# DOES IMPELLER AFFECT NPSHR?

by

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## ABSTRACT

The relationship between centrifugal pump impeller diameter, specific speed, noise and NPSH required is presented in this paper. A theoretical explanation is given to explain why NPSHR will significantly vary with impeller diameter for some pump designs.

The degree of cavitation, as indicated by NPSHR, will change at different flow rates. Also, a change in impeller diameter does not generally affect NPSHR, because cavitation occurs at the impeller inlet. Test results have proven that this assumption is not true for some high specific speed pumps.

High specific speed pumps are more sensitive to the effects of cavitation because of relatively shorter impeller blade lengths. When trimming the impeller diameter, the impeller blade length is also reduced. In some cases, the effects of cavitation blockage inside the impeller flow path can be more pronounced, because they interface with pressure recovery. Therefore, the degree of cavitation must be reduced to ensure proper impeller hydraulic operation. Trimming the impeller diameter of some high specific speed pumps will require higher NPSH values.

## INTRODUCTION

Pump cavitation occurs at the blade inlet surface of the impeller where the lowest pressure zones exist. The cavities formed in this area are carried through the impeller by flow and when they reach an area of higher pressure, they collapse and immediately disappear. Since cavitation occurs at the impeller inlet and disappears in the central area of the impeller, the relationships of impeller outlet diameter of NPSH and pump noise have not been fully studied in the past.

This paper discusses the impeller diameter-noise relationship based on data obtained from recent pump performance tests. The five pumps used for these tests were double-suction, single-stage, axially split centrifugal pumps with different specific speeds.

The purpose of this study was to answer the following questions:

How does NPSH vary when the impeller outer diameter is trimmed?

How does this varying NPSH affect cavitating pump noise?

How are the rates of variation of NPSH and noise affected by specific speed?

In addition, the problems and theoretical aspects of cavitation noise levels and NPSH variations caused by impeller trimming are discussed.

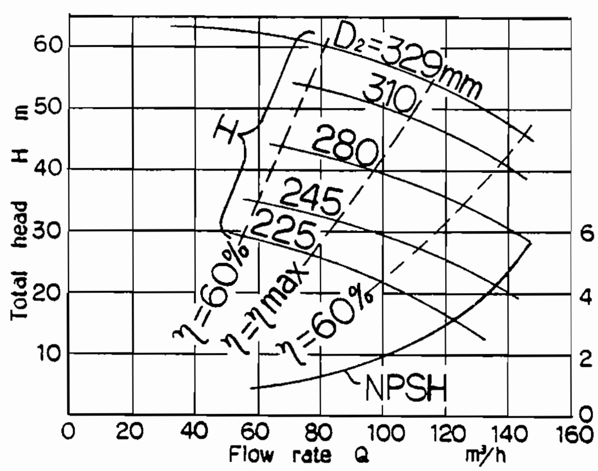



*Figure 1. Cavitation Test on a Centrifugal Pump with Various Impeller Outlet Diameters $D_2$ [1]; Rotational Speed $n = 2000$ $min^{-1}$; $\eta$; Overall Efficiency; $\eta$ max: Maximum of $\eta$.*

## EXPERIMENTS AND CONSIDERATIONS OF EXPERIMENTAL RESULTS

### *Test Apparatus*

A previous study by Krisam concluded that NPSH was not affected by trimming the outer diameter of an impeller [1]. In that study, tests were carried out on an end-suction centrifugal pump with a specific metric speed of about 140 ($min^{-1}$, $m^3/m$, m). The results of his cavitation tests, as shown in Figure 1, indicate that NPSH did not change even after the impeller outer diameter had been trimmed to 68 percent of full diameter.

In the present study, cavitation tests were performed using double-suction, single-stage, axially-split centrifugal pumps, as illustrated in Figure 2. The suction bore for all pumps was 350 mm and the pump specific speeds were 140, 200, 280, 400, and 560. In order to investigate pump noise during cavitation, a soundproof box was manufactured (Figure 3). Each test pump was covered with the soundproof box, as is shown in Figure 4, to prevent entry of noise from the driver and valves in the piping system. The soundproof box was equipped with a glasswool lining and designed to eliminate residual reverberation. Noise measurement methods conformed to API 615 standards.

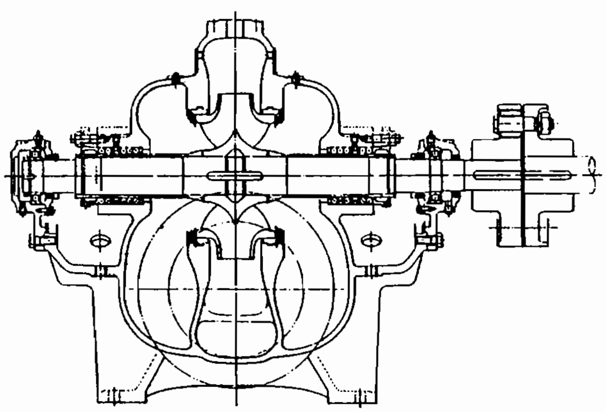


*Figure 2. Sectional Drawing of Double Suction Volute Pump.*

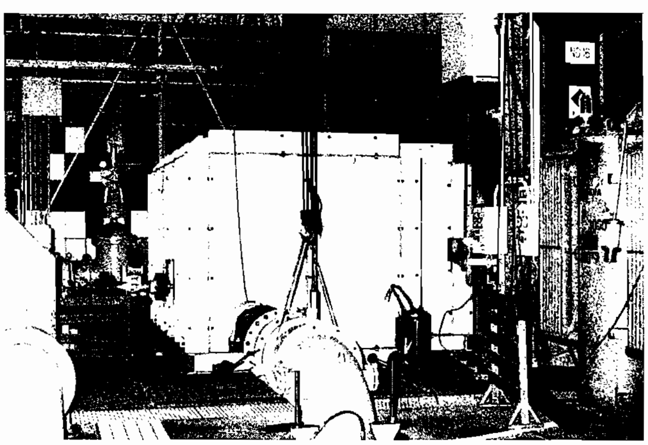


*Figure 3. Soundproof Box Viewed from Suction Side.*

### *Experimental Results*

The relationship between impeller outer diameter variation and NPSH versus head drop for 100 percent flow rate and Ns = 200 is shown in Figure 5. The same relationship for Ns = 560 is shown in Figure 6. In order to understand the relationship between NPSH and impeller outer diameter, these two

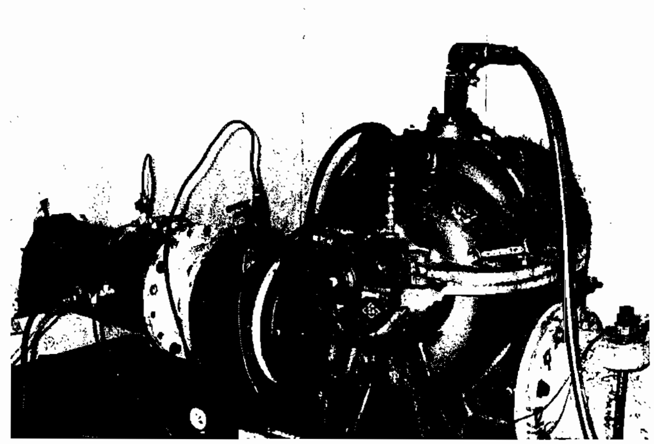


*Figure 4. Test Pump Covered with Soundproof Box.*

factors were compared at a 3 percent head drop rate, which is in conformance with the evaluation method stipulated by HIS.

The following relationships were derived from Figures 5 and 6:

1. When specific speed (Ns) is small, NPSH does not show significant variation, even when the impeller ouer diameter has been trimmed.

2. As specific speed increases, NPSH is largely controlled by variation of the impeller outer diameter.

3. The head drop patterns for low and high Ns impellers are completely different. In other words, the pump head drop falls off abruptly when Ns is low, it moves gradually when Ns is high.

The relationship between noise variations of cavitating pumps and head drop for maximum and minimum impeller diameters is shown in Figures 7 and 8. The specific speeds are Ns = 140 and Ns = 400, respectively, and both pumps are operating at 1790 RPM and at the best efficiency point. Noise and vibration were measured at various NPSH values. Although there are other reports of cavitation detection methods by analyzing noise and vibration spectra (for example, Deep-

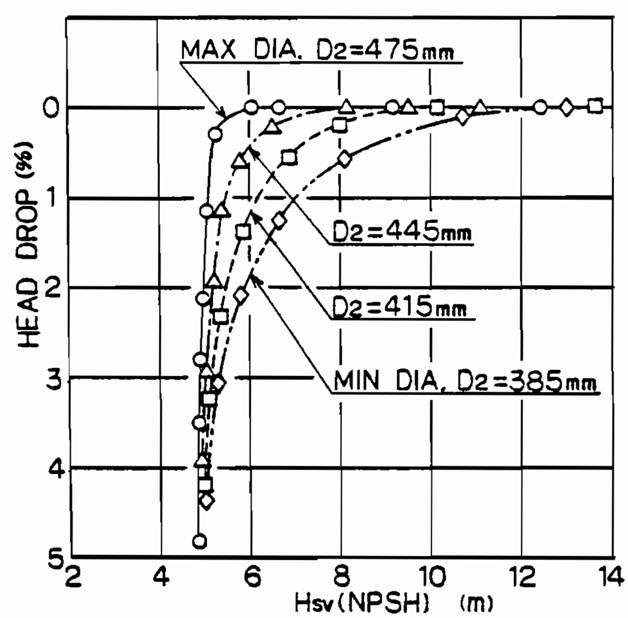



*Figure 5. Impeller Outer Diameter Variation and Hav-ΔH for 100 Percent Flow Rate with a Low Specific Speed.*

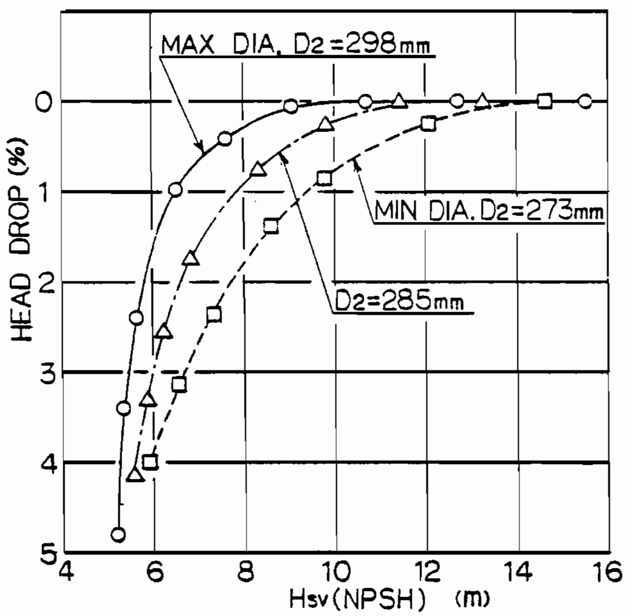
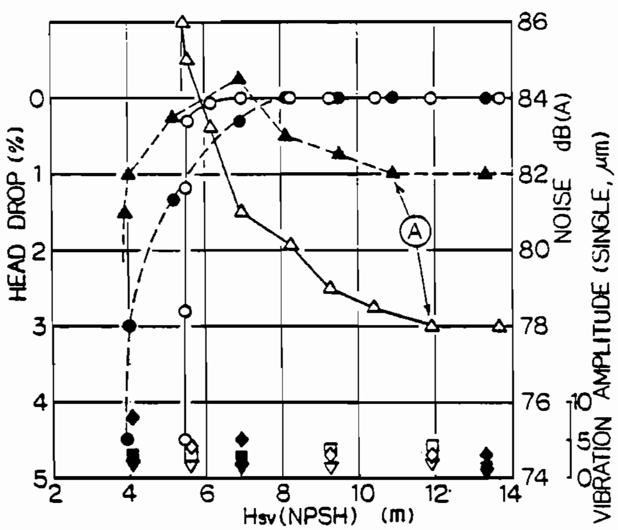



*Figure 6. Impeller Outer Diameter Variation and Hsv-ΔH for 100 Percent Flow Rate with a High Specific Speed.*




| | | Max. Dia. | Min. Dia. |
|---|---|---|---|
| Dia. | | 587 mm | 482 mm |
| Head | | ○ | ● |
| Noise | | △ | ▲ |
| Vibration | H | ◇ | ◆ |
| | V | ▽ | ▼ |
| | A | □ | ■ |

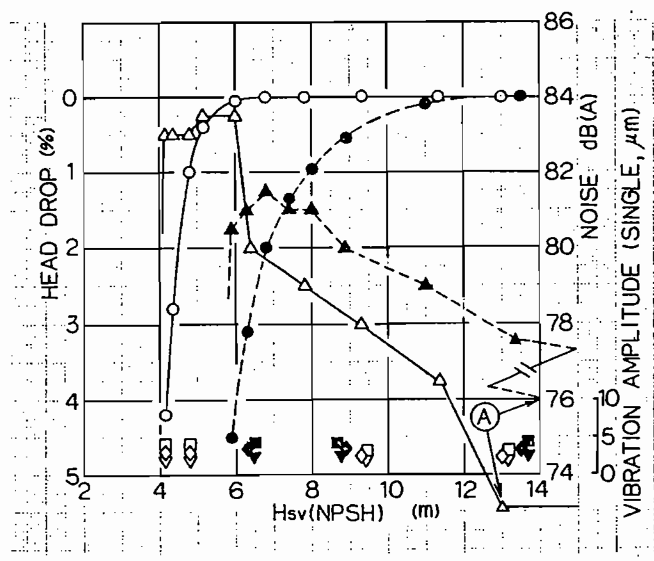
*Figure 7. Cavitation Noise, Vibration Noise, Vibration, and Head Drop for Maximum and Minimum Impeller Diameters with a Low Specific Speed. Rotational Speed n = 1790 min$^{-1}$*

rose [2] and Varga [3]), in this study, a condensor microphone was installed inside the previously mentioned soundproof box to measure the air-borne noise of the pump.

*Consideration of Experimental Results*

*Noise curves and impeller diameter*

The noise curves rise abruptly at Point A as shown in Figures 7 and 8. This NPSH value is much larger than the point at which the pump head begins to drop, irrespective of Ns and the impeller outer diameter. Therefore, Point A can be regarded as the point of incipient cavitation. However, the air-borne noise of the pump was measured, and transmission losses due to the casing thickness must be taken into consideration. The point of incipient cavitation would probably move to a slightly larger NPSH value, if the pump noise were measured with a microphone installed in the vicinity of the impeller.




| | | Max. Dia. | Min. Dia. |
|---|---|---|---|
| Dia. | | 334 mm | 284 mm |
| Head | | ○ | ● |
| Noise | | △ | ▲ |
| Vibration | H | ◇ | ◆ |
| | V | ▽ | ▼ |
| | A | □ | ■ |

*Figure 8. Cavitation Noise, Vibration, and Head Drop for Maximum and Minimum Impeller Diameters with a High Specific Speed. Rotational Speed n = 1790 min$^{-1}$*

The noise curve angles rising from the point of incipient cavitation are closely related to the impeller outer diameter and NPSH. The acutely rising curves are thought to be caused by the different quantities of air in the liquid entering the pump. In all cases, the noise level rises as NPSH decreases, and it falls off accompanying complete head drop.

However, there is an interesting paradox when comparing the noise levels of minimum versus maximum impeller diameters, as illustrated in Figure 9. Irregardless of specific speed, the noise levels just before cavitation occurs are higher for the minimum diameter impeller.

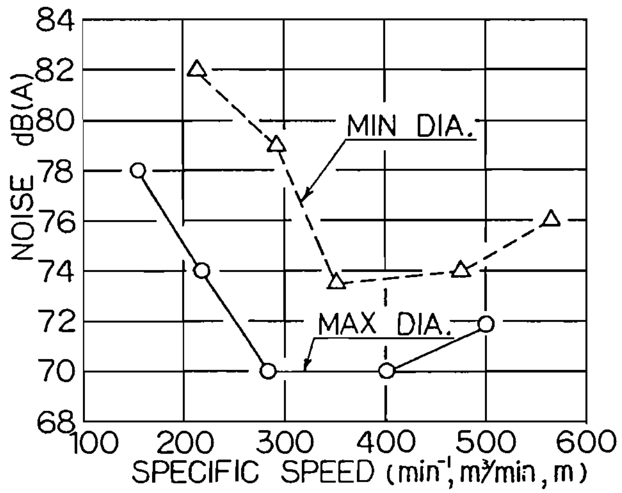



*Figure 9. Noise Level at NPSH Before the Occurrence of Cavitation in the Case of Maximum and Minimum Diameter Impellers.*

In the field of centrifugal fans, noise plays an important role and the following formula [4] has been derived:

$$TN = SN + 10 \log_{10} \frac{(QP_T^2)}{60} \text{ [db]} \tag{1}$$

where

TN : Total noise [db]
SN : Specific noise [db]
Q : Capacity [$m^3$/min]
$P_T$ : Total pressure [mmHg]

The specific noise (SN) is determined by matching the impeller and casing charcteristics of fans, and it is peculiar to each fan. This equation indicates that the total level of noise is basically proportional to the capacity of the fan and to the square of the total presure. Consequently, the noise level is in direct proportion to the fan impeller outer diameter. These relationships are corroborated by Neise [5].

According to Suzuki, these relationships hold true for a volute casing as shown in Figure 10a—however, he points out that if the impeller is trimmed beyond a given amount, TN will rise [6]. It has also been shown experimentally that, for casings such as the guide vane type depicted in Figure 10b, timing the impeller produces a drop in efficiency accompanied by turbulent flow. Compensating for this drop in efficiency, lost energy is changed to heat, noise, and vibration, which causes the value of SN in Equation 1 to increase. In some cases, there is also an increase in the value of TN, even if values of Q and P decrease.

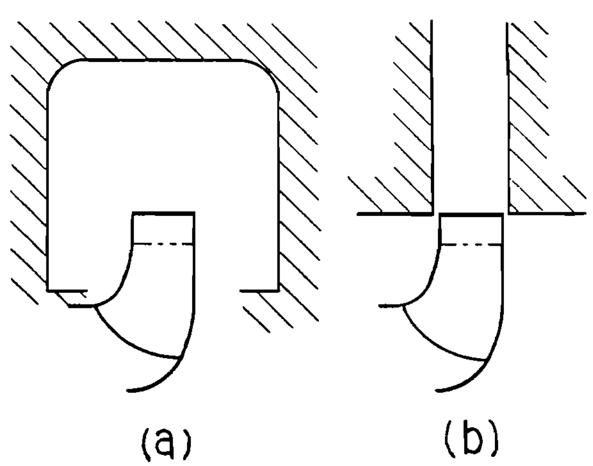



*Figure 10. Volute Casing Type. Guide Vane Type.*

The specific noise was calculated using Q and $P_T$ at the best efficiency points for the same pumps shown in Figure 9 and the results of these calculations are shown in Figure 11. The SN values increase as Ns becomes larger and the minimum diameter SN values are larger than those of the maximum diameter. In other words, as the impeller outer diameter is trimmed, the clearance between the impeller outer diameter and the casing tongue increases, resulting in a drop in efficiency. The power consumption corresponding to the efficiency drop is thought to be one of the causes for increases in specific noise. We are not aware of any reports dealing with pump specific noise versus impeller outer diameter. Future research concerning this topic will be necessary to completely define the specific noise of a pump.

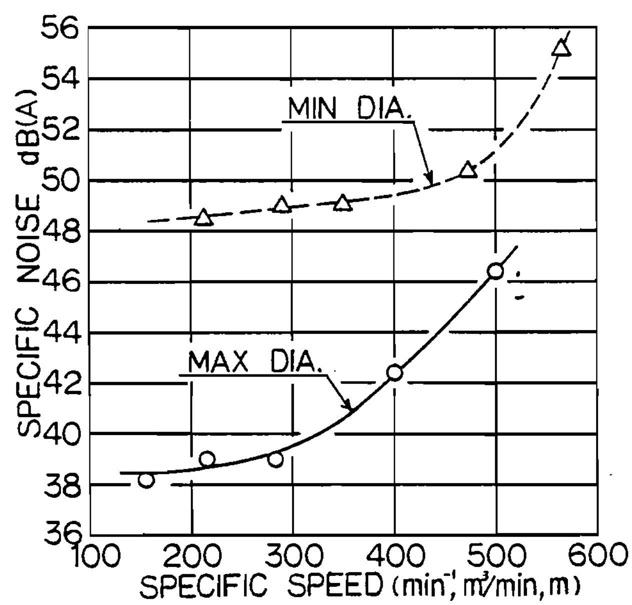



*Figure 11. Diagram of Ns Versus SN of Pump.*

*Vibration*

The vibration at each NPSH value and different impeller diameters was measured and the results are shown in Figures 7 and 8. Measurements were made at five locations, but the figures show only the values taken on the outboard bearing housing. However, the vibration levels obtained at the other measurement locations showed little variation. There was a slight increase in vibration as NPSH decreased, but this was not significant. Even when the impeller outer diameter was changed, there were no indications of notable phenomena such as the previously described noise variations.

The evaluation of noise and vibration level variations is subject to discussion. However, cavitation can be understood as a force which does not have as strong an influence on vibration as it does on noise.

## THEORETICAL CONSIDERATIONS

### *Reasons for Variation of NPSH Due to Trimming of Impeller Diameter*

What actually happens inside an impeller during operations under conditions of cavitation is now considered. Cavitation occuring in a mixed-flow impeller under 3 percent head drop conditions is shown in Figures 12 and 13. In Figure 12, cavitation is occuring from the blade tip to the pressure surface while operating in the large-flow range, namely, 117 percent of BEP capacity. Cavitation in the suction surface area in the partial-flow range, that is, 86 percent of BEP capacity is shown in Figure 13. The condition of cavitation, at 3 percent drop point, does not occur to any significant degree in the case of a centrifugal impeller. Cavitities collapse and disappear at zones of higher pressure inside the impeller. It is well known, that when the suction pressure decreases, large cavities full of vapour can form on the walls of the impeller flow passages. A sketch showing conditions of cavitation in a low-specific-speed impeller is presented in Figure 14.

*Figure 12. Cavitation Aspects Under 3 Percent Head Drop Conditions. (In the Large-Flow Range)*

*Figure 13. Cavitation Aspects Under 3 Percent Head Drop Conditions. (In the Partial-Flow Range)*

The following reasons can be considered when concerned with why NPSH changes when the impeller outer diameter is trimmed:

1. The relative velocity $W_1'$ of the flow through the impeller inlet in Figure 14, when compared with that under normal (noncavitating) conditions, increases through the narrowed portion of the flow passage. On the other hand, the relative flow velocity $W_2$ at the impeller outlet remains almost constant. This results in a loss due to flow expansion produced when cavitation occurs. In other words, head drop is dependent on the value of this loss, as determined by

$$\Delta H = \xi \frac{(W_1' - W_2)^2}{2g} \tag{2}$$

where

- $\Delta H$ : Internal loss of impeller due to occurence of cavitation [m]
- $\xi$ : Loss coefficient of flow passage
- $W_1'$ : Relative velocity at impeller inlet when cavitation occurs [m/sec]
- $W_2$ : Relative velocity at impeller outlet [m/sec]

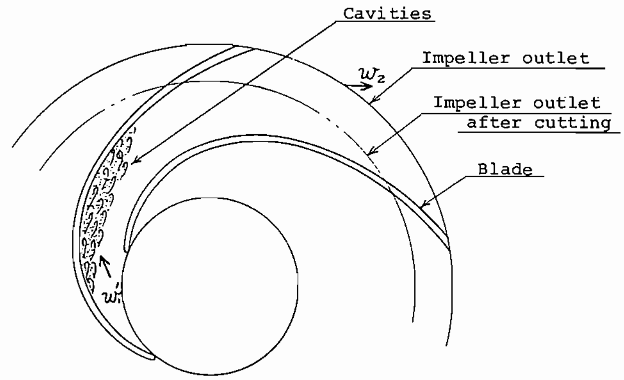



*Figure 14. Impeller with Low Specific Speed.*

When the impeller is trimmed, the absolute length of the flow passage is reduced, because the impeller blade length has been reduced. This remains true even if the pump suction conditions or the location of the cavitating region are unchanged. As a physical phenomenon, trimming the impeller produces the same effects as the expansion rate in an expansion

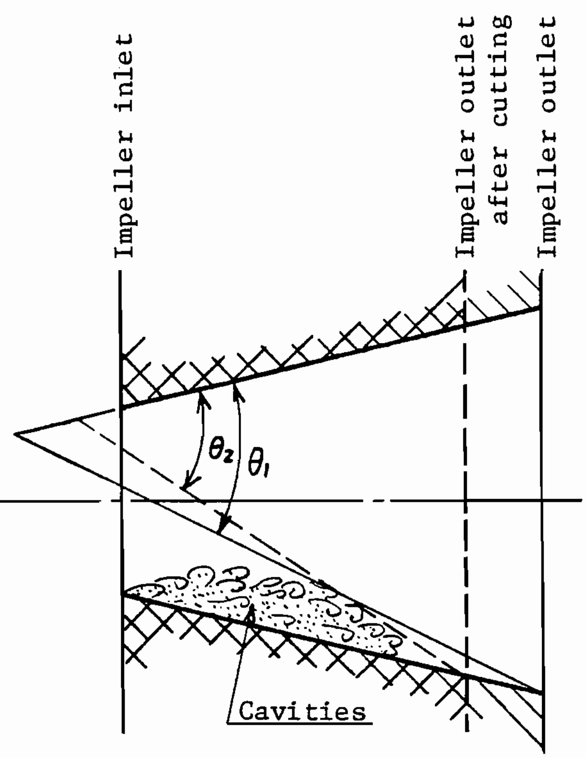



*Figure 15. Variation of the Expansion Rate of Trimmed Impeller.*

pipe, which is shown in Figure 15. That is, the flow passage loss coefficient increases when the impeller outer diameter is trimmed.

2. The second reason is related to the problem of ratios. At present, performance impairment caused by insufficient NPSH is expressed in terms of a 3 percent head drop in both HIS and API 610. Accordingly, since NPSH is related to a ratio of head, it is not surprising that NPSH deteriorates when the impeller outer diameter is reduced. Let's consider, for example, the case of a pump impeller whose head is 100 m at maximum diameter and 70 m at minimum diameter. If the suction pressure is reduced until cavitation occurs, the head drop will then be, 2.1 m. At this point, the maximum-diameter impeller has experienced only 2.1 percent head drop, but the minimum-diameter impeller has already reached the 3 percent head drop stipulated by the above mentioned codes.

In accordance with the above reasons, the criteria for evaluating suction performance is expressed in terms of head drop, and NPSH will deteriorate when a pump impeller is trimmed.

*Low Ns Versus High Ns*

In comparing low Ns versus High Ns impellers, actual values of Ns and impeller outer diameter under 3 percent head drop conditions are shown in Figure 16. As a note, suitable impeller blades and angles have been determined for each pump Ns, to provide an optimum design. With a low-specific speed impeller, the outer diameter is essentially large, and the blade length is greater than in the case of a high-specific speed impeller. Thus, the impeller flow passages of a low Ns pump are not as apt to be blocked, even when cavitation develops, because there is more distance between the impeller inlet and outlet as shown in Figure 14 versus Figure 17. For this reason, trimming the impeller to an extent that will not adversely affect pump efficiency will not cause NPSH to notably change.

In the case of high-Ns centrifugal pumps, the number of impeller blades cannot be increased when considering overall pump efficiency. In addition, the impeller outer diameter of such pumps is relatively small in comparison to the diameter of a low-Ns impeller. Therefore, the distance between the impeller outlet and cavitating zone of a high Ns pump is shorter. This is shown in Figure 14 versus Figure 17. In other words, the relationship between the impeller outer diameter and the block-

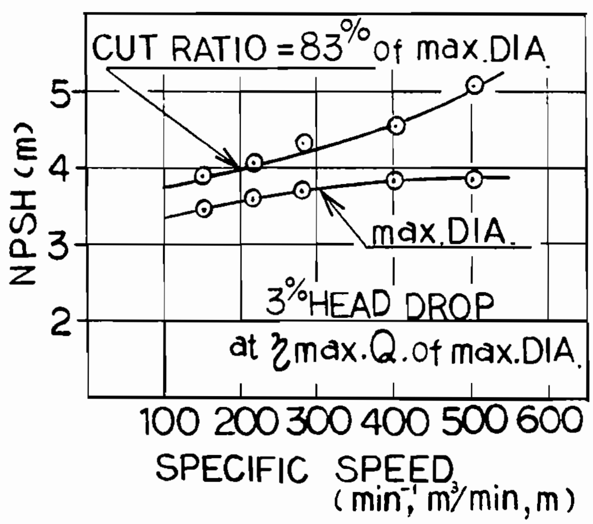



Figure 16. Specific Speed and Impeller Outer Diameter Under 3 Percent Head Drop Conditions.

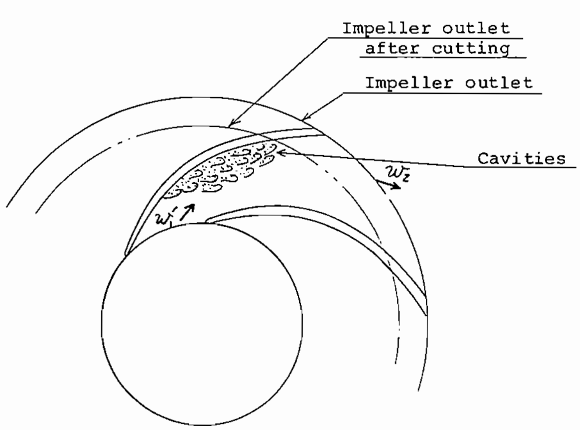



Figure 17. Impeller with High Specific Speed.

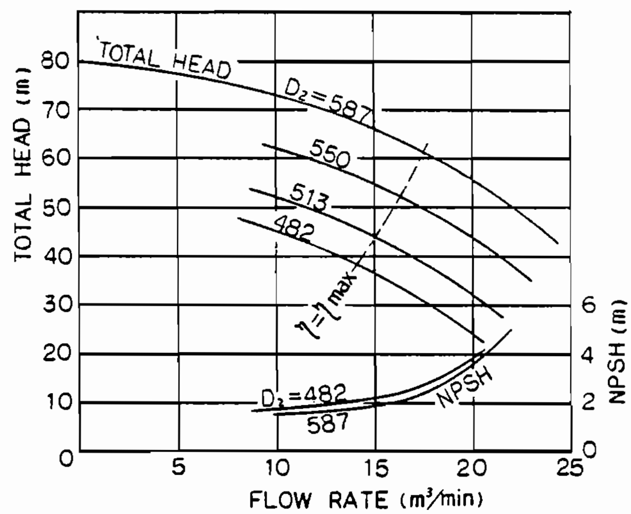



*Figure 18. NPSH for Various Impeller Outlet Diameters $D_2$ of a Centrifugal Pump with a Low Specific Speed; Rotational Speed $n = 1190\ min^{-1}$*

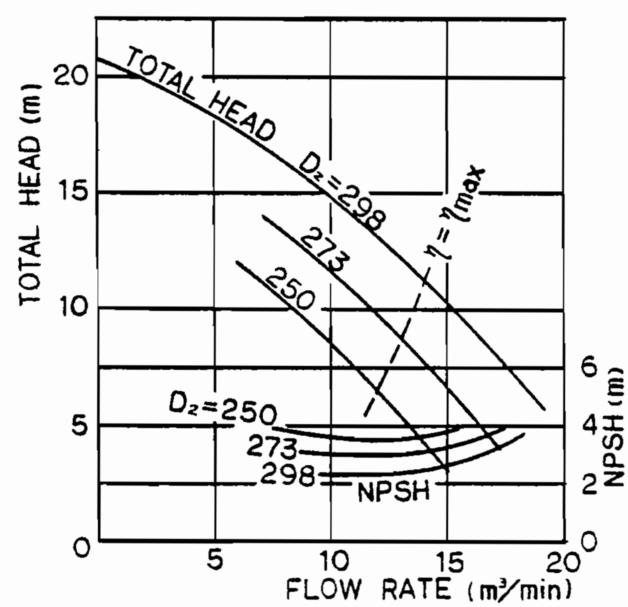



*Figure 19. NPSH for Various Impeller Outlet Diameters $D_2$ of a Centrifugal Pump with a High Specific Speed; Rotational Speed $n = 1190\ min^{-1}$*

age of flow passage produced by the cavities becomes more sensitive in a high Ns impeller. Depending on the amount of trimming, the impeller outer diameter approaches the cavitating zone and the complete restoration of pressure after the cavities have collapsed does not become sufficient. This results in a relatively large head drop and a marked deterioration in suction performance, when compared with a low-Ns impeller.

It has been shown that the impeller diameter of a centrifugal or mixed-flow pump has a strong influence on pump NPSH, and that the rate of NPSH variation is largely controlled by specific speed. The effect of impeller trimming on NPSH due to the specific speed is shown in Figures 18 and 19. Figure 18 shows the results on a pump with Ns = 140 and Figure 19 illustrates the results on a pump with Ns = 560, which is very large for a centrifugal pump.

## CONCLUSION

It has been proven that impeller diameter and NPSH are related and this relationship becomes more acute as suction specific speed increases. The authors' theoretical justifications have also been presented.

This paper illustrates that Pump Specific Noise, for single-stage double-suction volute pumps, increases in proportion to the extent that impeller diameter is trimmed from full size.

As a result of the above findings, the pump user and/or engineer should thoroughly investigate the pump suction conditions when considering a reduced impeller diameter for this type pump.

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