TUTORIAL SESSION
on
CAVITATION AND RECIRCULATION FIELD PROBLEMS

TUTORIAL LEADER

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ABSTRACT

High reliability and large rangeability are required of pumps in existing and new plants which must be capable of reliable on-off cycling operations and specially low load duties.

The reliability and rangeability target is a new task for the pump designer/researcher and is made very challenging by the cavitation and/or suction recirculation effects, first of all the pump damage. The present knowledge about the: a) design critical parameters and their optimization, b) field problems diagnosis and troubleshooting has much advanced, in the very latest years.

The objective of the pump manufacturer is to develop design solutions and troubleshooting approaches which improve the impeller life as related to cavitation erosion and enlarge the reliable operating range by minimizing the effects of the suction recirculation. This paper gives a short description of several field cases characterized by different damage patterns and other symptoms related with cavitation and/or suction recirculation. The troubleshooting methodology is described in detail, also focusing on the role of both the pump designer and the pump user.

INTRODUCTION

From the late 50's until the early 70's there was a continuous growth of the industrial countries economy, which led to the design and installation of larger and larger plants (fossil power plants and process plants).

Both energy level (head/stage) and capacity of pumping equipment increased drastically. In the area of power plants the head/stage increased by a factor 4 and the impeller peripheral speed by a factor 2. In the area of process plants pump maximum capacity was more than doubled and also the head increased to meet higher piping losses associated with large and complex plants and high capacity as well.

Moreover plant engineering contractor put high emphasis on low capital cost. Plant availability, which would require redundancy and so several units in parallel, received less consideration. Also pump users gave keen attention to low energy operating cost at full and maximum load of pumps.

Therefore the pump designer was urged to develop smaller and consequently faster pumps and also maximize efficiency at high flows. In addition lower NPSHA levels, which were dictated by low plant installation costs, had to be met at maximum capacity (runout condition). Impeller designs were finalized for high suction specific speed, S, and also inducers were widely applied. Moreover it was required to avoid sharp increase of the NPSHR curve (as based on 3% head drop) at high capacity, still maintaining high efficiency. The design solution was then to open the throat area of the blade channel at the impeller inlet in order to reduce the impeller sensitivity to cavitation blockage on through flow. Consequently both the impeller eye diameter was enlarged and also the blade inlet angle was increased, moving the shockless capacity well above the b.e.p. capacity. As a result the pump behavior was penalized at part flows, in combination with more severe requirement in terms of cavitation at duties above the best efficiency capacity.

In the late 70's and early 80's a slowdown of world economy followed. The overall production capability, for which plants and equipment were optimized, was exceeding the market demand. Then process plants were forced to operate at reduced capacity. In the utility area basic loads were picked by nuclear plants, while fossil power plants moved to cycling load along with their pumping equipment (namely feed water and booster).

As a result more and more high energy pumps were operating in a broad range of capacity including long duties at flows substantially below the b.e.p. capacity, and, especially, much lower than the impeller inlet optimum capacity or shockless capacity.

Then frequent failures surfaced which were characterized by heavy damage at the impeller inlet and outlet, pressure pulsations and vibrations with wide band and random frequency spectra.

Key experimental research data on pump behavior at low flows, which showed the occurrence of internal flow recirculation (1,2,3,4,5,6,7,8,9) and local cavitation with associated metal damage, were meanwhile published and gave some clear insights on the above failures. Also, other pump failures were observed with increasing frequency which presented classical cavitation damage aspect, as surface pitting, even with NPSHA level well above the NPSHR. A basic study of cavitation inception and growth had already been published since 1941 (10). However, extensive, experimental research on cavitation in pumps has been carried out only in the last two decades mostly concentrating on cavitation visualization (transparent test models and stroboscopic light) (1, 11, 12, 13, 14, 15) and acoustic detection of high frequency cavitation noise spectra (16). Then it became fully evident that cavitation inception occurs at NPSHA level (NPSHi) 5 to 20 times higher than the conventional value of NPSH corresponding to 3% head drop while cavitation damage occurs for NPSH level below the inception point but still higher than NPSHR, depending on various factor (17), first of all peripheral velocity at the impeller eye and pump operating capacity as fraction of the shockless capacity (11, 12).

While new criteria for establishing adequate NPSHA level started to be developed, (11, 12, 13, 17, 18, 19) pump designers were more and more called to solve field troubles related with cavitation and/or suction recirculation.

CAVITATION MODES

Blade Attached (or Sheet) Cavitation

A large number of cavitation visualization studies in pumps can be found in the literature. They are aimed at detecting the cavitation inception point, when the first cavitation bubble becomes visible, by using special transparent experimental models and a stroboscope light. According to the classic experiments of (1) with end suction pumps the curve of the NPSH at the condition of visual inception versus the pump capacity has a very peculiar shape, as shown by the top curves in Figure 1. The NPSHi (i = inception) has a minimum at a capacity which corresponds to shockless inlet flow (Qsl), i.e., the relative flow reaches the blade leading edge with an incidence angle around 0°. The NPSHi increases at $Q > Qsl$ and $Q < Qsl$, with cavitation starting on the pressure (hidden) and suction (visible) side of the blade respectively. At part flow the NPSHi peaks at a capacity slightly higher than the critical suction recirculation onset capacity, Qcrs (cr - critical, s - suction) (1, 20). A similar V-shape for the incipient cavitation curve was also found by using a small head drop criterion (0.5% of the head of twice the impeller eye peripheral velocity) for overhung pumps (10). Again, the curve exhibited at part capacity a peak, which is attributed to a critical incidence angle causing flow separation (10) or "stalling incidence angle" (8, 20).

At the point of the visual cavitation inception, the rate of the erosion damage is practically zero. Visual studies of cavitation show that more and more vapor is generated while the suction pressure or NPSH is continuously decreased during classical σ tests (i.e. test of head decay at constant rotational speed and constant capacity with decreasing NPSH). The vapor tends to coalesce and then forms a large cavitation bubble of increasing length. Pumps operate in the field with a NPSHA (A = Available) which is higher than the NPSHR but significantly below the NPSHi. Therefore they operate with bubble length at the NPSHA which varies widely from 0.5 inches to 4 inches or more depending upon the operating point (speed, flow, temperature) and the impeller design.

In order to produce damage the vapor bubbles must collapse in the vicinity of the metal surface. Normally it occurs for the regime characterized as "blade attached (or sheet) cavitation", which is more common in the usual capacity operating range. In this cavitation mode the curve of the cavitation erosion rate (ER = MDP/Time, where MDP = Mean Depth Penetration) versus capacity at constant speed/NPSHA has a peculiar V-shape, with minimum at the shockless capacity (Figure 2), which is similar to the NPSHi curve. The damage develops as pitting on the blade pressure side for flowrates above the shockless capacity. At flowrates below the shockless one the cavitation damage occurs on the visible side of the vane. Recent research (21) has demonstrated that in this cavitation regime the erosion rate, expressed as damage depth-to-operating time ratio (inches/year), is proportional to the bubble length (exponent 2.7) and the NPSHA (exponent 3.0), for given fluid properties and impeller material. Therefore for given impeller life, say 40000 hours, the acceptable cavitation bubble length is very much shorter for pumps running at high impeller eye speed and so operating with high NPSHA than for small pumps running with low impeller eye speed, which implies a low NPSHA. Correspondently, the acceptable cavitation bubble length can vary from 0.5 inches to 4 inches.

Cavitation Induced by Suction Recirculation (Vortex Cavitation)

Visual observations with stroboscopic light show that the cavitation bubble on the blade suction side becomes more and more unstable as the capacity is continuously decreased below the suction recirculation point towards shutoff. The bubble length changes more or less periodically with time, even disappearing for a fraction of time. Moreover, cavitating bubble clouds separate from the blade suction surface and move into the blade channel.

Essentially a new flow regime takes place which is characterized by "Strongly intermittent cavitation -suction recirculation" (15). As a generic indication such very unsteady flow regime occurs in the capacity range from 0% to 50%, roughly. However, the upper capacity limit can reduce or increase even close to the suction recirculation onset capacity depending on the impeller design and impeller eye peripheral velocity (Ueye) and NPSHA level.

Experimental investigation by means of a high speed movie camera along with stroboscope (22) clearly shows that at low flowrate two different patterns of cavitation i.e., "sheet cavitation" and "vortex cavitation" occurred alternatively near the leading edge of the impeller blades, as schematically shown in Figure 3. The cavitation started on the blade suction surface far away from the leading edge and moved upstream with an abrupt stroke and collapsed on the pressure surface of the next blade. This cavitation called "vortex cavitation" is attributed to the impeller suction recirculation. In fact, a vortex is generated by the shear forces at the interface between the reverse flow leaving the impeller near the front shroud and the ordinary forward flow entering into the impeller near the hub, as shown in Figure 4a. Moreover streams of both backward and forward flow can be suspected to occur also in the blade-to-blade plane in the inlet region of the blade channel, as sketched in Figure 4b. Then shear forces components exist also in this plane and contribute to generate a complex vortex in the three-dimensional space. When the inlet pressure (therefore, NPSHA) is low enough and also the strength of the vortex (therefore, intensity of the suction recirculation) is high enough, then the pressure in the vortex core drops below the saturation pressure and cavitation conditions are reached. A filament of cavitating flow develops which is starting on the suction side of the blade and is ending on the pressure side of the next blade, as shown in Figure 4c. Moreover, this vortex oscillates in a direction normal to the blade surface i.e., more or less in the direction of the main flow as sketched in Figure 4d. Consequently, damage is caused in form of a single large crater at the midspan of the blade on the pressure side.

A typical curve of NPSHd (d = damage) which can produce significant erosion damage throughout the whole range of operations is shown in Figure 1. The NPSHd is not unique and depends upon the desired impeller life, the pump design, the material characteristics, the fluid density and temperature. The basic engineering problem is to determine how much erosion damage can be allowed, under the operating conditions, to get a reasonable impeller life of several thousand hours and so an economically acceptable pump reliability. Moreover, rangeability and efficiency have to be included in the balance.

Inlet Flow Influence

A strong influence on cavitation inception (NPSHi) and damage (NPSHd) is produced by the flow distribution at the impeller eye, as induced by the upstream geometry i.e., inlet chamber and/or suction piping.

For side suction pump the shape of NPSHi curve can be strongly altered by the suction casing, which tends to displace and smooth the minimum and the peak.

The degree of distortion of the inlet flow becomes stronger with increasing capacity. Visual observations clearly show (15) that both the shape and the size of the cavitation bubble on each impeller blade is a periodic function of time, as the blade crosses flow zones with positive flow swirling either in the same direction of the impeller rotation (less intense cavitation) or in the opposite direction (more severe cavitation). Thus pressure pulsations and vibrations are induced by the inlet flow distortion. They reach the highest peak-to-peak values at the runout operating point, which usually is close to the pump maximum brake-horsepower. Areas of damage can be produced on both the surfaces of the blade but the erosion is prevailing on the pressure side.

Moreover, field experience indicates that with double suction impeller, the cavitation damage pattern (like pitting) may be different for each impeller eye, thus suggesting there is a flow imbalance forced by the upstream geometry (suction piping and/or pump inlet bay).

Cavitation Due to Secondary Flow at Blade Fillets (Corner Vortex)

In many cases cavitation damage has been found at the fillet between the blade suction side and the impeller hub surface. The damage appears to be caused by a strong vortex, which is confined in the blade root-to-hub corner and generates a drilling action leading to rapid perforation of the impeller hub and in many cases to shaft damage. The flow sources of such corner vortex are the intense shear forces associated with the secondary flows pattern due to the interaction of the blade surface velocity profile and the boundary layers of the impeller hub surfaces. Flow separation may be a contributory source, but not necessarily.

CASE NO. 1 - BOILER FEED PUMP FOR 660 MW UNIT

Case Failure Analysis

A cavitation erosion problem is presented which occurred in four full capacity main boiler feed pumps operating in the same power plant. Each pump was turbine driven with its booster pump in closed loop.

The plant, having four generating units of 660 MW each, had been planned for basic and high load duties. Large full capacity pumps were selected as peak efficiency was valued more than reliability at part load operations. The pump design target was: N = 5200 RPM, Q = 10,400 GPM, H = 12,850 FT, NPSHR = 325 FT, NPSHA = 560 FT, fluid temperature = 304°F. The stage specific speed, selected for high peak efficiency was around 1700 (US units) leading to 6 stages. Therefore, the BHP/stage was about 6300 HP indicating a very high energy pump. The design suction specific speed of the first stage impeller was around 6,900 (US units). The minimum continuous flow was specified at 50% b.e.p. capacity.

The four units started to operate in the early 80's and had different service history also with inactive periods. In March 1986 the main feedwater pump of Unit No. 1 with the longest service time was internally inspected and a severe cavitation damage was noticed in the first stage impeller. Thereafter the inspection of the main pump was extended to Units No. 2-3-4 revealing similar damage characteristics for all four pumps. The failure analysis indicated that (23):

- The erosion area was localized near the corner between the suction side of the blade root and impeller hub, as shown in Figure 5. The total operation time was 14,000 hours for Unit No. 1 and 6,000 hours for Unit No. 4. Both the location and the pattern of the eroded area were quite unusual. Contrary to literature indications, the blade surface was only slightly damaged, while the cavitation erosion perforated all the way thru the hub thickness and also penetrated into the shaft.

- The impeller of Unit No. 4 operated for about 70% of the time around its b.e.p. capacity (close to 500 MW output), while the impeller of Unit No. 1 operated for more than 50% of the time at part load (Figure 6). The impeller eye speed was ranging from 170 FT/S to 200 FT/S.

- The NPSHA at pump b.e.p. was about 1.7 times the NPSHR, which is inadequate for this application according to some recent literature (12).

Case Solution

A new design impeller (impeller B) which was considered as the most effective step to reduce the cavitation damage, was urgently developed and installed in the plant in October 1986. Also a better material from cavitation resistance standpoint was used i.e., CA6NM instead of CA15 (damaged impellers). However, a parallel program of cavitation visualization tests on several design impeller variants models was planned to cover various hypothesis about the peculiar cavitation pattern and so identify the best solution for late field implementation. This experimental investigation was carried out in early 1987.

Several geometrical configurations have been selected and tested, including the one (impeller A) which experienced in the plant the cavitation erosion pattern described above plus three impeller variants (B, BM, B1) of new design. (23).

A key design goal was to reduce the sensitivity to cavitation erosion in a wide range of plant loads from 300 MW to 660 MW corresponding respectively to 50% and 120% of pump b.e.p. capacity. The analysis of the variation of NPSHA and pump rotational speed with plant load, showed that the most critical condition was at the maximum load. At part loads more large margins develop between the NPSHA and the impeller eye speed, which strongly determines cavitation erosion rate (12). Moreover, the future load spectrum versus operating hours, as expected by the user, was close to the one in Figure 6b. Then the shockless capacity giving minimum cavitation erosion was selected around 110% of the b.e.p. capacity for the impeller B, which was planned for immediate field installation (Conf. 2). Moreover, the overall vane loading was reduced for the impeller B, by lowering the head coefficient and also increasing the blade solidity (overlap).

The cavitation bubble length (on blade suction side) Lc,b from model tests simulating the plant NPSHA is compared for all the variants in Figure 7, assuming as reference the baseline impeller A at b.e.p. capacity.

For impeller B (Conf. 2) the bubble length Lc,b was drastically reduced across the entire operating range. Moreover the suction recirculation point was also lowered (65% Qbep) with respect to impeller A (90% Qbep). Further reduction of Lc,b was obtained with impeller BM (Conf. 4) by grinding the vane suction surface at inlet. The cavitation bubble length was also reduced with impeller B1 (Conf. 2) but less than for impeller B and BM.

The analysis of the shape of the cavitation bubble at the site conditions from model tests (24) produced clear insights about damage mechanisms. With reference to impeller A, the bubble pattern is peculiar, showing a triangular shape with high cavitation activity at the blade root and zero cavitation activity at the blade tip. This indicates the existence of a highly three-dimensional flow (even existing at b.e.p. capacity and above), which produces high shear forces. Therefore, intense local vortices are generated, which collapse at the impeller hub and cause the peculiar erosion pattern (Figure 5). With reference to impeller B, the cavitation bubble showed in the model tests a rectangular shape with more or less constant length from hub to tip which was indicating a relatively two dimensional flow field around the blade leading edge. Some experiments with soft paint indicated that the damage was spread over a band parallel to the vane edge with zero damage at the impeller hub. Thereafter, the blade was cut back at the hub causing a high positive incidence angle and then the shape of the cavitation bubble became triangular with the largest cavitation erosion at the hub, as shown by soft paint endurance test.

Impeller Life Expectancy (Theoretical).

The theoretical erosion rate (ER) has been calculated by using a correlation based on cavitation bubble length, which was first published in September 1986 (21). The trend versus capacity of the erosion rate for the blade suction side (ER) has been derived by using the visual cavity length Lc,b from model tests and is plotted in Figure 8 for all the four impellers. It appears that:

• The erosion rate has been reduced across the full range of continuous duties by at least one order of magnitude with impeller B and BM, and also with impeller B1.

• The cavitation erosion on the blade suction side is remarkably increasing at part capacities as compared to shockless capacity by a factor 3 to 5 for the best impellers B and BM. Then the impeller life is reduced for large boiler feed pumps operating for long time at part capacities. Therefore, the minimization of the cavitation erosion at part loads is an inevitable but challenging task which must be accomplished by the pump designer/researcher in order to meet the needs of widely cycling power plants.

It is worthy to notice that the efficiency has been improved with all the new impellers at part capacity. One test variant has presented higher efficiency throughout the whole operating range.

Moreover, a newly published method (25) for predicting with probabilistic approach the expected impeller life based on cavitation bubble length has been used for all the three new design impellers (24, 26). Plant data have been used for the load spectrum along with other operating conditions. According to this theory the probability of reaching the target impeller life of 40,000 hours with the impeller B is about 65%. The probability raises to about 80% if a maximum erosion depth EDmax equal to the full blade thickness is allowed. The impeller BM with very smooth thickness distribution at the blade inlet is even better than impeller B, with a theoretical probability of 90% to reach a 40,000 hours impeller life. On the opposite, the impeller B1 shows only a probability of 30% to 40%, due to higher erosion rate (Figure 8) as consequence of enlarged impeller eye (S = 10,100 US Units) even if the NPSHA-to-NPSHR margin is higher across the operating range than for impeller B (S = 8150) and BM (S = 8100).

The boiler feed pump of Unit No. 1 has been overhauled (steam turbine problems) in September, 1990 after 20,306 hours of operation with the new impeller B. The plant operation profile resulted to be very close to the one anticipated at the design stage. The inspection of the impeller has shown that:

• Some cavitation erosion has developed on the suction side of each blade at the midspan. The average of the maximum erosion depth (EDmax) is 0.08 inches. The average Lc,ed is 0.70 inches, while the average erosion length Lc,ed is 1 inch. Total absence of damage is noticed on the pressure side.

• The shaft damage has been totally eliminated, while some minor cavitation erosion at the fillet between the blade suction surface and the impeller hub is existing. The average maximum erosion depth is 0.04 inches and the average erosion length is 0.15 inches.

• On the basis of the actual field erosion depth developed in 20,306 hours the probability to reach the target life of 40,000 hours is close to 95% (EDmax equal to 75% of the blade thickness), while the theory (25) has indicated a probability of 65%.

More details about the comparison of the theoretical prediction of the erosion rate with field data for this pump case are presented in another paper (26).

CASE NO. 2 - BOILER FEED PUMP FOR 330MW UNIT

Case Failure Analysis

This case history is regarding a single main boiler feed pump (100% capacity) of a 330MW power plant (26). The pump, which is motor driven, operates with variable speed. The pump suction line is directly fed by the deaerator (open loop) and a double suction impeller is used in the first stage. The pump, which has eight stages was originally designed in 1965 for basic and high load duties. The original C.O.S. was: N = 3420 RPM, Q = 6250 GPM, H = 9140 FT, NPSHR = 51 FT, NPSHA = 110 FT, fluid temperature = 345° F. The design suction specific speed per eye of the first stage impeller was about 9.900 (US units). The power/stage is 1900 HP. The original first stage impeller has eye peripheral velocity around 145 FT/S. Thus it can be considered as relatively high energy/high speed stage.

A field survey in August 1988 showed that the first stage impeller has suffered some metal damage. The damage area was located on the pressure side (hidden side) of each blade, but only on the inboard impeller eye (pump coupling side). No damage was noticed on the impeller eye at the outboard side. A panel of experts concluded that the damage was caused by the suction recirculation due to both operation at part loads and not equal repartition of the total capacity between each impeller eye. They recommended to redesign the first stage impeller to lower the suction recirculation and also reduce the tendency to flow imbalance between the two impeller eyes.

Solution Methodology

Step 1 - Operator Input (Plant Data)

The plant operator was requested (August 1989) to:

• Supply data on the expected pump operating profile, as key input to the impeller design target. Such data included the base operating mode plus two potential alternatives.

• Revise the original c.o.s. and possibly limit the highest flow capacity (at plant full load) to the really expected service, in order to help the designer in optimizing the impeller geometry for part flow operations. The pump operating line, as expected, is shown in Figure 9 along with the pump performance map.

Step 2 - Impeller Design Strategy

An impeller design strategy was developed aimed at achieving the specified life of 40,000 hours with high probability. The strategy was focused on the following aspects:

• The full range of the operational parameters (Q, N, NPSHA) was analyzed as shown in Figure 10.

• The erosion rate (ER) prediction curve was derived for a preliminary impeller geometry. A theoretical correlation based on cavitation bubble length (21) was used, while the variation of the cavitation length with capacity and NPSHA and impeller geometry was inferred from existing internal data base of cavitation visualization model tests. It is clear from Figure 10, that the tendency to cavitation erosion is higher at the base load (300MW) and, especially, the full load (330MW), while it is lower at part load. However, the average level of the erosion rate is much lower than for Case No. 1 (Figure 8), due to lower impeller eye velocity and also higher NPSHA-to-NPSHR margins. This shape of the ER-curve rapidly decreasing with plant load seems to be peculiar of a pumping configuration with variable speed main feedwater pump and deaerator (i.e., essentially constant suction pressure) which leads to the amplification of the NPSHA margin as the load is reduced. It is also important to note that the erosion rate has tendency to sharp rise at high flow/high speed. This peculiarity suggests that an overflow at the impeller eye, which is due to a flow imbalance between the inboard and outboard eye of the impeller and produces a negative incidence angle, is the most likely cause of cavitation damage over the blade pressure side ("blade attached cavitation") in this installation, rather than a suction recirculation related damage ("vortex cavitation").

• The cumulative damage (EDmax corresponding to MDP = Mean Depth Penetration) was compared for the three cases of the pump operating profiles i.e., base case, Alt. 1 (full load at 330 MW) and Alt. 2 (Unit Operating in Partial Load Condition) as shown in Figure 11. It is clear that the Alt. 1 is the most severe in terms of cavitation damage. Therefore the shockless capacity for the new design impeller was selected for the full load operation at 330MW. However, the expected cumulative damage should not reach 75% of the blade thickness, even after 60,000 hours.

• The probability of achieving an impeller life of 40,000 hours (operator realistic target) and even 60,000 hours (operator ultimate goal) was also analyzed as shown in Figure 12. In general, the situation is more than satisfactory with lower probability for Alt. 1/60,000 hours. (26)

• An addition requirement (beside impeller life) from the operator was to extend pump reliable continuous operations down to 100 MW load. Then, the suction recirculation onset was also lowered to 60% of the shockless capacity i.e., Qcrs = 2650 GPM at 2606 RPM (100MW load). Then the minimum continuous flow of 2146 GPM corresponds to 80% of the recirculation onset capacity. This is acceptable as the new impeller has more mild recirculation intensity (smaller eye diameter and optimized blade geometry as enlightened by previous research data (8, 20, 27) than for the existing impeller which has Qcrs = 3200 GPM at 2606 RPM.

• Additional key operating requirements at full load were considered too in optimizing the impeller geometry (e.g. acceptable NPSHA/NPSHR to withstand transients caused by sudden loss of the suction pressure).

The suction specific speed was finally optimized for 8,500 (US units) as the best compromise between the above conflicting requirements at high loads and part loads.

Step 3 - Suction Casing Modification

In order to equalize the capacity between the inboard eye and the outboard eye of the impeller and mostly reduce the risk of negative incidence, it was necessary to streamline in a better way the inflow to the impeller on the inboard side of the inlet bay. This imposed to (Figure 13a,b):

• Reshape by machining the twin volute inlet flange.

• Remachine the suction end cover for modifying the edge to match the contour of the shaft sleeve.

• Give a better contour to the wear ring surface facing the suction channel.

• Produce a more smooth meridional contour near the impeller with shallowed shaft sleeve.

Step 4 - Advanced Design Flow Straightener

A flow straightener of advanced design was also included in the solution strategy in order to eliminate a possible flow distortion due to the suction piping layout. However, the installation of the flow straightener was left to future action by the operator, if needed.

Field Response

The boiler feed pump with the above modifications has started in July 1990. According to the plant operator, the pump can now be operated reliably from 100MW to a full load of 330 MW, while previously the operating range was more narrow from 150 MW to 330 MW. Presently, the operations at part load down to 100 MW are satisfactory without any indication of audible cavitation, contrary to the previous situation.

CASE NO. 3 - SCRUBBER RECYCLE PUMPS

Case Description

This case is concerning two recycle pumps (scrubber service) operating in a power plant. The pump (6 inches suction) is single stage with top-top flanges and side suction casing and double suction impeller. Each pump operates at fixed duty (Figure 14) i.e., N = 3550 RPM, Q = 1350 GPM, H = 520 FT, NPSHR = 12 FT with cold water, which is drawn from an open tank (NPSHA = 36 FT). The power absorbed is 253 HP. The peripheral speed at the impeller eye is 93 FT/S.

The operation mode includes full time service with hot weather (mostly summer) and no service at all with cold weather (mostly winter).

After about two years of service the pumps were inspected due to high vibration and the impellers were found to be damaged.

The erosion pattern was characterized by the following aspects:

• The metal damage was located on the blade suction side (visible side) near the hub and also at the corner between the blade and the hub.

• The damaged area over the blades show the pitting aspect which is typical of cavitation attack. Heavy damage penetration was concentrated at the corner between the blade and the hub.

• The above damage was present on all the blades at both the eyes of the impeller. But the extension of the damage was much different on each eye, being much worse on the outboard side. This indicates that a) the capacity through the outboard impeller eye is below 675 GPM (i.e., 0.5 x Qduty), b) the capacity through the inboard is higher than 675 GPM, but still producing high positive incidence angle and so cavitation.

• There was no significative damage on the pressure side of the blade, which at part flow (Qduty = 0.69 Qbep) usually is caused by the suction recirculation.

Failure Analysis

The failure analysis (pump and system geometry, impeller material, fluid properties) enlightened the following aspects:

• The suction piping layout (Figure 15) with two elbows produces a flow distortion with uneven distribution of the total flowrate at the pump suction flange.

• The impeller would have at 675 GPM (0.5 x Qduty) a relative flow incidence of 6° at the tip and 15° at the hub. Then the hub blade section is prone to flow separation, which produces a cavitating vortex (corner vortex). A strong separation/cavitation can be expected for capacity below 675 GPM (outboard side). On the other hand, cavitation can occur, even less intensive for capacity above 675 GPM (inboard side) due to positive incidence angle (blade attached or sheet cavitation) i.e. the flow angle is lower than the blade angle for the operating conditions, leading to high velocity peak in the first portion of the blade on the suction side. This causes a very low pressure which gets locally below the vapor tension. Consequently local concentrated cavitation is generated even if the margin NPSHA-to NPSHR is apparently high (NPSHA/NPSHR = 3.0) and the peripheral speed at the blade inlet near the hub is relatively low (57 FT/S).

• The NPSHR curve (Figure 14) measured with the impeller at maximum diameter (13") does not likely change at the duty diameter (11.25"), as the impeller blades have still large overlap at the trim diameter.

• The impeller material (CA15) has low resistance to intense cavitation attack, although it is an adequate choice in most cases for pumps of this size running at 3550 RPM.

• The suction chamber geometry (nozzle plus inlet bay) is not directly responsible for the unequal distribution of the total flowrate, because of its symmetry, as clearly shown by the pump cross section. However, it might enhance any flow distortion present at the suction flange.

Solution Strategy

The following design changes were implemented:

• Eliminate/reduce the flow distortion in the suction piping by installing a flow straightener of advanced design.

• The existing design impeller with appropriate modifications was used as urgent temporary fix. The modifications were derived from cavitation visualization data which were previously obtained with a boiler feed pump old design impeller characterized by incidence angle distribution, similar to the one of the impeller of the recycle pumps i.e. low incidence at the tip but much higher at the hub. The cavitation bubble is shown in Figure 16a, indicating a very intense cavitation zone from the hub to the blade midspan. By using a special throttling cone at the impeller hub it was possible to drastically reduce the cavitation length, especially in the hub region, as shown in figure 16b. This result is due to a net improvement of the flow pattern and a reduction of the incidence angle at the hub. Therefore, a hub throttling cone was recommended as temporary fix for this specific case and applied in the outboard impeller eye along with grinding the impeller blades (Figure 17) on the suction side to reduce the blade angle.

• As ultimate solution, it was strongly recommended to install a new design impeller and also upgrade the impeller material for higher cavitation resistance. Moreover, the installation of some baffles inside the suction nozzle and/or the inlet volute was suggested in order to improve the flow distribution, although, they might have only marginal impact.

CASE NO. 4 - SMALL RAW SEWAGE PUMPS

Case Description

This field case is concerning three raw

sewage pumps operating in a sewage pumping station. Each pump (6 inches suction by 4 inches discharge), consists of a single suction impeller and a single volute. The three pumps are vertically mounted, connected to individual horizontal suction line starting from the wet well and ending with a 90° elbow just at the impeller eye. The pumps have a discharge line to a common header for parallel operation. The rated C.O.S. for each pump were specified for: N = 1770 RPM, Q = 750 GPM, H = 78 FT, NPSHR = 12 FT, NPSHA = 35 FT approx. The power absorbed is 21 HP. The peripheral speed at the impeller eye is 42 FT/S. Impeller material is cast iron.

During normal plant operation only a single pump was running at the above duty. However, when increased flow requirements were occurring, two pumps were operating in parallel. At this condition each pump operated at reduced capacity around 450 GPM. The operation for each pump was intermittent, calling for a service of 8 hours/day.

Field data taken by the operator at the plant first start up in January 1989 indicated that the pump operation was quiet and smooth with a single pump running at the duty capacity (750 GPM), while the pumps became noisy during parallel operations (about 400 to 500 GPM). The noise level increased progressively in the following months, also extending in more wide capacity range. In November 1989, the pump user complained with the manufacturer about the pump noise problem, while the pumps were still under warranty. The user expressed his concern that the pump noise with associated increase in vibration levels may lead to premature failure, including damage of impeller as well as wear on the pump shaft and bearings, with ultimate effect of reduced life.

A subsequent field report in March 1990 from the manufacturer service man indicated that:

• Noise, characterized as "gravel sound" was present, which could be generated by the check valve at the pump discharge.

• The noise level was around 82db to 83db for all the three pumps and various settings of the check valve.

• The vibration level at the bearings (lower/top) was ranging from 0.5 to 1.0 mils peak-to-peak at maximum for all the three pumps and various setting of the check valves.

• No damage (pitting or other aspect) could be seen in any area of the impeller, after approximately one year of operation.

In September 1990, the pump user insisted that the pump was not acceptable, in consideration of the expected life span of the bearings and rotating parts, and asked for a time extension of warranty. Then, the pump manufacturer agreed with the user to commit the solution of the trouble to corporate engineering specialist, starting from a fresh and deep failure analysis.

Failure Analysis

The actual parameter of the pumps in the field which were measured during a new field inspection in November 1990, are shown in Table 1. Only one pump was operated across a wide range of capacity. The isolation valve, which was located far away from the pump, was used for throttling the pump instead of the check valve, in order to clarify the source of the noise, either the pump or the check valve.

The field indications (Table 1) showed that:

• The noise was occurring across the entire range of capacities, even at the rated one. Clearly, the noise was originating from the pump.

• The noise reached a maximum level at capacity of 400 GPM and nearly disappeared at shutoff, thus indicating the presence of a noise peak at part capacities.

• The NPSHA level was about 3 times higher than the NPSHR. Such a ratio is apparently more than adequate for this size of pump according to usual practice.

• The noise was audible only in the pump room and relatively close to the pump (5 to 6 feet).

• The noise sound was as "crackling" as typical for cavitation noise.

The analysis of the pump performance at the design impeller diameter (10 inches) indicated that the best efficiency point, was corresponding to: N = 1770 RPM, Q = 1050 GPM, H = 73 FT, NPSHR = 16 FT (Ns = 2300 and S = 7200). The rated capacity (750 GPM) is at 71% of the b.e.p. capacity, while the second duty capacity of 450 GPM (two pumps in parallel) is at 43% of the b.e.p. capacity. Therefore the pump is running at capacities well below the design one.

The analysis of the impeller geometry pointed out that:

• The shockless capacity was slightly above the b.e.p. capacity.

• The impeller had two long vanes with large throat areas (inlet/ outlet) between the blades, as dictated by solids particle handling requirement for raw sewage pumps. However, this implies a small overlap between vanes, which promotes high NPSHi - to-NPSH ratio (incipient cavitation) especially at Q < Qsl and also high suction recirculation onset capacity (50% to 70% of b.e.p. capacity).

• The impeller was only slightly trimmed at 9.82 inches.

Then the group of the field observations along with the analysis of the pump performance and geometry pointed clearly out that the noise was due to cavitation inside the pump, because the NPSHA level was below the NPSHi corresponding the incipient acoustic cavitation. Moreover, for operations around 450 GPM suction recirculation was also present interacting with the cavitation (vortex cavitation).

The cavitation noise was not originally observed for single pump operation at maximum capacity during the plant start up because the NPSHA was higher. In fact, the water level in the wet well was orginally maintained about 1 FT higher, and also the head loss in the suction piping was lower with new and clean pipe surface.

Case Solution

The various concerns of the pump user about pump distress and life needed to be addressed.

A. *Cavitation Damage and Impeller Life*

At the point of the visual or acoustic cavitation inception the rate of the erosion damage is practically zero.

Therefore the right question is "what is a detrimental cavitation intensity level?". In fact most of the pumps operate in the field with a NPSHA (A = available) which is higher than NPSHR but significantly below the NPSHi i.e. they operate with cavitation.

There is experimental indication that the erosion rate due to cavitation is function of the impeller eye peripheral speed, Ueye, and the length of the cavitation bubble length attached to the blade Lc,b (12,21).

On the other hand the cavitation bubble length Lc,b is dependent from the impeller eye peripheral speed. For similar hydraulic condition (roughly, same margin NPSHA/NPSHR) the cavitation erosion rate varies with about the sixth power of the peripheral velocity at the impeller eye Ueye (19, 21). Therefore for the cavitation erosion and the impeller life the impeller eye peripheral speed is the most important factor.

Moreover, there is some statistical experimental evidence that cavitation erosion is practically zero if Ueye is below 60 FT/S to 70 FT/S (threshold value). Such a threshold value is probably related with the physical mechanism of cavitation damage along with the amount of the impact energy which is due to bubble collapse and is related with the pump energy level.

The impeller eye peripheral velocity for the specific case is Ueye = 42 FT/S, which is very low and below the statistical threshold level for cavitation metal damage. Thus impeller damage is very unlikely, as also already indicated by the first field inspection after more than one year operation and also confirmed in November 1990, after 1½ years operation. There is high probability (above 90%) that the impeller life expectancy and the pump reliability will not be impaired.

Cavitation - Suction Recirculation Interaction

As discussed above when a pump operates at part capacities suction recirculation occurs at the impeller inlet which can cause cavitation ("vortex cavitation" or "recirculation cavitation").

The intensity of such "vortex cavitation" is related with both the onset capacity and the intensity of the suction recirculation, which are very much dependent from the impeller design (8, 20, 27), and mostly the pump energy level (capacity, head, brake horsepower). According to an internal statistical chart derived from field data and covering thousands of pumps no field problem has been caused by suction recirculation with small pumps below the combined limit of 1000 GPM and 100 FT for capacity and head, respectively, at the best efficiency point.

Therefore it is very likely that the reliability of these specific pumps is not impaired, because the energy level of the pump is very low (brakepower below 30 HP). Moreover, the pump head of 70 FT at the best efficiency point is below the statistical head boundary, above which field problems could be expected according to the above statistical internal chart for troubles related with suction recirculation.

Cavitation Pressure Pulsations, Noise, Vibrations

When cavitation bubbles collapse they produce pressure pulsations, which increase the sound pressure level (or "cavitation noise") and also may induce some vibrations. If the bubbles collapse inside the liquid stream far away from the metal surface (blade, shrouds), there is no metal damage but only noise, which does not have any impact on pump reliability.

Moreover, the intensity of the pressure pulsations induced by the cavitation is determined by the pump energy level (power, head) and the suction pressure. For low energy pumps (roughly brake horsepower below 50 HP and head below 100 FT) the intensity of pressure pulsations induced from the cavitation is practically negligible and not harmful for the pump.

The dynamic load which is related with such pressure pulsations is very low and consequently the level of vibrations is only very marginally increased. In fact the key dynamic load for bearings (design load) is determined by:

- The pressure distribution at the impeller discharge, due to the impeller - volute interaction. Such a pressure level is proportional to the pump head and so is higher than the suction pressure.

- The surface at the impeller outer periphery. This area is much larger than the impeller eye area, which is interested by the suction pressure pulsations.

Therefore the extra dynamic load due to cavitation pressure pulsations is only a few percent of the bearing design load. It is more or less comparable to the accuracy degree of the bearing life calculations.

The vibration level at the pump bearing housings, which is ranging from 0.5 to 1.0 MILS P-P at maximum, is well below the acceptable limits given by various charts, including Hydraulic Institute Standards. Such a low vibration level indicate that the cavitation occurring in the impeller has zero or marginal impact on the bearing loading. Therefore the probability to reach the specified bearing life is very high (above 90%).

The overall pump noise level of 82-83db is low and fully acceptable according to standards. Moreover, the pump station is completed isolated and far from habited areas, while no noise is audible from outside the station and even in the pump room at a distance of 5 to 6 feet.

Final Approach

As a general conclusion, some cavitation is occurring in the pump, which produces some acoustic noise but at acceptable level. On the other hand such cavitation intensity is not detrimental for both the impeller life (damage) and the bearing life (additional dynamic loading and vibration increase). Therefore any action aimed at eliminating the cavitation/ recirculation would be unnecessary and only costly and possibly detrimental for the efficiency.

The above considerations were basically accepted by the pump user. The case was settled without any modifications to the pumps.

The latest field information after 2½ years of operation confirmed both the absence of impeller damage and the full reliability of these small raw sewage pumps.

CASE NO. 5 - CHEMICAL PLANT PUMP SERVICE

Case Description

This case history is regarding four units of the same centrifugal pumps operating in the same chemical plant for two different service (one pump operating and one pump on stand-by). The pump is a single stage end suction type (20 inches suction and 16 inches discharge) with a discharge volute casing. The suction piping of each installation start from a tank and has two elbows close to the pump inlet. The two service conditions are:

Duty	A	B
N (RPM)	980	980
Q (GPM)	10570	8800
H (FT)	98.5	105
BHP (HP)	326	303
NPSHR (FT)	16.6	16.1
NPSHA (FT)	22.8 (20.5)	22.8 (20.5)
Process fluid	water + Na_2Co_3	water + CaOH
Fluid temp (°F)	192	192
Specific gravity	0.97	0.97

Basically each service calls for a single operation at nearly fixed capacity.

According to the plant operator, all the four pumps exhibited typical cavitation noise since start-up (December 1982). Then the NPSHA was increased from the original level of 20.5 FT up to 22.8 FT by rising the fluid level in each tank. However, the noise was noticed even during a cold water field test with an approximate NPSHA level of 29 FT.

Then after a few months of operation (February 1983) the pumps were opened for inspecting the impellers. The inspection revealed that:

- Absolutely no sign of damage was present on both the two impellers of the pumps for duty A (dilute water solution of sodium carbonate at the saturation point), after 1100 hours of continuous service.

• Some damage was already evident on both the two impellers of the pumps operating for duty B (dilute water solution of calcium hydroxide at the saturation point) after 1000 hours (pump B-1) and 300 hours (pump B-2) of continuous service. The damage which was still at an early stage, clearly showed the typical aspect of cavitation pitting. The erosion was located on the visible side (suction side) of each blade, starting from the leading edge with length increasing from the tip to the root of the blades.

The highest concern of the plant operator was about the cavitation erosion of the pumps on duty B.

Failure Analysis

All the four units used the same pump size with the impeller trimmed at 20.1 inches (85% of the design diameter). The peripheral velocity at the impeller eye was the same (Ueye = 80 FT/S). Also, the impeller material was the same (cast iron).

Basically, the only significative difference between duty A and duty B was the operating capacity. This was higher for the pumps on duty A, which showed cavitation noise but not damage. The NPSHA-to-NPSHR ratio is 1.37 for duty A (cavitation noise but no damage) and 1.40 for duty B (cavitation noise and damage) but can be considered pratically the same.

The audible cavitation noise in the field was not continuous like persistent crackling indicating intense cavitation, but rather the noise was intermittent like random bubble collapse more characteristic of small degree of cavitation, close to the inception point. Moreover, the location of the damage on the blade suction (visible) side and visual appearance as pitting clearly indicated that the pumps were operating in the regime of "blade attached cavitation" corresponding to high positive incidence angle (Figure 1).

The analysis of the pump performance at the full (design) diameter (23.6 inches) showed for the best efficiency point: N = 980 RPM, Q = 15200 GPM, H = 118 FT, BHP = 530 HP, NPSHR = 23 FT, Ns = 3380, S = 11,500. Then the duty A corresponds to 70% of the b.e.p. capacity (Qbep-des, des = design diameter of the impeller), which indicates part flow operation. The duty B corresponds to 58% of Qbep-des, thus a further reduced operating capacity.

The analysis of the impeller geometry indicated that:

• The shockless capacity, Qsl, is at 17350 GPM i.e. at 114% Qbep-des, which is a design choice quite usual for high efficiency and reasonable suction specific speed. Then, with comparison to this very important capacity (zero incidence angle, minimum NPSHi) the duty A is at 58% and the duty B at 51%. At these conditions the incidence angle is very high, producing incipient cavitation (visual and acoustic) for NPSHA-to-NPSHR ratio much higher than the 1.4 value of this specific case (16).

• The theoretical suction recirculation capacity is around 50% of Qbep-des. Then the operating capacity for duty B is close to the one at which both the curve of incipient cavitation, NPSHi, and the curve of damaging cavitation, NPSHd, reach a peack (Figure 1).

The comparison of the duty A and duty B with Qbep-des and Qsl is shown in Figure 18a. Moreover, the comparison is extended to the best efficiency capacity at the actual field impeller diameter, Qbep-dd (dd = duty diameter). When the impeller is trimmed, the shockless capacity which is exclusively determined by the impeller geometry at the inlet, does not reduce while the b.e.p. capacity decreases with linear proportion. Then duty A and duty B correspond to 82% and 68% of Qbep-dd. Such values, even if indicates part flow operations, are apparently not critical with reference also to the NPSHA-to-NPSHR ratio of 1.40 (Figure 18b), which is commonly considered more than adequate for this relatively low speed pump (980 RPM, Ueye = 80 FT/S). Also, these values of Qduty/Qbep-dd may erroneously suggest that the pump operation is for both the duties well above the suction rcirculation point. In fact, this is customarily defined as fraction of the b.e.p. capacity and simply indicated Qrs/Qbep, but in the current practice (pump selection, operations, troubleshooting) it is tendencially straightway referred to the b.e.p. capacity at the impeller diameter selected for the duty.

The NPSH test curve at the capacity of duty A is shown in Figure 17c. There an indication that the cavitation inception point is at NPSH level about 2 times the NPSHR, i.e. much above the NPSHA available in the field (NPSHA/NPSHR = 1.4). Moreover, the virbrations were measured on the external side of the volute discharge casing. The RMS (Root Mean Square) level of the casing vibrations is shown versus decreasing NPSH for the rated capacity B in Figure 18c. The vibration level starts to increase at NPSH = 1.4 x NPSHR which gives a very rough indication that the cavitation intensity has grown enough to initiate some damage in the impeller.

Then it was concluded that the NPSHi curve was above the field NPSHA for both duty A and duty B, thus generating noise for all the pumps. On the other hand, the cavitation damage curve NPSHd, which starts to rise at Q<Qsl (Figure 1), is likely below the field NPSHA for the duty A (no impeller damage), while NPSHd becomes higher than field NPSHA for the duty B (with impeller damage).

Case Solution

A new design impeller (reduced capacity) was selected for solving the cavitation damage in the two pumps for duty B. The operating conditions of the two pumps for duty A were considered still acceptable, due to the absence of impeller damage and the low level of the cavitation noise.

The design approach for the new impeller included the following criteria:

- Lower the shockless capacity (Qsl = 12100 GPM, i.e. 30% smaller than for the large capacity impeller).

- Reduce the peripheral speed at the impeller eye (Ueye = 70 FT/S, which increased the impeller life by a factor 2.0).

- Increase possibly the NPSHA-to-NPSHR margin by reducing the NPSHR and so increasing the design suction specific speed, as much as allowed by other considerations (peak efficiency and suction recirculation).

- Obtain a satisfactory matching with the existing volute casing in terms of peak efficiency and flow stability.

- Meet the operating conditions of duty A for future replacement of the impeller if needed.

- Upgrade the impeller material (CA6NM).

The small capacity impeller had a design diameter of 22.8 inches for maximizing efficiency with the existing volute casing. Then the impeller was trimmed for the duty B at 89% of the design diameter. As shown in Figure 19a the rated capacity has moved to 73% of the shockless capacity, and to 88% of the b.e.p. capacity at design diameter (and to 93% of the b.e.p. capacity at the duty diameter). Moreover the suction recirculation onset capacity (and so the capacity giving the peak of NPSHi and NPSHd) has been lowered down to 6000 GPM, well below the operating capacity for duty B. The NPSHR at the duty B has been lowered to 14.2 FT giving a higher NPSHA-to-NPSHR ratio of 1.62 (Figure 18b). This value combined with lower incidence angle and lower Ueye and higher resistance material has drastically reduced and nearly eliminated the risk of cavitation damage.

The new design impeller was installed in the summer of 1983 and is still working satisfactorily.

It is worth to note that the efficiency at the operating point has been drastically improved by about 7 points with the reduced design capacity impeller.

A further remarkable observation is that the suction specific speed has been increased to a value of 11,800 (US units) at Qbep-des. Such a value should be considered not acceptable according to some literature recommendations (29) and several current pump specifications, but these pumps have operated for more than 8 years without any further problems.

The suction specific speed changes with capacity, as shown in Figure 19c. Although the most common definition is referred to the b.e.p. capacity at the design diameter, a more safe and meaningful definition should refer to the shockless capacity. This capacity, which determines the optimum geometry of the impeller inlet, strongly characterises the impeller behavior, especially at part flow with reference to cavitation (damage, noise) and suction recirculation (damage, vibrations) effects. It is also evident that specifying any maximum limit for acceptable value of the suction specific speed without any reference to the ratio Qsl/Qbep-des is not truly effective for avoiding unreliable pump operation at part flows.

CASE NO. 6 - INDUCER FOR CHEMICAL PLANT SERVICE

Case Description

The following field trouble was related with the inducer of two pump units operating in a chemical process plant. The pump was a single stage suction type (12 inches suction and 10 inches discharge) with a double volute discharge casing. The pump was equipped with an axial flow inducer to provide very low NPSHR. The pump was selected for the following rated conditions: N = 2980 RPM, Q = 3720 GPM, H = 657 FT, BHP = 1015 HP, NPSHR = 14.8 FT (inducer), NPSHA 20 FT, fluid temperature = 226°F, specific gravity = 1.30. The process fluid was a lye rich solution of potassium carbonate. Moreover, the minimum flow specified was 1320 GPM. The material for both the inducer and the impeller was a Cr. Ni. 26-2 S.S. similar to ASTM A890 Grade 5A.

The two pumps were started in the early 1978 and very quickly exhibited high noise. An internal inspection was made after 1000 hours about of continuous operation and showed that:

• The inducer blades were heavily damaged. The location of the erosion was on the suction (visible) side of each blade at 50% of the chord and also on the pressure (hidden) side of the next blade at 25% of the chord. The damage length was a few inches on both the blade suction side (more) and the blade pressure side (less). The damage areas were located at the outer periphery of the blade (suction and pressure sides) extending radially downward from the tip for about 20% of the blade height. In the axial direction (rotation axis) the zone of the deepest erosion on the blade suction side was nearly in front of the corresponding maximum depth of erosion on the pressure side of the next blade. The visual aspect of damaged area on each surface of the blades was like a sponge cloth with innumerous and irregular and deep minicraters in the central region, while presented a more a less uniform pitting in the peripheral zone.

• The outer sleeve was also damaged as pitting, but much less severely, in areas facing in the axial direction the damaged zones of the inducer blade.

• The main impeller was totally free of any damage at the inlet and also the outlet of the blades.

It was immediately clear that each inducer would have very short residual life and had to be replaced.

Failure Analysis

The audible noise was similar to cavitation crackling sound, but intermittent with random loud bubble collapse. The visual appearance of the damaged areas clearly indicated the presence of cavitation. The radial and axial locations of the damage areas on each side of the blade channel suggested that the cavitation was occurring blade-to-blade in the tip stream tube.

The pump performance at the full (design) impeller diameter had best efficiency point at: N = 2980 RPM, Q = 4900 GPM, H = 569 FT, BHP = 845 HP, Nsimp = 1790, NPSHRimp = 57 FT, Simp = 10,000, Hind - 32 FT, Nsind = 15500, NPSHRind = 25.0 FT, Sind = 18660. Then the operating capacity is at 76% of the b.e.p. capacity, indicating that both the impeller and the inducer were running at part flow.

The analysis of the inducer geometry indicated that:

• The shockless capacity was well above the design one of 4900 GPM. This is a pecularity of constant pitch inducers, which use a flat (not combered) airfoil profile for the blade geometry.

• At the duty capacity the incidence angle at the tip was +11°. This is a very high value which can easily promote cavitation on the suction side of the blade.

• The peripheral velocity at the inducer tip was 134 FT/S, which is high and susceptible of developing cavitation even at very high NPSHA.

• The vane loading of the blade at the tip section was too high at the duty capacity of 3720 GPM, indicating that flow separation was more than probable with consequent reverse flow inside the vane channel ("suction crecirculation").

The analysis of the main impeller geometry indicated that at the duty point the incidence angle was still below the stalling incidence. On the other hand an experimental investigation about suction recirculation was in course in the same time, which included this specific impeller (7, 8). The experimental data had clearly shown that the suction recirculation was occurring below the duty capacity of 3720 GPM.

Therefore it was concluded (Summer 1978) from the above analysis that the inducer alone was subjected to cavitation on the suction side of the vane along with flow separation. Then the tip fluid stream filled with vapor bubbles was turned by the internal reverse flow (suction recirculation) toward the pressure side of the blade where the residual vapor bubbles were collapsing. The same mechanism was visualized in centrifugal flow pumps in the same period by other researchers (22), as also shown in Figure 4.

Then the peculiar damage pattern appeared as a combined action of both the blade sheet cavitation (damage on the inducer blade suction side) and also the vortex cavitation due to the suction recirculation (damage on the inducer blade pressure side). The inducer was working just below the capacity giving the peak of NPSHi (Figure 1).
It was thought that the damage rate was quite intense because:

• The peripheral velocity Ueye was remarkably high.

• The high positive incidence angle was producing a large cavitation cloud, with many collapsing bubbles.

• The NPSHA-to-NPSHR ratio around 1.35 was very marginal.

• The specific gravity around 1.3 was an aggravating factor. In fact, it should be expected that the implosion of the vapor bubbles would produce high impinging pressure in high density liquid, as lately published (21, 25).

Solution Strategy

It was clear from the above analysis that a new design inducer with lower shockless capacity would alleviate the damage problem.

However as urgent and temporary fix it was considered to make new casting of the same design of the inducer in the field and overlay the blades with Stellite Grade 6 by welding. But the stellite coating did not resist too long because of its intrinsic brittleness and mainly an inadequate bond (welding) with the base metal. Moreover, it happened that the hot welding process generated a distortion of the blade geometry thus changing the blade angle at inlet which is extremely critical for the inducer cavitation characteristics of NPSHi and NPSHd. In fact these curves (minimum and peak value from Figure 1) are drastically changed by a deviation of the inlet blade angle even less than one degree.

The design strategy for the new inducer was focused on the following criteria:

• Reduce the incidence angle at the tip for the duty capacity from 11° down to a few degree. Then the shockless capacity was chosen around 4500 GPM (20% higher than the duty capacity).

• Increase the NPSHA-to-NPSHR margin by lowering the NPSHR and thus increasing the suction specific speed of the inducer. At the inducer shockless capacity the S value was 24,400 (US units) with NPSHRdes = 16.5 FT. Then the NPSHR at the duty capacity was 12.5 FT, thus giving a margin NPSHA-to-NPSHR = 1.60 approximately.

• Keep the same NPSHR for the impeller at the duty capacity. Then the head generated from the inducer needed to be slightly increased. As a consequence, the tip diameter of the inducer was unchaged, although a reduction would have been beneficial for lowering the cavitation damage rate. Also the exit blade angle needed to be increased.

• Change the blade shape by using a combered profile at the tip to meet the above design needs for low incidence angle and lower shockless capacity and higher inducer head. Then a variable pitch inducer was selected.

• Increase the blade thickness from tip to the hub.

• Impose very narrow tolerances on geometrical deviations. Then the inducer was made by using a 5-axis NC machine.

• Maintain the inducer material.

The new design inducer was installed in the early 1979 and operated for more than two years. The noise was eliminated while the cavitation erosion rate was drastically reduced, but not completely eliminated.

Thereafter, in 1982, the inducer design was finally optimized to possibly eliminate the cavitation erosion. The final inducer was still of variable pitch type with reduced tip diameter giving a peripheral velocity at the tip of 124 FT/S (previously 134 FT/S). The shockless capacity for the inducer was maintained at 4500 GPM while the suction specific speed was reduced to 21,500. The NPSHR duty capacity was about 13.5 FT giving a NPSHA-to-NPSHR ratio around 1.50, which appeared to be adequate. Also the final inducer was made by NC-machining in the same material as the previous one. This inducer was installed in the summer of 1982 and is still operating after more than 9 years.

Inducer Reliability and Suction Specific Speed

The inducer is a special device purposely designed for very low cavitation requirements (NPSHR) and so high S value from 15,000 to 25,000 (or higher) for industrial application (29, 30).

The high S value is obtained by combining special blade geometry and very low head (and thus low energy level) and axial flow configuration (specific speed Ns from 15,000 to 20,000 U.S.). Typically, the brake horsepower of the inducer is ranging from only 5% (at b.e.p. capacity) up to 10% (at 50% b.e.p. capacity) of the full pump horsepower.

Some published analysis and conclusions based on a statistical survey of centrifugal flow pumps (28) about a limit critical S value of 11,000 do not absolutely apply to inducers, which have much higher specific speed (axial flow) and much lower brake horsepower than the values characteristic of the pump population used for the survey.

Rather, the inducer cavitation erosion limits are strongly related to both the inducer design and the rotational speed as clearly shown in Figure 20 (31), which was obtained for water. Field experience has shown that S limits up to 20,000 U.S. for water and even 25,000 U.S. for hydrocarbon fluids can be reached with negligible or zero cavitation erosion.

Extensive research both theoretical and experimental on inducers since 1930's (first inducer patent historically) to 1990's (cavitation visualization and acoustic measurements plus internal flow measurements by Laser Doppler Anemometry (32) (Figure 21), has generated good insides about inducer design optimization. Moreover a very large industrial population of inducers (around a few thousands) have gained a depth of knowledge and wide experience in inducer technology.

Only a few inducer failures (quick incidence damage) were reported to the author's knowledge in the last 15 years which were caused by misapplication of inducer under critical duties i.e. high inducer tip speed (around 100 FT/S) plus large size inducers (10" and 14") plus high specific gravity (S.G. = 1.0 to 1.3) plus low flows duties (60% to 80% of b.e.p. capacity) plus inadequate NPSHA/ NPSHR margins (from 1.15 to 1.30) plus inducers material with low resistance to cavitation attack. All these field cases wee successfully fixed by using a new design inducer optimized for part capacity (inducer B) and a more resistant material (CA6NM, Duplex S.S.).

The key factors determining the inducer reliability as related to cavitation and/or suction recirculation are:

- Impeller eye peripheral speed (Ueye).
- Duty capacity as percent of the "shockless capacity" (which can be different from the b.e.p. capacity).
- Impeller design (Deye, blade geometry).
- NPSHA (and so NPSHA-to-NPSHR margins).
- Fluid density.
- Fluid thermodynamic properties.
- Fluid temperature.
- Impeller material.

It should be remarked that the above factors do not show any direct influence of the suction specific speed on the cavitation damage rate and so on the inducer life.

CONCLUSIONS

Various pump field problems related with cavitation and/or suction recirculation characterized by different degree of damage have been widely discussed. The damage pattern has been clearly interrelated with the cavitation mode and the flow mechanism and the key geometrical parameter of the impeller and the operating conditions plus the time spectrum of the plant load.

The field problems were fully cleared applying new design criteria for the impeller along with other pertinent modifications and recommendations as suggested by a thorough failure analysis, including system aspects and operator input.

The selection of the shockless capacity with respect to both the pump operating range and also the expected plant load distribution versus operating hours is a key design choice. In this regard a close cooperation between the pump designer and the pump user is fundamental in order to harmonize the hydraulic design criteria of the impeller or inducer with the expected operating mode of the plant, as basis for reaching the impeller/inducer life target.

Inducers can operate with suction specific speed very high, if they are properly selected and their design is optimized for the actual duties with adequate NPSHA.

NOMENCLATURE

BHP	brake horsepower
D	diameter
ED	erosion depth
ER	erosion rate
IL	impeller life
Lc	cavity length
H	total dynamic head
NPSHA	net positive suction head available
NPSHR	net positive suction head required (3% head drop)
Ns	specific speed (US unit)
P	power plant load
Q	capacity
RL	life factor
S	suction specific speed (US units)
Tb	blade thickness
U	peripheral speed
W	probability
τ	operating time (given load)-to-total service time ratio

Subscripts

b	bubble
bep	best efficiency point
cal	calculated
d	damage
dd	duty diameter
ed	erosion maximum depth
el	erosion length
eye	impeller eye
i	incipient
imp	impeller
ind	inducer
max	maximum

ref reference (at best efficiency point of impeller A - Case No.1)
req required

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TABLE 1 - RAW SEWAGE PUMP - FIELD OBSERVATIONS

TEST #	PUMP	Ps (PSIG)	Pd (PSIG)	Pv (PSIG)	H (FT)	Q (GPM)	NPSHR (FT)	NPSHA (FT)	NPSHA / NPSHR	ΔPv (PSI)	REMARKS
1	2	-1.0	36	36 (33)*	86	650	~ 12	36.9	3.07	~ 0 (3)*	CAVITATION NOISE FROM PUMP - SLIGHT INCREASE WITH CLOSING CHECK VALVE
2	1	-0.5	37	37 (34)*	87	580	~ 12	35.6	2.97	~ 0 (3)*	AS FOR POINT 1 (ABOVE)
3A	1	-0.15	35	~ 35	81	750	~ 12	35.2	2.93	~ 0	CAVITATION NOISE (PUMP) CHECK VALVE OPEN.
3B	1	-0.3	38	~ 38	89	560	~ 12	34.7	2.89	~ 0	CAVITATION NOISE INCREASES.
3C	1	0.2	44	~ 44	102	(150)	~ 12	33.5	2.79	~ 0	CAPACITY NEAR SHUTOFF (ISOLATED VALVE CLOSED) NO NOISE.
3C	1	-0.8	40	~ 40	94	400	~ 12	36.1	3.00	~ 0	MAX LEVEL OF CAVITATION NOISE.

Ps = SUCTION PRESSURE
Pc = DISCHARGE PRESSURE
Pv = PRESSURE AFTER CHECK VALVE
ΔPv = PRESSURE DROP ACROSS THE CHECK VALVE
* = CHECK VALVE PARTIALLY CLOSED

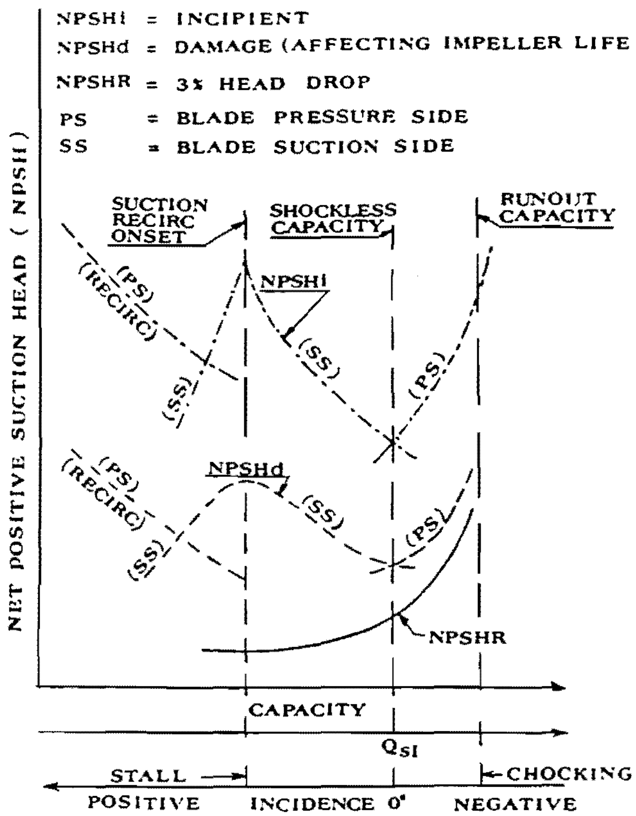

Figure 1. NPSH = Peculiar Curves Defining Various Cavitation Modes.

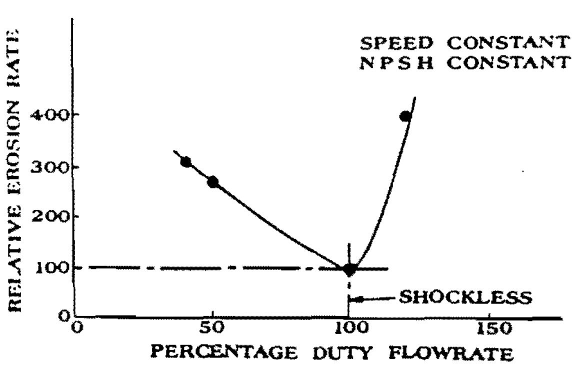

Figure 2. Variation of Erosion Rate with Capacity (11).

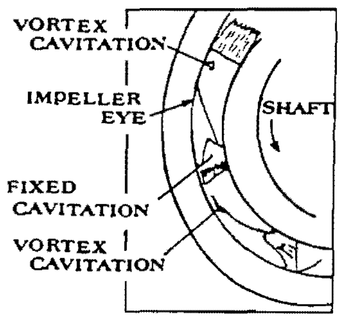

Figure 3. Alternating Sheet Cavitation with Vortex Cavitation(22).

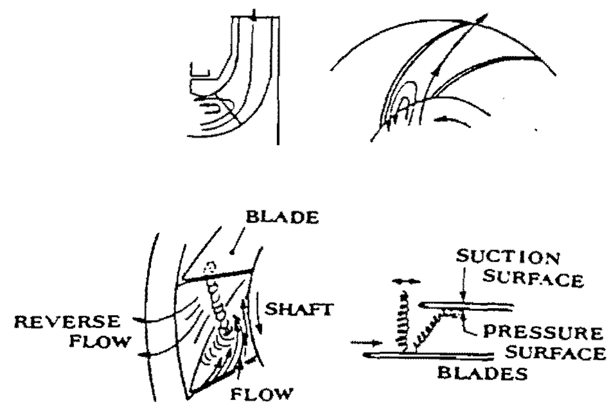

Figure 4. Suction Recirculation (a-b) as Source of the Vortex Cavitation (c-d)(22).

UNIT	Lc,ed	Lc,el	EDmax	A	B	C	D	[ ]	Δ%
1	2.05	2.36	1.18	0.16	0.39	0.59	0.01	in	10
4	1.97	2.28	0.98	0.08	0.20	0.39	0.01		

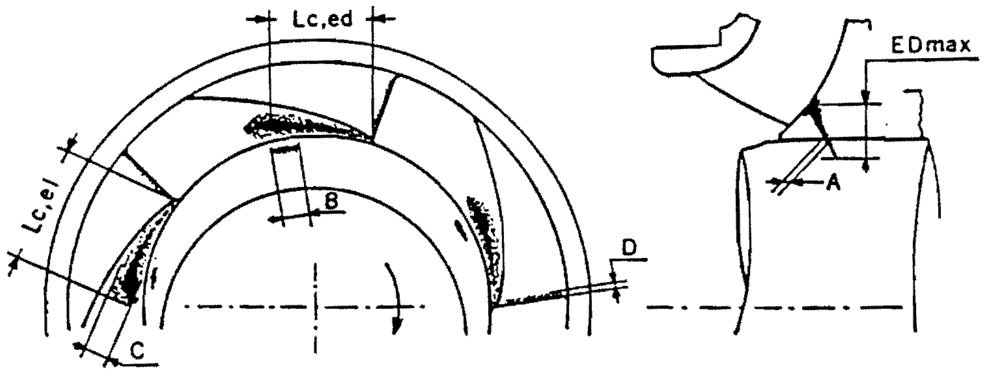

Figure 5. Field Cavitation Erosion Pattern (Impeller A).

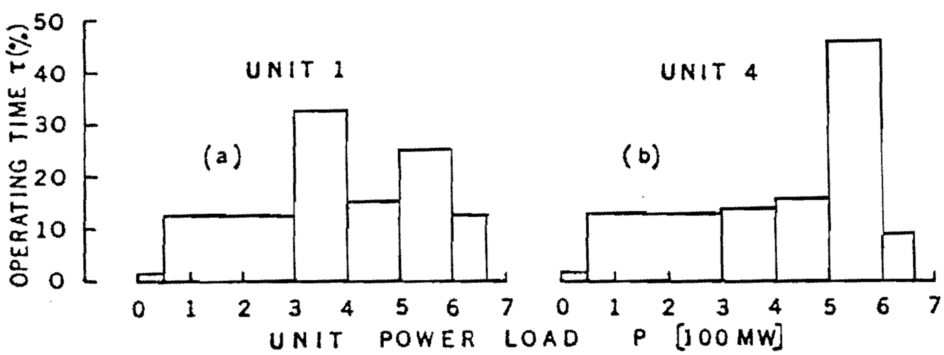

Figure 6. Operating Hours (% Total) vs. Power Loads.

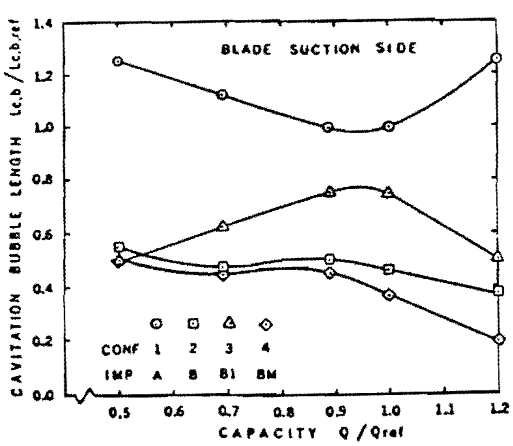

Figure 7. Cavitation Bubble Length at Plant NPSHA.

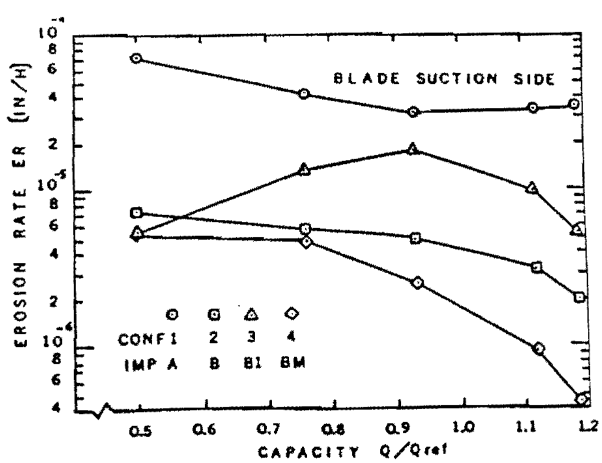

Figure 8. Erosion Rate on Blade Suction Side vs. Capacity.

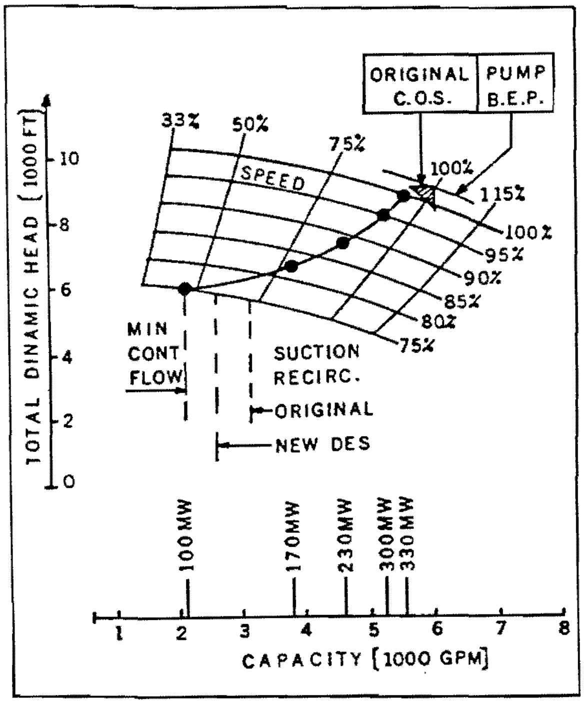

Figure 9. Pump Performance Map and Operating Line.

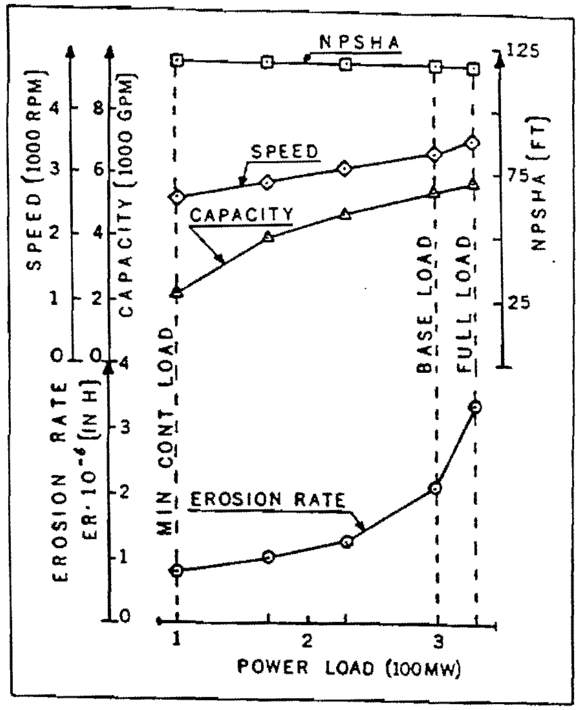

Figure 10. Operating Parameters and Cavitation Erosion Rate.

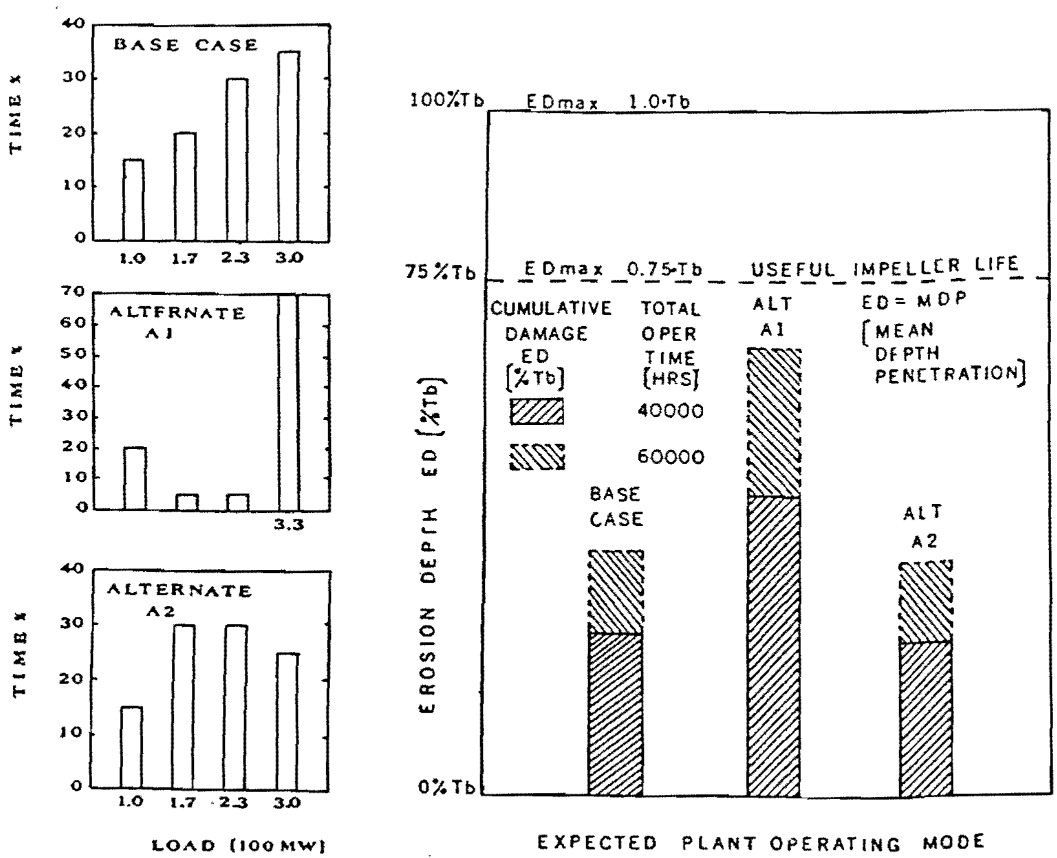

Figure 11. Cavitation Cumulative Damage with Various Plant Operating Modes.

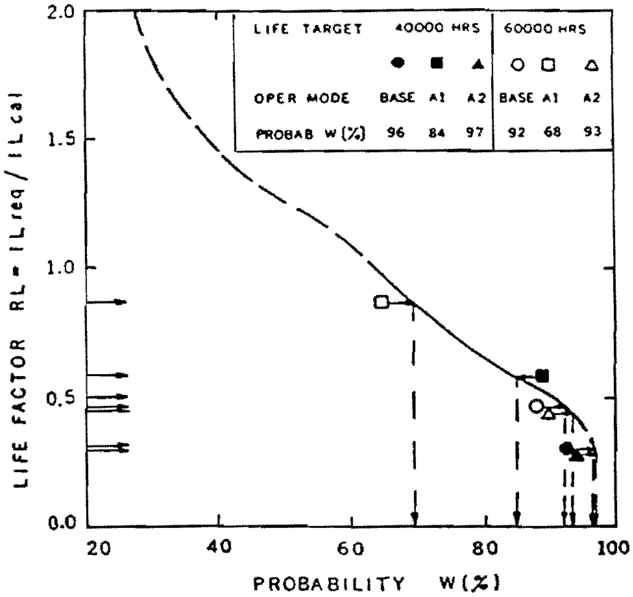

Figure 12. Impeller Life Prediction.

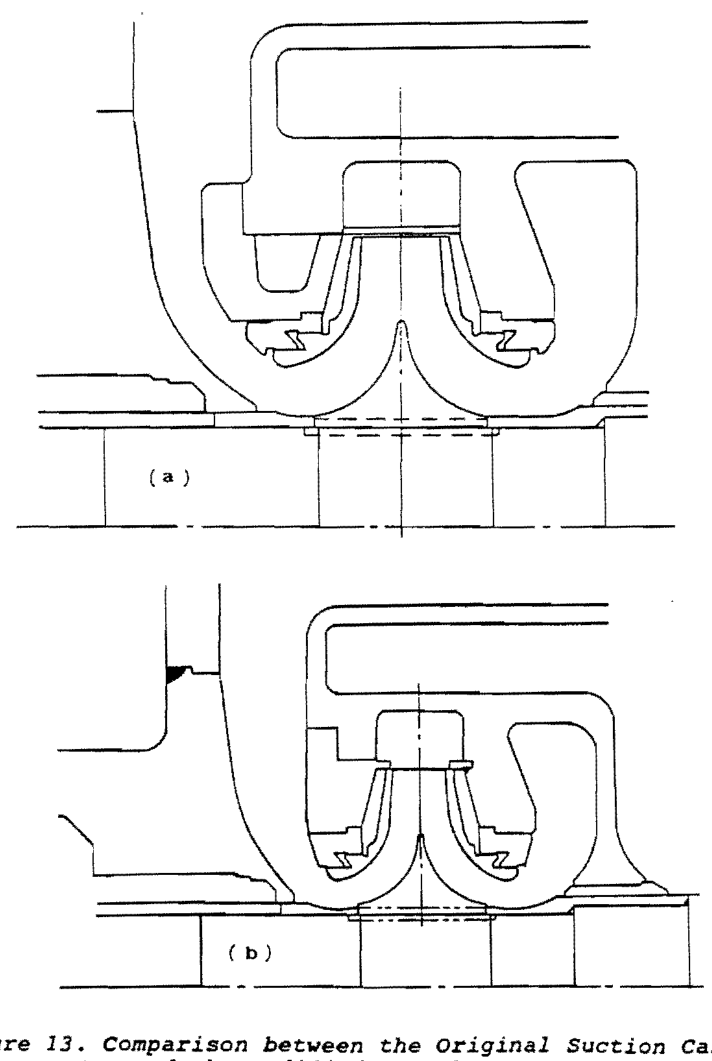

Figure 13. Comparison between the Original Suction Case Geometry (a) and the Modified One (b).

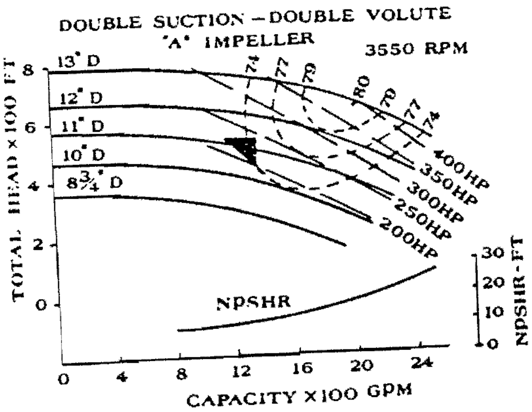

Figure 14. Scrubber Recycle Pump: Performance and Operating Point.

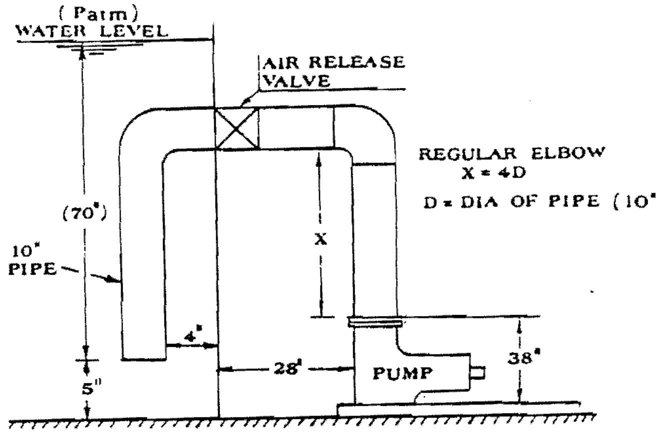

Figure 15. Scrubber Recycle Pump: Suction Piping.

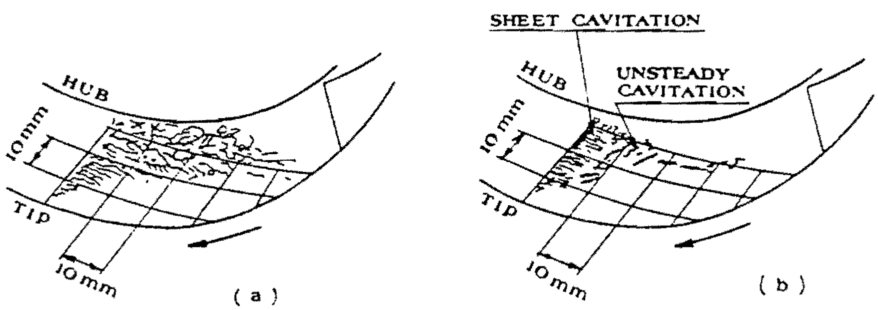

Figure 16. Cavitation Bubble Visualization with High Incidence Angle at the Hub Case a: free flow at the impeller inlet; (b): throttling cone at the impeller hub.

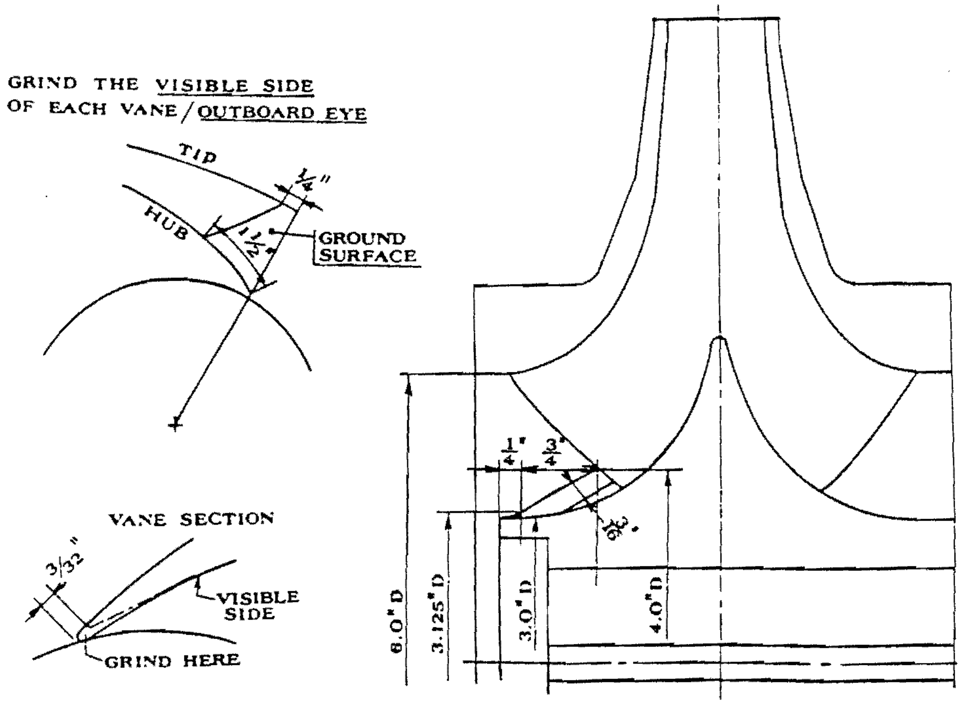

Figure 17. Impeller Modifications as Temporary Fix.

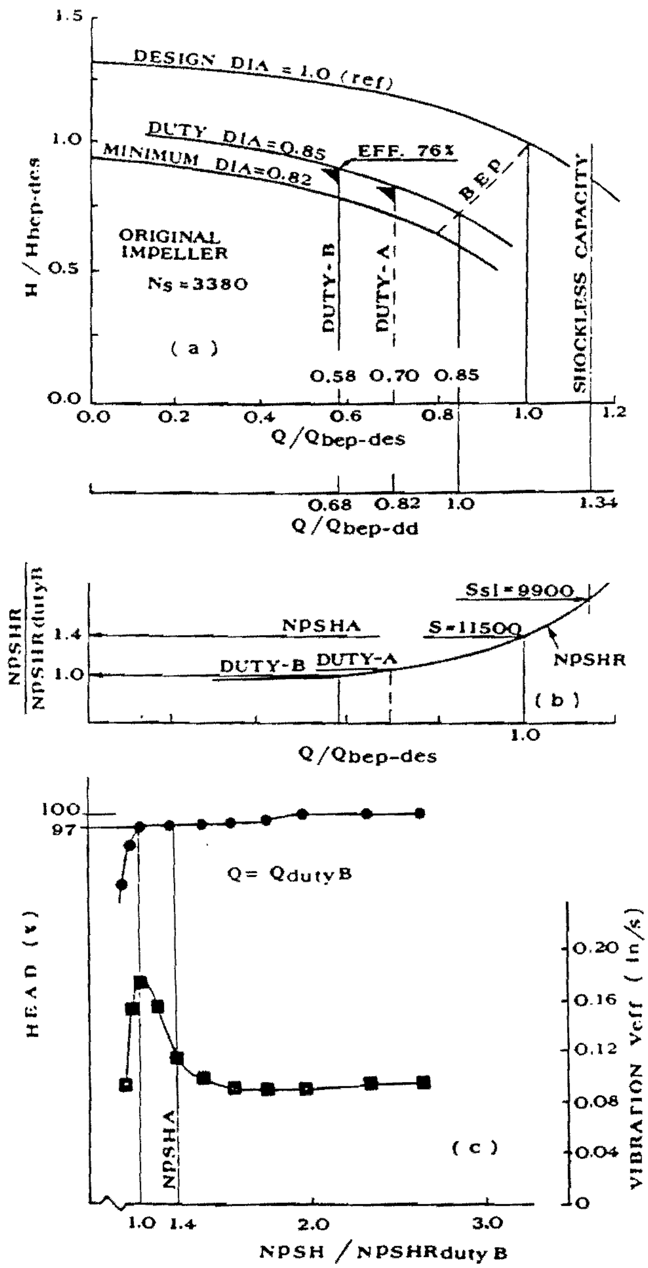

Figure 18. Unreliable Off-design Duty with Oversized Impeller: (a) Head and Efficiency, (b) NPSHA/NPSHR, (c) Cavitation Intensity.

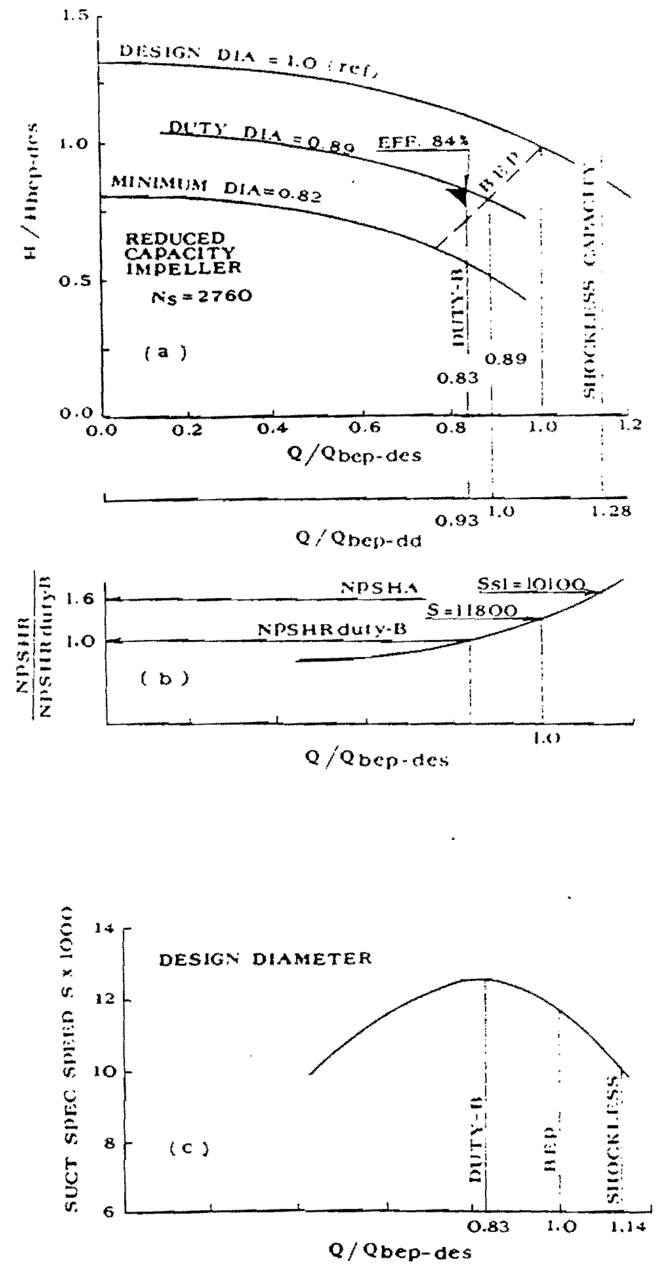

Figure 19. Performance and Reliability Improvement for Off-design Duty with Reduced Capacity Impeller: (a) Head and Efficiency, (b) NPSHA/NPSHR, (c) S versus capacity.

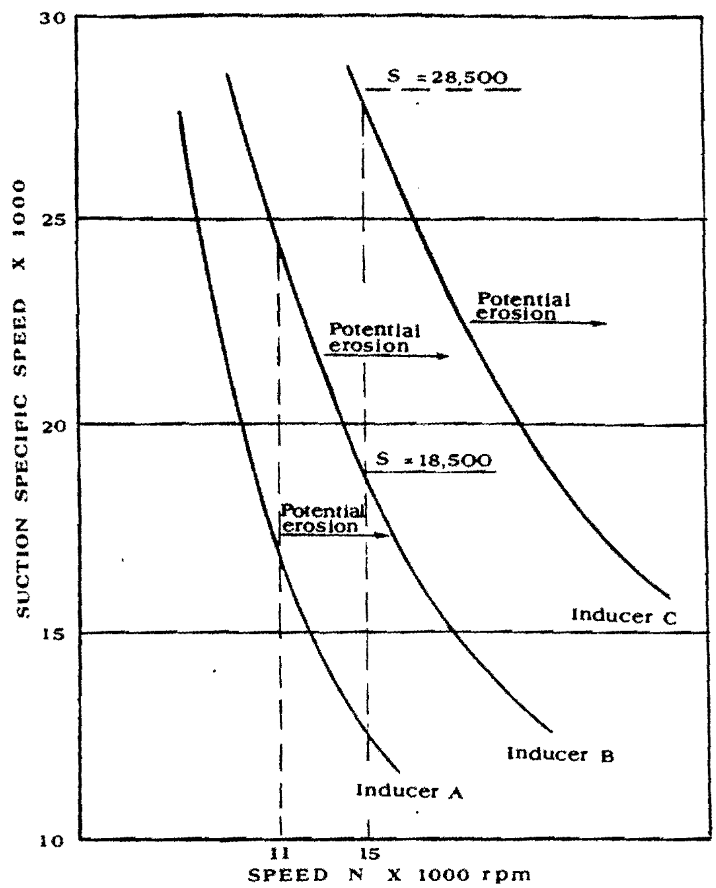

Figure 20. Inducer Cavitation Erosion Limits (31).

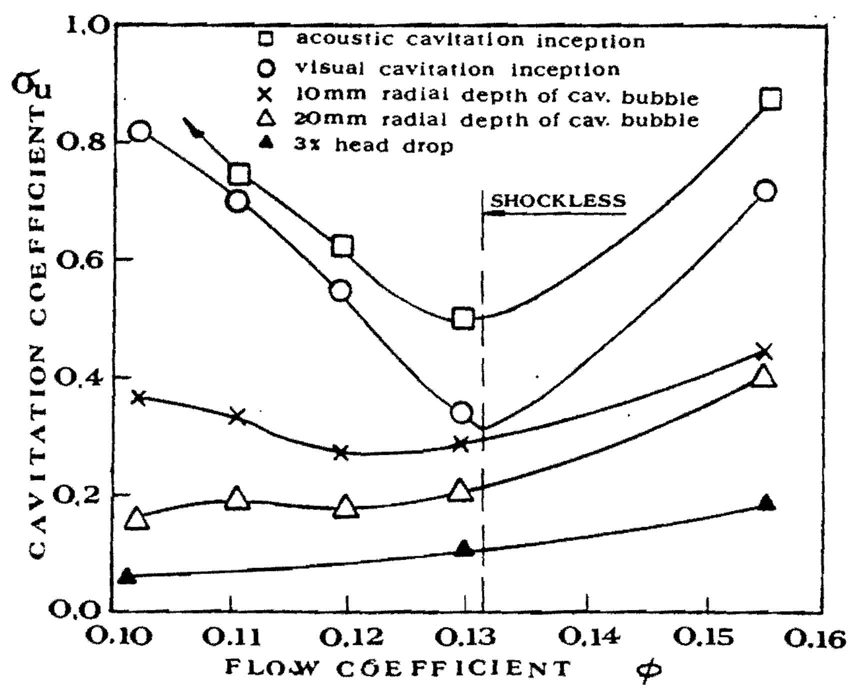

Figure 21. Inducer Cavitation Characteristics from Inception to Head Drop by 3%.