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A Study of Supercavitating Pumps*

(1st Report, Pump Performance and Cavitation on Impeller Blades)

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Experiments on two supercavitating pumps, pumps A and B, were carried out to investigate the characteristics of supercavitating pumps. The two pumps were designed for the specific speed N_s of 3,300 (in units $m, m^3/min, rpm$) and the suction specific speed S of 2,000. In the design of the impeller of pump A experimental data obtained by Wade and Acosta on supercavitating cascades of planoconvex hydrofoils were used. Wu's theory on supercavitating circular arc hydrofoils was applied to design the impeller of pump B. Following results were obtained by experiments. The performance of pump A is N_s 3,030, S 2,320 and efficiency 59.4%, and that of pump B is N_s 3,440, S 3,260 and efficiency 51.8%, under supercavitating conditions. It was confirmed that specific speed and suction specific speed of supercavitating pumps were much higher than those of conventional axial flow pumps.

1. Introduction

Cavitation causes loss in the performance of hydraulic machines, i.e. pumps, water turbines, ships' propellers etc., and the collapse of cavity causes erosion of the machines. Studies of the machines are usually carried out to avoid cavitation as far as possible and to obtain the good performance of the machines under non-cavitating conditions. However, it is very difficult to avoid cavitation in the high speed operating region. This is a severe limitation on the speed of machines.

To solve this severe problem and obtain high speed hydraulic machines, the application of supercavitation to machines has been made. It is not to avoid cavitation, but to use cavitation bubbles. Whole suction side of the impeller blades is covered with fully developed cavity. The collapse of the cavity occurs downstream of the impeller and causes no erosion of the blades. Many theoretical studies of supercavitating hydrofoils and cascades have been carried out. Experimental studies, however, have been scarce, since high power is necessary to produce supercavitation in cavitation tunnels. The application of supercavitation to ships' propellers and a hydrofoil running below the free water surface has been investigated, but a few papers ^{(1)~(3)} on supercavitating pumps and turbines have been presented.

A series of experiments are carried out to investigate characteristics of supercavitating pumps. This paper deals with design and experiments of two supercavitating pumps A and B.

2. Nomenclature

The following nomenclature is used in the paper.

- c: absolute velocity, m/s
 - C_L : lift coefficient
 - D: impeller diameter, m
 - g: acceleration due to gravity, m/s^2
 - H: total head across pump, m
 - K_b : cavitation coefficient for impeller blades
 - K_{b0} : cavitation coefficient for isolated blade
 - N: rotational speed, rpm
 - N_s : specific speed = $N/\sqrt{Q}/H^{3/4}$, (rpm, m^3/min , m)
 - P: pump input, kW
 - Q: capacity, m^3/min
 - r: radius, m
 - S: suction specific speed = $N/\sqrt{Q}/(NPSH)^{3/4}$, (rpm, m^3/min , m)
 - t/l: pitch chord ratio
 - u: peripheral velocity, m/s
 - w: relative velocity, m/s
 - α : angle of attack
 - β : angle between relative velocity and peripheral velocity
 - δ : dimensionless impeller diameter
 - ϵ : drag/lift ratio
 - η : pump efficiency
 - η_h : hydraulic efficiency
 - λ : depression ratio of lift coefficient
 - ν : hub/tip ratio
 - σ : dimensionless specific speed
- Subscripts
- 1: impeller inlet
 - 2: impeller outlet
 - u: peripheral component
 - m: mean value of values at impeller inlet and outlet.

3. Design of test pumps

Test pumps were designed as follows.

- (1) Supercavitating pumps were considered as a kind of axial flow pumps of which specific speed was higher than that of conventional axial flow pumps. A design method of axial flow pumps was used.

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Table 2 Lift coefficient of planoconvex hydrofoil cascades

Pitch chord ratio t/l	Angle of attack α	Supercavitating condition		Non-cavitating condition	Depression ratio of lift coefficient $\lambda = \frac{CLSC}{CLNC}$
		Cavitation coefficient K_b	Lift coefficient $CLSC$	Lift coefficient $CLNC$	
0.8	4° *	0.22	0.17	0.235	0.72
1.6	8°	0.22	0.32	0.53	0.61
	9°	0.22	0.30	0.60	0.50

* Angle of attack where value of drag/lift ratio is minimum.

Table 3 Design specifications of pump A impeller

Design sections	I	II	III	IV	V
Diameter mm	102.0	144.5	177.0	205.4	228.5
Lift coefficient	0.716	0.458	0.351	0.294	0.261
Pitch/chord ratio	1.15	1.50	1.72	1.95	2.16
Thickness/chord ratio %	9.34	7.16	5.68	4.75	4.18
Hydrofoil	Planoconvex

Fig. 2 Impeller of pump A

Number of guide vanes was six. Pitch/chord ratio of guide vanes was so determined that their lift coefficient was approximately unity. Hydrofoils of NACA 6-series were used for the guide vanes. The mean camber line is that of NACA 64 hydrofoil. The thickness of the design section near the hub and that near the tip are that of NACA 63A010 and that of NACA 63A006, respectively. Change in thickness from the hub to the tip is smooth.

3.2.3 Structure of pump A

Fig. 3 shows a sectional elevation of the pump. Cavitation in the impeller can be observed through the apparent part of the casing.

Fig. 3 Sectional elevation of test pumps

3.3 Design of pump B

3.3.1 Impeller

A supercavitating hydrofoil theory was applied to the design of each section. In this theory a blade cavitation coefficient K_b , which is a measure of cavitating conditions, is used, in order to estimate the performance of supercavitating blades. Wu's isolated hydrofoil theory⁽⁵⁾ was used, since its validity was verified by experiments. In this case the same assumptions as those in case of pump A impeller were made.

The selected supercavitating hydrofoil was circular arc, since it would have better performance than the planoconvex one, which was equivalent to the flat plate under supercavitating conditions. The theory is applied with ease in the case of circular arc hydrofoils. The shape of the suction side of blades was determined by reference to TMB modified section⁽⁷⁾. Since the shape of trailing edge at suction side would have no effect on pump performance under supercavitating conditions, the trailing edge was cut at right angles to the blade chord. A cavitation coefficient K_b for impeller blades was defined by Eq. (7). The coefficient K_b corresponds to a blade cavitation coefficient K_{b0} defined by Eq. (6) in Wu's theory.

$$K_{b0} = \frac{h_0 - h_v}{c^2/2g} \quad \dots \dots \dots (6)$$

where h_0 is static head in uniform flow, h_v is vapor pressure (head) of water, and c is velocity in uniform flow.

$$K_b = \frac{NPSH - (c_1^2/2g)}{w_1^2/2g} \quad \dots \dots \dots (7)$$

where NPSH is net positive suction head, and c_1 and w_1 are absolute velocity and relative one at impeller inlet, respectively. Values of blade cavitation coefficient in five design sections are shown in Table 4. Angle of attack necessary for cavitation and for the lift coefficient was determined from the performance of blades, which had been expressed by a parameter of arc angle of supercavitating circular arc hydrofoil. Values of lift coefficient, which were initially assumed equal to those of pump A, were so modified that arc angle of blades had the same value at each design section. Design specifications and a photograph of the impeller are shown in Table 4 and Fig. 4, respectively.

3.3.2 Guide vanes

The downstream guide vanes of pump A were also used for pump B for the following reason. The difference between the outlet angle of pump B impeller and that of pump A impeller is only a few degrees, and the outflow angle from impellers under supercavitating conditions can not be estimated in the early stage of the study.

Fig. 4 Impeller of pump B

Table 4 Design specifications of pump B impeller

Design sections	I	II	III	IV	V
Diameter mm	102.0	144.5	177.0	205.4	228.5
Blade cavitation coefficient	0.515	0.299	0.211	0.161	0.132
Lift coefficient	0.645	0.426	0.341	0.288	0.257
Pitch/chord ratio	1.00	1.38	1.68	1.93	2.15
Thickness/chord ratio %	7.48	6.82	6.36	5.85	5.40
Hydrofoil	Pressure side: Circular arc

Fig. 5 Test arrangement

4. Test arrangement

Fig. 5 shows the test arrangement. A Venturi tube was used to measure the capacity. The pressure differences of the Venturi and of the total head across pump were measured with mercury manometers of one column type. The power input to the pump was calculated from the pump torque measured with the electric dynamometer and the speed measured with an electric pulse generator attached to the shaft and an electronic pulse counter.

5. Experimental results

Experiments were carried out under conditions where NPSH was maintained at constant values. Total head, power input and capacity were measured at various NPSH.

5.1 Pump performance

Figs. 6 and 7 show the performances of pumps A and B at the values of NPSH where whole suction side of blades is under supercavitating conditions. The pumps have the following performances at the best efficiency point under supercavitating conditions.

Fig. 6 Performance of pump A

Fig. 7 Performance of pump B

Pump A: Specific speed $N_s = 3,030$, suction specific speed $S = 2,320$, and efficiency $\eta = 59.4\%$.
Pump B: $N_s = 3,440$, $S = 3,140$, and $\eta = 51.8\%$.

NPSH = 10.0 m
 $Q = 20.9 \text{ m}^3/\text{min}$
 $H = 6.29 \text{ m}$ $\eta = 58.3\%$

NPSH = 20.2 m
 $Q = 23.0 \text{ m}^3/\text{min}$
 $H = 11.2 \text{ m}$ $\eta = 78.2\%$

Fig. 8 Cavitation in pump A

It was confirmed that specific speed and suction specific speed of supercavitating pumps were higher than those of conventional pumps.

5.2 Cavitating conditions

Fig. 8 shows photographs of cavitation at the best efficiency points of pump A at two values of NPSH. The photograph at the NPSH of 10.0 m shows the supercavitating condition on the whole suction side of blades. It shows that the length of cavity near the tip is approximately twice that of the blade chord length. In Fig. 9 a sketch is presented, which shows cavitating conditions at low and high NPSH.

Fig. 9 Cavitating conditions

NPSH = 6.5 m
 $Q = 18.0 \text{ m}^3/\text{min}$
 $H = 6.68 \text{ m}$ $\eta = 51.9\%$

NPSH = 21.0 m
 $Q = 22.4 \text{ m}^3/\text{min}$
 $H = 11.46 \text{ m}$ $\eta = 68.8\%$

Fig. 10 Cavitation in pump B

Supercavitation starts on impeller blades near the tip and spreads to the hub as NPSH decreases. Fig. 10 shows photographs of cavitation at the best efficiency points of pump B at two values of NPSH. The photograph at the NPSH of 6.5 m shows the supercavitating condition on the whole suction side of blades. Since cavity would have touched the guide vanes if they had been located at the design position, they were moved 30 mm downstream from the design position. The length of cavity near the tip of pump B impeller is approximately 2.5 times as great as the blade chord length, and is greater than that of pump A impeller.

Fig. 11 Cavitation performance of pump A

Fig. 12 Cavitation performance of pump B

5.3 Cavitation performance

Figs. 11 and 12 show the cavitation performances of pumps A and B. One curve shows the relation between NPSH and performance at the best efficiency points which were obtained from the pump performance curves measured at various values of NPSH. The other curve shows the cavitation performance at a constant capacity at the best efficiency point under slightly cavitating conditions.

In the region indicated by supercavitation in the figures the whole suction side of blades is under supercavitating conditions. The figures show characteristics of supercavitating pumps, e.g. the rate of decrease in performance as NPSH decreases is smaller in supercavitating pumps than in conventional axial flow pumps. Figs. 6, 7, 11 and 12 show that the expected performance was satisfied with the performance at the best efficiency points under slightly cavitating conditions, i.e. at a high NPSH, but not satisfied with that under supercavitating conditions on the whole suction side of blades. Under the latter conditions the capacity was much reduced by cavities. For design of supercavitating pumps it must be taken into account that the blade passage area is reduced by cavities.

Under supercavitating conditions the best efficiency of pump B is lower than that of pump A. However, decrease in performance of pump B is less than that of pump A, according as supercavitation develops. Therefore, the circular arc hydrofoil used for pump B impeller is one with the characteristics suitable for supercavitating hydrofoil, and is expected to be a good supercavitating hydrofoil if its drag/lift ratio is reduced.

Fig. 13 Effects of blade setting angle on cavitation performance of pump A

5.4 Effects of setting angle of blades

Cavitation performances at various setting angles of blades of the two impellers are shown in Figs. 13 and 14. Under slightly cavitating conditions, i.e. at high NPSH, effects of blade setting angles of supercavitating pumps on capacity are as much as that of conventional axial flow pumps, but under supercavitating conditions the effects are less. The causes of these results may be considered as follows.

Increase in blade setting angle corresponds to that in angle of attack and that in capacity under slightly cavitating conditions. However, the increase in capacity is canceled by growth of cavities in blade passages under supercavitating conditions. Moreover, increase in blade setting angle under supercavitating conditions may cause decrease in efficiency, since increase in angle of attack causes increase in drag/lift ratio of hydrofoils.

6. Comparison of supercavitating pumps with conventional axial flow pumps

Since experimental results show that specific speed and suction specific speed of two supercavitating pumps are high, high speed pumps are possible by application of supercavitation. For given specifications (i.e. capacity, total head) of a pump prototype, supercavitating pumps and a conventional axial flow pump are designed from the models obtained by this study and from an axial flow pump model of the specific speed of 1,600, respectively. Table 5 shows impeller diameter and speed of the designed conventional axial flow pump under non-cavitating conditions and those of the designed supercavitating pumps. The supercavitating pumps were compared with the conventional axial flow pump at the same NPSH. The pumps designed from pump B at the same specific speed and the same suction specific speed as those of pump A are also shown in the table. Table 5 shows that the speed and impeller diameter of the supercavitating pumps are approximately two times and 75% as large as those of the conventional pump. However, efficiency of supercavitating pumps is less than that of conventional ones under non-cavitating conditions, as predicted.

Fig. 14 Effects of blade setting angle on cavitation performance of pump B

Table 5 Speed, size and performance of supercavitating pumps and conventional ones

		Specific speed	Efficiency	Speed ratio	Ratio of impeller diameter **
Conventional axial flow pump (Non-cavitating)		1,600	84%	1.0	1.0
Pump A	Supercavitating	3,030	59	1.89	0.78
Pump B	Supercavitating	3,440	52	2.25	0.75
	At the same specific speed as that of pump A *	3,030	64	1.89	0.77
	At the same suction specific speed as that of pump A *	3,160	62	2.00	0.76

* Part of impeller blades is not under supercavitating conditions.

** Ratio of impeller diameter: $\frac{D_{sc}}{D_c} = \frac{[\sigma\delta]_{sc} (N_s)_c}{[\sigma\delta]_c (N_s)_{sc}} \dots\dots\dots (8)$

where $\sigma\delta$ is a speed factor ($= u_t/\sqrt{2gH}$), which is obtained by experiments.

Subscripts $_{sc}$: supercavitating pump
 $_c$: conventional axial flow pump
 $_t$: values at impeller diameter

7. Conclusions

Supercavitation was applied to axial flow pumps, in order to increase specific speed and suction specific speed of pumps. Two pumps A and B were designed and tested. The following results were obtained.

(1) Under supercavitating conditions, the performance of pump A is: specific speed N_s 3,030, suction specific speed S 2,320 and efficiency η 59.4%, and that of pump B is: N_s 3,440, S 3,260 and η 51.8%. It was confirmed that specific speed and suction specific speed of supercavitating pumps were high.

(2) The capacity was much reduced under supercavitating conditions. This may be caused by decrease in blade passage area with cavities, and must be taken into account in the stage of design.

(3) Under supercavitating conditions the best efficiency of pump B is lower than that of pump A. However, decrease in performance of pump B is less than that of pump A, according as supercavitation develops. Therefore, the circular arc hydrofoil used for pump B impeller is one with the characteristics suitable for supercavitating hydrofoil, and is expected to be a good supercavitating hydrofoil if its drag/lift ratio is reduced.

(4) Effects of blade setting angle on capacity under supercavitating conditions are less than those under slightly cavitating conditions. However, their effects on efficiency are great, and efficiency decreases as setting angle increases.

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