INLET FLOW AND ASPECTS OF CAVITATION IN CENTRIFUGAL IMPELLERS

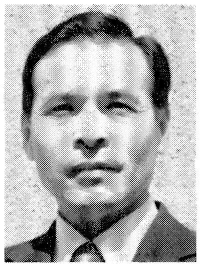
by

Masao Oshima

Pump Research and Development Department

Haneda Plant, Ebara Corp.

Haneda, Ota-ku, Tokyo, Japan



Masao Oshima graduated from Yokohama National University in 1956, and since then has been working at Ebara Corporation. After the first three years during which he worked in the design department, he has been in the pump research and development department. After presenting his first paper at the 1962 IAHR Symposium at Sendai, he has published many papers mainly on cavitation in centrifugal, mixed and axial flow impellers. He received the doctor degree for his study on pump cavitation in 1970.

ABSTRACT

This paper describes the relationships of inlet velocity distribution, incipient cavitation and suction performance of centrifugal pumps. It also discusses, in particular, the aspects of cavitation as related to inlet reverse flow (internal recirculation).

There are many ways of improving the suction performance of centrifugal pumps. Enlarging the inlet diameter or inlet angle are two of the most commonly used methods. However, this may increase the capacity at which inlet reverse flow begins. Inlet reverse flow normally starts to occur at about half of the shock-free inlet capacity, and this has no relation to the best efficiency flow.

Pump total head begins to drop only during advanced cavitation, especially at partial flow operation. When the impeller is designed with a large inlet angle to improve suction performance, there is a possibility of increasing the capacity where inlet reverse flow occurs, which may even be at the best efficiency point.

There are still other ways of improving suction performance without increasing the capacity at which inlet reverse flow occurs, e.g. reducing the number of blades, sharpening inlet edges, etc., and these are also referred to in this paper.

INTRODUCTION

Cavitation in a centrifugal pump is caused by vaporization of water flowing on the impeller blades and casing surfaces; however, the extent and location of its occurrence depends not only on NPSH, but also on flow rate.

The development of cavitation does not directly cause a reduction of pump head and there are significant differences between incipient cavitation and critical NPSH values; i.e., the point at which head reduction occurs. This difference increases as flow is decreased.

When inlet reverse flow (internal recirculation) occurs during reduced capacity operation, the inlet flow pattern and cavitation develop very abnormal characteristics. Cavitation then sometimes gives rise to severe pressure pulsation and vibration, and in such cases, the initial inlet recirculation flow, denoted hereafter by Qr, is sometimes defined as the minimum continuous flow. In addition, some impellers show lower noise and vibration levels while operating with inlet recirculation in comparison with operation at normal flow.

This paper describes the velocity distribution at the impeller inlet, cavitation and its effect on pump performance. Emphasis is placed on these phenomena in relation to high suction performance pumps.

VELOCITY DISTRIBUTION AT IMPELLER INLET

The inlet flow of end-suction type impeller has no prerotation, and therefore, the velocity at every point is completely proportional to the flow rate when operating above the flow rate Qr.

Figure 1 shows the velocity distributions measured at an axial flow impeller [1], where:

$\phi_{dm} = V_{dm}/U_t$

$\phi_s = V_s/U_t$

$\chi_s = V_{us}/U_t$

V_s and V_{us}: meridional and peripheral components of inlet velocity

V_{dm}: mean meridional velocity at impeller outlet

U_t: impeller tip speed

R: radius ratio to impeller tip.

As seen in the figure, inlet reverse flow occurs at $\phi_{dm}<0.11$. It returns with a strong rotation and the velocity of incoming flow near the hub is boosted to almost normal velocity. This causes the incidence angle of the incoming flow on the blade to increase with decreasing flow rate at $\phi_{dm}<0.11$. However, the reverse flow can cause the incidence angle at radii near the hub to approach zero or become negative for $\phi_{dm}<0.11$, as shown in Figure 2.

The inlet reverse flow begins to occur at the inlet critical flow rate Q_r, which is dependent on the inlet blade angle. Figure 3 [2] shows inlet meridional velocities for various blade angles (β) measured at two different radii, i.e., R = 0.48 and 1.06. When operating above the critical flow rate, it is clearly shown that the meridional velocity completely changes in proportion to the flow rate. Below the critical flow rate, the velocity near the hub increases to its normal value and remains constant. An interesting contrast is the reduction of the velocity near the blade tip as the flow rate decreases. The critical flow rate is less for a smaller blade angle although it is

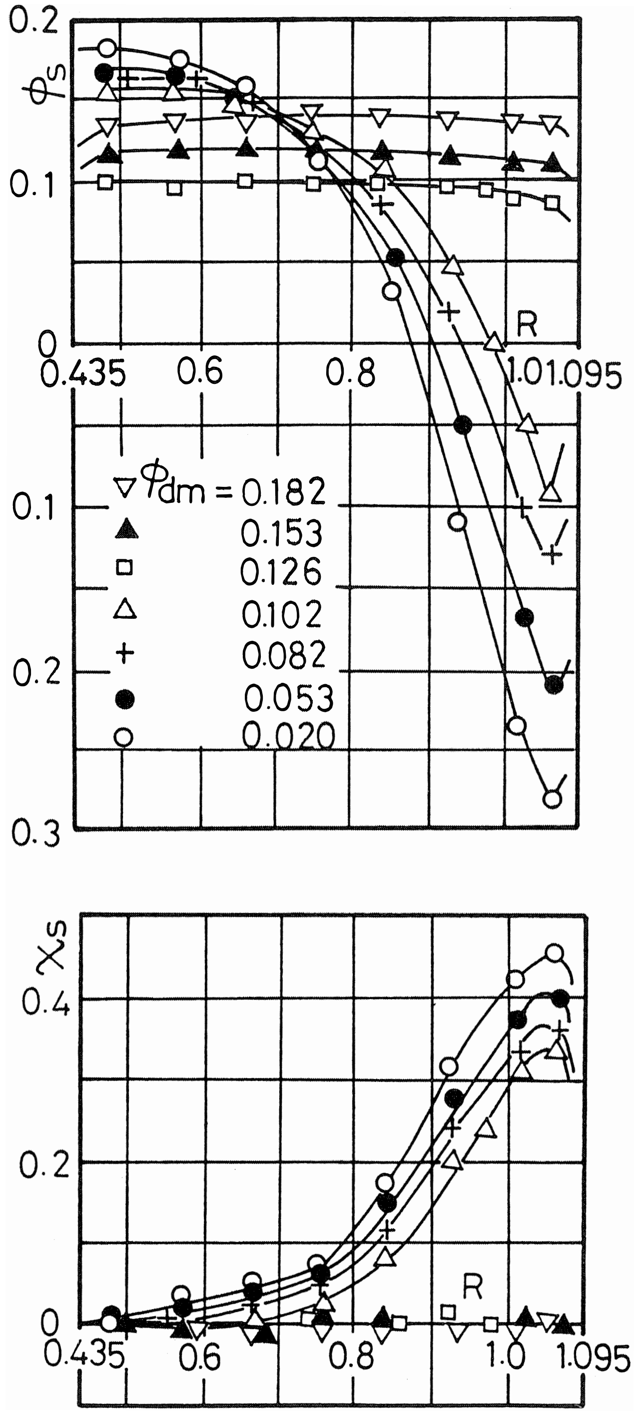


Figure 1. Velocity Distributions at Inlet of an Axial Flow Impeller [1]; Best Efficiency at ϕ_{dm} = 0.191.

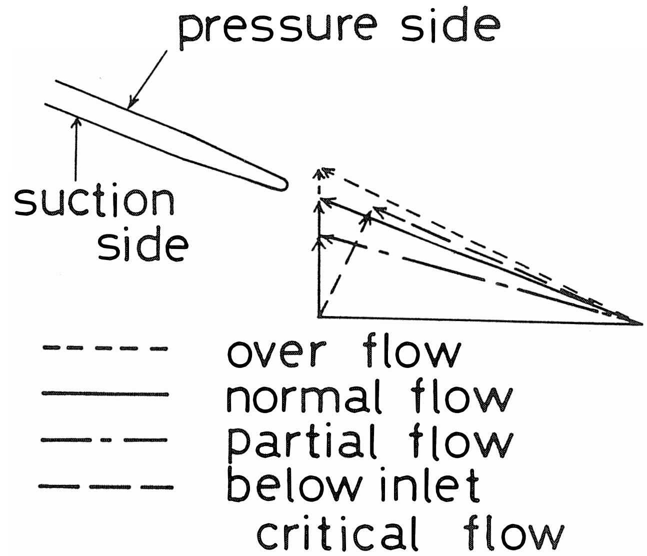


Figure 2. Velocity Triangles at Various Flows.

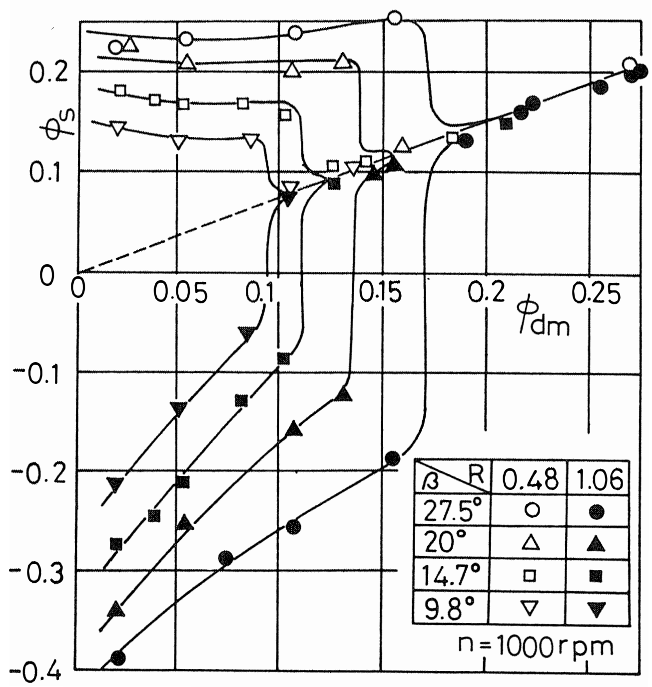


Figure 3. Meridional Inlet Velocity Near Hub and Tip of an Axial Flow Impeller for Various Blade Angles β [2].

independent of the number of blades, as illustrated in Figure 4[2].

The inlet edge geometry has a strong effect on the critical flow rate, which becomes larger when the inlet edge at the hub is extended in the upstream direction [3, 4]. In any case, the critical flow rate is between 40 ~ 60% of the shockfree inlet flow rate for usual standard centrifugal impellers [3] (ref. Figure 5).

CAVITATION IN IMPELLERS: INCIPIENCE, DEVELOPMENT AND EFFECT ON PERFORMANCE

When the available NPSH (Net Positive Suction Head) is reduced to a certain value, cavitation begins to occur on the impeller blade surface, its location and extent varying with the flow rate. Figure 5 [4, 5, 6] shows the incipient cavitation parameters against flow rates for centrifugal, mixed and axial-flow impellers, where:

λ : cavitation parameter $= 2gH_{sv}/(w_1^2 + v_0^2)$

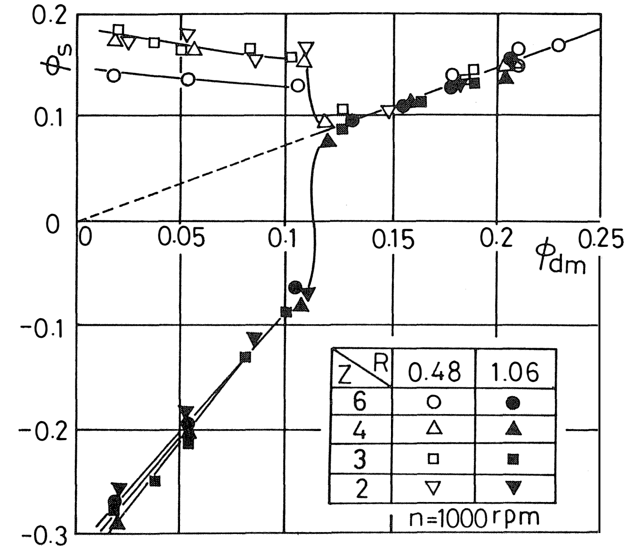


Figure 4. Meridional Inlet Velocity Near Hub and Tip of an Axial Flow Impeller for Various Number of Blades Z [2].

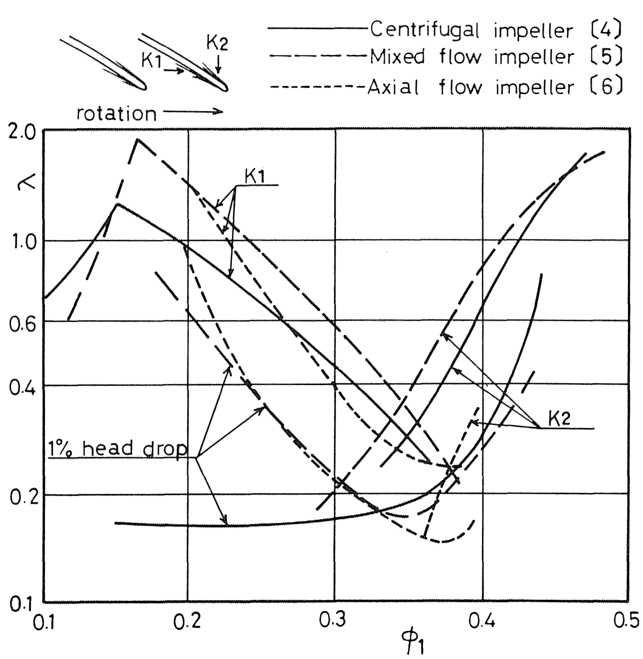


Figure 5. Cavitation Parameter for Inception and 1% Head Drop [4, 5, 6].

g : gravitational acceleration

H_{sv} : NPSH

w_1 : relative inlet velocity just after the blade inlet

v_0 : meridional inlet velocity just before the blade inlet

ϕ_1 : inlet flow coefficient $= v_1/u_1$

v_1 : meridional inlet velocity just after the blade inlet

u_1 : blade peripheral velocity at the inlet tip

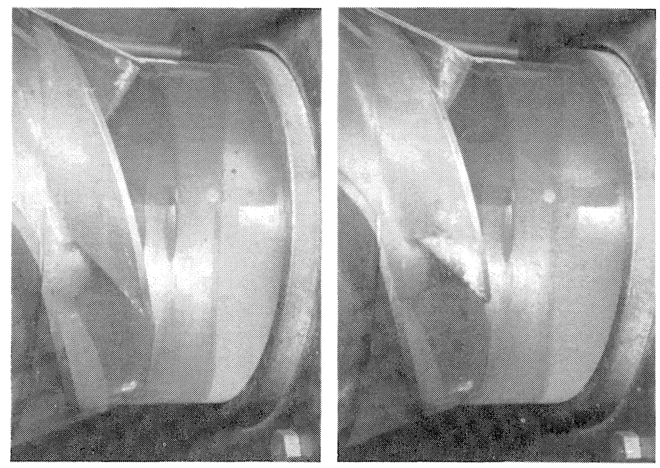
In this study, cavitation is classified according to its region of occurrence as stated below;

K_1: cavitation occurring at the leading edge on the suction side of the blade

K_2: cavitation occurring at the leading edge on the pressure side of the blade

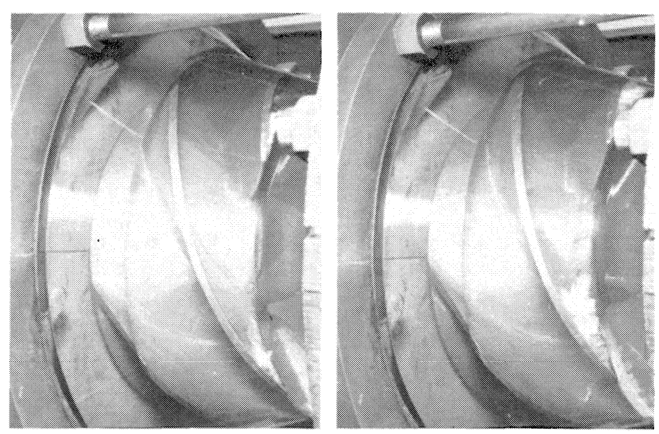
In the over-flow range ($\phi_1>0.34$), the flow enters the impeller with a negative angle of incidence. That is, the flow hits the suction surface of the blades, and cavitation occurs first on the pressure side of the blade (Figure 6i). At a reduced NPSH, cavitation is also observed on the suction surface.

On the contrary, in the partial flow range ($\phi_1<$●.34), the flow comes into the impeller with a positive angle of incidence. This causes cavitation to occur on the blade suction surface first (Figure 6ii). Cavitation in the tip clearance zone of the blades is not discussed because it does not significantly affect the performance of the impeller; however, cavitation in this region can occur.



(a) $\lambda = 1.2$ *(b) $\lambda = 0.6$*

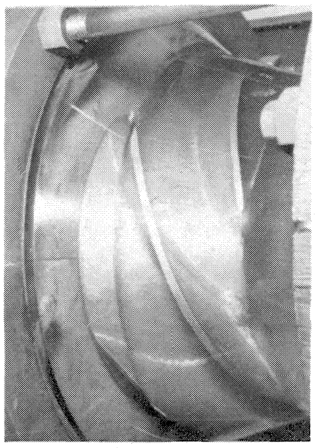
Figure 6i. Cavitation Aspects; $\phi_1 = 0.476$



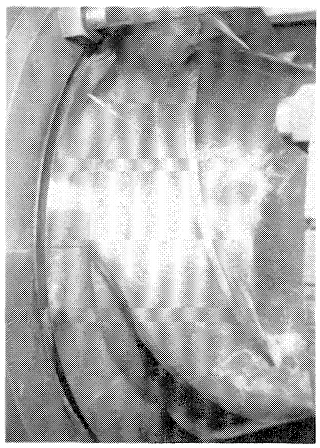
(a) $\lambda = 0.54$ *(b) $\lambda = 0.37$*

Figure 6ii. Cavitation Aspects; $\phi_1 = 0.241$

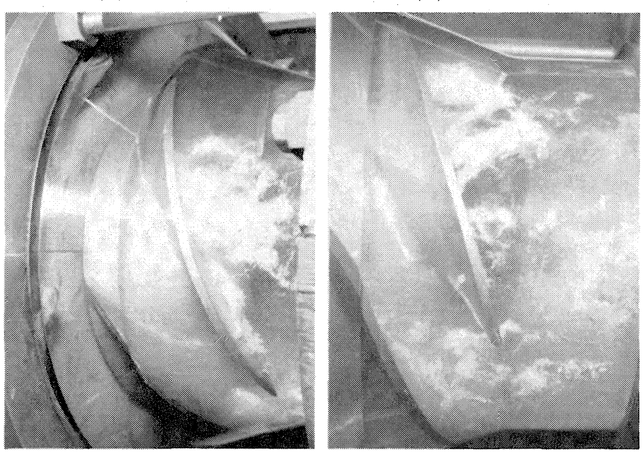
At the critical flow, $\phi_1=0.17$ (for the mixed-flow impeller), the λ curve shows a break-off point resulting from the occurrence of inlet reverse flow. Above this flow rate, cavitation originates at the outermost part of the blade inlet edge. Below the critical flow, cavitation starts at a point between the outermost part and the root of the inlet edge. The occurrence of cavitation is prevented near the peripheral part of the impeller inlet, because it is occupied by the reverse flow which causes a higher pressure in this region. Furthermore, cavitation arises in the sheared layer between the out-coming peripheral reverse flow and the central inlet flow. The cavitation in this zone is not steady but intermittent and closely resembles "lightning bolts" (Figure 6iii).



(a) $\lambda = 0.54$



(b) $\lambda = 0.26$



(c) $\lambda = 0.23$

Figure 6iii. Cavitation Aspects; $\phi_1 = 0.123$

With the development of cavitation, choking begins in the passage between the impeller blades (Figure 7), thus decreasing the pump total head. The λ values for the 1% head-drop points are shown in Figure 5. In the case of the mixed-flow impeller, the head-drop curve runs almost parallel to the incipient cavitation curves.

In the case of the centrifugal impeller, the total head within the over-flow range begins to drop at almost the same λ values as in the case of the mixed-flow impeller. Within the partial flow range, however, the pump head does not drop even under a marked development of cavitation and there is quite a difference between the two curves, incipient cavitation and 1% head drop curves, especially at reduced flow rates. This difference tends to be more significant as the pump head increases because the blade passage becomes longer and it becomes more difficult for the pump head to be affected by the cavitation.

The λ characteristics discussed above are determined by the blade inlet angle, and not by the flow rate of the best efficiency point (b.e.p.). This means that the λ characteristics

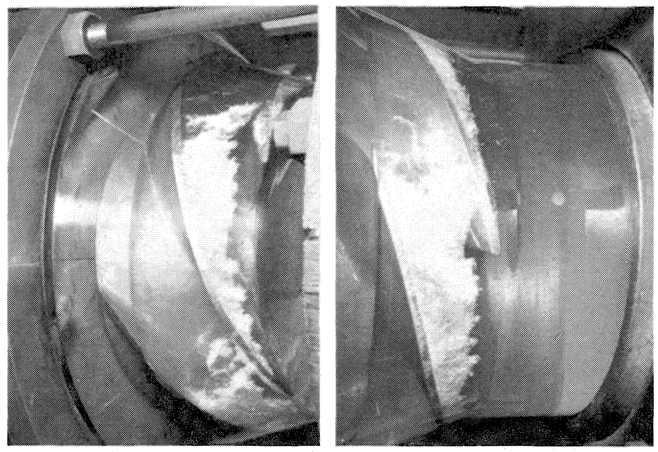


Figure 7. Cavitation Aspects, $\phi_1 = 0.34$, $\lambda = 0.127$.

can be adjusted independently of the bep flow rate. This is also the reason why the required NPSH remains unchanged for centrifugal impellers even when the best efficiency point is moved towards smaller capacities by cutting the impeller outlet diameter (Figure 9 [7]).

HIGH SUCTION PERFORMANCE IMPELLER

There are several methods of improving the suction performance of pumps. The following are the most commonly utilized and effective means:

1. Decreasing the incipient cavitation parameter, that is, sharpening the inlet edges of the blades [5], and,
2. Broadening the blade passages to lessen their susceptibility to being choked by cavities by:
   - i) Decreasing number of blades [5],
   - ii) Increasing blade inlet angle, or
   - iii) Increasing impeller inlet diameter [8].

Method 1 is commonly applied to the standard design of pump impellers. Method 2 (i) is most effectively used in the design of two or three-blade inducers.

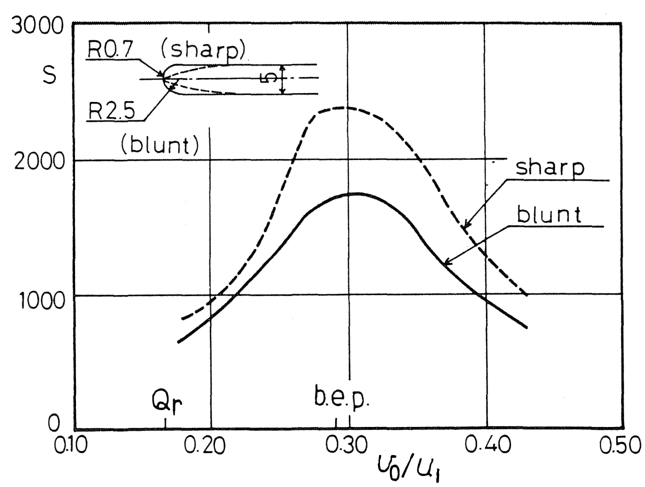


Figure 8. Effects of Sharpened Blade Inlet Edges on Suction Performance of a Mixed Flow Impeller; Impeller Inlet Diameter: 235 mm, Speed of Rotation: 1200 min^{-1}.

When Methods 1 and 2 (i) are applied to a centrifugal impeller design, the inlet blade angle does not change at all and, thus, the shock-free inlet flow and the initial recirculation flow (Qr) remains unchanged.

Figure 8 shows the effects of the sharpened inlet edges on the suction performance of a mixed flow impeller [5], where,

v_0/u_1 : refer to notation for Figure 5, and

S : suction specific speed (m/min).

The maximum suction specific speed was obtained near the b.e.p. flow and it exhibited an impressive increase from 1750 to 2400 (m/min) (from 11700 to 16000 (gal/ft/min)) due to the sharpened inlet edges; at the same time there was still no increase in the initial inlet recirculation flow.

Krisam's results (Figure 9) [7] certify the validity of Method 2 (ii). The trimmed impeller produces a higher suction specific speed at b.e.p., because the blade inlet angle becomes larger in comparison with the inlet flow angle at b.e.p. As seen in Figure 9, there is a possibility of improving the suction specific speed to about 2000 (m/min) (13300 (gal/ft/min)) by enlarging the blade inlet angle, resulting in the initial inlet recirculation flow coming nearer to the b.e.p. flow.

Where the blade inlet angle is increased, the shock-free flow, and hence, the initial recirculation flow inevitably increase. In an extreme case, the inlet reverse flow may be observed even at the b.e.p. while the pump is still showing very good suction performance [9].

As for Method 2 (iii): increasing the impeller inlet diameter, which is also commonly used in the design of centrifugal impellers, does not necessarily increase the shock-free inlet flow, as the blade inlet angle is independent of the inlet diameter. When the inlet angle is decreased as a result of increasing the inlet diameter, the shock-free inlet flow can be kept the same, and the suction performance can be improved without increasing the initial inlet recirculation flow.

In short, all the methods for obtaining higher suction performance do not increase the initial inlet recirculation flow, unless the blade inlet angle is made too large for the pump b.e.p. flow.

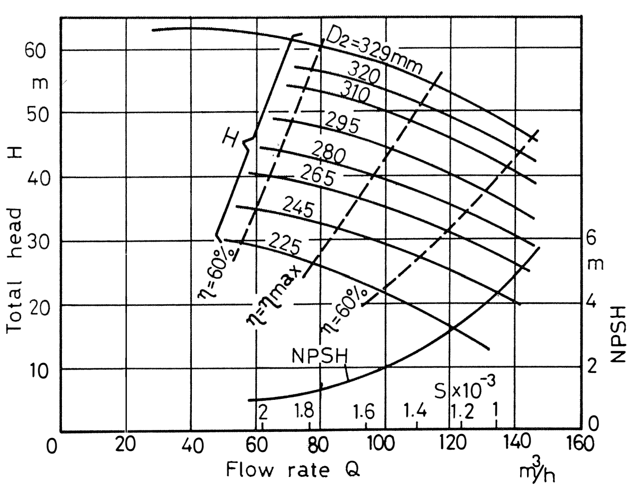
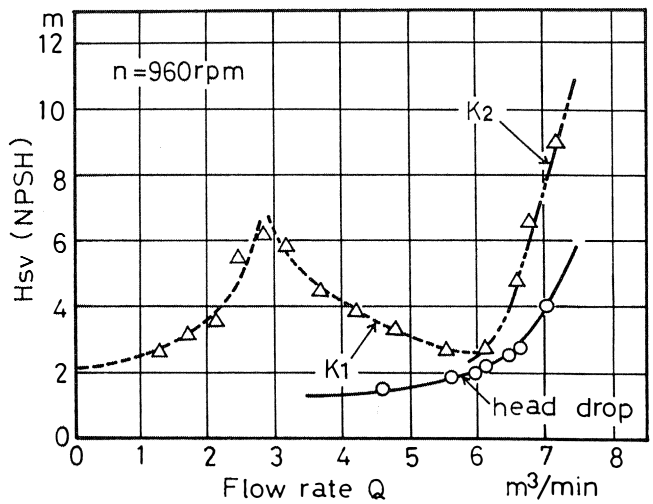


Figure 9. Cavitation Test on a Centrifugal Pump with Various Impeller Outlet Diameters D_2 [7]; Rotational Speed n = 2000 min^{-1}; η : Overall Efficiency; η_{max}: Maximum of η ; S: Suction Specific Speed (m/min).

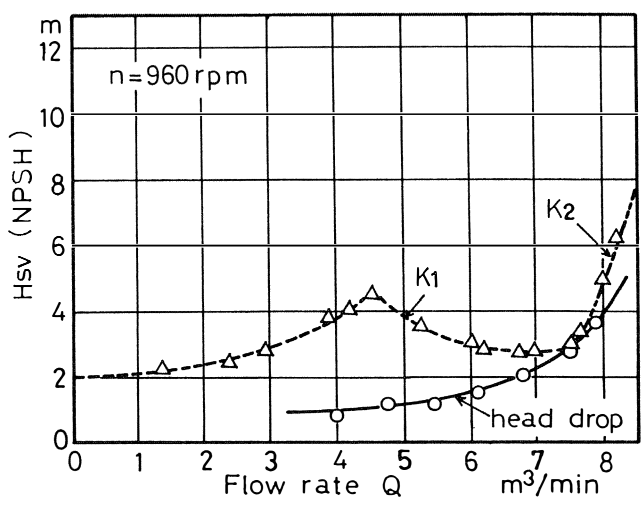
INLET RECIRCULATION AND ALLOWABLE OPERATION RANGE

The inlet reverse flow returns unsteadily to the suction pipe at a very high velocity, which is about the same as the peripheral speed of impeller eye. Sometimes it can bring about very severe pressure pulsation and vibration [10]. For this reason, the initial recirculation flow is often taken as the minimum continuous flow for centrifugal pumps.

Figure 10 [4] shows the results of cavitation tests on two centrifugal impellers with the same outlet dimensions. Impeller A is the normal impeller and Impeller D has inlet edges which extend further in the upstream direction than Impeller A. This causes impeller D to have a larger initial recirculation flow (Q = 4.6 m^3/min) than Impeller A (Q = 2.9 m^3/min). It also shows NPSH values for the inception of cavitation that are smaller than those of Impeller A over the whole flow range



(i) Impeller A



(ii) Impeller D

Figure 10. NPSH for Cavitation Inception Based on 1% Head Drop [4]; Inlet Diameter, D_1 = 200 mm.

with the exception of Q = 4 ~ 6.2 m^3/min. Moreover, it should be noted that Impeller D gives much lower NPSH values for inception than Impeller A at partial flows. It may be safely concluded that a higher initial recirculation flow does not always bring about lower suction performance.

Figure 11 gives the results of noise measurements on a mixed-flow impeller that were carried out at a constant available NPSH. At Q = 6.8 m^3/min the noise level shows a break-off, indicating the onset of the inlet recirculation. Actually, the noise level is much smaller in the flow range below the initial recirculation point than in the flow range above it.

The noise characteristics also indicate that it may not necessarily limit the allowable operation range to the flow rate above the initial recirculation capacity.

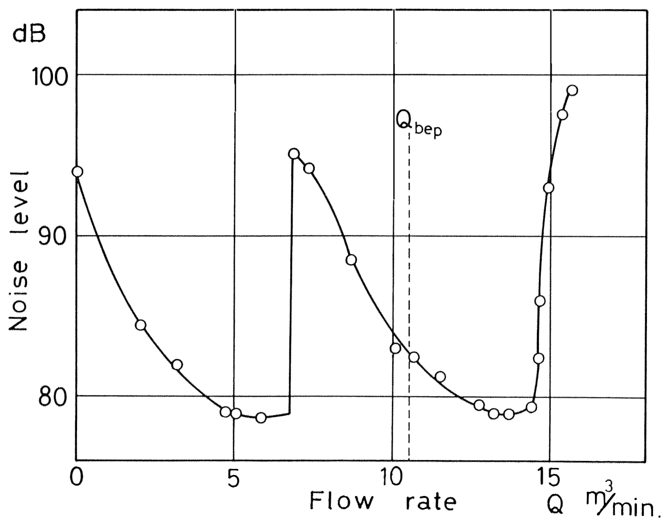


Figure 11. Noise Level (SPL) of a Mixed Flow Pump; Rotational Speed; n = 800 min^{-1}, Specific Speed n_s = 560 (m/min), NPSH = 11 m, Inlet Diameter D_1 = 250 mm.

As indicated in Figure 1, the peripheral velocity component (χ_s) of the reverse flow increases considerably as the flow rate decreases. This causes a notable increase in the energy transferred to the reverse flow which raises the level of noise. However, the energy imparted to the reverse flow is considerably small in the flow range near the initial recirculation point and allows the pump to be operated in this flow range.

After all, the minimum continuous flow should be determined by evaluating all the factors which will affect the pump operation, including not only the initial recirculation flow but also the energy received by the reverse flow.

CONCLUSION

The extent and location of cavitation in centrifugal impellers vary with the flow rate. Therefore, a meaningful understanding of pump cavitation requires some knowledge of the velocity distribution at the impeller inlet.

This paper has pointed out that higher suction performance does not always cause a higher initial recirculation flow, and that the minimum continuous flow should be determined taking into account all the factors affecting the pump operation, including the energy received by the inlet reverse flow.

In addition, inlet blade angle is the controlling factor for altering the initial inlet reverse flow during pump design.

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