THE SHORTCOMINGS OF USING PUMP SUCTION SPECIFIC SPEED ALONE TO AVOID SUCTION RECIRCULATION PROBLEMS

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ABSTRACT

There is a mistaken belief in the pump industry (fueled by many publications) that the best way to avoid suction recirculation problems is to design and/or apply only pumps having low suction specific speed values (below 8,500 to 11,000). Attempts are made to correct this overly simplified treatment of a complex problem, which if used alone, can still lead to field problems, or, unnecessary and expensive over design. A more accurate, but still simple, method is presented as an alternative for identifying pumps which may be susceptible to suction recirculation problems (damage) during reduced capacity operation. Pump type, impeller inlet tip speed, impeller vane overlap and fluid specific gravity have been added to pump suction specific speed to greatly improve the prediction process.

INTRODUCTION

There are many published articles which infer a direct link between suction specific speed and suction recirculation damage, with little or no recognition of other equally important factors. Suction specific speed is defined as:

$$S = \frac{N\sqrt{Q}}{(\mathrm{NPSHR})^{.75}} \quad \text{(@ BEP Capacity \& Max. Impeller Dia. (US Units))}$$

Hallam [1] states that "centrifugal pumps with a suction specific speed (S) greater than 11,000 failed at a frequency nearly twice that of centrifugal pumps with suction specific speed less than 11,000." He blames this higher failure rate on "A centrifugal pump impeller with a high S, has a large impeller eye or an inlet vane angle, B, such that the impeller is susceptible to inlet eye recirculation." Lobanoff and Ross [2] state that one of the steps to avoid cavitation in a centrifugal pump is "not (to) select a pump with a suction specific speed above 10,000." Fraser [3], states that "There is no question that many pump installations are operating today either continuously or intermittently in suction or discharge recirculation. This is especially true for pumps designed for *high suction specific speeds.* The history of these pumps will show a pattern of cavitation damage, noise, rotor oscillation, shaft breakage or surging in varying degrees depending on the pump design and application. Many of these problems can be avoided by designing the pump for lower suction specific speed values and limiting the range of operation to capacities above the point of recirculation." The 14th edition of the "Hydraulic Institute Standards" (4) states that "Increased pump speeds without proper suction conditions can result in abnormal wear and possible failure from excessive vibration, noise, and cavitation damage. Suction Specific Speed Available, SA, has been found to be a valuable criterion in determining the maximum permissible speed. The curves presented in this standard are based on SA of 8500; this represents a practical value." It states that "Suction specific speed required, S, must equal or exceed the suction specific speed available, SA, to prevent cavitation."

Suction specific speed alone has not, however, been able to explain why pumps with suction specific speed values as low as 6,900 have experienced recirculation damage, while pumps with suction specific speeds as high as 18,000 and higher can operate over their entire flow range without any detrimental effects. The preceding publications are correct to the extent that the suction specific speed of a pump is one factor in determining whether or not a pump will experience suction recirculation damage, but it is not the only factor. Budris [5] lists four other (equally important) factors which, when properly combined with the pump suction specific speed, have proven to be able to predict suction recirculation damage with a higher degree of accuracy. The additional factors are the pump type, determined by the number of right angle turns the liquid must make at the inlet to the pump, the tip speed of the impeller inlet, the specific gravity of the fluid pumped, and existence of impeller vane overlap. Even though there is a certain relationship between the suction specific speed of a pump and the size of the impeller inlet, as suggested by Hallam, there are other design methods for improving the $NPSH_R$ of a pump than by just increasing the impeller eye diameter, so it is only a loose tie in.

Further, the suction specific speed of a specific pump remains generally constant, regardless of the speed of operation, while the impeller inlet vane tip velocity changes directly with pump speed. This is especially important in today's age of variable speed drives. The impeller eye diameter (tip speed) must, therefore, remain an independent factor from suction specific speed in determining the likelihood of damage from suction recirculation. For example, a particular pump with a suction specific speed of, say, 12,000 may not experience any suction recirculation problems at low flowrates when operated at 1200 rpm or 1800 rpm. However, when this same pump is operated above 3000 rpm it may begin to experience suction recirculation damage, even though the suction specific speed has basically not changed.

PUMP TYPE

The relative susceptibility of a centrifugal pump to damage from suction recirculation at low flow operation is affected by, among other things, the number of right angle turns the liquid must make in the inlet of the pump. Thus, an axial flow impeller and inducer (Figure 1), which has no turns at the vane inlet has the best performance. An end suction radial flow impeller (Figure 2), which has one right angle turn as the liquid is picked up by the vanes, is second best. Finally, a radial suction, radial flow impeller (Figure 3) is the worst performer, as it has two right angle turns, one in front of the impeller, and one as the liquid is picked up by the vanes. The effect of pump type is shown in Figure 4 [5] for the susceptibility of a pump to damage from suction recirculation at reduced capacities. Above the upper suction recirculation factor (SRF) lines, many pumps can be expected to experience problems if allowed to operate at or below the start of suction recirculation without providing an adequate NPSH safety margin, while below the lower SRF curves, minimum flow restrictions are not required, except to prevent thermal build up. Between the "upper limit of no pump problems" and the "lower limit of potential pump problems" is a gray (shaded) area where damage cannot be accurately predicted, due to the many influences not included in the SRF method. Mean lines, based on plots of approximately 200 actual pumps (as shown in Figures 6, 7, and 8), are also presented for each pump type in Figure 4. It is interesting to note that the mean radial suction pump line crosses the lower suction recirculation factor (limit) line for radial suction pumps at a suction specific speed of about 9,000, close to the Hydraulic Institute 8,500 recommended limit. Further, the mean end suction pump line crosses the lower suction recirculation factor line for end suction pumps at an S value of 12,000, not far from Hallam's 11,000 start of increased field problems with API type pumps. Most pumps in a refinery are of the end suction type.

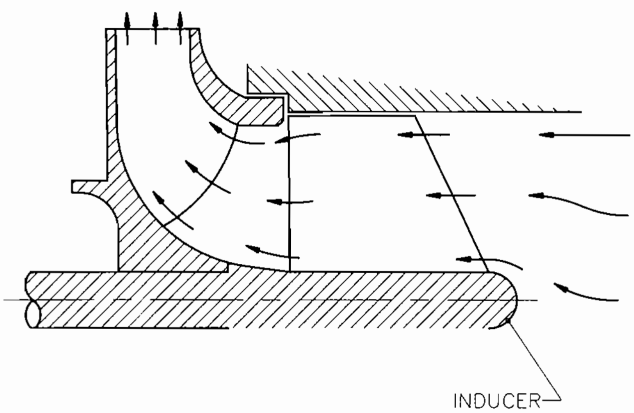

Figure 1. End Suction Induced, with Impeller.

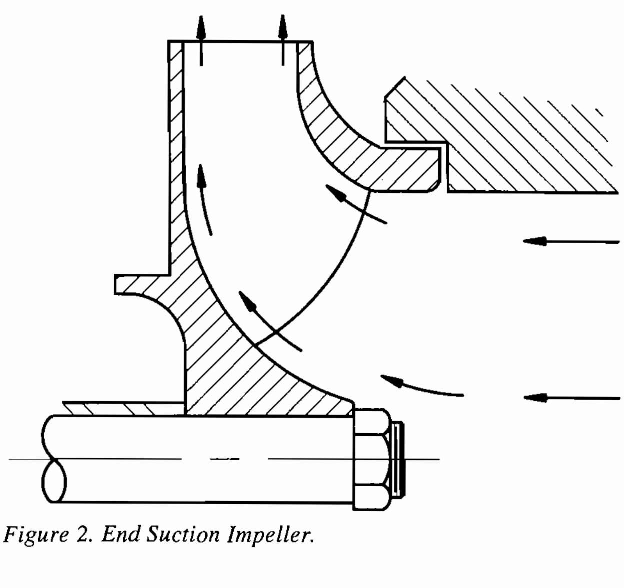

Figure 2. End Suction Impeller.

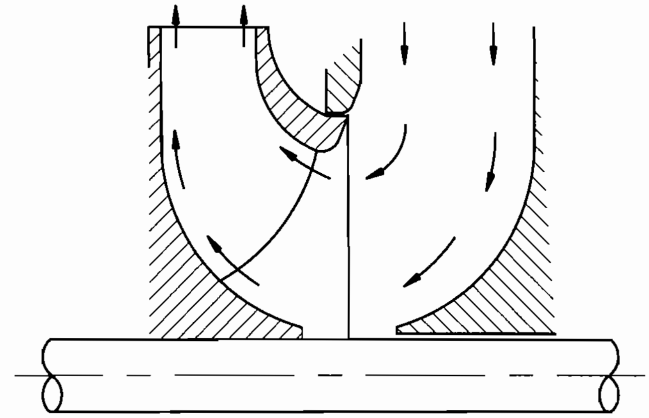

Figure 3. Radial Suction Impeller.

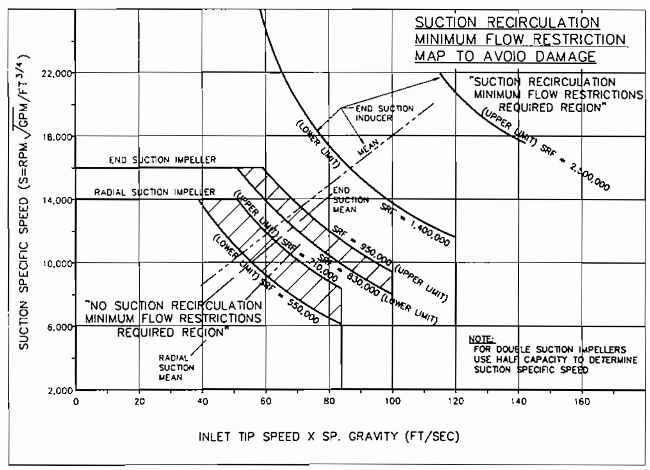

Figure 4. Suction Recirculation Minimum Flow Restriction Map to Avoid Damage.

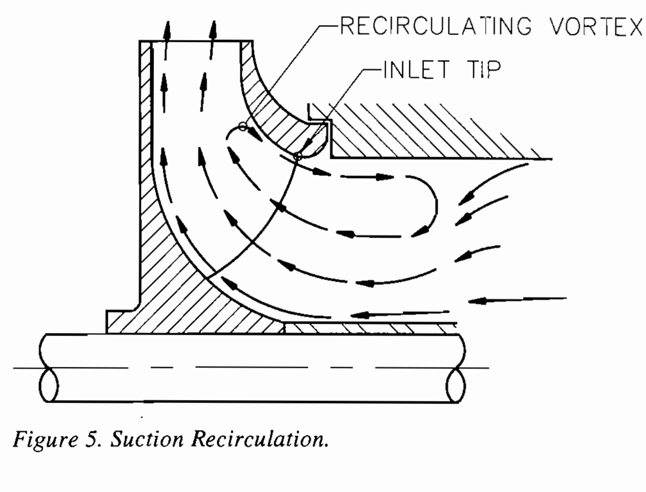

Figure 5. Suction Recirculation.

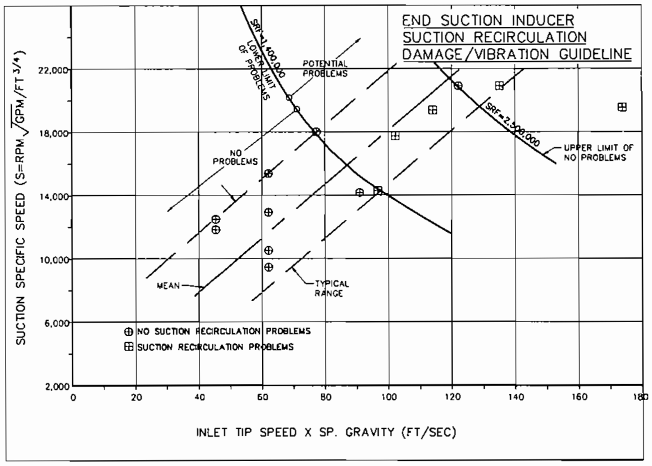

Figure 6. End Suction Inducer, Suction Recirculation Damage/ Vibration Guide Line.

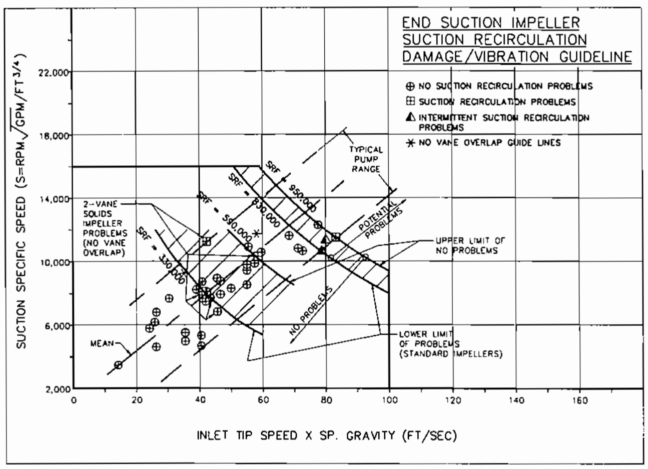

Figure 7. End Suction Impeller, Suction Recirculation Damage/ Vibration Guide Line.

FIG. 8

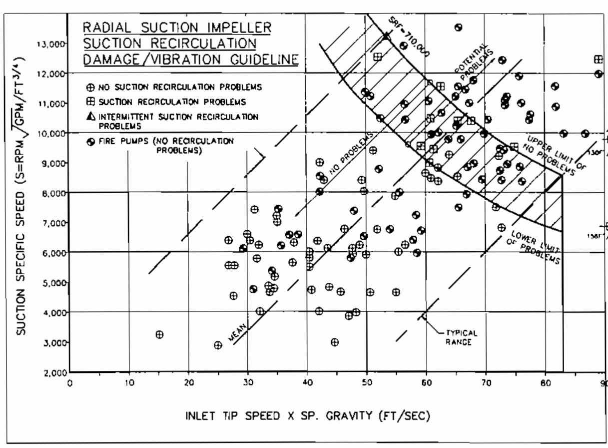

Figure 8. Radial Suction Impeller, Suction Recirculation Damage/ Vibration Guide Line.

Not all pumps fall into one of these three basic pump types, however. A vertical in line or self priming pump has a right angle turn in front of the impeller, like a radial suction pump, but does not have a shaft through the eye. The performance of these in-between pump types would be expected to fall somewhere between that of the two basic types, depending on how sharp the inlet turns really are.

INLET TIP SPEED

Within any given pump type, the inlet tip speed of the impeller vanes is the most critical single factor which determines recirculation intensity in a pump. If the tip speed (energy imparted to the liquid) is too high in relation to the through flow velocity momentum (energy) of the approaching liquid, the liquid in contact with the outer portion of the inlet vanes will be thrown back into the inlet of the pump (suction recirculation), Figure 5. Many publications substantiate this basic fact. The point of contention is not that a large impeller eye will cause suction recirculation and possibly damage, but that tip speed is independent, to a large degree, of suction specific speed. If all pumps fell on the mean pump lines shown in Figure 4, then inlet tip speed would not have to be considered separately. However, as shown in Figures 6, 7, and 8, not all pumps fall on these mean lines. There is a wide spread in performance, due in part to the fact that a given pump, with a given suction specific speed, can operate at more than one speed (inlet tip velocity). Further, such factors as vane blockage, inlet vane angle (*vs* flow angle), and inlet throat area can also affect the $NPSH_R$ and suction specific speed of a pump, without affecting the size of the inlet eye or inlet tip speed. For a given value of suction specific speed, the spread in tip speed is such that the highest tip speed is approximately double or greater than the lowest value. For example, given a 10,000 suction specific speed radial suction impeller pump (Figure 8) with a low inlet tip speed, say 40 ft/sec, one would not expect to experience any significant damage from suction recirculation. On the other hand, given had a second radial suction impeller pump, also with a suction specific speed of 10,000, but with an inlet tip speed of 80 ft/sec, there would be a high probability of experiencing problems from suction recirculation at low capacities with this second pump. Schiavello [6] shows several examples of cavitation/suction recirculation field problems, where the solution was actually an increase of the pump suction specific speed. He accomplished this with better matching of the impeller inlet vane angle with that of the incoming flow, while actually reducing the inlet tip speed.

VANE OVERLAP

For most impeller designs, the inlet performance is unaffected by the discharge of the impeller, except when there is very little or no vane overlap, such as in a two vane solids handling impeller, or an impeller with a severe impeller cut down (trim), Figure 9. With this type of design and under low flow conditions, a portion of the impeller discharge flows back between the vanes to the suction, causing suction recirculation. This explains why end suction two vane impellers are prone to suction recirculation damage, at a much lower suction recirculation factor value, as shown in Figure 7, than more typical end suction impeller designs.

SPECIFIC GRAVITY

Given the fact that virtually all centrifugal pumps will experience suction recirculation at reduced capacities (normally in the 50 percent to 85 percent of best efficiency capacity range), the likelihood of damage from suction recirculation becomes a matter of energy level at the suction of the pump. As already indicated, the tip speed of the leading edge of the impeller vanes is a measure of the inlet energy. Further, the number of turns the liquid must make at the inlet adds turbulence (energy), and the lack of vane overlap allows the high energy discharge liquid to enter the suction.

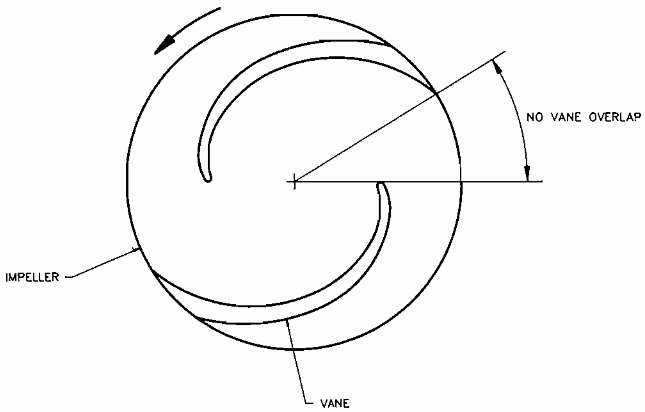

Figure 9. Impeller Vane Overlap.

Following the same logic, the specific gravity of the liquid which determines the energy level for a given head of liquid, is, therefore, another factor adding to the energy level at the inlet of the pump, and must be included. Entrained air will reduce the net specific gravity of the fluid pumped and cushion the cavitation bubble collapse, which can reduce recirculation damage.

COMBINING/USING THE KEY FACTORS

The best way to use the above factors is to combine them by taking the product of the suction specific speed (S), impeller inlet tip speed (U1), and liquid specific gravity (SG) for the specific pump type application in question. This combination is referred to [5] as the "Suction Recirculation Factor" (SRF).

Radial suction impellers can be either of the single or double suction type. For double suction impellers (normally found in radial suction impeller pumps), the pump capacity must be divided by two for the suction specific speed calculation.

The resulting SRF should then be compared with the limits for the pump type/vane overlap in question (i.e., Radial suction, end suction, axial inducer, or no vane overlap impeller), as shown in Figures 6, 7, and 8. Below the lower SRF limit, pumps have not been found to experience damage from suction recirculation when operated at reduced capacities. On the other hand, a large number of pumps with SRF values above the upper SRF limits have been found to experience problems when they were operated in the suction recirculation flow region, especially when they had inadequate NPSH safety margins. This simplified suction recirculation damage prediction method leaves a gray area where risk of damage can not be accurately predicted due to other factors not included (see "other factors" below).

Exceptions to the SRF upper limits set down in this paper are, however, possible. Radial split case, radial suction *Fire Pumps* (Figure 8), due to their limited actual operating time and bronze impellers, are able to operate at SRF values as high as 1,100,000. Eighteen percent of the fire pumps investigated had SRF values above the SRF upper limit (Note: 78 additional fire pumps with SRF values below the lower SRF limit are not shown in the graph due to insufficient data available). Pumps in other applications can also operate in the "forbidden" region if NPSH margins are high enough, highly cavitation resistant materials are used, and/or some damage can be tolerated. Other approaches should be used to evaluate the quantitative effects of these and other factors on pump reliability.

The SRF limits by pump type and vane overlap are listed below. The "lower SRF limit" should be used for critical or severe services, or where no prior successful experience is available (above the lower SRF Limit) with the pump at reduced capacity operation. Otherwise, the "upper SRF limit" should be chosen, also unless successful field experience is available. When operating with a pump SRF value above the selected SRF limit, a suction recirculation related *minimum flow restriction* is normally required. In other words, the capacity at which suction recirculation starts in the pump should be calculated using Fraser [3] or Gopalakrishnan [7] (or another proven method), and the operation of the pump restricted to flowrates above this calculated recirculation inception point, unless a substantial NPSH safety margin is provided. If, on the other hand, the calculated SRF value falls below the appropriate SRF limit, then the pump should not experience damage from suction recirculation, even when operated at low capacities (assuming the flowrate is maintained above the minimum flowrate required to prevent over heating of the pump). A suction recirculation minimum flow restriction should then not be necessary.

	SRF Lower Limit Problem Pumps	**SRF Upper Limit No Problem Pumps**
No Vane Overlap End Suct.	330,000	590,000
Radial Suction	550,000	710,000
End Suction Impeller	830,000	950,000
Axial Flow Inducer	1,400,000	2,500,000

Where:

The above SRF guidelines are presented graphically in Figures 4, 6, 7 and 8.

SRF = S × U1 × S.G. (Suction Recirculation Factor)

In which:

$$S = \frac{N\sqrt{Q}}{(NPSHR)^{.75}}$$ (Suction Specific Speed @ BEP Capacity and Max. Impeller Dia.- US Units)

N = Pump / Impeller Speed (RPM)

Q = Pump Capacity (USGPM)

NPSHR = Net Positive Suction Head Required (Feet)

U1 = Peripheral velocity in impeller eye at vane leading edge maximum radius (ft/sec.)

S.G. = Specific gravity of liquid pumped.

OTHER FACTORS

The above five key factors (including suction specific speed) are not the only parameters affecting the likelihood of damage from suction recirculation, however, they seem to have a key influence and are the easiest to determine. Other factors which help explain the gray areas between the upper and lower suction recirculation Factor limits, and why some pumps can operate above the upper SRF limits are:

- Suction recirculation damages pump impellers by generating high velocity vortices which initiate cavitation. The overall amount of cavitation is determined by the margin between the NPSHA (available from the system) and the NPSHR (required by the pump). It takes an NPSH margin (ratio of NPSHA/NPSHR) of from two to 20 to fully suppress cavitation in a pump. Within the "no suction recirculation minimum flow restriction required regions" (Figure 4) a margin in the 1.0 to 1.7 range (depending on inlet tip speed, incidence angle, and liquid pumped) is normally adequate to avoid damage. Above the upper SRF limit, a margin of 2.0 or greater is normally required.

- The greater the velocity distortion into the impeller, the lower will be the local pressure, and the more likely the formation of cavitation bubbles. This is especially important with radial suction impellers.

• The materials of construction are important, since harder/tougher materials, like stainless steel, are more resistant to cavitation damage than softer, less ductile materials, such as cast iron. The SRF limits listed here are most applicable for cast iron construction. The SRF limits can be increased for superior materials.

• The greater the impeller vane incidence angle, with the approaching flow, the greater the turbulence and cavitation. The incidence angle increases as pump operation moves away from the shockless capacity, which is usually at or above the bep capacity. The percent bep capacity is, therefore, a factor.

• The number, shape, thickness, location, and curvature of the inlet portion of the impeller vane, and the resulting throat area, all affect the smoothness of the pickup of the fluid, and local velocity levels. These all play some small part in the amount of cavitation formation in the impeller inlet.

• The size, head per stage, and total energy of the pump are additional factors suggested by some authors as having an influence on suction recirculation damage. High energy pumps, especially pumps with high suction energy, are more prone to damage when operated above the SRF limits listed.

• Finally, recirculation damage is a function of time, and the less a pump is operated the longer it will survive at higher SRF values.

CONCLUSIONS

It has been shown that avoiding centrifugal pump suction specific speed values over 8,500 to 11,000 is an imperfect way of preventing damage from suction recirculation, or selecting the optimum pump for an application. Using the SRF (suction specific speed × inlet tip speed × specific gravity) limits based on one of four basic pump types/vane overlap has proven to be a more comprehensive and realistic method. Applying this SRF approach to the six cavitation/suction recirculation field problems covered by Schiavello [6], showed excellent agreement. The one complaint (case) where no damage was found had a SRF value below the lower limit. The case where one pump had a problem and the identical sister pump did not was between the upper and lower SRF limits. The other four verified cavitation/suction recirculation problems were all well above the upper SRF limits. On the other hand, only two of these six field complaints had suction specific speed values above 11,000. The solutions to these field problems was to *increase* the $NPSH_R$ (suction specific speed) value to provide an increased margin by reducing the incidence angle, and at the same time reducing the eye diameter. This actually allowed the pumps to operate above the SRF limits in some cases due to the higher NPSH margin.

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