Performance Tests of a Centrifugal Pump*

Shoichiro FUKUSAKO**
and
Mikio ARIE**

Abstract

The results of performance tests of a centrifugal pump in the cavitation region were presented in terms of the variation in head, driving power and efficiency. The effect of air leakage in the suction system was also examined. It was found, in the present experiments, that the existence of a prerotation showed its effect of improving the cavitation performance of a pump.

1. Introduction

The present paper presents some experimental results on the performance test of a centrifugal pump installed in the system of a cavitation tunnel recently constructed at the Fluid Mechanics Laboratory of the Hokkaido University.

Cavitation characteristics of the pump were first examined in terms of the reduction of total head and efficiency as well as a trend of increasing the driving shaft horse power in the cavitation region under a constant load of discharge quantity.

It is generally accepted, these days, that an introduction of a certain amount of air to the upper part of the draft tube of a reaction type of turbine, diminishes unfavorable features of the cavitation phenomena. On the other hand, the leakage of air in the suction system of a pump causes considerable hindrance in the maintainance of its function in raising liquid to the required level. With special regard to this, the influence of air leakage in the suction system on the performance of the pump was examined as a second step. It was demonstrated that air leakage is invariably accompanied by an unfavorable effect which results in a drop in head, an increase in the required shaft horse power which leads to a lowering of efficiency.

Finally, the prerotation at the suction eye of the impeller was estimated, based on an ideal velocity triangle at the inlet of the vane to investigate the relation between cavitation characteristics and the value of prerotation.

The majority of the results were prepared in acceptable dimensionless forms for the purpose of providing material for the model study of a pump.

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** Faculty of Engineering, Hokkaido University, Sapporo, Japan.

Nomenclature

G_a : flow rate of air (gr./sec)
G_w : flow rate of water (kg/sec)
H : total head of pump (mAq.)
H_a : total head of pump when air is introduced into the suction system (mAq.)
H_{sv} : net positive suctive head (mAq.)
N_i : shaft horse power (horse power)
Q : capacity of pump (m^3/sec)
g : acceleration of gravity (m/sec^2)
n : revolution of impeller (r.p.m.)
ρ : density of fluid (kg$\cdot$$sec^2$/$m^4$)
η : efficiency of pump ($\gamma QH/75N_i$)
u_1 : peripheral velocity at the inlet (m/sec)
u_2 : peripheral velocity at the exit of impeller (m/sec)
v_{m0} : average velocity of fluid in suction pipe (m/sec)
ϕ : capacity coefficient defined at the inlet (v_{m0}/u_1)
ϕ_a : flow rate ratio between air and water (G_a/G_w)
ϕ_{am} : capacity coefficient for the mixed liquid of water and air ($\phi_{am}=\phi_a\cdot\phi$)
ψ : head coefficient ($H/u_2^2/2g$)
ψ_a : head ratio (H_a/H)
ζ : shaft horse power coefficient ($N_i/\rho A_2 u_2^3/75$)
$n\sqrt{N_i}/H_{sv}^{\frac{5}{4}}$: suction specific speed

2. Experimental apparatus

A schematic view of the circulation type of water system is given in Fig. 1 to show the arrangement of water tank, centrifugal pump to be tested, electric dynamometer, vacuum pump, pipe lines and manometers. The capacity of tank is 6.63 m^3, which is sufficient to fill the tap water available at the laboratory for the whole system. The water level in the tank was carefully watched to maintain a free surface in such a way that the vacuum pressure there evacuated by a wet type vacuum pump could be adjusted to give various values of the net positive suction head to the pump. Baffle plates are provided in the tank in order to avoid the direct inflow of the cavitation bubbles into the suction system. The specifications of the 150 mm single suction type of centrifugal pump are $H=7$ m, $Q=2{\cdot}8$ m^3/min, $n=960$ r.p.m., and n_s(m$\cdot$$m^3$/sec)$=47.6$. The shroud of impeller and the casing cover at the suction side were transparent so that the cavitation bubbles in impeller could be observed by the aid of a stroboscopic flash light. The main dimensions of the impeller are shown in Table 1. All pressures were read on the mercury columns of various manometers, and the capacity of pump was measured on an orifice meter. Fig. 2 shows the main dimensions of the impeller tested.

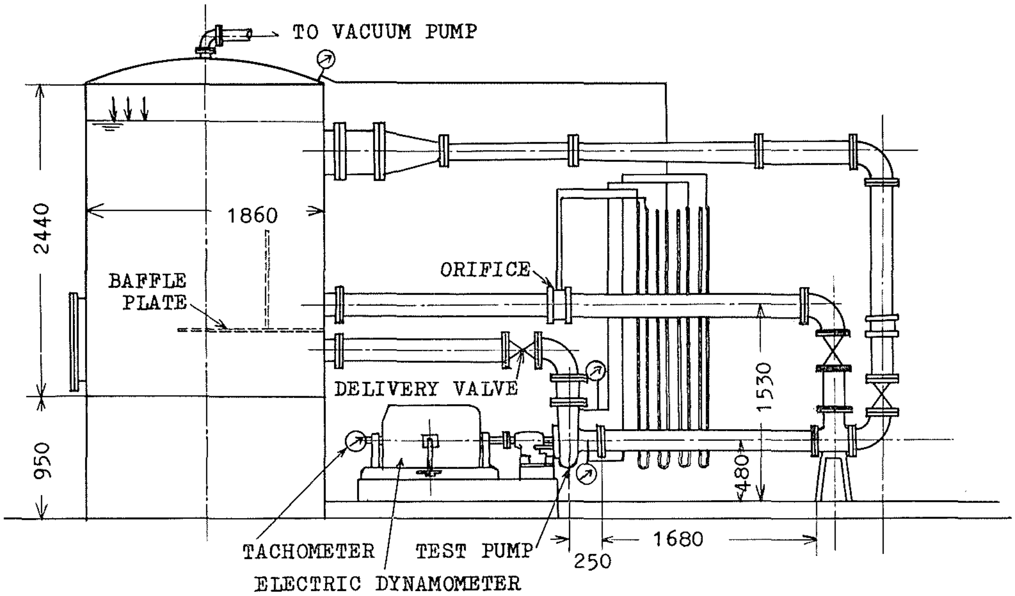


Fig. 1. A schematic view of experimental apparatus.

Table 1. Mean dimensions of the impeller tested.

exist diameter		$D_2 = 270.0$ mm
inner diameter	on the suction side shroud	$D_{1c} = 152.0$ mm
	on the impeller disc	$D_{1i} = 100.0$ mm
width of exist		$b_2 = 45.0$ mm
width at inlet		$b_1 = 65.0$ mm
vane angle at the exist		$\beta_2 = 20°0'$
angle of vane at the inlet	on the suction side shroud	25°30′
	at the midway	32°27′
	on the impeller disc	39°25′
number of vane		6

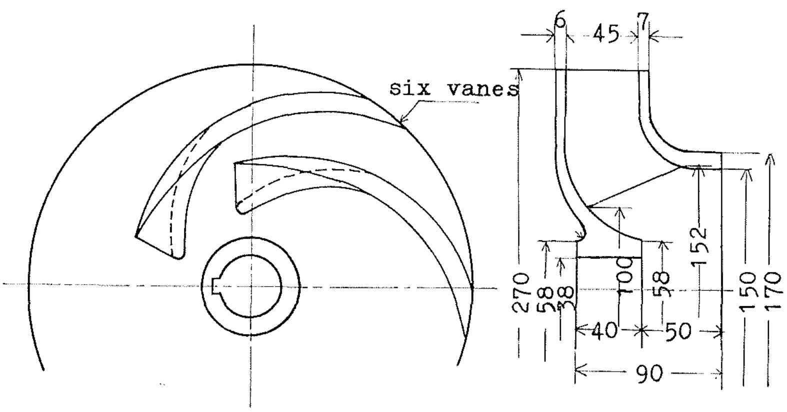


Fig. 2. Dimension of impeller.

3. Cavitation performance

Six series of experiments were first undertaken for different constant speeds of 700, 800, 900, 1000, 1100 and 1200 r.p.m. by changing the discharge quantity to examine the nature of similarity in performance curves in the cavitation-free region of practical operation. The results are plotted in Fig. 3 in terms of head coefficient, capacity coefficient, power coefficient and pump efficiency. As may be

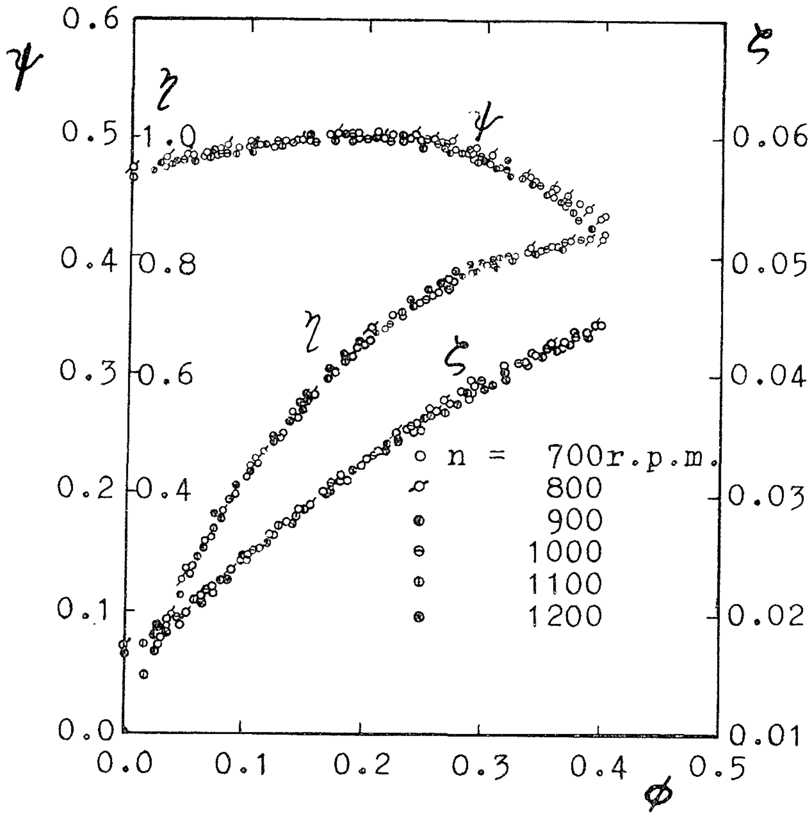


Fig. 3. Head-capacity, power-capacity and efficiency-capacity curves in normal operation.

seen in this figure, the similarity of performance can be accepted for various speeds of pump now selected. Therefore, any further results to be obtained for these pump speeds by changing the parameter concerned was expected to show the effect of this particular parameter.

The cavitation and its effect on the function of a pump may be defined in several ways: the production of cavitation bubbles somewhere inside the pump, the generation of cavitation sound and vibration, the degree of cavitation erosion on the material, the drop in head and capacity, etc. However, there is no doubt that the value of suction head governs a larger part of the initiation of these cavitation features. Fig. 4 (a) and Fig. 4 (b) are give to illustrate the change of

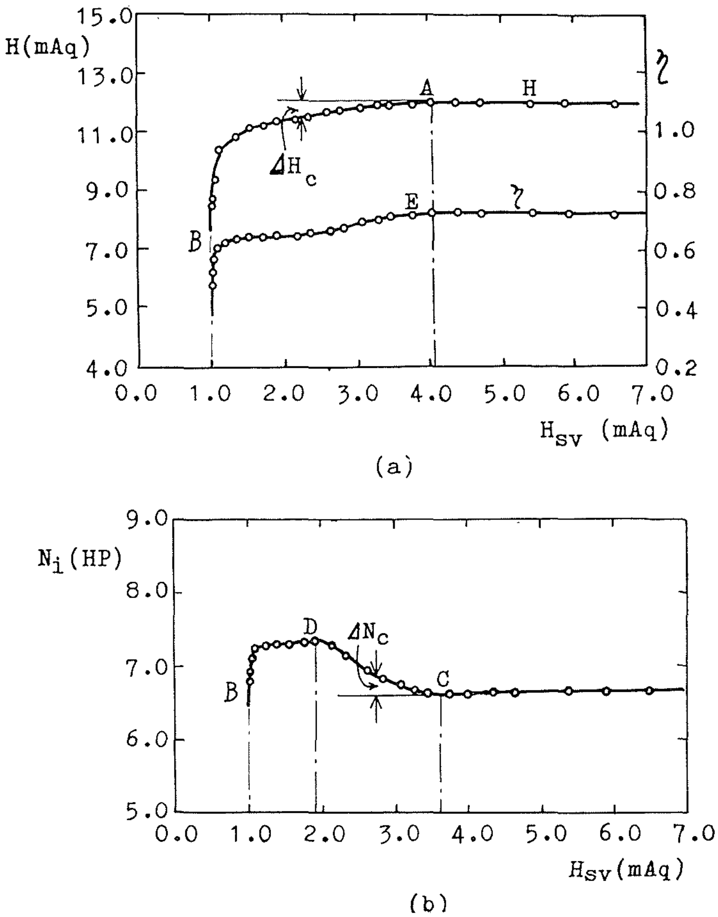


Fig. 4. Definition of the characteristic points in cavitation region ($n=1100$ r.p.m., $Q=32.533$ lit/sec).

characteristics, based on the interests of mechanical performance of the pump, obtainable by loading the same capacity of discharge for a constant speed of impeller and by changing only the suction head for the purpose of clearly exaggerating the effect of cavitation. Fig. 4 (a) is a plot of head against various values of net positive suction head. As may be seen in this figure, the head starts to lower at A when the suction head increases to a certain value and finally it falls suddenly at B where the pump fails to lift water any more. Since all the operating parametrs except the suction head are kept constant, ΔH_c may be considered to be the head loss due to cavitation. In the same way, the shaft horse power begins to increase at C in Fig. 4 (b) and attains its maximum value at D, and the pump efficiency begins to decrease at E. ΔN_c in Fig. 4 (b) may be considered to be the increase of shaft horse power due to cavitation in the same way as ΔH_c. A series of experiments were undertaken to reveal the general trend of these characteristic points for four different speeds of 900, 1000, 1100 and 1200 r.p.m. (The experiments for 700 and 800 r.p.m. are discarded, because these

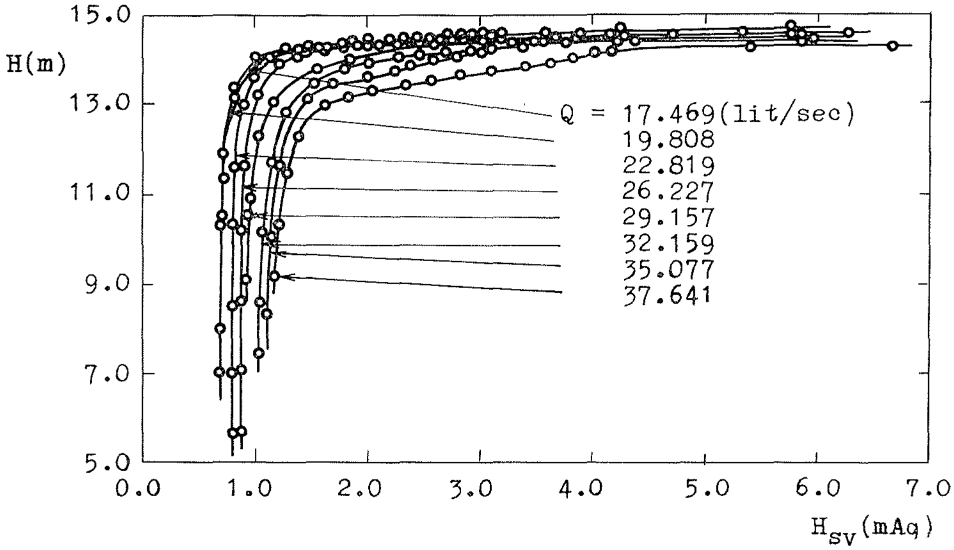


Fig. 5. Head-NPSH relation ($n=1200$ r.p.m.).

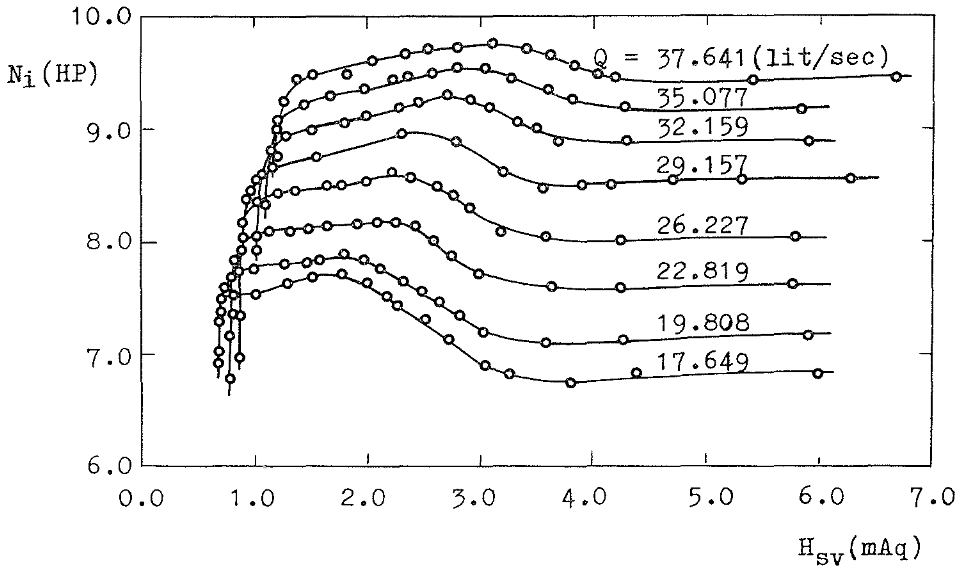


Fig. 6. Shaft horse power-NPSH relation ($n=1200$ r.p.m.).

characteristic points could not clearly be detected in the present installation). The results for 1200 r.p.m. are shown in Figs. 5, 6 and 7, as an example.

Two characteristic natures were found in these experiments in the cavitation region produced by decreasing the net positive suction head. One is a decrease in total head ΔH_c in Fig. 4 (a) that gives the loss of power $\Delta N_{ch}=\gamma \Delta H_c Q/75$ in the cavitation region, and the other is an increase in shaft horse power indicated by ΔN_c in Fig. 4 (b). The sum of these two powers ΔN_t may be considered to be the loss of power due to cavitation :

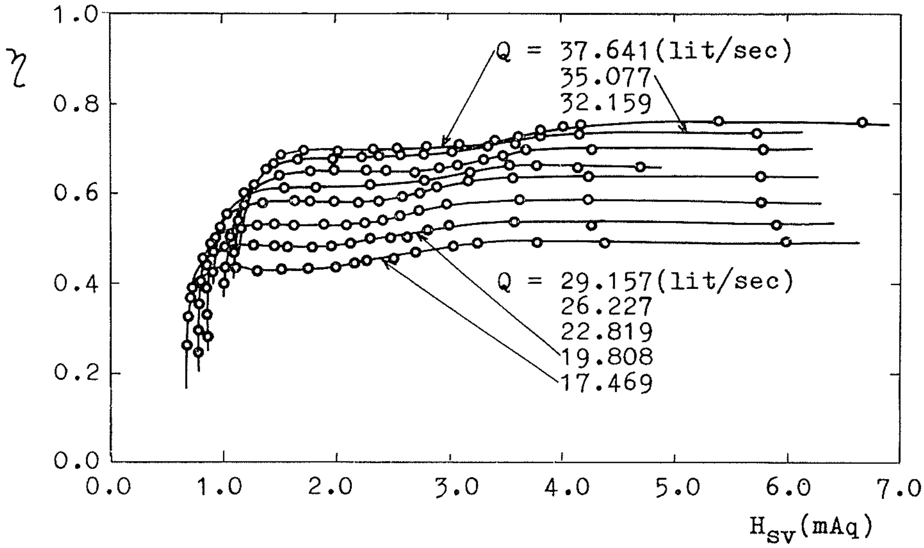


Fig. 7. Efficiency-NPSH relation ($n=1200$ r.p.m.).

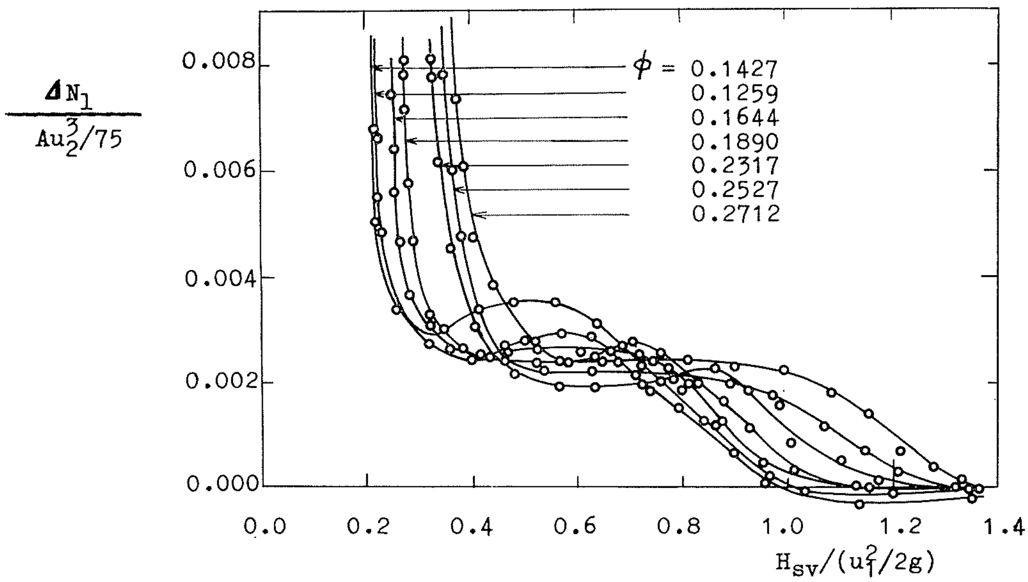


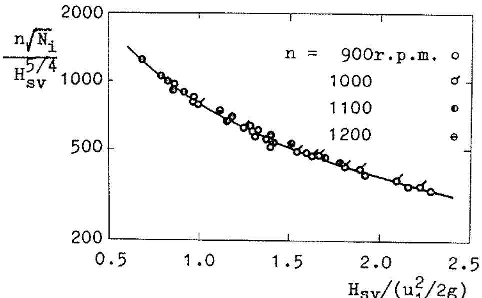
Fig. 8. Dimensionless plot of power loss vs. NPSH ($n=1200$ r.p.m.).

$$\Delta N_l = \Delta N_c + \Delta N_{ch} = \Delta N_c + \gamma \Delta H_c Q/75$$

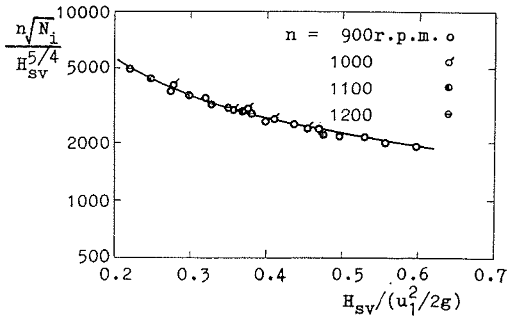
where, ΔN_c is positive when it increases and $\gamma Q \Delta H_c/75$ is positive when it decreases. The power loss of cavitation thus computed for $n=1200$ r.p.m. is shown in Fig. 8, as an example. It may be seen in this figure that the loss in the cavitation region shows a trend to increase with the decrease of the net positive suction head when the load of discharge is positively kept constant. This fact may be understood in two ways:

(a) The production of cavitation bubbles resulted in an increase of the apparent specific volume of the fluid, and hence the velocity triangles at the inlet and exit of the impeller were deformed to be accompanied by a reduction of the hydraulic efficiency.

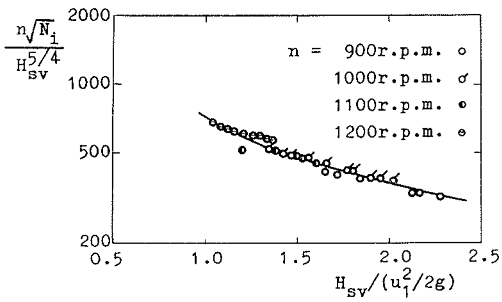
(b) Insofar as insistance of the experimental technique of keeping the discharge of the pump constant, the increment of the power exceeding the drop in head must be consumed in some way directly related with the production of cavitation bubbles.



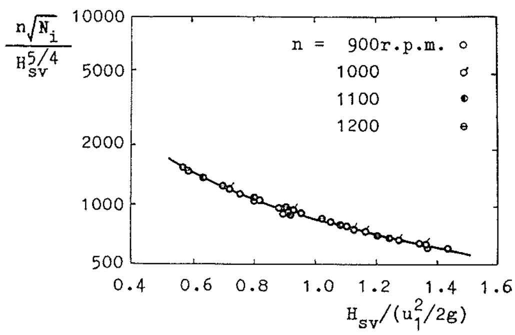
(a) The point where the head begins to drop.(A)



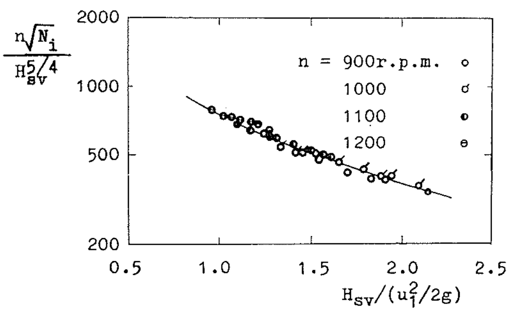
(b) The point where the pumping action stops.(B)



(c) The point where the shaft horse power begins to increase.(C)



(d) The point where the shaft horse power attains its maximum value.(D)



(e) The point where the efficiency begins to lower.(E)

Fig. 9. Suction specific speed vs. characteristic points.

Perhaps, these two processes of energy dissipation may be combined together to increase the loss of power in the cavitation region with the decrease of the net positive suction head.

The distribution of the characteristic points indicated by A, B, C, D and E is shown in Fig. 9 on a plane of dimensionless net positive suction head $H_{sv}/(u_1^2/2g)$ and suction specific speed $n\sqrt{N_i}/(H_{sv})^{\frac{5}{4}}$. Since the characteristic points on the characteristics in the cavitation region represented by A, B, C, D and E are fairly well distributed respectively on one curve, as may be seen in these figures, the mode of the plots may be useful in the estimation of the limiting value of the suction height to be given to a centrifugal pump.

4. Effect of air leakage in the suction system of a centrifugal pump on its performance

Fig. 10 shows a schematic view of the air injection device at the suction pipe. Thirteen holes of 4 mm diameter were drilled as the air injection ports in a brass tube of 9.8 mm diameter inserted into the suction pipe along its diameter. The discharge load and the speed of the pump were again kept constant as in the previous case of cavitation test and a certain amount of constant weight of air was given through these holes from the surrounding atmosphere into the pump for each series of experiments to measure the total head and the driving horse power for various values of suction head.

The experimental results obtained for $n=1200$ r.p.m. are shown in Figs. 11, 12 and 13, as examples with respect to the variations of head, shaft horse power and efficiency, respectively. The relation between the head ratio H_a/H and the

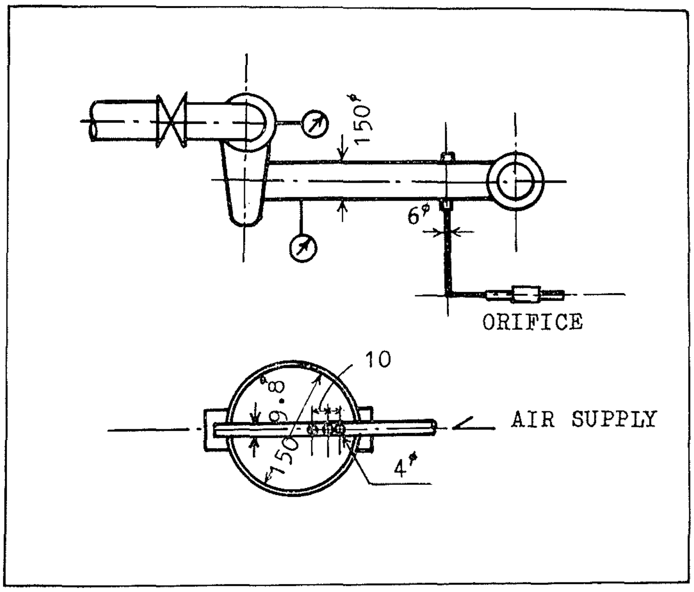


Fig. 10. Air injection device.

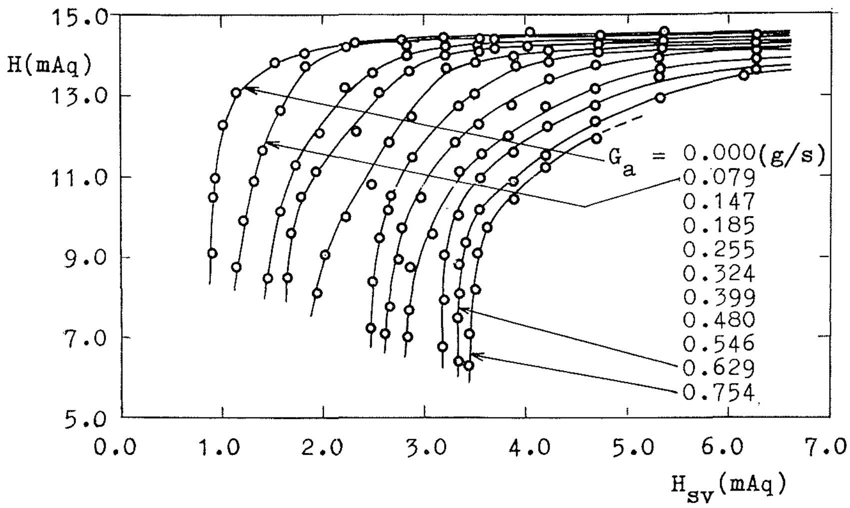


Fig. 11. Head-NPSH relation with air injection ($n=1200$ r.p.m., $G_w=29.157$ kg/sec).

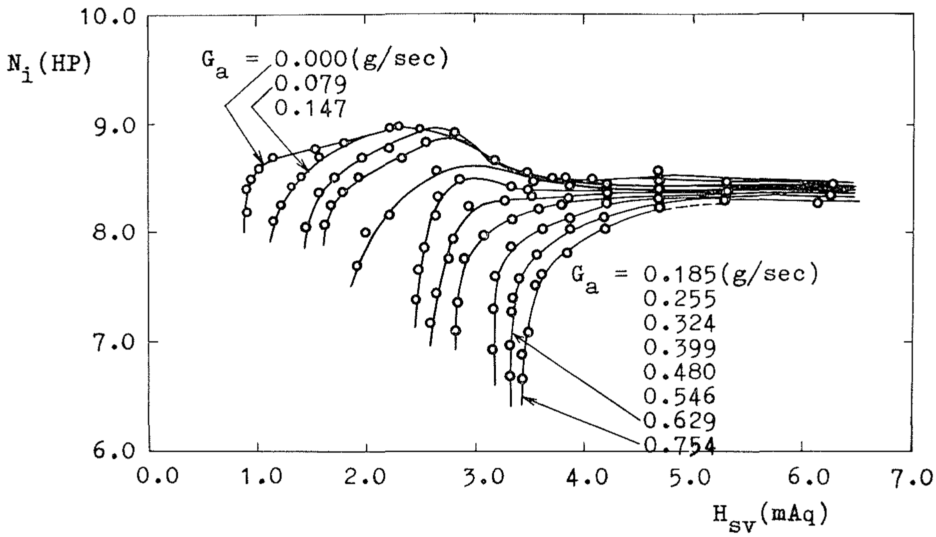


Fig. 12. Power-NPSH relation with air injection ($n=1200$ r.p.m., $G_w=29.157$ kg/sec).

weight ratio of air and water G_a/G_w is plotted for $n=1200$ r.p.m. to show the effect of air leakage on pump characteristics. The introduction of air in the suction system shows no sign of improving the nature of cavitation at least on the characteristics of a pump. However, it can clearly be seen that even the same quantity of air leakage gives a different value of effect on the pump performance depending on the value of the net positive suction head required. This result must be caused by the difference in volume occupied by the air bubbles—when the suction head is large, the air bubbles may be considered to expand under the

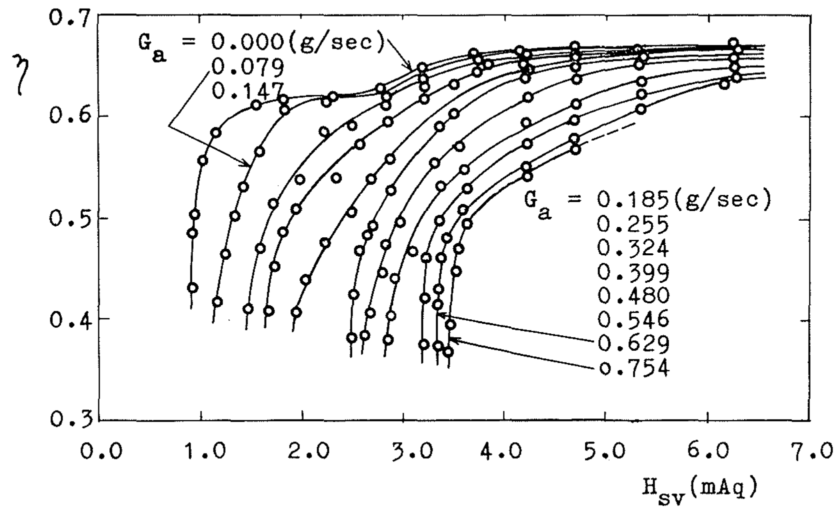


Fig. 13. Efficiency-NPSH relation with air injection ($n=1200$ r.p.m., $G_w=29.157$ kg/sec).

low pressure of the surrounding water.

The head ratio H_a/H obtained in the present investigation is given in Figs. 14 and 15 to show the effect of air leakage on a pump performance in a plot of (H_a/H) vs. (G_a/G_w) and $(n\sqrt{G_a\phi}/H_{sv}^{\frac{3}{4}})\cdot(H_a/H)$ vs. ϕ_{am}. As may be seen in this figure, the effect of air leakage in the suction system of a centrifugal pump can be given rather pertinently in a customarily acceptable dimensionless form. Since all the parameters to describe the pump operation with an air leakage are taken into consideration, this manner of expressing the characteristics of a pump can be used for the estimation of pressure drop when a mixture of air and water is pumped.

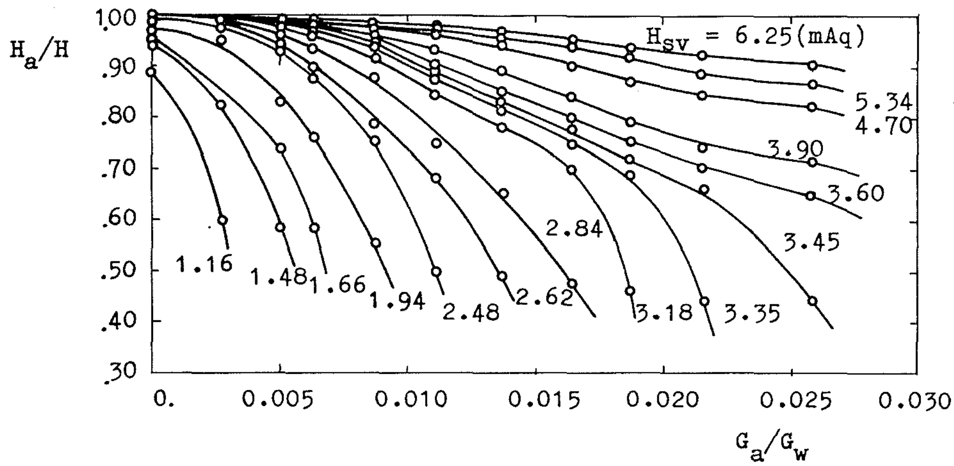


Fig. 14. Head ratio vs. flow rate ratio ($n=1200$ r.p.m., $G_w=29.157$ kg/sec).

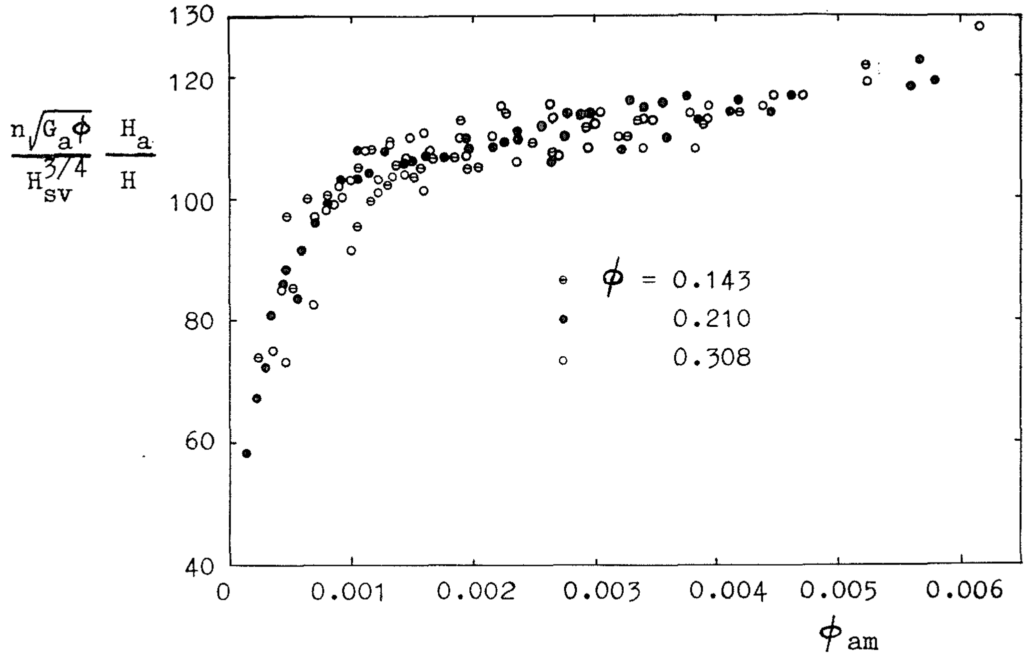


Fig. 15. $(n\sqrt{G_a\phi}/H_{sv}^{\frac{3}{4}})(H_a/H)$ vs. ϕ_{am}.

5. Qualitative nature of the effect of prerotation on cavitation characteristics of a centrifugal pump

An impeller of pump is usually designed to give $\alpha_1=90°$ at its rated capacity so that the flow of fluid onto the vane would not have the componet of prerotation from the standpoint of an efficient momentum transmission between the impeller and fluid. However, there exists a certain amount of prerotation when the operating point deviates from the design point as will be expected on the velocity triangle at the inlet of the impeller: the component of prerotation is in the direction of the rotation of the impeller when the capacity is less than the rated discharge, and it is in the reverse direction when the discharge demand exceeds

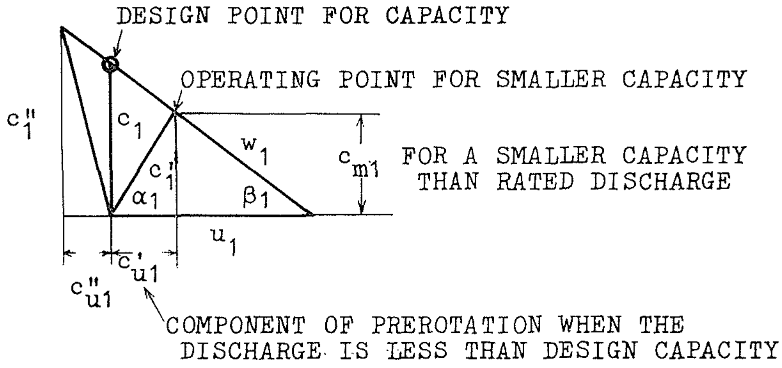
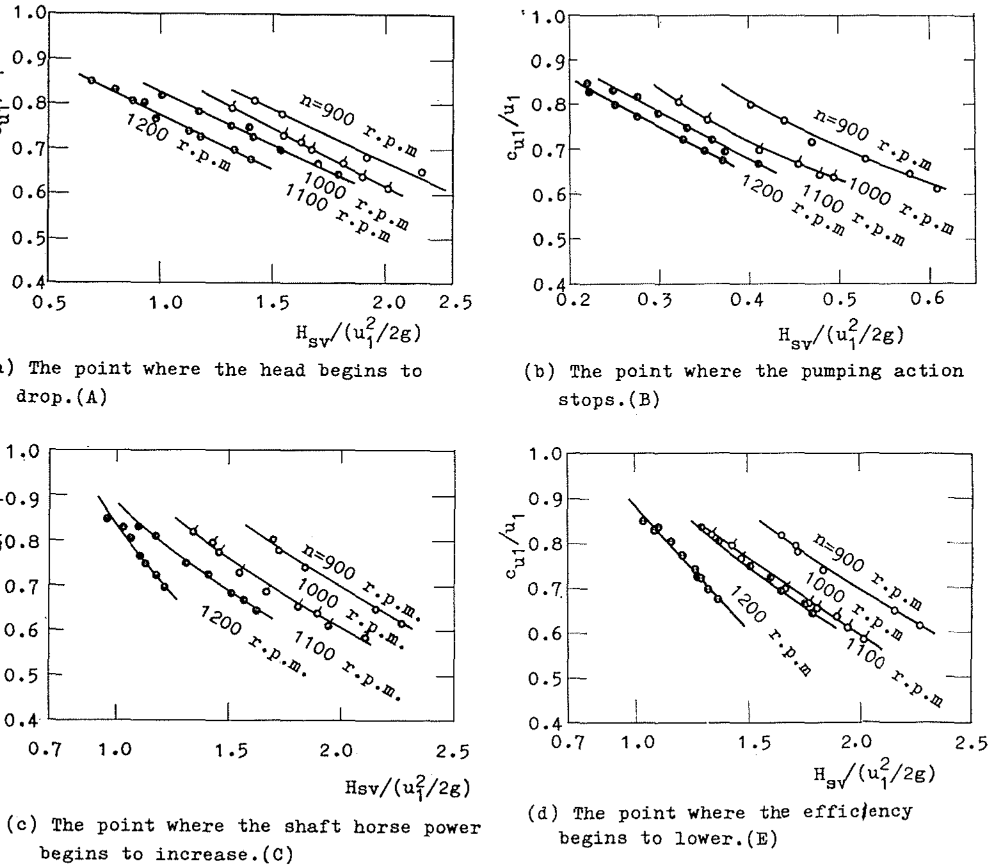


Fig. 16. Velocity triangle at the inlet.



(a) The point where the head begins to drop.(A)

(b) The point where the pumping action stops.(B)

(c) The point where the shaft horse power begins to increase.(C)

(d) The point where the efficiency begins to lower.(E)

Fig. 17. Effect of prerotation on characteristic points in cavitation region.

the rated capacity as is shown in Fig. 16. The existence of these types of prerotation can easily be detected by an observation of the flow in transparent suction pipe. Since there might occur a certain amount of shock loss at the inlet when the operating point is different from the design point, a quantitative consideration of the prerotation can not be expected in the present experiment. However, a qualitative effect of prerotation on the pump performance is derivable when the component of shock velocity at the inlet is neglected.

By constructing the velocity triangle at the inlet, the peripheral component of inflow c_{u1} was estimated for each series of experiments, and the results are shown in dimensionless form in Fig. 17 to show the relations with the characteristic points A, B, C and E on cavitation performance. As may be seen in this figure, the increase of prerotation results in a movement of the characteristic points to a smaller value of the net positive suction head. Therefore, it may be permissible to conclude that the existence of a prerotation in the direction of u_1 has a trend to improve the cavitation performance of an impeller of a pump.

References

1) Krisam, F.: "Neue Erkenntnisse in Kreiselpumpenbau," V.D.I.-Z., Vol. 95, 320 (1953).
2) Kasai, T. and Takamatsu, Y.: "Studies of cavitation aspect and suction performance of centrifugal pumps, 3rd, 4th and 5th reports," (in Japanese), Trans. J.S.M.E., Vol. 29, 1294 (1963).
3) Minami, S., Kawaguchi, K. and Homma, T.: "Experimental study on cavitation in centrifugal pump impellers," (in Japanese), Jour. J.S.M.E., Vol. 62, 881 (1959).
4) Gonger, G. A.: "A theory of cavitation flow in centrifugal pump inducer," Trans. A.S.M.E., Vol. 63, 29 (1941).
5) Wislicenus, G. F.: "Test stand for centrifugal and propeller pump," Trans. A.S.M.E., Vol. 64, 619 (1942).
6) Siebrecht, W.: "Untersuchungen über Regelung, theoretische und wirkliche Förderhöhe von Kreiselpumpen," V.D.I.-Z., Vol. 30, 87 (1930).