may be more accurate and simple, if we use a better torque meter and a more convenient thrust meter. We are researching the blade characteristics by the other testing turbine, measuring the torque and thrust with the strain gauge of resistance wire.

6. Conclusion

(a) In the impulse turbine of partial admission, the characteristics of the rotor blades may be obtained with little effort, by the measurements of the forces acting on the blades.

(b) The deviation angles of exit flow from

the rotor blades are larger than the results obtained in the cascade tunnel test.

References

- (1) K. Nakamura and S. Fukunaga: *Trans. Japan Soc. Mech. Engrs.*, Vol. 18, No. 67 (1952), p. 234.
- (2) S. Sugawara and K. Nakamura: *Trans. Japan Soc. Mech. Engrs.*, Vol. 5, No. 14 (1939), p. 104.
- (3) D. G. Ainly: *Internal Combustion Turbines* (1949), p. 232.
- (4) J. A. Dunsby: *R & M*, No. 2728 (1953), p. 5.
- (5) I. H. Johnson and L. R. Knight: *R & M*, No. 2927 (1956), p. 5, and 11.

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Cavitation Tests on an Axial Flow Pump*

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Cavitation tests on an axial flow pump were made over a wide range of operating conditions. Through these tests, the aspects of cavitation in the impeller were observed with a strobolight and variation of the pump characteristics was studied. A parameter σ_s to represent the suction condition was introduced to give the cavitation limits for any flow rate.

The pump head showed little change until the cavitation on the blade surface was developed considerably, while the shaft horse-power began to increase slightly with the incipient cavitation on the blade, decreasing abruptly as the pump head dropped due to the growth of cavitation. So that the efficiency showed gradual decrease as the incipient cavitation grew.

The cavitation limits concerning the incipience on the blade and variation in pump performance are presented against the flow rate in terms of the parameter σ_s as well as in the suction specific speed.

1. Introduction

The cavitation limits of the axial flow pumps at the design point have been known experimentally using either the Thoma's cavitation coefficient or the suction specific speed. But it is not clear whether the limits actually refer to the incipience of cavitation in the pump or to the performance deterioration due to cavitation. As to the limits concerning the operating range out of the design point, very few results have been reported.

The axial flow pump, however, is often used for agricultural or drainage purposes and in such cases both suction and discharge levels fluctuate frequently owing to the weather variation of different seasons, so that it is almost impossible

to operate the pump only at the designed duty. It becomes important to know the relation between the suction head and cavitation in the case when the pump is operated out of the design point.

Recently the authors made a series of cavitation tests on an axial flow pump specially built for the purpose of clarifying cavitation characteristics over a wide range of the pump capacity. Through the tests occurrence of cavitation was observed and variations of the pump performance were studied. In this paper the test results have been presented in a manner applicable to the field of practice.

2. Nomenclature

The following nomenclature is used in the paper:

A_0 : annular area of impeller m^2

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H : total head	m
H_a : atmospheric pressure	m Aq.
H_n : total head at design point	m
H_{sv} : net positive suction head	m
H_v : vapor pressure of water	m Aq.
h_d : delivery pressure	m Aq.
h_s : suction pressure	m Aq.
h_s' : pressure at suction bore	m Aq.
Δh : height between pump center line and points h_s' measured	m
K_d : cavitation coefficient for hydrofoil	
L_s : shaft horse power	H.P.
N : speed of revolution	r.p.m.
Q : capacity	m ³ /min
S : suction specific speed	m ³ /min, m, r.p.m.
u_t : peripheral velocity at impeller tip	m/sec
V_d : mean velocity at points delivery pressure measured	m/sec
V_s : mean velocity at impeller inlet	m/sec
v_{m0} : axial velocity in front of blade inlet	m/sec
w_{0t} : relative velocity in front of blade inlet at impeller tip	m/sec
γ : weight of water per unit volume	kg/m ³

ζ : correction factor for suction pressure

η : efficiency %

μ : shaft horse power coefficient

σ : Thoma's cavitation coefficient

σ_s : parameter to represent suction conditions

ϕ : flow coefficient

Ψ : head coefficient

Ψ_n : head coefficient at design point

3. Testing apparatus and methods

3.1 Testing apparatus

The test pump was provided with a casing partly made of transparent acril resin in order to observe the inside of the pump, and the curved casing of suction side was also provided with a glass plate glazed in order to observe along the direction of shaft as shown in Figs. 1 and 2.

The major dimensions of the test pump are as follows; Specific speed $N_s=1450$ (m, m³/min, r.p.m.), Impeller outside diameter $D_i=370$ mm, Hub ratio $m=0.5$, Number of blades $z=4$, Clearance at impeller tip $a=0.75$ mm, Number of guide vanes $z_g=6$. At the impeller tip the camber of the blade c/l was 4%, the blade setting angle

Fig. 1 Sectional view of test pump

Fig. 2 External view of test pump

Fig. 3 Test impeller with casing removed

Fig. 4 Scheme of test circuit

$\theta=16^{\circ}50'$ and the pitch cord ration $t/l=1.43$, respectively.

Fig. 4 shows the outline of the testing apparatus. The test pump was driven by a motor; 30 H.P., variable speed, direct current, through V pulleys. The suction pressure of the pump was adjusted by controlling the opening of the suction valve. A straight pipe long enough behind the valve and rectification lattices were installed in order to prevent influence of unilateral current caused by the valve. A double suction volute pump was provided at the discharge side of the test pump with an ample distance to fill up lack of head and to extend the scope of the tests. The pump capacity was controlled by the discharge valve.

3.2 Testing methods

In the tests, we clarified the state of cavitation and changing phases of performance concerning several flow rates at three kinds of speed, i.e. 600, 750, and 900 r.p.m. in order to investigate the influence of the speeds of revolution. The suction pressure of the pump was slowly lowered keeping the capacities constant by controlling the valves, while the variations in the pump head and shaft horse power were measured. At the same time the state of cavitation was observed by a strobolight.

During the tests, revolution of the shaft was controlled by rheostats of the direct current motor and was checked with an electric tachometer. For the shaft horse power measurement, a strain meter of electric resistance type was attached directly to the pump shaft, and the shaft torque was measured by an oscillograph. The horse power was derived from this torque and the speed of revolution.

To measure the pump capacity a venturi meter was used (opening ratio 0.326), its flow coefficient being properly corrected beforehand. For the measurement of pressure difference a mercury U-tube was used as seen in Fig. 4.

The suction head was obtained by measuring the pressure with a mercury U-tube at two points, right and left sides of suction bore of the curved casing (H_s' in Fig. 1). Pressure difference between right and left sides was always within the experimental error. According to the above the suction pressure head of the pump will be as follows:

$$h_s = h_s' - \Delta h + \zeta \frac{V_s^2}{2g} \dots\dots\dots (1)$$

The term $\zeta \frac{V_s^2}{2g}$ is considered to express the minute static pressure difference between those at the suction bore and inlet of the impeller*.

For the investigation of cavitation, it is reasonable to use the Net Positive Suction Head, which is expressed as follows:

$$H_{sv} = H_d + h_s + \frac{V_s^2}{2g} - H_v \dots\dots\dots (2)$$

The discharge head was found by measuring pressure with a mercury U-tube at two points, right and left sides of the guide vane outlet (H_d in Fig. 1). Then the total head of the pump was calculated as follows:

$$H = h_d - h_s + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} \dots\dots\dots (3)$$

In order to observe cavitation occurring inside the pump, strobolight was given synchronously with the speed of revolution and one shot flash of the strong strobolight was used for photographing. Useful duration of the light being about $3/10^5$ sec., photographs of the impeller rotating at high speed were taken as if it were stationary.

The water temperature was measured at the suction pipe inlet and the atmospheric pressure was measured by a mercury barometer.

4. Test results and discussion

4.1 Means for presenting cavitation test results

In performing cavitation tests on a pump, Thoma's cavitation parameter in Eq. (4) is generally used to express the suction conditions in a manner irrespective of the pump speed.

$$\sigma = \frac{H_{sv}}{H} \dots\dots\dots (4)$$

The parameter σ which estimates the critical cavitation condition for the best efficiency point of a pump has close relation with the specific speed of the pump and variation of the critical σ versus the specific speed has been obtained experimentally⁽¹⁾.

In expressing the variation of the cavitation limits against the capacity of a pump, it is not appropriate to use the parameter σ as given by Eq. (4) at any point on the $H-Q$ characteristic curve of the pump. For the value of σ varies according to the pump head at given capacities

* Actual value of ζ was derived from the pressure difference taken at the suction bore (H_s' in Fig. 1) and at four points along the inlet periphery (H_{s1} , H_{s2} etc. in Fig. 1), measured at a capacity near the b. e. p. where the rotating component of flow at the impeller inlet did not accompany.

Fig. 5 Cavitation characteristics at constant flow rates
(A-line in each figure indicates incipient cavitation on blade surface)

even if the required NPSH be the same. It is preferable to use a ratio of the NPSH at given capacities to a quantity which is proportional to the square of the pump speed. For instance the dimensionless quantity which is the NPSH divided by u_t^2/g can be used⁽²⁾. In expressing the results of present tests, a parameter σ_s , which is the ratio of NPSH to the pump total head at the design point, has been used to represent the suction conditions for all capacities, that is:

$$\sigma_s = \frac{H_{sv}}{H_n} \dots\dots\dots (5)$$

When variation of the critical value of σ_s versus the capacity is obtained, it expresses the cavitation characteristics of a pump in terms of the head at the design point in a manner proportional to the required NPSH at the critical conditions*.

The capacity, head and shaft horse power were expressed by dimensionless coefficients ϕ , Ψ , and μ respectively to indicate the cavitation characteristics of the pump in the manner independent of the speed of rotation. These coefficients are defined as follows:

$$\begin{aligned} \text{Flow coefficient} \quad \phi &= \frac{v_{m0}}{u_t} \\ \text{Head coefficient} \quad \Psi &= \frac{H}{u_t^2/2g} \\ \text{Power coefficient} \quad \mu &= \frac{75L_s}{\gamma A_0 u_t^3} \end{aligned}$$

Besides the Thoma's cavitation parameter σ , the suction specific speed S defined by Eq. (6) is used in practice to give the criterion of cavitation limits in pumps⁽³⁾.

$$S = \frac{n\sqrt{Q}}{H_{sv}^{3/4}} \dots\dots\dots (6)$$

In this report the suction specific speed has been used to give the cavitation limits of the pump not only for its best efficiency point but for all flow rates tested.

4.2 Pump performance affected by cavitation

As the overall pump characteristics unaffected by cavitation, namely the head, shaft horse power and efficiency versus the capacity had been confirmed to satisfy the law of similarity with

* When the cavitation coefficient K_d , which is commonly used in expressing the cavitation characteristics of hydrofoils, is applied to the blade element at the impeller tip, it takes, for example, the following form:

$$K_d = \frac{H_{sv} - V_{m0}^2/2g}{W_{0t}^2/2g} \dots\dots\dots (I)$$

The parameter σ_s is related to the cavitation coefficient K_d in the following manner:

$$\sigma_s = \frac{K_d(1+\phi^2)+\phi^2}{\Psi_n} \dots\dots\dots (II)$$

respect to the pump speeds tested, each of them has been plotted in Fig. 6 by a solid line in the dimensionless quantity, the efficiency being expressed in percentage ratio to the maximum efficiency taken as 100 per cent.

Cavitation tests were carried out at constant capacities corresponding to several flow coefficients previously chosen by decreasing the value of σ_s . Fig. 5 shows the dimensionless cavitation characteristics plotted against σ_s for each flow coefficient. At the upper part in each figure, variation of efficiency has been shown by η_s which was expressed in percentage ratio to the efficiency unaffected by cavitation taken as 100 per cent at each flow rate. In the middle and the lower parts in each figure, the characteristics of the head coefficient Ψ and the shaft horse power coefficient μ have been presented respectively.

The incipient cavitation on the blade surface, which was observed clearly by the strabolight on every blade, took place practically at the same value of σ_s at each flow rate regardless of the pump speed. This critical value of σ_s has been indicated by (— · — · —) line in each figure. It can be seen from Fig. 5 that each characteristics affected by cavitation hardly showed the influence of pump speeds or heads changed in the tests. This fact was also anticipated from the cavitation test results on hydrofoils with the varying static pressure in front of them⁽⁴⁾.

From the above results, cavitation characteristics plotted against the flow rate was obtained and have been presented in Fig. 6 for three values

Fig. 6 Pump performance affected by cavitation

of σ_s in comparison with the overall characteristics unaffected by cavitation.

4.3 Cavitation aspects in the impeller and their effects on pump performance

Some of the photographs taken to investigate the cavitation aspects in the impeller have been presented in Fig. 7 (Photos. 1~10). In Fig. 8 variation of the head coefficient Ψ against σ_s has been shown at each constant flow rate. The numbers in the circles on these curves refer to the photographs in Fig. 7 indicating how the aspects of cavitation affected the Ψ characteristics. In the same figure, A-line gives the values of σ_s at which the incipient cavitation on the blade surface sets in, B-line indicating the cavitation fully developed over the blade surface. Fig. 9 illustrates various kinds of cavitation taking place in the impeller which are observed in the photographs in Fig. 7.

Photos 1 to 5 in Fig. 7 show the development of cavitation at the flow rate of $\phi=0.229$. In Photo. 1 a slender row of bubbles is seen taking place in the core of the vortex shed from the tip of the blade. This phenomenon used to appear before cavitation on the blade set in, but seemed to be too minute to have any influence on the pump performance.

Photo. 2 shows incipient cavitation on the blade suction surface near the impeller tip at a lower value of σ_s . At the same time the leakage flow at the tip clearance began to cavitate and form so called marginal cavitation⁽¹⁾ which was affected by the pressure difference across the blade. At this stage some objectionable noise accompanied, as the marginal cavitation was located near the pump casing.

With further decrease in σ_s , Photo. 3, cavitation occurred near the middle part of the blade surface, accompanied by the growing marginal cavitation. As is seen from the corresponding point in Fig. 8 and Fig. 5, the head coefficient hardly shows any change, and the horse power seems to increase slightly, the drop in efficiency being manifested at this stage of cavitation.

At the point where the head began to deteriorate, cavitation was developed over the greater part of the blade suction surface and the marginal cavitation became quite disturbing as seen in Photo. 4. The positions of the tails of cavitation on the blade suction surface, which are hardly found in the photographs, coincided roughly with the trailing end of the marginal cavitation.

Photo. 5 shows how cavitation sets in on the pressure side of the blade inlet at a reduced value

Photo. 5 (Blade pressure side)
 $\phi=0.229$, $\sigma_s=0.87$, $N=900$ r.p.m.

Photo. 4 (Blade suction side)
 $\phi=0.229$, $\sigma_s=1.05$, $N=750$ r.p.m.

Photo. 3 (Blade suction side)
 $\phi=0.229$, $\sigma_s=1.20$, $N=900$ r.p.m.

Photo. 2 (Blade suction side)
 $\phi=0.229$, $\sigma_s=1.60$, $N=600$ r.p.m.

Photo. 1 (Blade suction side)
 $\phi=0.229$, $\sigma_s=2.58$, $N=750$ r.p.m.

Photo. 10 (Blade suction side as taken from window at suction bend) $\phi=0.183$, $\sigma_s=2.12$, $N=900$ r.p.m.

Photo. 9 (Blade suction side) $\phi=0.137$, $\sigma_s=2.57$, $N=750$ r.p.m.

Photo. 8 (Blade pressure side) $\phi=0.298$, $\sigma_s=1.84$, $N=600$ r.p.m.

Photo. 7 (Blade pressure side) $\phi=0.259$, $\sigma_s=0.90$, $N=600$ r.p.m.

Photo. 6 (Blade suction side) $\phi=0.259$, $\sigma_s=0.90$, $N=600$ r.p.m.

Fig. 7 Aspects of cavitation in impeller (The photographs showing the suction and pressure sides of the blade were taken from the suction and delivery sides of the impeller, respectively.)

of σ_s with the head dropped.

The aspects of each kind of cavitation were observed to have hardly different forms with respect to the pump speeds tested, provided that the parameter σ_s and the flow coefficient were the same.

When the flow rate was increased to a value just beyond the best efficiency point, cavitation on the blade surface took place first at the zone near the middle part on the suction surface. The Photos. 6 and 7 show the states of cavitation on the suction and pressure sides of the blade taking place at a reduced σ_s -value at which the head dropped already.

With further increase in capacity, as shown in Photo. 8 cavitation took place only at the pressure side of the blade inlet, with the marginal cavitation fading out.

At reduced capacities as in Photo. 9 cavitation was apt to take place at the leading edge of the blade tip on the suction side on account of the increased incidence angle of the blade, as well as the marginal cavitation being developed with the increased pump head. Photo. 10 is an example taken from the window attached to the suction bend of the pump, showing an incipient cavitation on the blade.

Relationship between the cavitation aspects and variation in the pump performance was generally as follows:

The pump head varied very little until the cavitation developed to cover the greater part of the blade suction surface, and showed an abrupt drop with a further development. In some cases as for $\phi=0.183$ in Fig. 5 it showed a slight increase prior to the head drop.

The marginal cavitation seemed to have no apparent effect on the head characteristics. The shaft horse power showed gradual increase with the growth of incipient cavitation on the blade surfaces, and soon after the head began to drop the shaft horse power reached to a maximum value, a few per cent greater than at the normal operation, decreasing abruptly with the further drop in the head. The efficiency began to drop soon after the incipient cavitation on the blades took place, as the shaft horse power tended to increase while the head showed little change at the incipient stage of cavitation.

4.4 Variation of cavitation limits with capacity

In order to clarify variation of the cavitation limits with the pump capacity, the values of σ_s

Fig. 8 Head coefficient versus σ_s (A-line indicating cavitation incipience on blade surface and B-line fully developed cavitation over blade surface)

Fig. 9. Kinds of cavitation in impeller

which gave the various limits have been plotted in Fig. 10 against the flow coefficient. First the σ_s value which gave the incipient cavitation on the blade has been indicated, showing a minimum at the shockless entry flow rate ($\phi \approx 0.27$) and increasing at both sides. The minimum σ_s occurred a little beyond the b.e.p. for some angle of incidence generally accompanied at the b.e.p. The fine solid line shows the σ_s value at which the marginal cavitation was developed over the blade tip. The variation of σ_s -value which gave one per cent drop in the efficiency ratio η_s is presented by the dotted line as a limit detectable from performance tests. The heavy solid line indicates the σ_s -value which gave the limit of the pump head as taken at the point where the head began to fall (If Ψ showed a slight increase prior to the drop, the σ_s -value giving the maximum Ψ was taken).

Fig. 11 shows the cavitation limits expressed in terms of the suction specific speed explained in Section 4.1. As seen in the figure, the maximum value of each limit occurred approximately at the shockless entry flow rate.

Fig. 10 Variation in critical values of σ_s versus flow rate

Fig. 11 Variation in critical values of suction specific speed

As seen from Figs. 10 and 11, each critical value lies close to each other at or near the shockless entry flow rate but tends to depart from each other at both sides.

5. Conclusions

A series of cavitation tests on an axial flow pump was made at various flow rates. The parameter σ_s to represent the suction condition for any flow rate was introduced. Through the tests the relation between the conditions of cavitation and the performance variation was clarified by observing the state of cavitation using a stroboscope. Cavitation occurred at the suction or pressure sides of the blade surface and at the tip clearance according to the flow rate and suction conditions. The pump head varied very little until cavitation was developed considerably and it suddenly dropped when the cavitation occurred on the greater part of the blade suction surface. The shaft horse power, however, began to increase slightly when the cavitation set in on the blade surfaces, decreasing abruptly after reaching to a few per cent larger value than at the normal

operation.

In the scope of the tests, it was found that if the suction condition represented by σ_s -value was the same, cavitation characteristics of the pump showed similar features at a given flow coefficient regardless of the pump speeds tested. The limiting values of σ_s for cavitation inception and pump performance were obtained as seen in Fig. 10. Fig. 11 shows the same limits expressed in terms of the suction specific speed.

The axial flow pump performance against cavitation, however, is generally considered to be

affected by several factors, such as the number of blades, radial distribution of head at the impeller, the blade element and the pump dimensions. Each case needs to be studied in the future.

References

- (1) A. J. Stepanoff: *Centrifugal and Axial Flow Pumps* (1948 John Wiley & Sons), Chapter 12.
- (2) R. D. Bowermann: *Trans. ASME*, Vol. 78, (1956), p. 1723.
- (3) G. F. Wislicenus: *Fluid Mechanics of Turbomachinery* (1947 McGraw-Hill), Chapter 4.
- (4) F. Numachi and I. Chida: *Rep. Ist. H. S. Mech.*, Vol. 5, No. 42 (1955), p. 37.