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DESIGN PARAMETERS AFFECTING THE PERFORMANCE OF THE IRRI-DESIGNED AXIAL-FLOW PUMP

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DESIGN PARAMETERS AFFECTING THE PERFORMANCE OF THE IRRI-DESIGNED AXIAL-FLOW PUMP[1]

ABSTRACT

Among modern pumps, the axial-flow pump comes nearest to meeting the low lift, high capacity requirements of most rice-producing areas. Improving the operating characteristics of this type of pump requires optimal flow on both sides (suction and discharge) of the impeller in addition to correct impeller design. This paper describes the effects of the more important design parameters on pump performance.

Results show that pump efficiency depends greatly on the shape of the inlet. The impeller vane discharge angle or pitch relates directly to the head-capacity characteristics. The larger the clearance setting between the impeller and the stator casing, the greater the loss in capacity due to leakage. The diffusion vane angle is not a critical design parameter.

The two-stage pump was compared with the single-stage pump. At constant capacity the two-stage pump has double the head and power input of the single-stage pump but operates at the same efficiency as the one-stage pump. The optimum spacing between the impellers of the two-stage pump is 150 mm. The two-stage pump is more economical at pumping lifts above the operating head of the single-stage pump, ie, higher than 2.5 m.

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DESIGN PARAMETERS AFFECTING THE PERFORMANCE OF THE IRRI-DESIGNED AXIAL-FLOW PUMP

Finding the optimum flow conditions is the objective of every pump designer. To design an axial-flow pump requires experimental data. Karassik and Carter (1) stated that pump output depends on the angle and length of the impeller vanes as well as on the number of vanes. Hence, specifying the characteristics of the axial-flow impeller requires detailed information on the impeller and other pump parts.

In fine-tuning the design of the IRRI axial-flow pump, IRRI engineers realized that finding the optimum flow conditions on the suction and discharge sides of the impeller was as important as having correct impeller design. Improving the pump's efficiency or operating characteristics requires considering many variables. Each design parameter can affect the pump's output and several performance curves can be developed from each combination of these parameters. Among the design parameters needing further study are the inlet shape, the impeller vane discharge angle or pitch, the diffusion vane angle, and the clearance between the impeller and the stator casing. Knowing the effects of these parameters would provide pump designers with the necessary data to design for optimum hydraulic performance.

The axial-flow pump operates at a greatly reduced capacity when the operating head is above the design point and may overload the prime mover due to the increase in power input. To overcome overloading, Linsley and Franzini (3) suggested that the head may be distributed among pumps in series or by a multistage unit. A two-stage pump would provide a higher pumping lift.

This study examined the effects of these parameters on the performance of the IRRI-designed axial-flow pump. A two-stage unit was designed and compared with the single-stage pump. A cost comparison determined which pump is more economical at different pumping lifts.

DESIGN AND DEVELOPMENT OF THE IRRI AXIAL-FLOW PUMP

In 1978, the IRRI Agricultural Engineering Department began to develop an efficient, low-cost, and portable axial-flow pump that can lift water 1-4 m from lakes, rivers, or irrigation canals, and can be driven by an internal combustion engine or electric motor. The objective was to improve the efficiency and simplify the design of a Thai-designed axial-flow pump, to reduce its cost and enable fabrication by rural workshops (4). The Thai axial-flow pump used a boat propeller and shaft encased in a long tube, coupled with an engine. The propeller rotation sucked water in and through the tube.

The first prototype used a bamboo discharge tube, 4.2 m long and 12 cm in diameter, a steel impeller coupled to a lineshaft of a 1.9-cm (0.75 inches nominal) pipe, a bamboo stator casing, and a 180° steel elbow. The 180° steel elbow connects the bamboo stator casing and discharge tube. The lineshaft was supported by clamp-type wooden bearing holders parallel to the bamboo tube. The pump was driven by a 5-hp (3.73 kW) gasoline engine through a flexible coupling. Preliminary tests showed the pump could pump up to 1,350 litres water/min at 1.5-m lift.

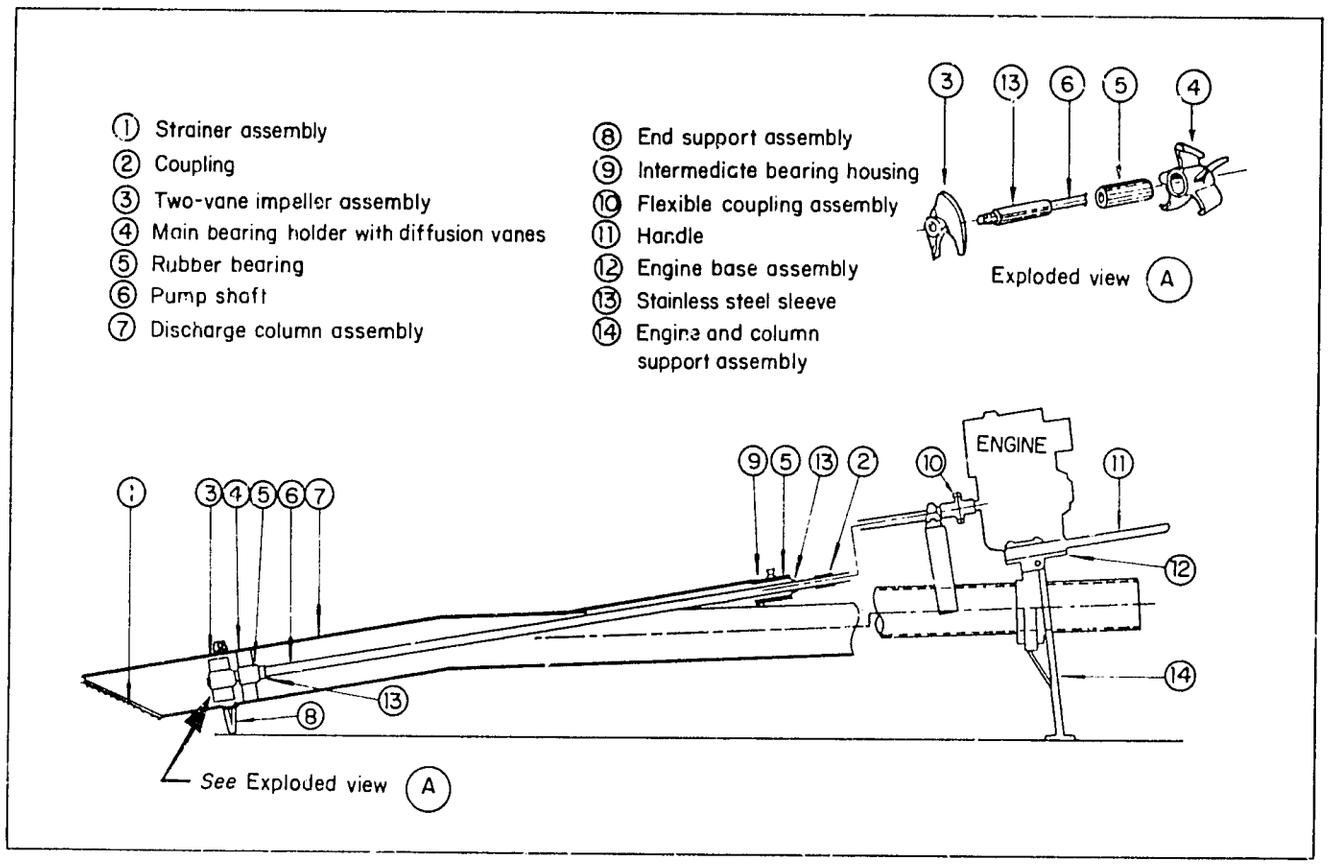
The pump performed satisfactorily during a 100-h test but the bamboo tube began cracking and deflecting and bearings wore excessively. This led to the second prototype, using a steel discharge tube and redesigned wooden bearings. Inlet guide vanes and diffusion vanes were incorporated to increase efficiency, and an oblique inlet was added to reduce inlet head loss. The highest efficiency was 40% at an output of 1,900 litres/min, a speed of 2,330 rpm, and total head of 1.8 m.

After the performance test on the second prototype, a value analysis team began work to reduce the pump cost and improve its versatility. They eliminated the inlet guide vanes, added an engine stand that doubles as a handle during transport, and added a belt-drive attachment for using a power tiller as power source.

More performance tests evaluated the effects of removing the inlet guide vanes. The absence of inlet guide vanes induced water prerotation in the direction of the impeller rotation, which unloaded the impeller and reduced capacity and brake horsepower. However, this did not appreciably change pump efficiency.

To increase the capacity and efficiency, the impeller vane pitch was changed from 15° to 20°; efficiency increased 10%. Field performance and life tests showed a 5-hp (3.73 kW) gasoline engine could drive the 15-cm-diameter pump. Drawings of the pump were then released to manufacturers in late 1979.

The final design (Fig. 1) consists of an axial-flow impeller secured to a pump shaft and encased inside a 15-cm-diameter steel discharge tube. The pump is supported by an engine stand at the discharge end, and a support near the pump inlet. A flexible coupling connects the engine and the pump shaft (alternately, the pump may be driven by a separate engine such as power tiller, using V-belts and pulleys). The pump inlet area is widened to reduce suction losses and vortex formation through a 30° cut across the



1. Schematic drawing of axial-flow propeller pump, 1979.

tube axis. A wire mesh inlet strainer prevents entry of foreign materials. Behind the axial-flow impeller, a set of diffusion vanes straightens the spiral flow from the impeller for improved efficiency, and holds the primary rubber bearings. To simplify replacing the intermediate bearing, the discharge tube is bent at approximately 70 cm from the inlet. This allows the pump shaft to exit from the discharge tube so the bearing can be located outside the tube.

REVIEW OF DESIGN PARAMETERS

According to Kovats (2), effective hydraulic design depends on the interaction of the individual elements. Stepanoff (6) noted that designers need experience and skill to select design parameters and to discern the effects of these variables leading to optimum pump performance. The literature review concerned the following design parameters:

1. *Inlet approach.* Kovats (2) mentioned that the emphasis of previous research on pump design was always on "exact" calculations of impeller blade angles; inlet shape was often overlooked and little attention was given to parts downstream from the impeller. Obviously, "correct" blade angles are not of much use if the flow enters the impeller from the side at a very narrow angle or the flow is restricted.
2. *Impeller vane curvature.* Stepanoff (6) stated the impeller alone generates the head of a centrifugal pump. The head produced is essentially the same for all vane settings and thus is a function of vane curvature alone. This means that although the tangential component of the absolute velocity of flow at the impeller discharge is higher at higher values of vane discharge angle, the tangential component of the absolute velocity of flow at inlet is increased by approximately the same amount. The peripheral velocity being the same at inlet and outlet results in no change in head. Capacity varies almost directly with the inlet pitch.
3. *Diffusion vane curvature.* Shepherd (5) stated that the efficiency obtained with diffusion vanes depends on the vane setting and the flow area. The diffusion vane angle or curvature is selected so that the water enters the diffusion vanes with minimum loss and leaves the casing axially. Because of uncertain knowledge of field flow conditions, it is necessary to test several arrangements and obtain the optimum results empirically. High velocity can reduce efficiency. Pump parts operating at high speeds require more refined design than pumps operating at moderate speeds and head per stage.
4. *Clearance between the impeller and the stator casing.* Loss in pump capacity and efficiency is a function of

clearance. Leakage losses take place primarily through the clearance setting between the impeller and the stator casing. The capacity available at the pump discharge is smaller by the amount of leakage than that passing through the impeller. In IRRI tests, a Thai axial-flow pump with a 6-mm clearance had less capacity than the IRRI axial-flow pump with a 3-mm clearance.

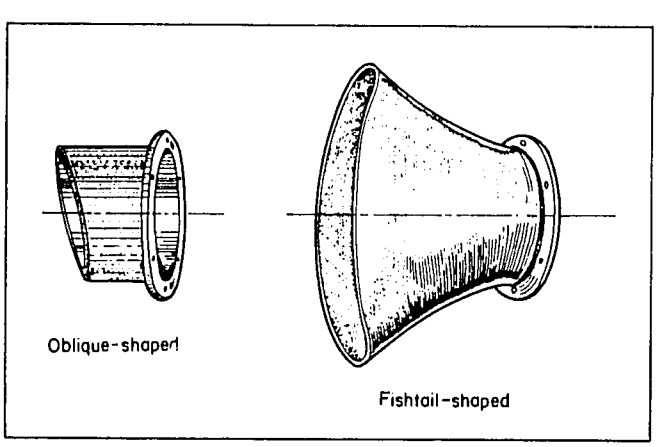
MATERIALS AND METHODS

Dimensional analysis permits detailed examination of blades and rotors of turbomachines. We used dimensional analysis to determine the relationships of several design parameters. The effects of the parameters on the pump capacity and power input (bhp) were determined by varying one variable, while holding all others constant.

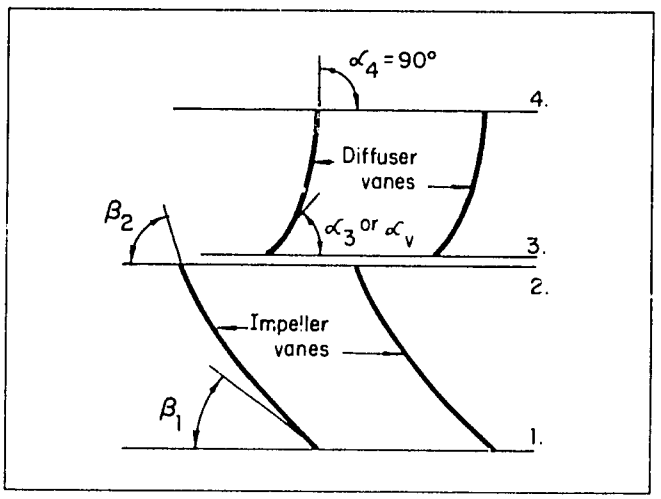
Two shapes of inlet approach, the original oblique and fishtail shape (Fig. 2), were compared. The optimum design of each shape was first determined before efficiencies were compared.

The impeller vane discharge angle or pitch β_2 (Fig. 3) was varied from 15° to 30° at 5° intervals. The pitch was varied up to 30° only because of the limited power of the pump motor. The prime mover used throughout the experiment was a calibrated 15-hp (11.19 kW) electric induction motor. The diffusion vane angles were set at 40°, 50°, 60°, and 90°. The 90° vane angle was of particular interest because it is easier to fabricate. The clearance between the impeller and the stator casing was varied from 2 mm to 10 mm at 2-mm intervals.

The two-stage pump design was basically the same as the single-stage, plus a 152-mm-long flanged series case with the diffusion vanes and main bearing holder welded inside it, an impeller, a rubber bearing, and an additional shaft and coupling (Fig. 4). The performance curves of the two-stage pump were developed from test data and compared with the performance of the single-stage pump. Spacings of 125 mm, 150 mm, 175 mm, and 1,250 mm between the two impellers were tested.



2. Two inlet shapes.



3. Development of cylindrical sections showing impeller vane discharge angle, β_2, and diffuser vane angle, α_v.

Cost analysis used discounting to express the costs over the machine life. The lower present worth of cost was used as the selection criterion.

RESULTS AND DISCUSSION

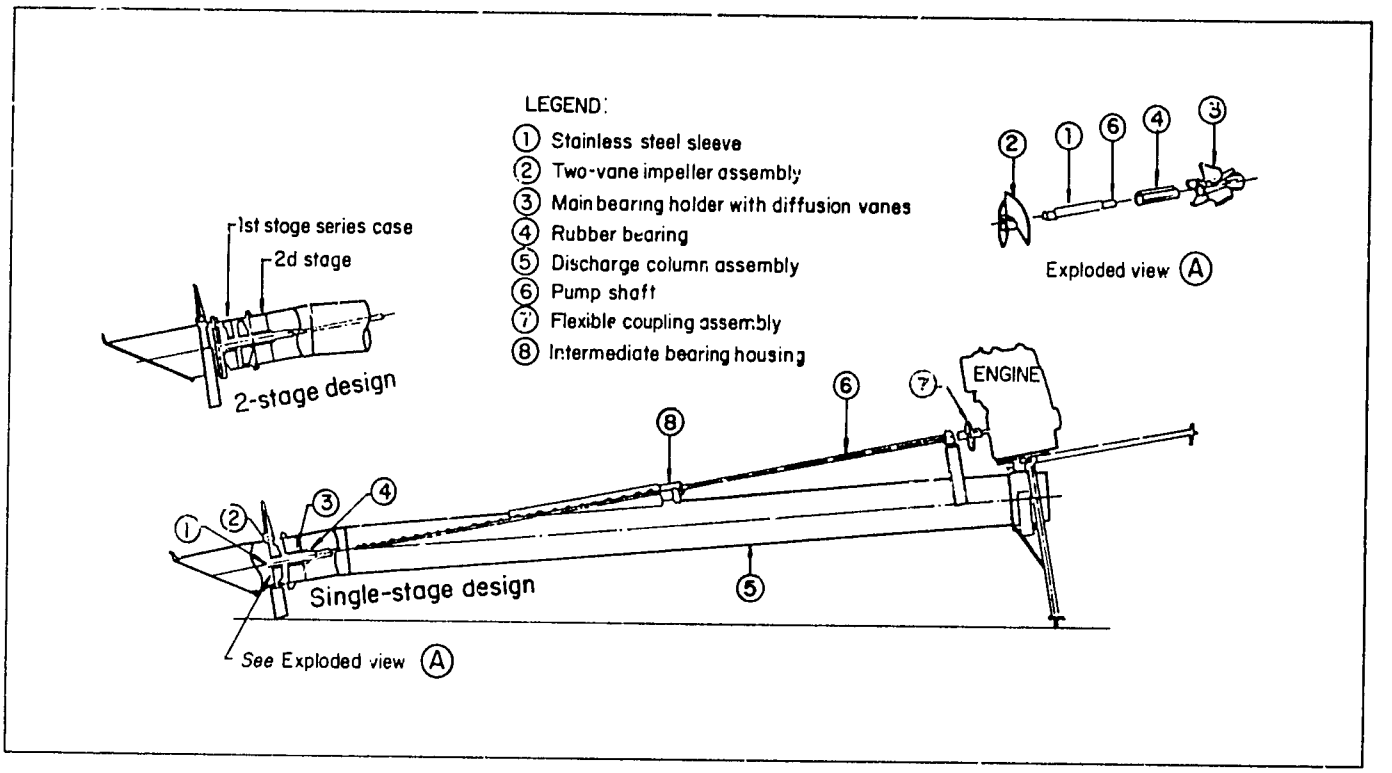
Effect of inlet shape on pump efficiency

Two inlet shapes were tested. With the fishtail-shaped inlet, maximum efficiency was obtained with a 40-cm radius of curvature of the flared sides. With the oblique-shaped inlet the 60° angle of cut across the cylindrical column was most efficient. Both approaches were attached close to the impeller.

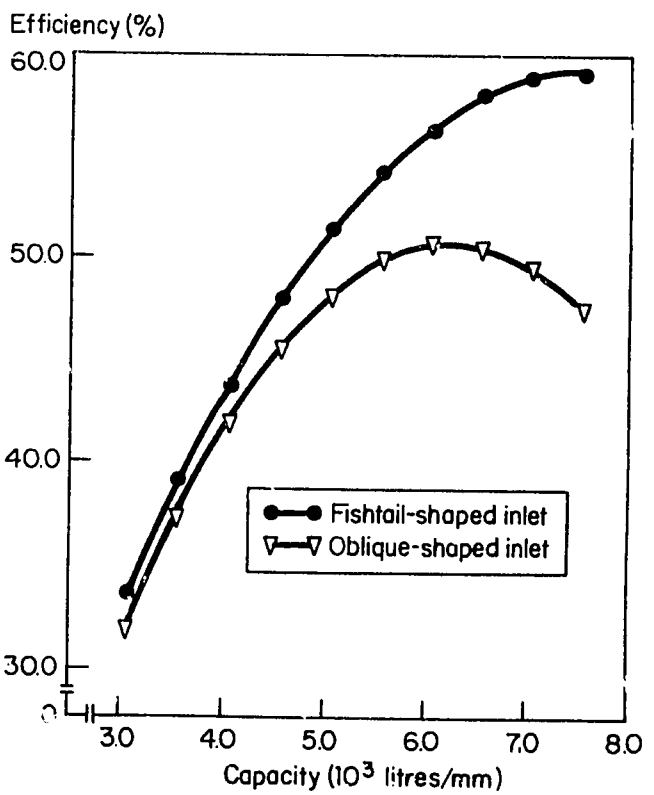
The effect of inlet shape on pump efficiency is shown in Figure 5. The fishtail-shaped inlet has higher peak efficiency at a much higher capacity (7,500 litres/min) than the oblique-shaped inlet with a peak efficiency obtained at 6,250 litres/min capacity. The better performance of the fishtail-shaped inlet could be attributed to a more uniform velocity distribution attained as water approaches the pump suction, compared to that produced by the sharp turn of the oblique-shaped inlet.

Effects of impeller vane discharge angle, diffusion vane angle, and clearance between the impeller and stator casing on the capacity and bhp of the pump

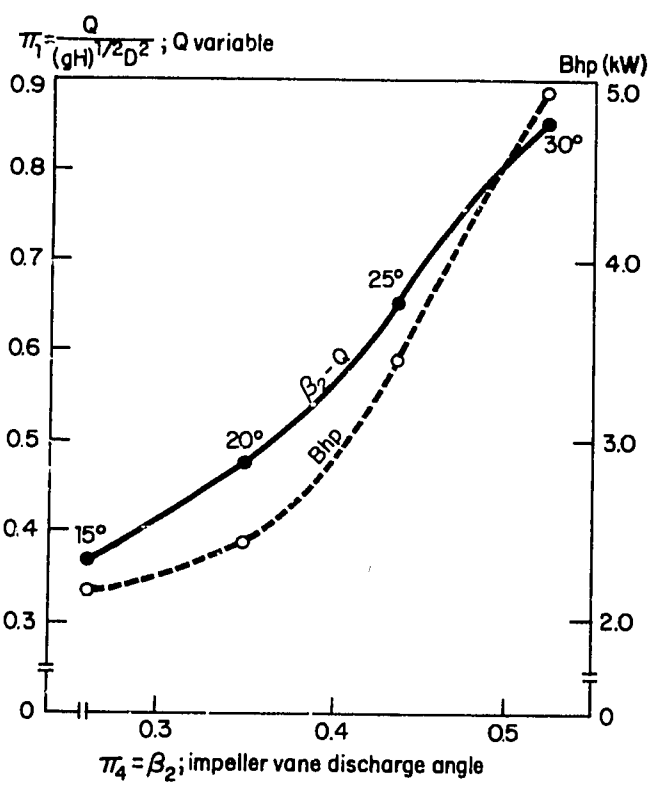
1. *Impeller vane discharge angle or pitch, β_2.* The capacity (Q) and brake horsepower (bhp) of the pump varied directly with pitch. The variation in capacity with β_2 was due to the change in the net area swept by the impeller vane. Consequently, the bhp increased with the increase in Q. Such relationship as represented in dimensionless terms, π_4 on π_1, is shown in Figure 6. As β_2 increases, Q increases and correspondingly bhp increases as a result of the increase in Q (bhp varies directly with Q). The increase in bhp varies significantly with the increase in β_2. The increases in Q and bhp at



4. The single-stage axial-flow pump design can be converted into a two-stage unit by attaching a series case assembly consisting of a flanged casing 152 mm long, with the diffusion vanes and main bearing holder welded inside it, a rubber bearing, an additional shaft, and another impeller.



5. Effect of inlet shape on the efficiency of IRRI-designed axial-flow pump.



6. Effect of impeller vane discharge angle on capacity, Q, and brake horsepower, bhp, of axial-flow pump.

pitch greater than 22.5° are greater than the increases in Q and bhp at pitch lower than 22.5°.

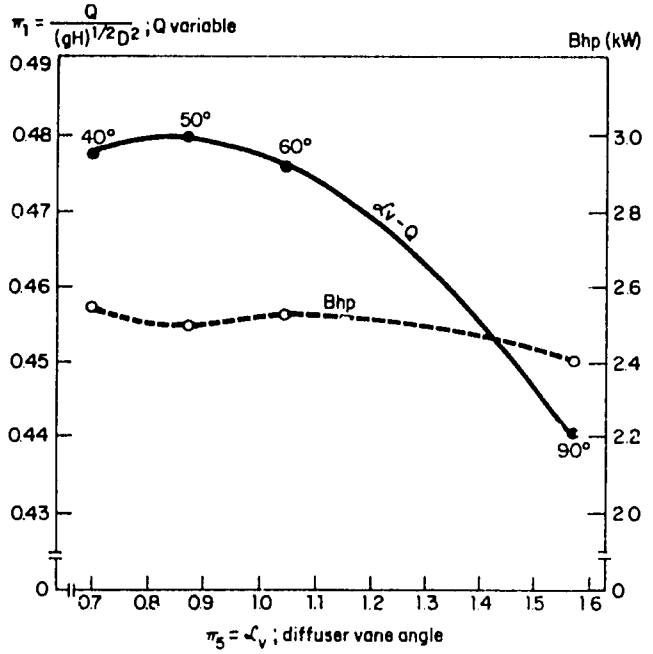
2. *Diffusion vane angle, α_v.* Figure 7 shows slight variation in capacity as the diffusion vane angle was varied from 40° to 60°, but a large drop in capacity in using a straight vane or 90° vane angle. Bhp decreased correspondingly but not significantly. Therefore, even though the straight diffusion vane (90°) was simpler to fabricate, its low performance eliminated it. Although variations from 40° to 60° vane angle have no noticeable effect on pump performance, a diffusion vane angle of 47° was found optimum for a 20-cm-diameter impeller.
3. *Clearance between impeller and stator casing.* The capacity increased significantly from 2-mm to 4-mm clearance, but dropped at larger clearances, as shown in Figure 8. The unexpected increase in Q from 2-mm to 4-mm clearance could be explained by the impeller vane tips rubbing against the stator casing due to the long pump shaft deflecting more than the required 2-mm clearance. The long shaft was mainly supported by two water-lubricated rubber bearings at the diffuser hub and the middle-bearing housing. Beyond 4 mm, Q dropped due to leakage losses between the impeller and the stator casing. Leakage increased as clearance increased, as presented in Figure 9, excluding the 2-mm sample point which was unaffected by clearance. The optimum clearance for the 20-cm-diameter impeller is about 4 mm. Bhp decreases as Q decreases because of increase in clearance.

Effects of spacing between impellers

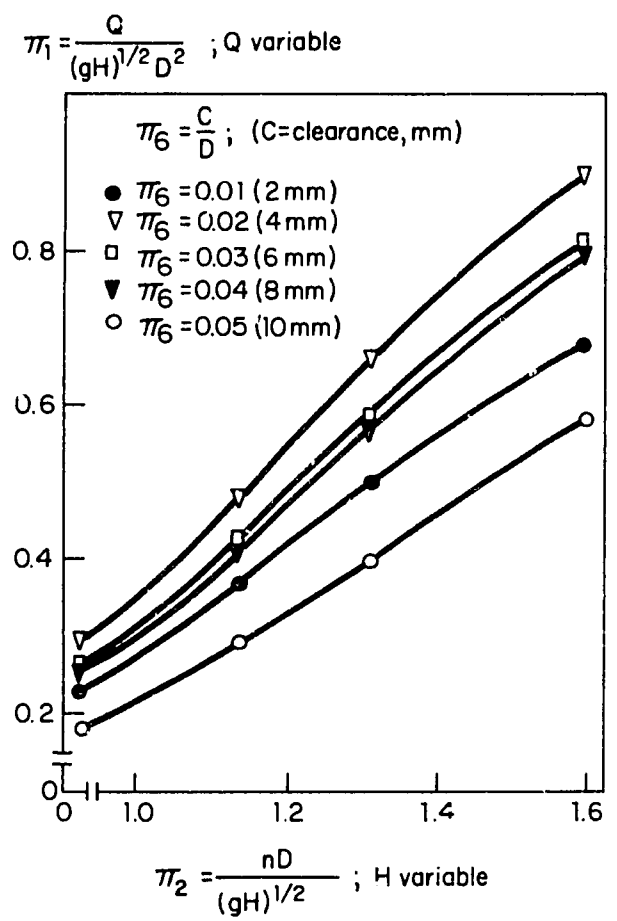
Spacing between impellers did not significantly affect capacity, but the bhp requirement was significantly different at spacing greater than 150 mm (Table 1), the spacing of highest efficiency (Fig. 10). The increase in bhp was due to the difference in stage pressures caused by interstage leakages. Between two adjacent stages, the interstage leakage is the principal cause of the difference in stage pressures. The high pressure casing has a capacity greater by the amount of interstage leakage than the low pressure casing.

Comparison of performance of the two-stage and single-stage pumps

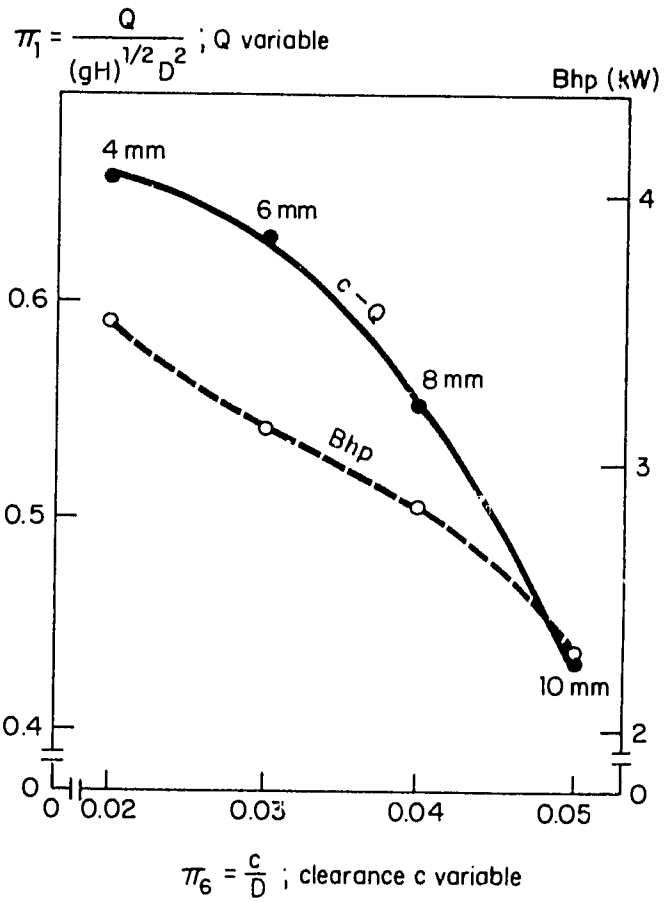
The two-stage pump was compared with the single-stage pump in capacity, brake horsepower, and efficiency. The performance curves of both pumps at 1,800 and 1,500 rpm are shown in Figures 11 and 12. At 1,800 rpm, the capacity of both pumps at best efficiency point was about the same (5,400 litres/min). At this capacity, the total static head and the bhp of the two-stage pump were more than twice those of the single-stage pump. The greater percentage of increase in head than in bhp resulted in a slightly higher peak efficiency of the two-stage pump. Similar observations were



7. Effect of diffuser vane angle, α_v, on capacity, Q, and brake horsepower, bhp, of axial-flow pump.



8. Relationship between π_1 and π_2 at different levels of π_6.



9. Effect of clearance on capacity, Q, and brake horsepower, bhp, of axial-flow pump.

Table 1. Pump capacity and brake horsepower (bhp) input at best efficiency point of two-stage axial-flow pump as affected by impeller spacing at 1,500 rpm.

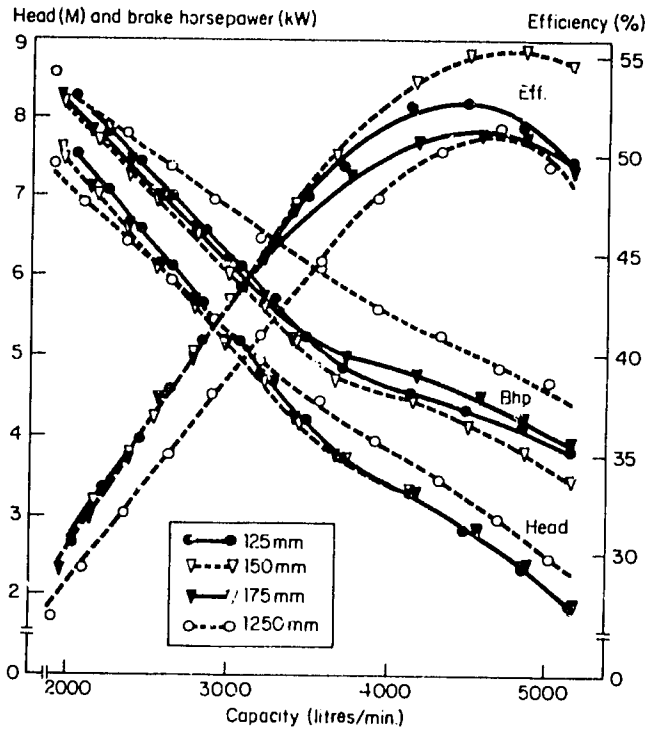
Impeller spacing (mm)	Capacity mean[a] (litre/min)	Bhp input mean[a] (kw)
125	4375 a	4.42 c
150	4375 a	4.30 c
175	4470 a	4.66 b
1250	4690 a	4.90 a

[a]Av of three readings. Means in a column having a common letter are not significantly different at the 5% level.

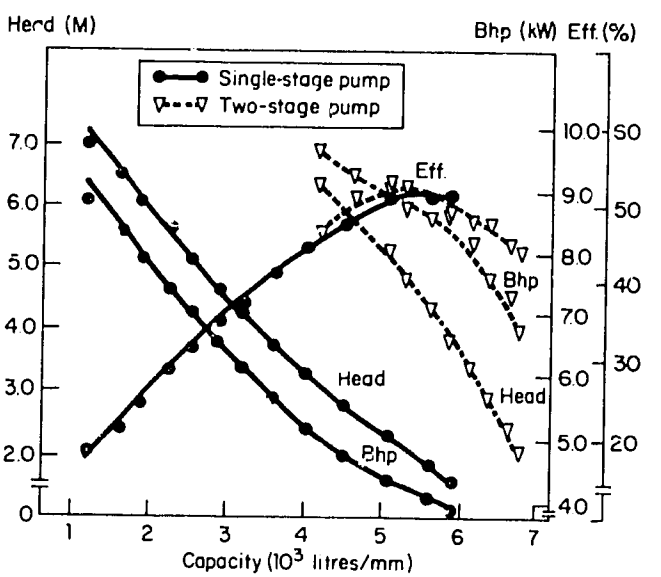
made at 1,500 rpm. However, two-stage pump efficiency decreased at this speed. The average efficiencies of both pumps, therefore, were similar.

Cost comparison of two-stage and single-stage pumps

At a pumping lift of up to 2.5 m, the single-stage pump can be driven by a 5-hp (3.73 kW) engine and the two-stage pump by an 8-hp (5.97 kW) engine. Above 2.5-m lift, the prime mover of the single-stage pump would be an 8-hp (5.97 kW) engine while that of the two-stage pump would be a 10-hp (7.46 kW) engine. Table 2 shows the pump capacity, pumping hours required per hectare per season, and area capacity per year at different lifts of both pumps. The



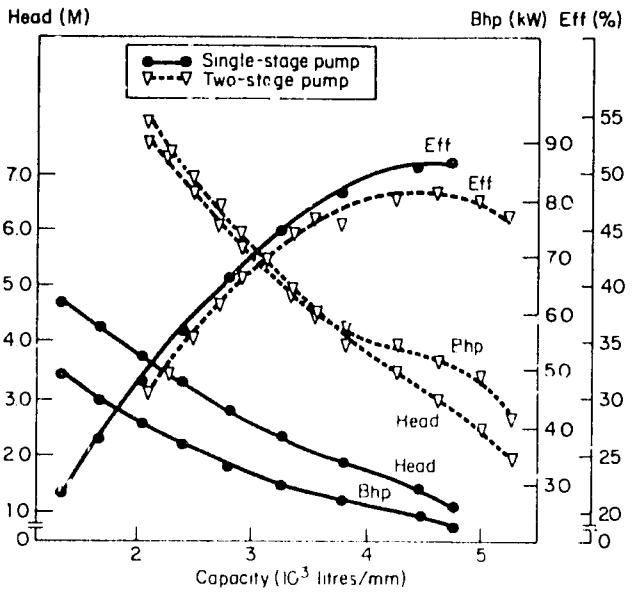
10. Effect of spacing between impellers on the performance of two-stage axial-flow pump.



11. Performance of single-stage, and two-stage IRRI axial-flow pumps at 1,800 rpm. Bhp = brake horsepower.

two-stage pump has greater Q and consequently lesser pumping hours required per hectare and greater area capacity per year than the single-stage pump at the same static head or lift.

The two-stage pump costs more initially than the single-stage pump due to the added series case assembly and bigger engine (Table 3, 4). It has greater total fixed cost per year and variable cost per hour, but lower total variable cost per hectare because the total pumping time per hectare is shorter.



12. Performance of single-stage and two-stage IRRI axial-flow pumps at 1,500 rpm. Bhp = brake horsepower.

Table 2. A comparison of the pump capacity, pumping requirement, and area capacity of the 20-cm-diameter single-stage (SS) and two-stage (TS) axial-flow pumps operating at 1,500 rpm at different pumping lifts.

Pumping lift	Pump capacity (m^3/h)		Pumping requirement[a] (h/ha per season)		Area capacity (ha/yr)[b]	
	SS	TS	SS	TS	SS	TS
2.0 m	219.0	316.5	54.8	37.9	28.0	40.5
2.5 m	186.0	297.6	64.5	40.3	23.8	38.1
3.0 m	157.8	277.5	76.1	43.2	20.2	35.5
3.5 m	133.5	253.5	89.9	47.3	17.1	32.5
4.0 m	111.0	240.0	109.1	50.0	14.1	30.7

[a]Average water requirement per hectare per season is 12,000 m^3.
[b]Assuming 8 h/day of operation, 6 days/wk, 4 wk/mo, 4 mo/season, and 2 seasons/yr, or 1,536 h/yr.

Cost comparisons based on present worths were made at 2.0-, 2.5-, and 3.0-m lifts and the calculations are summarized in Table 5. At 2.0-m lift, the single-stage pump has a lower present worth of cost for all 3 discount rates used. This indicates that at 2.0-m pumping level and below, it is cheaper to use a single-stage pump. At 2.5-m lift, the two-stage pump is cheaper to use up to 30% discount rate. At 40% rate, however, the single-stage is more economical. The two-stage pump continues to have lower cost at 3.0-m lift, and hence is cheaper to use for lifting water at 3.0 m or higher level. This is because the single-stage pump's capacity declines as the head or lift increases, entailing longer pumping hours per hectare and additional operating cost. As Figure 12 shows, at 2.0-m lift the single-stage pump has reached its peak and the efficiency starts to drop as the head increases. On the other hand, the two-stage pump is just starting to increase its efficiency at 2.0-m lift and reaches its peak at a lift approximately twice the optimum head of the single-stage pump. At this pumping level, the single-stage pump's efficiency has dropped almost 10%.

The costs at 2.5-m lift have different shapes (Table 5). The two-stage pump has a lower cost up to 30% and the single-stage pump at 40% discount rate. In this case, the choice would depend on the opportunity cost of capital. A point where the two pumps have the same present worth, known as the crossover or equalizing discount rate, is shown in Figure 13. At 39.17% opportunity cost of capital, the choice between the single-stage and the two-stage pumps would be unimportant.

Table 3. Comparative cost analysis of the IRRI 20-cm-diameter single-stage and two-stage axial-flow pumps operating at 2.0-m and 2.5-m lifts and at 1,500 rpm.

	Single-stage pump (US$)	Two-stage pump (US$)
Capital outlay		
Pump	197.18	259.15
Engine	335.21	478.87
Total capital outlay	532.39	738.03
Fixed cost per year		
Depreciation – pump and engine	95.83	132.85
Interest on average capital investment, 30%	87.85	121.77
Repair and maintenance, 10% of initial cost (IC)	53.24	73.80
Insurance, 2% of IC	10.65	14.76
Total fixed cost per year	247.56	343.18
Variable cost per hour		
Gasoline, oil, and grease	0.69	1.04
Labor	0.14	0.14
Total variable cost per hour	0.83	1.18
Total variable cost per hectare at 2.0-m lift	45.62	44.68
2.5-m lift	53.72	47.52

1US$ = ₱17.75 (Oct 1984 conversion rate).

Table 4. Comparative cost analysis of the IRRI 20-cm-diameter single-stage and two-stage axial-flow pumps operating at 3.0-m lift at 1,500 rpm.

	Single-stage pump ($)	Two-stage pump ($)
Capital outlay		
Pump	197.18	259.15
Engine	478.87	673.24
Total capital outlay	676.06	932.39
Fixed cost per year		
Depreciation – pump and engine	121.69	167.83
Interest on average capital investment, 30%	111.55	153.85
Repair and maintenance, 10% of initial cost (IC)	67.61	93.24
Insurance, 2% of IC	13.52	18.65
Total fixed cost per year	314.37	433.56
Variable cost per hour		
Gasoline, oil, and grease	1.04	1.29
Labor	0.14	0.14
Total variable cost per hour	1.18	1.44
Total variable cost per hectare at 3.0-m lift	89.62	62.09

1US$ = ₱17.75 (Oct 1984 conversion rate).

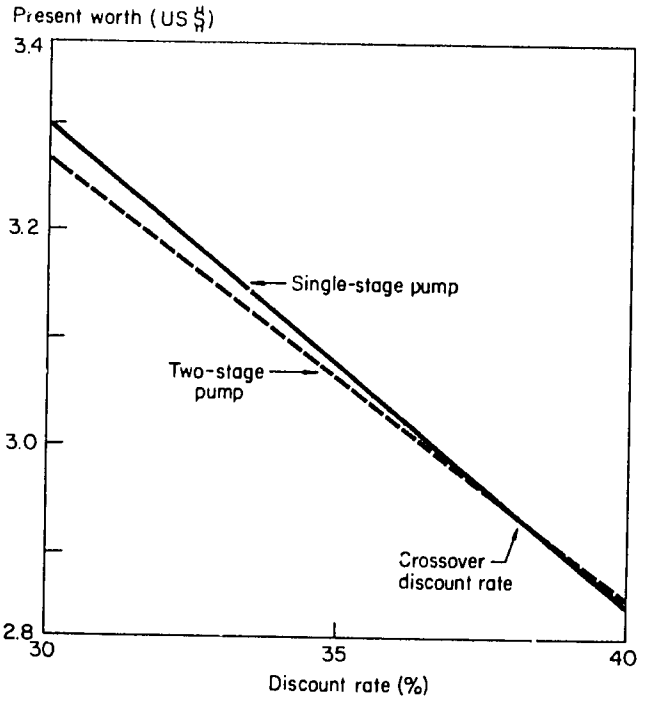
Table 5. Comparison of total present worth of costs between the single-stage and the two-stage pumps using 20, 30, and 40% discount rates for 2.0-m, 2.5-m, and 3.0-m pumping lifts at 1,500 rpm.[a]

Pumping lift (m)	Discount rates (US$)					
	20%		30%		40%	
	Single-stage pump	Two-stage pump	Single-stage pump	Two-stage pump	Single-stage pump	Two-stage pump
2.0 a	3451.66	3680.39	2909.80	3130.20	2519.38	2736.90
2.5 b	3927.99	3845.24	3305.01	3268.56	2869.24	2852.51
3.0 c	6281.30	4979.60	5240.85	4228.39	4491.27	3687.15

[a]Assumptions: 1. Estimated useful life of pumps and components is 5 yr with 10% salvage value. 2. Total area that could be served by both pumps per year at 3 m or below is 20 ha.

Computed costs per year using the single-stage (A) or the two-stage (B) pumps, in US$ are:
a. A: IC (532.39); FC/year = R&M (53.24) + insurance (10.65); Annual VC (912.45)
B: IC (738.03); FC/year = R&M (73.80) + insurance (14.76); Annual VC (893.58).
b. A: IC (532.39); FC/year = R&M (53.24) + insurance (10.65); Annual VC (1074.48)
B: IC (738.03); FC/year = R&M (73.80) + insurance (14.76); Annual VC (950.42).
c. A: IC (676.06); FC/year = R&M (67.61) + insurance (13.52); Annual VC (1792.39).
B: IC (932.39); FC/year = R&M (93.24) + insurance (18.65); Annual VC (1241.69).

1US$ = ₱17.75 (Oct 1984 conversion rate). IC = initial cost, FC = fixed cost, VC = variable cost, R&M = repair and maintenance.



13. Crossover discount rate between single-stage and two-stage pumps.

CONCLUSIONS AND RECOMMENDATIONS

1. Inlet shape affects optimum flow conditions. A fishtail-shaped inlet gives higher pump performance than the oblique-shaped inlet.
2. The impeller vane discharge angle β_2 affects the head-capacity characteristics. Because the head-capacity curve varies directly with β_2, the value of β_2 depends on the desired head capacity. If power is not limited, β_2 may be increased up to 30°, otherwise the power input must be considered.
3. The diffusion vane angle α_v is not a critical design parameter. Variations as great as ±5° from the optimum value of 47° can be used without any considerable effect on performance.
4. The greater the clearance between the impeller and the stator casing, the greater the capacity loss due to leakage. The minimum clearance is limited by the maximum deflection of the pump shaft.
5. The optimum spacing between impellers of the two-stage pump is 150 mm. Greater spacing requires greater power input due to interstage leakage.
6. A two-stage pump is basically two single-stage units in terms of performance. At constant capacity at the best efficiency point, the head and bhp requirement is doubled with approximately the same efficiency as that of a single-stage pump.
7. The single-stage pump is cheaper to use up to 2-m lift and at 2.5-m lift if the opportunity cost of capital is 40%. However, the two-stage pump is more economical at 2.5-m lift if the opportunity cost of capital is up to 30% and at higher levels.

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