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A Study on the Improvement of Performance of the Axial-Flow Pump*

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The axial-flow pump was examined from the point of view of cavitation characteristics. The studies clarified the relationship between specific speed N_s and impeller diameters. According to this relationship, the suction performance of the axial-flow pump was expected to be improved by making its inlet diameter smaller than that of the outlet.

Impellers of axial- and mixed-flow types, of the same design point and the same mean outlet diameter, were tested to prove the above supposition. The experiments revealed the following: in pump performance the mixed-flow impeller showed higher efficiency by 1 to 2% at normal flow and by about 5% at partial flow, and the shut-off horsepower decreased in proportion to the square of inlet diameter in the case where the mean outlet diameter was kept the same. In suction performance the mixed-flow impeller was superior in the incipience, except for K_1 cavitation, and showed about 10 to 20% higher critical S -values for drop in head and efficiency as compared with the axial-flow.

1. Introduction

Under the present trends there is a prevailing demand for making machines and equipment smaller and lighter. In the field of pumps this naturally calls for units with higher rotating speeds (rpm) and specific speeds. The axial-flow pump, wellknown for many years as a high specific speed type, has been used widely because of its simple construction and smaller dimensions when compared with low specific speed types. Recently, however, the mixed-flow pump has been taking the place of the axial-flow type because it has better suction performance, smaller shut-off power ratio and higher efficiency at partial flows. Specific speed of the mixed-flow pump is 700 to 1200 (m^3/min , m, rpm) which is very low in contrast with the specific speed of 1000 to 2000 of the axial-flow type. Accordingly the mixed-flow type results in a bigger and heavier unit than the axial-flow type for equivalent service conditions. Therefore it can be readily seen why, because of smaller size and lower cost, the prevailing demand is for the axial-flow high specific speed type pumps.

The axial-flow impeller was examined from the viewpoint of suction performance. This report clarifies, first, the relationship between specific speed and impeller diameter, and, second, shows

that high specific speed pumps which have so far been confined to the axial-flow type can be expected to be improved in weak points by designing as modified mixed-flow type without change in specific speed. With confirmation tests it was shown further that the mixed-flow type, high specific speed pumps had smaller shut-off power ratio and higher efficiency as well as better suction performance than the axial-flow type.

Nomenclature

A :	impeller outlet area	m^2
D :	impeller diameter	m
d_n :	impeller inlet hub diameter	m
g :	gravitational acceleration	$=9.8 \text{ m/sec}^2$
H :	total head	m
H_{sv} :	available net positive suction head	m
Δh :	required net positive suction head	m
k :	reduction factor for impeller inlet area	$=1-(d_n/D_{1a})^2$
k_d :	cavitation parameter	$=(H_{sv}-v_{m0}^2/2g)/(w_{0a}^2/2g)$
k_u :	design constant	$=u_2/\sqrt{2gH}$
L :	power input	kW
n :	pump speed	rpm
N_s :	specific speed	$=nQ^{1/2}/H^{3/4}$
Q :	capacity	m^3/min
S :	suction specific speed	$=nQ^{1/2}/H_{sv}^{3/4}$
s :	blade thickness	m
t :	pitch of blade	m

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- u : peripheral velocity m/sec
 v : absolute velocity m/sec
 w : relative velocity m/sec
 β : angle between $-u$ and w
 γ : specific weight of water kg/m³
 η : efficiency %
 η_h : hydraulic efficiency
 ϕ : flow coefficient $= v_{m2}/u_2$
 ψ : head coefficient $= gH/u_2^2$
 λ_1, λ_2 : coefficient of velocity head
 μ : power coefficient $= 102 gL/\gamma Au_2^3$
 ν : hub ratio
 σ : blade thickness measured in peripheral direction $= s/\sin \beta$ m

Subscripts

- 0: just upstream of blade inlet
 1: at blade inlet edge
 2: (when used singly) vane outlet at geometrical mean diameter (when used with another subscript) blade outlet
 a: blade tip
 i: impeller hub
 m: meridional component
 u: peripheral component

2. Specific speed and impeller diameters

The general range of specific speed of mixed-flow pumps extends from 700 to 1200, and that of axial-flow pumps from 1000 to 2000. However there are little data available on these ranges. The relationship between specific speed and impeller diameter will be clarified through calculating the optimum inlet and outlet diameters for a pump at a design point; where $Q = 60 \text{ m}^3/\text{min}$, $H = 1 \text{ m}$

Fig. 1 Velocity triangles at impeller inlet

Fig. 2. Impeller

and $n = n \text{ rpm}$.

2.1 Impeller inlet diameter D_{1a} The velocity triangle at impeller inlet tip gives absolute inlet velocity v_{m0} (in cases of no prerotation and no inlet deviation angle⁽¹⁾), as follows:-

$$v_{m0} = v_{m1} \left(1 - \frac{\sigma_1}{t_1} \right) = u_{1a} \left(1 - \frac{s_1}{t_1 \sin \beta_{1a}} \right) \tan \beta_{1a}$$

$$= \frac{\pi n}{60} D_{1a} \left(1 - \frac{s_1}{t_1 \sin \beta_{1a}} \right) \tan \beta_{1a} \quad \dots (1)$$

Capacity can be written as

$$\frac{Q}{60} = \frac{\pi}{4} D_{1a}^2 k v_{m0} \quad \dots (2)$$

By combining Eqs. (1) and (2), D_{1a} can be expressed as

$$D_{1a} = \sqrt[3]{\frac{4Q}{\pi^2 k n \tan \beta_{1a} \left(1 - s_1/(t_1 \sin \beta_{1a}) \right)}} \quad \dots (3)$$

where β_{1a} , on condition of minimum Δh -value, can be obtained as follows:-

According to the results obtained through the cavitation tests on mixed-flow impellers by the author et al.⁽¹⁾, Δh is expressed as

$$\Delta h = \lambda_1 \frac{w_{1a}^2}{2g} + \lambda_2 \frac{v_{m0}^2}{2g} \quad \dots (4)$$

Assuming $1 \gg \sigma_1/t_1$, Eq. (4) becomes

$$2g \Delta h = \lambda_1 (v_{m1}^2 + u_{1a}^2) + \lambda_2 \left(1 - 2 \frac{\sigma_1}{t_1} \right) v_{m1}^2$$

$$= \left\{ \lambda_1 + \lambda_2 \left(1 - 2 \frac{\sigma_1}{t_1} \right) \right\} v_{m1}^2 + \lambda_1 u_{1a}^2 \quad \dots (5)$$

Practically $\beta_{1a} \leq 20^\circ$, so that $\beta_{1a} \approx \tan \beta_{1a} \approx \sin \beta_{1a}$, hence,

Fig. 3 Variation of optimum inlet angle

$$v_{m1} = u_{1a} \tan \beta_{1a} = u_{1a} \beta_{1a}$$

Accordingly Eq. (5) can be converted to:-

$$2g\Delta h = u_{1a}^2 \left\{ (\lambda_1 + \lambda_2) \beta_{1a}^2 - \frac{2\lambda_2 s_1}{t_1} \beta_{1a} + \lambda_1 \right\} \\ = \left(\frac{\pi n D_{1a}}{60} \right)^2 \left\{ (\lambda_1 + \lambda_2) \beta_{1a}^2 - \frac{2\lambda_2 s_1}{t_1} \beta_{1a} + \lambda_1 \right\} \quad (6)$$

Assuming $3D_{1a}^2 \gg d_n^2$, Eq. (3) gives β_{1a} and $d\beta_{1a}/dD_{1a}$ as follows:-

$$\beta_{1a} = \frac{4Q}{\pi^2 n D_{1a} (D_{1a}^2 - d_n^2)} + \frac{s_1}{t_1} \quad (7)$$

$$\frac{d\beta_{1a}}{dD_{1a}} = -\frac{12Q}{\pi^2 n (D_{1a}^2 - d_n^2)^2} = -\frac{12Q}{\pi^2 n D_{1a}^4 k^2} \quad (8)$$

Differentiating Eq. (6) with respect to β_{1a} and letting it be equal to zero,

$$\left(1 - \frac{k}{3}\right) (\lambda_1 + \lambda_2) \beta_{1a}^2 - \left(\lambda_1 + 2\lambda_2 - \frac{2k\lambda_2}{3}\right) \frac{s_1}{t_1} \beta_{1a} \\ + \lambda_2 \left(\frac{s_1}{t_1}\right)^2 - \frac{k}{3} \lambda_1 = 0 \quad (9)$$

Finally the optimum β_{1a} can be computed from Eq. (9). In a special case of $s_1/t_1=0$, β_{1a} can be expressed approximately as follows:-

$$\beta_{1a} \approx \sqrt{\frac{k\lambda_1}{2(\lambda_1 + \lambda_2)}} \quad (10)$$

Substitution of the optimum β_{1a} obtained from Eq. (9) into Eq. (3) yields the optimum impeller inlet diameter D_{1a} .

Fig. 3 shows the relationship between λ_1 and the optimum β_{1a} for the case where $\lambda_2=1.0$, $k=0.9$ and 0.94 , $s_1/t_1=0.02$, 0.03 and 0.04 . The impeller inlet diameter D_{1a} is shown in Fig. 5 for the previously mentioned design point, where $k=0.94$, $s_1/t_1=0.03$, $\lambda_1=0.3$, 0.25 , 0.2 and 0.15 .

2.2 Impeller outlet diameter D_{2a} The impeller outlet diameter D_{2a} can be obtained by assuming a velocity triangle for a finite number of blades at the geometrical mean outlet diameter. The velocities for this triangle are expressed with prime (') hereafter.

When the flowline lies on a cylindrical surface*

Fig. 4 Velocity triangles at impeller outlet

* When the flowline does not lie on a cylindrical surface at the impeller outlet, D_{2a} becomes larger, and D_{2i} smaller than for the above mentioned case. D_{2a} is compared with D_{1a} herein so that the above assumption is taken for simplicity.

at the impeller outlet, the impeller outlet diameter D_{2a} can be expressed as

$$D_{2a} = D_2 \sqrt{\frac{2}{1 + \nu_2^2}} \quad (11)$$

where

$$D_2 = \frac{60}{\pi n} u_2 \quad (12)$$

and

$$u_2 = k_u \sqrt{2gH} \quad (13)$$

Substituting Eqs. (12) and (13) into Eq. (11),

$$D_{2a} = \frac{60}{\pi n} k_u \sqrt{\frac{4gH}{1 + \nu_2^2}} \quad (14)$$

where k_u can be obtained as follows.

$$\tan \beta_2' = v_{m2} / (u_2 - v_{u2}') \quad (15)$$

$$v_{u2}' = gH / (u_2 \eta_h) \quad (16)$$

$$Q/60 = (\pi/4) D_{2a}^2 (1 - \nu_2^2) v_{m2} \quad (17)$$

By combining Eqs. (13) through (17), the following relation can be obtained.

$$N_s^2 = \frac{60^3}{2\pi} (2g)^{3/2} \frac{1 - \nu_2^2}{1 + \nu_2^2} k_u \left(k_u^2 - \frac{1}{27\eta_h} \right) \tan \beta_2' \quad (18)$$

Consequently D_{2a} -value can be computed from Eqs. (14) and (18) with an assumption of β_2' -, ν_2 - and η_h -values.

The D_{2a} -values are shown in Fig. 5 for the previously mentioned design point, where $\nu_2=0.5$, $\eta_h=0.88$, $\beta_2'=20^\circ$, 23° and 26° .

2.3 Specific speed and impeller diameters

As shown in Fig. 5, D_{2a} is larger than D_{1a} , except in the case where an unusually large outlet angle ($\beta_2' > 26^\circ$) is assumed, and a small value $\lambda_1 (< 0.1)$ is taken in Eq. (4) (e.g., very low λ_1 can be taken for the critical point of drop in head if due consideration is paid to the blade design.). Since a large part of the shut-off horsepower is consumed in circulating eddy on the suction side of the impeller in case of an axial-flow pump⁽²⁾,

Fig. 5 N_s versus impeller diameter

it is not desirable to make D_{1a} unnecessarily large.

Accordingly, by modifying a high specific speed pump, which has been so far designed as an axial-flow type, the suction performance can be expected to be improved and the shut-off horsepower to decrease.

3. Test apparatus and procedure

For the purpose of confirming the suppositions stated above, experiments were carried out on pumps of two types, axial- and mixed-flow, which were designed for the same design point and with the same geometrical mean outlet diameters and outlet angles.

Fig. 6 shows the testing floor layout. Figs. 7 and 8 show the test pumps of axial- and mixed-flow types respectively. Fig. 9 shows the test impellers. Tables 1 and 2 give the dimensions of the test impellers.

Suction and total heads were obtained from pressures measured at suction and discharge pressure taps as shown in Fig. 6. The pump capacities

Table 1 Dimensions of axial-flow impeller

Tip dia. of imp.	356 mm	Hub diameter	172 mm		
Hub-tip ratio	0.483	Number of imp. blades	4		
Flowline	I	II	III	IV	
Diameter	356 mm	296 mm	236 mm	172 mm	
Spacing-chord ratio	1.47	1.34	1.245	1.11	
Thickness ratio	4.21%	5.20%	6.71%	9.02%	
Blade angle	18°20′	22°05′	27°35′	37°00′	

The hydrofoils used were derived from Göttingen 428.

Fig. 6 Testing floor layout

Fig. 7 Sectional view of axial-flow pump

were measured with a Venturi flowmeter which had been calibrated in position in the test loop prior to the tests. The pump speed was obtained by a generator type tachometer calibrated by an E-PUT meter and the shaft torque was measured with a strain-gauge torquemeter. A transparent cylinder surrounding the impeller made it possible to observe cavitation. A strobolight with a spark duration period of 2.5×10^{-5} seconds was used for visual observations.

A jet-ejector was used for reducing the suction pressure to provide desired suction lift conditions. The dissolved air content of water was 8 to 9 ppm.

4. Test results

4.1 Overall pump performance Overall performance curves for the test impellers under non-cavitating conditions are indicated in Fig. 10. In this figure the performance curves for a supplemental mixed-flow impeller are indicated for the

Table 2 Dimensions of mixed-flow impeller

Inlet tip dia. of imp.	320mm	Outlet tip. dia. of imp. 348 mm			
Outlet hub dia. of imp.	188 mm	Number of imp.blades 4			
Flowline	I	II	III	IV	
Inlet diameter	320 mm	270 mm	208 mm	128 mm	
Outlet diameter	348 mm	305 mm	254 mm	188 mm	
Spacing-chord ratio	1.47	1.24	1.10	0.97	
Thickness ratio	3.65%	4.10%	5.42%	7.82%	
Maximum camber	2.92%	3.44%	4.04%	5.08%	
Blade angle	21°50′	23°50′	29°00′	41°50′	

The blade inlet and outlet angles were determined by Kováts' method⁽³⁾, while the hydrofoils were derived by modifying Göttingen 428 as required to meet the camber angles.

Fig. 8 Sectional view of mixed-flow pump

Mixed-flow type Axial-flow type
Fig. 9 Test impellers

purpose of showing the relationship between inlet diameter and shut-off horsepower. The supplemental impeller was also designed for the same design point and with the same geometrical mean outlet diameter and outlet angle as the prime impellers. Since the same guide vanes were used for both mixed-flow impellers, the flow rates at the best efficiency points were not equal. Dimensions for the supplemental impeller are shown in Table 3. Variations of shut-off power coefficient μ_s versus inlet diameters are given in Fig. 11.

The following remarks are set forth regarding Figs. 10 and 11.

I) Both mixed-flow impellers showed slightly higher values of maximum efficiency, but at partial flows about 5% higher efficiency than for the axial-flow type. Roundness of efficiency curves increased remarkably.

The efficiency increase is considered to be due to reduction of deceleration loss through blades of

Table 3 Dimensions of supplemental mixed-flow impeller

Inlet tip dia. of imp.	294mm	Outlet tip dia. of imp. 346mm			
Outlet hub dia. of imp.	192 mm	Number of imp. blades 4			
Flowline	I	II	III	IV	
Inlet diameter	294mm	248mm	192mm	124mm	
Outlet diameter	346mm	304mm	255mm	192mm	
Spacing-chord ratio	1.47	1.235	1.11	1.00	
Thickness ratio	3.70%	4.25%	5.70%	8.20%	
Maximum camber	3.26%	3.32%	3.42%	4.70%	
Blade angle	22°55'	26°10'	31°40'	43°30'	

The hydrofoils were determined in the same way as indicated in Table 2, below.

Fig. 10 Performance characteristics of impellers

the mixed-flow impellers resulting from decreased inlet relative velocities, Fig. 12. Increase in efficiency is most noticeable at partial flows.

II) The shut-off horsepower decreased noticeably for the mixed-flow impellers. As shown in Fig. 11 the shut-off horsepower changed in proportion to the square of inlet diameter in the case where the geometrical mean outlet diameter was kept the same ($\tan 63^\circ 25' = 2.00$). A considerable portion of the shut-off horsepower is consumed in circulating eddies on the suction and discharge sides of the impeller, as mentioned before. The variation shown in Fig. 11, therefore, can be attributed to the reduced power loss at the impeller inlet which is proportional to the square of inlet peripheral velocity where the same mean outlet diameters are used.

4.2 Suction performance

4.2.1 Critical points for cavitation inception

The variation of critical value of k_d for cavitation inception is shown against flow rates in Fig. 13. Classification of cavitation is explained in reference (1).

As seen in Fig. 13 the inception curves for K_1 and K_3 cavitation differ entirely for both types of impellers, although the same hydrofoil section was applied. The inception curves of K_1 and K_3

Fig. 11 Variation of shut-off power coefficient with decreasing inlet diameter

Fig. 12 Velocity triangles at impeller inlet

cavitation for the axial-flow impeller were given as a continuing curve whereas two different curves were given for the mixed-flow impeller.

The mixed-flow impeller gives smaller values of k_d near $\phi=0.35$ and slightly larger values at $\phi<0.33$ than for the axial-flow type. As for K_2 and K_4 cavitation, both types of impellers show similar inception curves.

4.2.2 Critical points for drop in head and efficiency A series of tests were made by reducing the suction pressure corresponding to different values of k_d , at constant speeds and capacities to observe the change in total head, shaft horsepower and efficiency. The results against k_d are presented in Fig. 14. Critical values of k_d at which the efficiency began to drop* and those at which the total head began to fall** were determined at several

flow rates, and the results are shown in Fig. 15.

In representing the pump suction performance, Δh -value and suction specific speed S are preferred in practice. The critical values of S for drop in head and efficiency against flow rate are given in Fig. 16.

Fig. 13 Critical value of k_d for cavitation inception

Fig. 15 Critical value of k_d for drop in head and efficiency

Fig. 16 Critical value of S for drop in head and efficiency

Fig. 14 Variation of efficiency, head and power coefficient with decreasing cavitation parameter

* The values of k_d were taken at the point where the efficiency dropped 0.5% from the noncavitating efficiency.

** The values of k_d were taken at the point where the total head dropped 0.5%; in case the head showed a slight rise, the point selected was where the head dropped again to the noncavitating value.

Study of Figs. 15 and 16 reveals comparison between impellers as follows:-

I) Regarding the critical values of k_d for drop in head and efficiency, both types of impellers gave nearly the same values. This is thought to be due to the independence of the critical k_d -values on the type of impeller, when the impeller blade angles and the spacing chord ratios are kept the same.

II) With respect to the critical values of S for drop in head and efficiency, both types of impellers showed different values. As the cavitation parameter k_d is made dimensionless with w_{0a} , the same value of k_d corresponds to different values of S for the two types of impellers owing to a difference in their values of w_{0a} , i.e., their inlet diameters. The mixed-flow impeller showed about 10 to 20 % higher values of S than the axial-flow, and proved its superiority with respect to suction performance.

5. Conclusion

The study was conducted to determine performance improvements of the axial-flow pump. The results are summarized as follows:

I) The relationship between specific speed and impeller diameter was clarified on the high specific speed pumps which have been so far designed as axial-flow pumps; where the inlet diameter was determined with regard to cavitation performance and the outlet diameter by the discharge conditions respectively (Fig. 5).

II) Performance was expected to be improved by making the inlet diameter smaller than the outlet, i.e., mixed-flow type even for the high specific speed pumps.

III) The axial-flow and the mixed-flow pumps

were designed for the same design point and with the same mean geometrical outlet diameter, and tested for the purpose of proving the above supposition. The test results revealed that the mixed-flow type showed 1 to 2 % higher efficiency at maximum value point, about 5 % higher efficiency at partial flow and 23 to 32 % less shut-off horsepower than the axial-flow type, respectively (Figs. 10, 11).

IV) With respect to k_d -values for cavitation inception, the axial-flow impeller gave slightly lower values for K_1 cavitation at partial flow, but a little higher for K_3 . As for K_2 and K_4 , both impellers showed about the same values (Fig. 13).

V) Regarding the critical values of k_d for drop in total head and efficiency, both types of impellers revealed almost the same values. The critical values of S , however, were different for both impellers due to the difference in their inlet diameters, and the mixed-flow type showed about 10 to 20 % higher values than the axial-flow type (Figs. 15 and 16).

Thus, the mixed-flow impeller proved its superiority over the axial-flow type, in both overall and cavitation performance.

Acknowledgment

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References

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Discussion

T. TOYOKURA: (1) J. Holba et al. studied a method of deriving D_{1a} which makes Δh minimum, by expressing Eq. (4) as a function of D_{1a} only. This method seems simpler.

(2) D_{1a} and D_{2a} are functions not only of N_s but also of Q , H , n . Therefore, Fig. 5 seems inappropriate.

(3) I think λ_1 and k_d are identical, physically, but they are treated by using w_{1a} or w_{0a} separately, in this paper. It would be more natural to express them by the same relative velocity.

(4) It is to be desired that a cavitation test be made on the impeller shown in Table 3, and that the optimum inlet diameter of impeller be

derived experimentally.

H. MURAI: (5) It is considered insufficient to determine the inlet diameter of the diagonal-flow impeller, which has a comparatively small number of blades and such sections of aerofoil profiles as adopted in this experiment, by using only the one-dimensional analysis, Eqs. (1) and (2).

(6) The correction of profile forms of blade sections might be referred to a three-dimensional analysis.

T. KYOGOKU: (7) The debater thinks that the assumptions $\beta_{1a} \approx \tan \beta_{1a} \approx \sin \beta_{1a}$ and $3D_{1a}^2 \gg d_n^2$ lead to a great error. The author differentiates

Eq. (6), taking λ_1 to be a constant. Is it not necessary to study the relation between λ_1 and β_{1a} , D_{1a} , etc?

(8) The author says that there is a proportional relation between shut-off horsepower and D_{1a}^2 . The debater would like to know the mechanism of input consumption at shut-off operation and the effects of other elements.

Author's closure

The author takes this opportunity to express his appreciation and thanks for the constructive discussions submitted by Doctors Toyokura and Murai and Mr. Kyogoku.

Doctor Toyokura pointed out that the determination of D_{1a} giving the minimum Δh without using β_{1a} as a parameter had already been introduced by J. Holba.

The blade blockage effect can be expressed as $t_1/(t_1 - \sigma_1) = t_1/\{t_1 - s_1/(\sin \beta_{1a})\}$ (19)

It is a function of β_{1a} , therefore, of D_{1a} . From this standpoint the author took β_{1a} for a parameter in expressing Δh , as shown in Eq. (6), whereas Holba considered the effect to be constant.

The values of D_{1a} and D_{2a} are not direct functions in themselves of N_s , as Doctor Toyokura indicated, but those for the design point: $60 \text{ m}^3/\text{min} \times 1 \text{ m} \times n \text{ rpm}$ might well be considered to show their relation to N_s by the similitude law. As Doctor Toyokura suggested, the ratio D_{1a}/D_{2a} is expressed as a function of N_s combining Eqs. (3), (4) and (18), as

$$\frac{D_{1a}}{D_{2a}} = \sqrt[3]{\frac{\pi N_s^2}{2 \times 60^3 \times k \tan \beta_{1a} \{1 - s_1/(t_1 \sin \beta_{1a})\}}} \sqrt{\frac{1 + \nu_2^2}{g k u^2}} \text{(20)}$$

Because of the many factors to be assumed in the equation, the values of D_{1a} and D_{2a} are given separately in the paper.

In section 2, equations are developed from Eq. (4), of which the rationality is shown in reference (1), taking into account the blade blockage effect and the case of low- N_s pump impellers. In high- N_s pump impellers, however, full application of Eq. (4) might be somewhat questionable owing to large inlet deviation angles and hydrofoils used for blade sections. Under these conditions the test results were given in terms of w_{0a} as is the case for ordinary cascade.

It was one of the objectives of this study to justify the expression of Eq. (4), i.e., to show the constancy of k_d -values for drop in total head and efficiency. Mr. Kyogoku was of the opinion that k_d could not be considered to be constant for all

types of pumps. The study, however, indicates that k_d can be considered to be constant. On this point the author considers that the prime aim has been attained. The study did not include investigation of optimum inlet diameters.

With regard to Doctor Murai's first remark some questions remain on Δh in case of obtaining the relationship between N_s and D_{1a} without starting from the one-dimensional equations such as Eqs. (1) and (2), when Δh is expressed by the relative and the absolute velocities as in Eq. (4).

The value Δh should be written, then, as

$$\Delta h = \lambda \frac{w_{\infty a}^2}{2g} + \frac{v_{m0a}^2}{2g} \text{(21)}$$

where $w_{\infty a}$ represents mean relative velocity at tip and v_{m0a} absolute inlet velocity at tip.

It is not proper to apply the results on single-hydrofoils or on straight hydrofoil cascades without tip clearance. However, complete test results on axial-flow impellers are available, so that any questions regarding the value of λ can be answered with available data.

From the viewpoint that an impeller is considered a three-dimensional cascade, Δh should be written with $w_{\infty a}$ as in Eq. (21). $w_{\infty a}$ is obtained rather easily for an axial-flow impeller, whereas calculation is very difficult for a mixed-flow impeller (the impeller is not limited to the axial-flow type in the case of giving the correlation of N_s to D_{1a}) because $w_{\infty a}$ is connected with the outlet diameter, total head, outlet and inlet velocities.

Even if $w_{\infty a}$ were replaced by $w_{0a} = (u_{1a}^2 + v_{m0a}^2)^{1/2}$, v_{m0a} would not be equal to the mean absolute inlet velocity. v_{m0a} would be affected greatly by the configuration of impeller and suction casing.

It may be comparatively easy to obtain the optimum inlet diameter for a given design point from the above three-dimensional relations and equations by a trial-and-error method. It is, however, much involved to state the general relationship between N_s and D_{1a} . Besides, the effect of tip clearance cavitation should be taken into account in high specific speed pumps. The effect as yet has not been clarified sufficiently.

Under these circumstances, w_{1a} , v_{m0} and Δh are written as Eqs. (1), (2) and (4), for simplicity in this study.

Although there are some minor discrepancies in ϕ -values for the points of $\eta_{\max}$ and $k_{d \min}$ Doctor Kovats' theory is considered to be one of the most satisfactory methods for modification of the hydrofoil section. The discrepancies are thought to be due to the inlet angle of the guide vanes used in

the tests which affects the ϕ -value for $\eta_{\max}$ point and the different v_{m0} -values along the impeller inlet edge under actual flow conditions.

The value of $k_d \min$ was smaller for the axial-flow impeller than for the mixed-flow type, as shown in Fig. 15. This is connected also with the velocity distribution along the impeller inlet edge, and further with the expression of cavitation parameter as

$$k_d = \frac{H_{sv} - v_{m0}^2/2g}{w_{0a}^2/2g}$$

where the second term of the numerator is given with the mean velocity along impeller inlet edge.

In a strict sense, therefore, the hydrofoil section should be modified by the two- or three-dimen-

sional method, as pointed out by Doctor Murai.

Eq. (9) is developed under the assumption of

$$\beta_{1a} = \tan \beta_{1a} = \sin \beta_{1a}, \quad 3D_{1a}^2 \gg d_n^2$$

Usually $\beta_{1a} \leq 20^\circ$ so that the error under the assumption does not exceed +2% or -4%; d_n/D_{1a} on the other hand is about 0.3 and $d_n^2/3D_{1a}^2$ becomes 0.03. Thus, the errors are considered to be negligible.

As for the shut-off horsepower, it should be mentioned that it varied in proportion to the square of inlet diameter within the range of this study. The shut-off horsepower is affected by many factors as is wellknown, and its correlation with inlet diameter should be adjusted for pumps with different N_s .

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Study of Prototype 2-Stage Jet Type Pump*

(1st Report, Characteristic Tests)

By Mitsuhiro HOSHI**, Iwao YOSHIKAWA***, and Koshiro SAKAI**

The jet pump works by means of tendency of a stream or jet of fluid to drive or carry contiguous particles along with it. In a jet pump the maximum possible realized suction capacity Q_s will be twice the driving capacity Q_0 , because of limitations in area ratio between nozzle and throat.

In the case of the newly designed 2-stage jet pump, the suction openings of the 1st and 2nd stage are connected on an even water level. Further, suction water of the 1st stage is also utilized for driving the 2nd stage. One primary objective of designing this trial pump was to achieve $Q_s/Q_0=4$.

Changes in construction design, based on experimentation, were incorporated in the test pump unit on which test results were obtained, as follows:-

- (1) $Q_s/Q_0 > 4$ was obtained as anticipated; further efficiency is favorable.
- (2) The pump without a diffuser in the first stage outlet showed higher performance than that of a pump having a diffuser.
- (3) The function of the 1st stage was not influenced by opening or closing the discharge valve. The performance of the pump was, however, affected critically by operation of the suction valve of the 1st stage.

1. Preface

In the jet pump it is known that the suction capacity Q_s does not exceed 2 times the driving

capacity Q_0 , because of limitation in area ratio between nozzle and throat. Therefore, an increase in suction capacity is held within close limitations for a given driving head, even though the driving head is much higher than the total jet pump head H_T . We could not increase suction capacity beyond these definite limitations, so we had to reduce the

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