shaft and hence it was called the "pump shaft". The intermediate shaft connecting the pump shaft with the driven shaft by a cacdan coupling was entirely placed in a tube along the casing. This restricts corrosion of the shaft by water. The third segment was a driven shaft connected to electric motor with the intermediate shaft by cacdan couplings. This shaft can be changed at any time as per the requirement of the operating conditions.

The *brass sleeve* was replaced by two ball bearings (No. 7313), one each to withstand axial force and concentric force. These bearings were placed at the lower end of the pump shaft which were held in the bearing cup placed inside the guide vanes. There were two other bearings (No. 6312) at the upper end of the pump shaft for orienting the center of the pump shaft.

The designed thickness of the *impeller* vanes was 0.8 mm at the outlet edge and 2.5 mm at the inlet edge. After casting, the thickness of the vanes at outlet and inlet were 2.0 and 4.2 mm, respectively. The impeller was put in use without any further processing. This led to a considerable increase in mechanical losses. For the impeller of the modified pump, these thicknesses were reduced to 1.2 mm at the outlet edge and 2.8 mm at the inlet edge by hand polishing machine. The surface of the impeller vanes was polished rather well before assembling.

In the modified *lubricating system,* a case was created by the pump shaft and guide vanes which was filled with oil either during assembling or during periodic servicing. This made lubricating simpler and led to increased operating life of the pump.

Performance of Modified Pump

After modifications, the pump was tested again for pumping water at similar conditions as that of the original pump. Experimental results at maximum efficiency point at different operating conditions are given in Table 3. Selected performance curves of the pump at 1.75 m total static head are shown in Fig. 3.

As the total static head increased from 1.45 m to 2.15 m, the efficiency decreased from 57.63 to 54.52%, the discharge decreased by 10.4% (from 388 to 346 l/s), total head increased by 8.1% (4.29 to 4.63 m) and the power input increased by 2.4% (from 28.23 to 29.91 kW), respectively.

Comparison of performance of the Original and Modified Pump

The above experimental results showed that the performance of the modified pump was improved when compared to the original one. The efficiency of the pump after modification increased by 1.3 - 1.6%. There was no significant change in discharge (0.8%), but the total head improved particularly at 2.15 m total static head. Decrease in power input was by 1.7 % for different total static heads.

Advantages and Disadvantages of Modified Pump Over the Original one

a) As the pump shaft was divided into three separate segments and connected with each other by flexible joints, assembling and detaching the pump for maintenance or transportation become much easier. For maintenance or to change the impeller or other parts of the pump, it is necessary to detach just 12 bolts of 16 mm diameter connecting the diffuser vane section with the pump casing. Moreover, flexible coupling can restrict the lateral misalignment of the shaft.
b) The shaft was completely protected from water corrosion in the new design.
c) The lubricating system was changed to an one-time lubricating system. This restricts daily lubrication requirement, especially in a condition where the pumping stations run out of lubricant frequently.
d) Normally bearing material has higher resistance than brass. In the same circumstances of lubrication and operation, bearings will wear out

considerably less than brass sleeves. Hence, the service life of the pump with bearing will be longer.

e) The requirement of water tightness of the stuffing box has been reduced.

The modified pump has the following disadvantages that must be further investigated:

a) Since the impeller was made from cast iron, it was difficult to make it with the required thickness, especially with technology currently available in Vietnam. This limits the pump performance and damages the impeller easily.

b) The shaft was placed in a tube, which created certain difficulties for fabrication of pump casing, guide vanes and diffuser vanes. Moreover, the joints of these parts need to be very water tight.

c) The pump shaft was connected to the electric motor by cacdan couplings and it required more time for assembling and dissembling.

Economical analysis

With the same total static head (1.45 m), total head and discharge at the maximum efficiency points, the modified pump consumes 0.483 kW less power than the original one. If the pump works for 20 hours per day and 25 days per month, then with the existing cost of electricity of US$0.05/kwh, the total saving per year would be US$144.9. This savings is many times higher than the cost of the pump itself. This figure shows that even with a little improvement in design, a considerable amount of money could be saved.

CONCLUSIONS

1. When the total static head varied from 1.45 to 2.15 m, the peak efficiency varied from 56.11 to 53.15% and 57.63 to 54.91% for the original and the modified pump respectively. The efficiency of both pumps was affected significantly by the total static head.

2. The efficiency of the modified pump was increased by 1.6% at the maximum efficiency point and it can be concluded that the modifications were effective.

3. When the total static head increased from 1.45 to 2.15 m, the total head developed by modified pump varied from 4.29 to 4.63 m and the discharge from 388 to 346 l/s whereas for the original pump these values were 4.25 to 4.59 m and 387 to 347 l/s, respectively.

4. Economic analysis suggested that the improved pump design should bring substantial savings to farmers and the pump life cycle will be longer than the original pump.

REFERENCES

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TABLES

Table 1. Technical specifications of the original pump

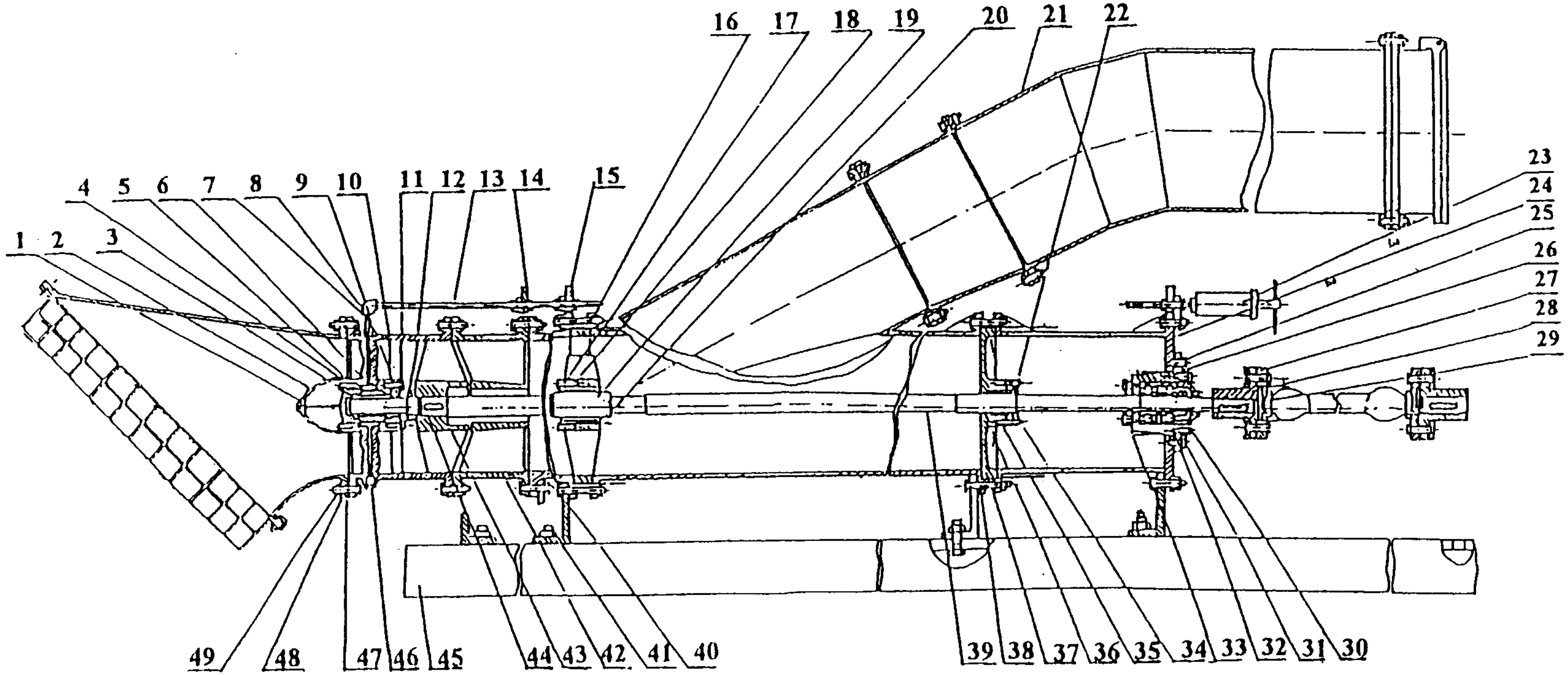
Characteristics	Specifications
Type	Inclined shaft axial flow
Power Source	Electric motor of 55 kW
Rotational speed, N	980 rpm
Discharge, Q	458 l/s
Head, H	5 m
Diameter of impeller, D	370 mm
Diameter of discharge pipe, d	400 mm
Number of vanes, Z	3

Table 2: Performance of original pump at the different operating conditions (at the maximum efficiency point)

No	Total static head (m)	Total head (m)	Discharge rate (l/s)	Power input (kW)	Efficiency (%)
1	1.45	4.25	387	28.72	56.11
2	1.75	4.35	372	28.99	54.81
3	2.15	4.59	347	29.38	53.15

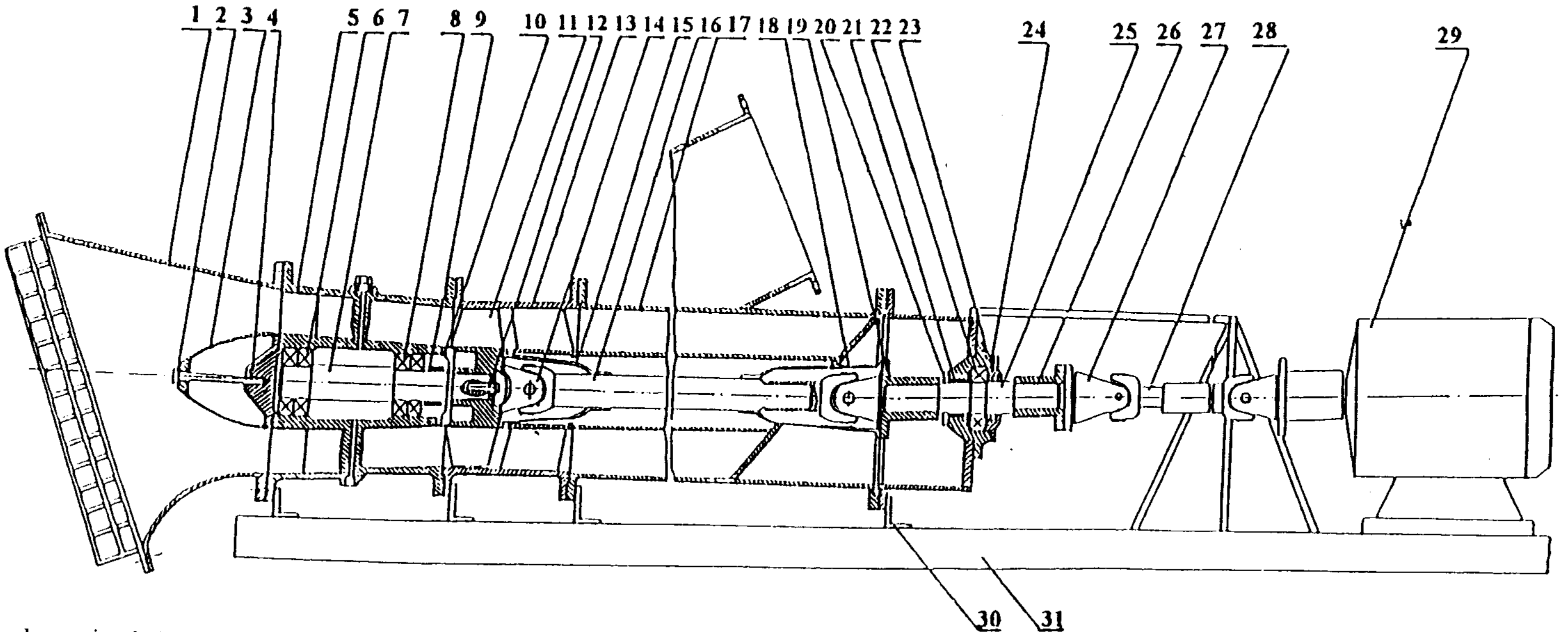
Table 3. Performance of modified pump at the different operating conditions (at the maximum efficiency point)

No	Total static head (m)	Total head (m)	Discharge rate (l/s)	Power input (kW)	Efficiency (%)
1	1.45	4.29	388	28.23	57.63
2	1.75	4.39	375	28.67	56.40
3	2.15	4.63	346	28.91	54.52



1 - bolt at shaft end
2 - guide plate
3 - flinger
4 - brass sleeve
5 - bolt 12
6 - brass sleeve
7 - joint pipe
8 - short oil pipe
9 - ring flinger
10 - oil flinger
11 - bolt 12
12 - arbor nut
13 - long oil pipe
14 - bolt and gland nut 10
15 - oil pipe support
16 - casing link pipe
17 - gland bolt 12
18 - shaft sleeve
19 - shaft link pipe
20 - shaft support
21 - bend discharge pipe
22 - bolt 12
23 - lubricating system
24 - casing joint pipe
25 - plate
26 - stuffing box
27 - coupling
28 - gland follower
29 - key
30 - arbor bolt
31 - bearing (No 7313)
32 - packing
33 - oil follower
34 - oil seal
35 - gland stud
36 - shaft 2
37 - pump support joint
38 - bolt
39 - shaft 1
40 - bearing stand
41 - bolt
42 - diffuser vanes
43 - impeller
44 - key
45 - pump support
46 - arbor bolt
47 - suction bell
48 - paper ring
49 - bolt $\phi 16$

Fig. 1 Original Pump



1- suction bell
2- bolt at shaft end
3- guide plate
4- flinger
5- guide vanes
6- inlet vane bearing (No 7313)
7- impeller shaft
8- upper inlet vane bearing
9 - packing
10- bearing cap
11- impeller
12- shaft end bolt 60x40
13 diffuser vanes
14- cacdan coupling
15- coupling sleeve brushing
16- intermediate shaft
17- pump casing
18- coupling sleeve brushing
19- shaft coupling
20- packing cap
21- packing
22- bearing (No 6313)
23- bearing cap
24- bearing cup
25- arbor shaft
26- shaft coupling
27- cacdan coupling
28- power driven shaft
29- electri motor
30- bearing stand
31- pump support

Fig. 2 · Modified Pump

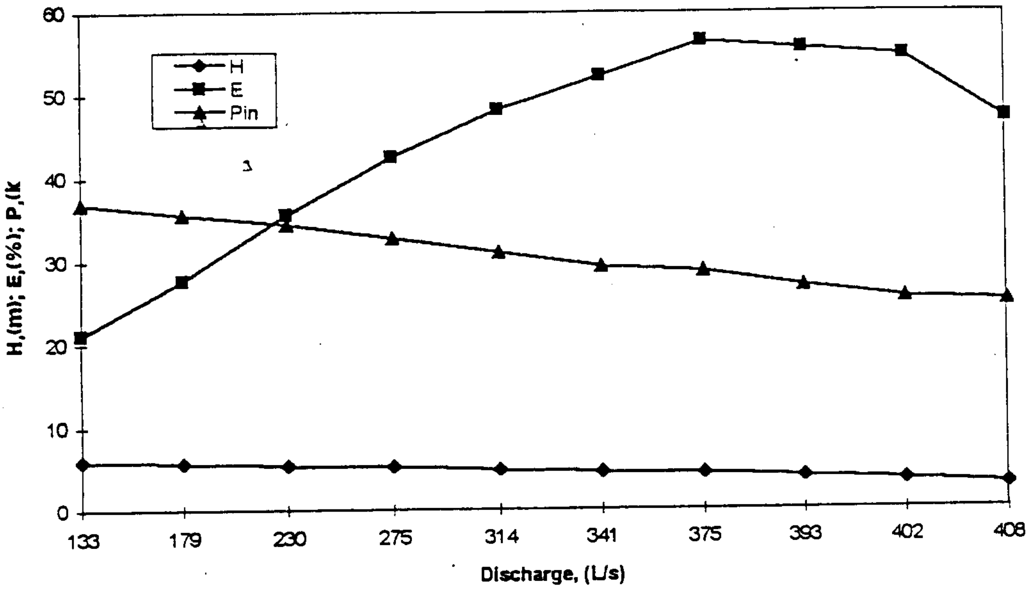


Fig. 3· Performance curves of modified pump at 1.75 m TSH
H - Pump Head; E - Pump efficiency; P_{in} - Power input