CAVITATION AND RECIRCULATION TROUBLESHOOTING METHODOLOGY

by

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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 especially low load duties.

The reliability and rangeability target is a new task for the pump designer/researcher and is made more challenging by cavitation and/or suction recirculation effects, primarily, the pump damage. Knowledge about design critical parameters and their optimization, and field problems diagnosis and troubleshooting has advanced considerably in recent years.

The objective of the pump manufacturer is to develop design solutions, and troubleshooting approaches that improve the impeller life as related to cavitation erosion, and to enlarge the reliable operating range by minimizing the effects of the suction recirculation. A short description of several failure modes, including damage patterns and other symptoms, which are related to cavitation and/or suction recirculation is presented. The troubleshooting methodology is described in detail, focusing on the various steps of failure analysis, diagnosis, and solution strategy, including quick fixes and ultimate solutions along with new material. The troubleshooting method essentially is focused on failures characterized by metal damage.

INTRODUCTION

From the late 1950s until the early 1970s, the size of various types of plants was greatly increased and, consequently, both energy level (head/stage) and capacity of pumping equipment increased drastically. In the area of power plants, the head/stage was increased by a factor of four and the impeller peripheral speed by a factor of two. In the area of process and chemical plants, the pump maximum capacity was more than doubled, and the head was increased to meet the higher piping losses associated with large, complex plants and high capacity plants.

Initially, a plant engineering contractor would put high emphasis on low capital cost. Plant availability, which would require redundancy and, therefore, several units in parallel, received less consideration. Pump users also gave keen attention to low energy operating cost at full and maximum pump loads.

Therefore, the pump designers have been urged to develop smaller and, consequently, faster pumps, and also maximize efficiency at high flows. In addition, lower $NPSH_A$ 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 inducers were widely applied, also. Moreover, pump suppliers needed to avoid sharp increase in the $NPSH_R$ curve (as based on three percent head drop) at high capacity while still maintaining high efficiency. As a result, the pump behavior was penalized at partial flows, in combination with more severe requirements in terms of cavitation at duties above the best efficiency point (BEP).

In the early 1970s and early 1980s, due to a contraction in the demand, an increasing number of process and chemical 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 loads, along with their pumping equipment (namely feedwater and booster).

As a result, more and more high energy and/or large size pumps were operating in a broad range of capacity, including long duties at flows substantially below the BEP capacity, and, especially, much lower than the impeller inlet optimum capacity (called "shockless capacity"). Frequent failures then 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] and local cavitation with associated metal damage, gave some clear insights on the aforementioned failures.

Other pump failures have been observed with increasing frequency, which presented a classical cavitation damage aspect, in the form of surface pitting, even with $NPSH_A$ level well above the $NPSH_R$. A basic study of cavitation inception and growth was published in 1941 [7]. However, extensive experimental research on cavitation in pumps has been carried out only in the last two decades, primarily concentrating on cavitation visualization (transparent test models and stroboscopic light) [1, 8, 9, 10, 11] and acoustic detection of high frequency cavitation noise spectra [12]. It then became fully evident that cavitation inception occurs at $NPSH_A$ level ($NPSH_i$) 5.0 to 20 times higher than the conventional value of $NPSH_R$ corresponding to three percent head drop, while cavitation damage occurs at for NPSH levels below the inception,

point but still higher than $NPSH_R$, depending on factors such as peripheral velocity at the impeller eye, and pump operating capacity as fraction of the shockless capacity [8, 9].

While new criteria for establishing adequate $NPSH_A$ level were being developed, [8, 9, 10, 13], pump designers were called on more and more solve field troubles related with cavitation and/or suction recirculation. A detailed history of several field cases were discussed along with solutions in previous papers [14, 15]. The scope herein is to describe, in detail, the various steps of a methodology for troubleshooting in effective manner (both technically and economically) pump field failures caused by cavitation/ suction recirculation, mostly pump metal damage.

CAVITATION MODES

Blade Attached (or Sheet) Cavitation

A large number of cavitation visualization studies in pumps are available 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 stroboscope lights. According to the classic experiments of Manami, et al. [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 $NPSH_i$ (i = inception) has a minimum at a capacity which corresponds to shockless inlet flow (Q_{sl}), i.e., the relative flow reaches the blade leading edge with an incidence angle around zero degrees. The $NPSH_i$ increases at $Q > Q_{sl}$ and $Q < Q_{sl}$, with cavitation starting on the pressure (hidden) and suction (visible) side of the blade respectively. At part flow, the $NPSH_i$ peaks at a capacity slightly higher than the critical suction recirculation onset capacity, Q_{rs} (rs - suction recirculation) [1, 16]. A similar V-shape

for the incipient cavitation curve was also found by using a small head drop criterion (0.5 percent of the head of twice the impeller eye peripheral velocity) for overhung pumps [7]. Again, the curve exhibited a peak at part capacity, which is attributed to a critical incidence angle causing flow separation [7] or "stalling incidence angle" [5, 16].

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 form a large cavitation bubble of increasing length. Pumps in the field operate with an $NPSH_A$ (A = Available), which is higher than the $NPSH_R$, but significantly below the $NPSH_i$. Therefore, they operate with bubble length at the $NPSH_A$, which varies widely from 0.5 in to 4.0 in 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) vs capacity at constant speed/$NPSH_A$ has a peculiar V-shape, with the minimum at the shockless capacity (Figure 2), which is similar to the $NPSH_i$ 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 [17] 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 (average exponent 2.7) and the $NPSH_A$ (exponent 3.0), for given fluid properties and impeller materials. Therefore, for a given impeller life, say 40000 hr, the acceptable cavitation bubble length is considerably shorter for pumps running at high impeller eye speed, and so operating with high $NPSH_A$, than for small pumps running with low impeller eye speed, which implies a low $NPSH_A$. Correspondingly, the acceptable cavitation bubble length can vary from 0.5 in to 4.0 in.
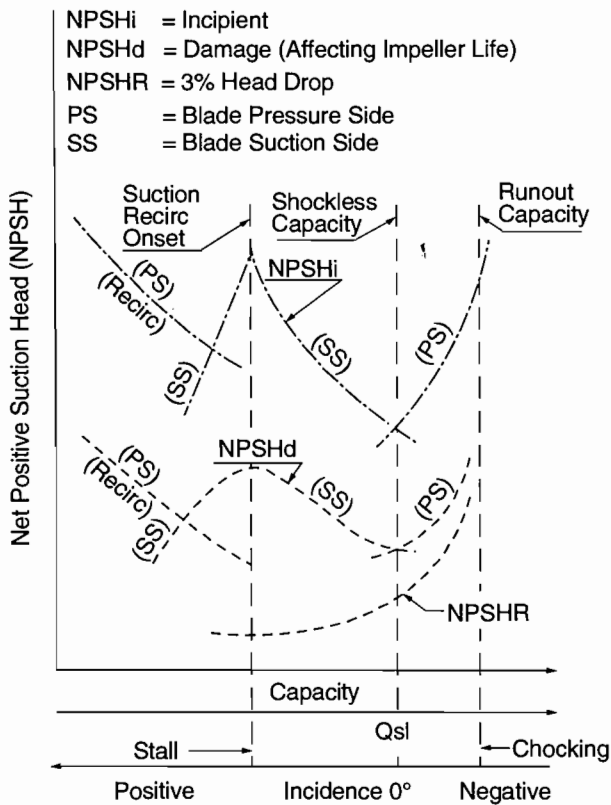

NPSHi   = Incipient
NPSHd   = Damage (Affecting Impeller Life)
NPSHR   = 3% Head Drop
PS   = Blade Pressure Side
SS   = Blade Suction Side

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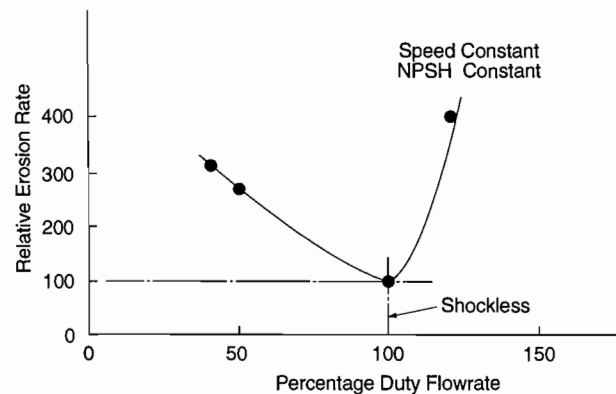

Figure 2. Variation of Erosion Rate with Capacity [8].

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" [11]. As a generic indication, such very unsteady flow regime occurs in the capacity range from zero percent to 50 percent, 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 (U_{eye}) and $NPSH_A$ level.

Experimental investigation by means of a high speed movie camera along with stroboscope [18] clearly shows that at low flowrate, two different patterns of cavitation, "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, 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 4 (a). Moreover, streams of both backward and forward flow also can be suspected to occur in the blade-to-blade plane in the inlet region of the blade channel, as sketched in Figure 4 (b). Then, shear forces components also exist in this plane and contribute to the generation of a complex vortex in the three-dimensional space. When the inlet pressure (therefore, $NPSH_A$) is low enough and also the strength of the vortex (i.e., intensity of the suction recirculation) is high enough, the pressure in the vortex core drops below the saturation pressure and cavitation conditions are reached. A filament of cavitating flow develops, starting on the suction side of the blade and ending on the pressure side of the next blade, as shown in Figure 4 (c). 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 4 (d). Consequently, damage is caused in the form of a single large crater at the midspan of the blade on the pressure side.

A typical curve of $NPSH_d$ (d = damage) which can produce significant erosion damage throughout the whole range of operations is shown in Figure 1. The $NPSH_d$ is not unique and depends upon the desired impeller life, the pump design, the material
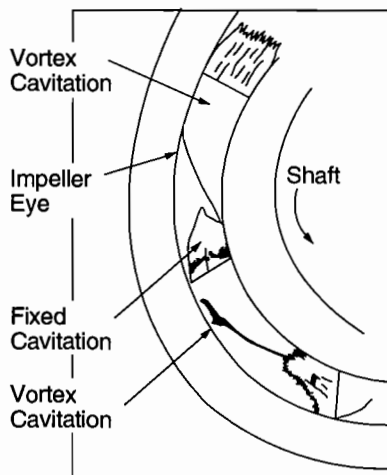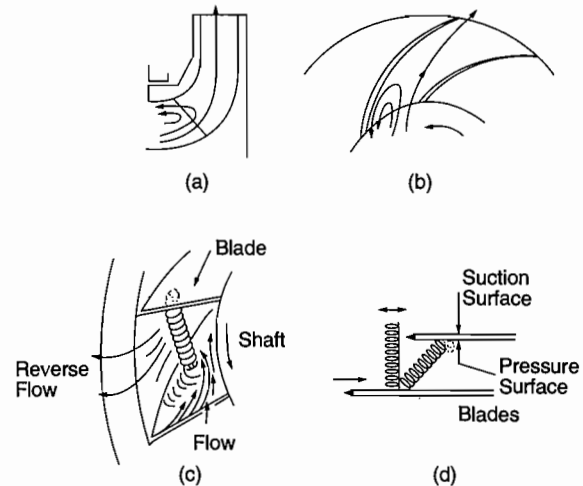

Figure 4. *Suction Recirculation (a-b) as Source of the Vortex Cavitation (c-d) (18).*

characteristics, the fluid density, and the 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 level of pump reliability. Rangeability and efficiency also have to be included in the balance.

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 flow patterns 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.

Inlet Flow Influence

Flow Distortion. A strong influence on cavitation inception ($NPSH_i$) and damage ($NPSH_d$) 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 pumps, the shape of $NPSH_i$ 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 [11] that both the shape and size of the cavitation bubble on each impeller blade are periodic functions of time, as the blade crosses flow zones with positive flow, swirling either in the same direction as 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 blade surfaces, but the erosion is prevalent on the pressure side.

Flow Imbalance. Field experience indicates that with double suction impellers, the cavitation damage pattern (like pitting) may be different for each impeller eye, thus suggesting there is a flow

Figure 3. *Alternating Sheet Cavitation with Vortex Cavitation [18].*

imbalance forced by the upstream geometry (suction piping and/ or pump inlet bay).

Cavitation Due to Impeller-Diffuser (Volute) Interaction

The interaction between impeller and diffuser generates pressure pulsations at the impeller exit/diffuser (or volute) inlet.

Pressure fluctuations inside the diffuser channel have been measured since 1936 [19]. At the normal operation point near the BEP and in absence of cavitation, the peak-to-peak amplitude is high at the diffuser throat, with the minimum level well above the pressure at the impeller suction. However, the presence of high cavitation at the impeller inlet amplifies the pressure fluctuations at the diffuser inlet (by making the fluid more elastic), and the instantaneous pressure at the diffuser throat becomes lower than the pressure at the impeller suction for a fraction of the vane passage time. Therefore, vapor bubbles are generated and collapse periodically, producing damage on all the metal surfaces that are exposed to the reflection of pressure waves caused by the implosion of the cavitation bubbles. This cavitation mode can be referred as "unsteady cavitation," due to impeller-diffuser interaction.

The amplitude of the pressure fluctuations at the impeller exit/ diffuser inlet depends mostly on the radial clearance between the impeller and the diffuser. It also increases at part capacity.

The damage is usually located at the impeller exit, both on the vane pressure (visible) side (from the hub to the tip), and also around the external edge of the shrouds (hub and tip). Moreover, the damage can be present at the diffuser inlet on the suction (visible) side of the vanes, and also on the two annular wall surfaces facing the impeller shrouds. The damage aspect is characterized by uniformly spread pitting with many minicraters of more or less constant diameter. This suggests a cavitation erosion mechanism with a cluster of thousands of small vapor bubbles, homogeneously distributed and individually imploding. It should be also noted that in some cases no damage at all was observed at the impeller inlet (vanes and shrouds).

FAILURE ANALYSIS

Problem Identification: Failure Mode

The very first step is to possibly identify what type of pump failure is occurring, i.e., one of the above failure mode or a combination of them or none of them. Therefore, the most characteristic symptoms (quantitative and/or qualitative) should be first analyzed.

Type of Symptoms/Effects

The symptoms that are currently considered in relation with the occurrence of the above cavitation/suction recirculation modes are:

• *Metal Damage.* The loss of metal or erosion in the wet parts is the parameter considered. The aspect of the damaged zone can look differently: pitting ("spongy surface"), craters (one large or several distributed), rough surface as effect of innumereous minicraters, striations, perforations. Potential areas for damage location are: impeller (mostly at inlet and in peculiar cases at the exit), shaft surface (at pump eye and under impeller bore), volute/ diffuser (at inlet), shaft sleeve (impeller eye), front wear ring (zone close to impeller eye), inlet guide vanes or splitters/ribs/stop piece in front of the impeller eye, and suction elbow. Damage can be discovered only by visual observation (direct inspection and/or borescope) or detected by means of special instrumentation, which at the present time is still under evaluation (lab/field).

• *Audible Noise.* The frequency of airborne noise is in the acoustic range up to 10 Khz, mostly concentrated up to 4.0 to 5.0 kHz. Then no special instrumentation is needed. This noise frequency range by itself does not imply occurrence of metal damage.

• *High Frequency Noise.* There is experimental evidence that cavitation metal damage rate (erosion intensity) is characterized by high frequency acoustic spectra [12] from a 5.0 to 10 kHz up to 180 to 200 kHz, with high concentration of noise activity in the range of 40 to 120 kHz. Then the detection of fluid borne noise at high frequency by a means of a flush-mounted piezoelectric transducer is an indication of erosion intensity due to cavitation. This method (which is still experimental with very few field applications [20]), has the potential to give good insights on a comparative basis by using a baseline spectrum, such as the one produced by the trouble pump configuration.

• *Suction Pressure Pulsations.* The amplitude and the frequency spectra of the pressure pulsations at the pump suction are affected by cavitation and/or suction recirculation phenomena. The most immediate aspect is a change of the amplitude level, compared to pump operating conditions free from cavitation and/ or suction recirculation (reference conditions) [16, 21]. Absolute values of the peak-to-peak pressure amplitude can give at the best only qualitative indications about the failure mode. Relative levels obtained by comparison to unaffected reference levels can be more indicative about the nature and intensity of the failure mechanism, especially in case of dominant suction recirculation and absent/ weak cavitation. In general, the peak-to-peak amplitude of the relative suction pressure pulsations increases with the intensity of the suction recirculation. The relationship between suction pressure pulsations and cavitation degree (level of $NPSH_A$ above $NPSH_R$) shown by laboratory clean data is too complex for effective use in troubleshooting. It is important to note that the level of measured pressure pulsations is very dependent on various specific factors (location and mounting of pressure transducers, fluid properties, gas content, geometry of suction casing, etc.), which exclude general quantitative criteria for failure identification. The frequency spectra can give more immediate insights into the nature of the failure, in case of suction rotating stall (frequency from 0.3 to 0.5 of the running frequency, but more research is needed) and surge (very low frequency up to 10 Hz, in most cases).

• *Casing Vibration.* There is experimental evidence that the casing vibrations measured by a means of accelerometer(s) mounted externally on the pump casing increase with the occurrence of cavitation and/or suction recirculation. Some experimental results indicate the existence of a quantitative relationship between cavitation erosion intensity and the casing ultrasonic vibrations (overall value) in the range from 1.0 kHz to 20 kHz [22]. The absolute value of the vibration level depends on numerous parameters, including stiffness/damping of the casing (geometrical shape, wall thickness, material), mounting position of accelerometer, and liquid properties. However, the casing ultrasonic vibrations method permits comparison of cavitation erosion intensity in a particular machine at various operating conditions (e.g., speed, capacity, $NPSH_A$), thus giving direct insights about the failure mechanism, especially for cavitation damage.

• *Bearing Vibrations.* There is general concern that cavitation would produce bearing vibration because of the pressure pulses generated by bubble collapse. However, in the author's experience with several field failures associated with heavily damaging cavitation (ultrasonic cavitation), no external sign of bearing vibration increase was evident. The change of bearing vibration is a symptom unlikely, or too weak, in order to identify pure cavitation failures.

• *Sudden Failure (Catastrophic).* There is a high statistical indication from a wide number of field troubles clearly related with cavitation and/or suction recirculation, that sudden catastrophic pump failures leading to a forced outage of the plant, or to heavy damage of the machinery equipment, are rare. The basic

reason is that these two flow mechanisms induce pump distress of fatigue nature and thus, accumulation and service time is a crucial factor. One or more of the symptoms previously outlined appear before reaching the critical point, which in the fatigue life curve, anticipates a rapidly increasing distress and, consequentially ,sudden failure. Such symptoms are easily detectable with periodic monitoring of the field operating parameters and/or inspection of the pump internal components.

Abnormal pump operating conditions, which in principle can cause rapid failures, are: a) pump flashing and consequent dry running (sudden loss or quick drop of the suction pressure, pump operating capacity below minimum thermal flow), and b) severe unsteady flows (e.g., surge), leading to vibrations above the alarm level.

Quantitative Analysis

The second step is quantitative analysis aimed at determining some key quantitative parameters, which are always required in combination with the aforementioned symptoms for correct identification of the failure. A second scope of the analysis is to evaluate how critical the failure is in terms of pump performance and reliability, giving good insights about the solution strategy.

Quantitative Parameters

• *Rated Conditions (Impeller duty OD).* As a starting point, the rated duty of the pump at the impeller duty diameter (Dduty) should be considered

Speed:	Nrated
Capacity:	Qrated
Head:	Hrated
Efficiency:	μrated
Required NPSH:	$NPSH_R$rated
Available NPSH:	$NPSH_A$rated
Peripheral vel. at imp. eye:	U_{eye}-rated
Operating time (% - year total):	$Trated/T_{tot}$
NPSH margin	$(NPSH_A/NPSH_R)$rated

These values, which are easily available from the order files, give preliminary insights about pump operations conditions to start the troubleshooting analysis. It is necessary to verify whether the actual basic duty is at the original rated point.

• *Best Efficiency Point Capacity.* The pump capacity at the best efficiency point (BEP) for the impeller design diameter is a first overall index of the best operating point with the lowest level of the pump distress. Also, the BEP capacity at the duty impeller diameter can be considered, although it is less important. Then the following data should be written for both the impeller design diameter (des) and the impeller duty diameter (duty):

Impeller diameter	D_{des}(duty)
BEP capacity	$Q_{BEP\text{-}des}$(duty)
$NPSH_R$	$NPSH_{rSEP,des}$(duty)
Diameter ratio	$Dduty/D_{des}$
Capacity ratio	$(Qrated/Q_{bep})_{des}$(duty)

The preceding values are available in the order files (test curve) and from the pump manufacturer. The diameter ratio and the capacity ratio give immediate independent indications at which

degree of off-design condition the pump has been selected and is likely operating.

• *Shockless capacity.* The most important and more specific index of best operating suction conditions is the impeller shockless capacity (Q_{sl}, sl = shockless), not the BEP capacity. Theory and experiments indicate that impeller cavitation and suction recirculation are primarily and crucially interrelated to the impeller shockless capacity. This capacity corresponds to zero incidence angle (smooth matching of the flow direction with the blade angle at inlet), and so is exclusively determined by the impeller geometry at the inlet (area, diameter, and angle of the blade at the leading edge mainly at the tip section). Thus, the shockless capacity is invariant with the impeller trimming diameter. On the contrary, the BEP capacity is determined by the geometry, and flow conditions in all the pump components, especially volute (diffuser) geometry which have none or weak impact on impeller inlet cavitation and/ or suction recirculation. Then the following quantitative data (sl = shockless) have keen important in the troubleshooting procedure:

Shockless capacity	Q_{sl}
Shockless-to-bep capacity ratio	$Q_{sl}/Q_{BEP\text{-}des}$
Rated-to-shockless capacity ratio	Qrated/Qsl
Required NPSH	$NPSH_{Rsl}$
Suction specific speed	S_{sl}

The value of the shockless capacity can be derived from impeller drawing data (available only to the pump manufacturer) or from direct measurements of the impeller geometry, as shown in Appendix A. For quick evaluation, it can be assumed for many current designs that Q_{sl} is from 110 percent to 130 percent of the $Q_{BEP\text{-}des}$. However, an accurate determination of Q_{sl} is recommended when the impeller eye peripheral velocity is above 80 ft/s and/or the suction specific speed at shockless flow is above 11,000 (US units).

• *Suction Recirculation Capacity.* Another very crucial parameter is the suction recirculation onset capacity, Q_{rs}. This capacity is quite peculiar, as clearly shown by accurate measurements (5, 16, 23). It is widely claimed in the literature that the suction recirculation onset capacity by itself is not a concern, because suction recirculation becomes damaging only at capacity below Q_{rs}. Such a statement is misleading, because it focuses only on suction recirculation intensity as potential cause of pump distress. This claim ignores two other critical flow conditions fully evident from experimental data, which occur at, and close to, the suction recirculation onset: a) the suction side sheet cavitation reaches the highest intensity level [1, 9, 16], as evident in Figure 1, and b) suction flow unsteadiness in the form of rotating stall is observed with the highest amplitude [24, 25, 26].

The determination of Q_{rs} is a key step in the troubleshooting procedure for part flow operations. Moreover, it represents a well distinct criterium for discriminating between pure cavitation and suction recirculation, thus avoiding some erroneous interpretations which in the past have contributed to create the high concern about suction recirculation with high suction specific speed impellers. The following parameters covering various peculiar capacities are crucial:

Suction recir. onset	Q_{rs}
Recirc. onset-to-bep (des)	$Q_{rs}/Q_{bep\text{-}des}$
Recirc. onset-to-shockless	Q_{rs}/Q_{sl}
Rated-to-recirc. onset	$Qrated/Q_{rs}$

The value of Q_{rs} is essentially determined by the impeller design, including various factors and three-dimensional flows effects, which prevent a simple and straightforward prediction correlation [8, 16, 26]. Some simple empirical methods for predicting Q_{rs} have been published in the last years, which are based on one-dimensional approach and using "calibrations" or "fitness curve" along with some impeller geometrical parameters. These formulas can give only rough indications with error of ± 30 percent, as shown by comparison with experimental values. But totally misleading results (100 percent error) are obtained for families of impellers optimized with design criteria, not contemplated in the backup data base. In fact, the author has verified that such empirical formulas predict $Q_{rs}/Q_{bep-des}$ ≈ 80 percent for new optimized design impellers that show experimental value of $Q_{rs}/Q_{bep-des}$ ≈ 40 percent or less. Other straightforward charts for predicting Q_{rs}, using suction specific speed as the main input parameter, give even rougher indications, which might be applicable for old design impellers only. Such global charts should be totally disregarded for state of the art design impellers using the latest knowledge about cavitation and recirculation.

In doubt, the value of Q_{rs} should be produced by the pump manufacturer who may have internal data. The onset recirculation capacity can be determined experimentally in various ways [5, 16, 23]. A very straightforward method for end suction pumps is to measure the wall static pressure at three axial locations along the suction pipe (e.g., x/D_{pipe} = 0.3/1.0/1.8, x = distance from blade leading edge at impeller tip). The plot of each local relative pressure (p_x - p_{suct}, suct = suction time), vs capacity, will clearly show the capacity Q_{rs}, x at which the suction recirculation flow pattern reaches the measurement axial station. Then the plot of Q_{rs}, x vs x/D_{pipe} should be extrapolated to x/D_{pipe} = 0 to obtain the value of Q_{rs}. Bear in mind that values of suction recirculation onset capacity measured at locations in the suction pipe far from the impeller blade leading edge are not accurate and below the actual value of Q_{rs}.

• *Actual Operating Range.* The actual range of operating conditions from minimum continuous flow up to maximum continuous capacity is very important for a successful diagnosis. The following data should be considered for range effect evaluation (oper: operating, cap = capacity, cont = continuous):

Plant continuous load	Minimum(Maximum)
Speed	N_{min}(max)
Capacity	$Q_{oper-min}$(max)
Required NPSH	$NPSH_R$min(max)
Available NPSH	$NPSH_A$min(max)
Operating time(%)	$T_{oper-min}$(max)/T_{tot}
Operating-to-rated cap.	$Q_{oper-min}$(max)/Qrated
Operating-to-shockless rec. cap.	(O_{per}/Q_{sl})min(max)
Operating to-suct. rec. cap.	(Qoper/Qrs)
Available to required NPSH	$(NPSH_A/NPSH_R)$min(max)
Periph. vel. at imp. eye	$U_{eye-min}$(max)

The preceding input about the plant-pump operation mode can be obtained from plant monitoring files and/or discussions with the plant operator. The percent of operating time in one year at various continuous loads (rated, minimum, maximum) is often difficult quantify from the plant operating files. However, even a rough estimate can be enough enlightening. Both the Q_{sl} and Q_{rs} at actual operating speed can be derived by linear proportion from the corresponding value at rated speed. Continuous operation means from five percent to ten percent, at least, of the total service hours during the year.

• *Intensity Factors.* While the analysis of the symptoms indicates the severity of the failure, the intensity of the root cause needs to be evaluated. Then the focus of the analysis moves to the operating parameters that determine the intensity of the flow mechanism involved in the failure. These parameters are listed in Table 1 for field cases presenting metal damage that seems to be related with cavitation and/or suction recirculation from the previous step of the failure mode identification. The relative importance (rank) of the various parameters is suggested by the current knowledge, including theory and experimental data and field experience. Moreover, the impact of each parameter from negligible to high influence which is presented in Table 1 should be assumed as a probabilistic guideline, and not as a quantitative prediction. The content of Table 1 is only indicative, and likely will change with the progress of knowledge. However, it is fully proven that the operating parameters having paramount importance are: peripheral velocity at the impeller eye (by and large the most dominant), for better design operating capacity as fraction of the shockless one, $NPSH_A$-to-$NPSH_R$ margin, fluid specific gravity, impeller material. The role of fluid temperature and gas content is not completely focused, as it includes positive and negative influences.

Table 1. Intensity Factors for Metal Damage (Cavitation and/or Suction Recirculation).

PARAMETER		LIKELY LEVEL OF IMPACT (QUALITATIVE)				
NATURE	SYMBOL (UNIT)	NEGLIGIBLE	LOW	MEDIUM	HIGH	IMPORTANCE RANK
PERIPHERAL VELOCITY AT IMPELLER EYE	Ueye (FT/S)	<50	50-70	70-100	>100	••••••••••
OPERATING CAPACITY RATIO						
ABOVE SHOCKLESS (PS - CAVIT.)	Q/Qsl	1.0-1.05	1.05-1.10	1.10-1.15	>1.15	••••••••
BETWEEN SHOCKLESS AND RECIRC. ONSET (SS-CAVIT)	(Qsl-Q) (Qsl-Qrs)	0.0-0.20 >1.30	0.25-0.45	0.50-0.70	0.75-1.25	•••••
BELOW RECIRC ONSET (PS-RECIRC)	Q/Qrs	1.0-0.9	0.85-0.75	0.70-0.50	<0.50	•••••
AVAILABLE NPSH MARGIN	NPSHA NPSHR	5.0-3.5	3.0-2.0	1.5-1.2	<1.10	•••••••
SPECIFIC GRAVITY	SG	<0.6	0.65-0.80	0.85-1.10	>1.10	•••••
FLUID TEMPERATURE	°F	CLOSE TO TRIPLE POINT	MEDIUM	LOW	COLD (<60)	••••
NON CONDENSIBLE GAS CONTENT	%	RELATIVELY HIGH	HIGH	MEDIUM	LOW < 50 ppm	••
MATERIAL		SPECIAL ALLOY	STAINLESS STEEL Ni-Al BRONZE	CARBON STEEL	CAST IRON TIN BRONZE	•••••
SUCTION SPECIFIC SPEED	S					
• SHOCKLESS		<7000	7500-9500	10000-12000	>12500	•••
• BEP-DES		<8000	8500-10500	11000-13000	>13000	•
• BEP-DUTY		IRRELEVANT	IRRELEVANT	IRRELEVANT	IRRELEVANT	NONE

The suction specific speed, which is a design indicator rather than an operating parameter, is also included in Table 1 with low ranking. There is a wide opinion that the suction specific speed is the most important index for explaining and solving pump field troubles of pumps operating at part capacity, on the basis of some statistical field data [27]. These field indications cannot be ignored, but they can be explained in a more solid and effective way by using truly physical parameters (U_{eye}, Q/Q_{sl}, Q_{rs}/Q_{sl}, $NPSH_A$/$NPSH_R$), rather than a conventional index such as suction specific speed. The definition of the S-parameter is based on "conventional" conditions (three percent head drop), which are not directly related with actual concern (damage), as shown by experimental data and theory. Troubled impellers of the old design with high S value are correctly valued in Table 1 through the aforementioned physical parameters, which permit avoiding erroneous troubleshooting conclusions with new design impellers of high S value.

The secondary role of the S-parameter, as included in Table 1, is to express the potential marginal influence of some impeller design parameters, which help to achieve high S values, and are independent from the impeller eye diameter (included as direct parameter through U_{eye}). It should be noted that the most appropriate definition of suction specific speed is at the shockless capacity

or S_{sl}. The use of the current definition with reference to the BEP capacity at impeller design diameter, or $S_{BEP-des}$, is not totally meaningful and should be used only as second choice. The calculation of the suction specific speed at the BEP capacity with trimmed impeller, or $S_{BEP-duty}$, is totally incorrect and always misleading.

The time factor is not included in Table 1, since it determines the severity of effects rather than the intensity of the cause.

Other Background Data

• *Pump Cross Section Drawing.* The cross section drawing of the pump should be reviewed always, especially for pumps with side suction casing. Attention must be given to the continuity and smoothness of the flow passage from the suction flange to the impeller eye. Sharp changes of flow direction are undesirable and act as magnification factors for lowering local $NPSH_A$ at impeller eye (eddies, flow acceleration) and promoting cavitation. Moreover, they can produce strong flow distortions, which contribute to the mismatch between flow angle and blade angle, thus altering the theoretical shockless capacity based on uniform flow distribution. Also, the onset of suction recirculation is affected by suction casing in an unpredictable way. Further, the presence and the location of any baffles and stop pieces should be reviewed and discussed with pump hydraulic design experts.

A radial type suction casing with centerline of the suction flange crossing the rotation axis tends to generate zone of positive and negative swirls at the impeller eye, which enhances cavitation at high flow and also likely promote earlier suction recirculation, thus narrowing the typical V-shape damage curve (Figures 1 and 2).

A tangential type suction casing, with centerline of the suction flange at offset from the rotation axis, tends to induce a global positive prerotation (in the same direction of the impeller rotation). Theoretically, this should increase cavitation at high flow and reduce suction recirculation. In practice, both the location of the stop piece and the internal shape of the casing has a strong influence, which in the worse case creates a zone of stagnant flow and eddies, raising the $NPSH_R$ level and narrowing the trouble free operating range.

• *Pump size and capacity (at design point).* As general criteria, the effects of cavitation and recirculation become worse with increasing pump size (impeller eye diameter) and design capacity. There is statistical indication that for design capacity below 600 US gpm (per eye), the impact of suction recirculation on pump reliability is practically zero, independent from suction specific speed.

• *Pump energy level.* It is frequently claimed that pump energy level has critical influence. However, a quantitative sound definition of this parameter relevant to cavitation/suction recirculation has not yet been given. Statistics from field data indicate that pump failures due to suction recirculation are unlikely for design head (stage) below 200 ft and design brake horsepower (eye) below 50 hp. It must be clear that such limits of capacity, head, and power are only indicative in giving a first feeling whether or not the problem might be serious or would require a careful analysis of symptoms and operating data.

• *Piping Configuration.* The layout of the suction piping should be always analyzed, with particular attention to elbow(s), flow straighteners, and valves.

DIAGNOSIS

The third step is a clear diagnosis aimed at identifying the root(s) of the problem, by analyzing both the symptoms or effects and all the quantitative parameters. Therefore, the main objective is to isolate the physical flow mechanism, which is the source of the failure mode, and also establish the severity of the failure, by correlating all the data collected in steps 1 and 2.

Basically, there are two categories of failures due to cavitation and/or suction recirculation: 1) metal damage and 2) pump distress due to highly unsteady flows.

Damage Failures

When the primary concern is metal damage, the diagnosis should first focus on the damage pattern (symptom) and the operating capacity (quantitative parameter). Additional attention should be put on geometry peculiarity (background data), if suggested by the damage pattern. The scheme of an effective diagnosis of the most frequent (but not the only ones) damage failures for cavitation and/or suction recirculation is enlightened in Table 2.

Table 2. Diagnosis for Damage Failures.

DAMAGE PATTERN		GEOMETRY PECULIARITY	OPERATING CAPACITY	FAILURE MODE	
LOCATION	ASPECT			FLOW MECHANISM	NAME
IMPELLER INLET					
BLADE PRESSURE SIDE(PS)	PITTING	-	Q > Qsl	SHEET CAVITATION (PS)	SCPS
BLADE SUCTION SIDE(SS)	PITTING	-	Qsl > Q > Qrs(s)	SHEET CAVITATION (SS)	SCSS
BLADE PRESSURE SIDE AT MIDSPAN	SINGLE LARGE CRATER	-	Q < Qrs	VORTEX CAVITATION (SUCTION RECIRCULATION)	VCSR
FILLET BETWEEN BLADE-SS AND HUB • HUB • SHAFT	DEEP CRATERS (DRILLING) PERFORATION PITTING/HDLES	-	Q < Qsl	CORNER VORTEX CAVITATION (INLET)	CVCI
BLADE-PS (MOSTLY)	PITTING	SIDE SUCTION CASING AND/OR SUCTION ELBOW(S)	HIGH (OFTEN)	FLOW DISTORTION CAVITATION	FDC
DIFFERENT AT EACH EYE	PITTING AND/OR CRATER	DOUBLE SUCTION IMPELLER	ANY	FLOW IMBALANCE (SHEET CAVIT. AND/OR VORTEX CAVIT.)	FIC
IMPELLER OUTLET AND/OR DIFFUSER (VOLUTE) INLET					
IMPELLER BLADE-PS AND/OR DIFFUSER (VOLUTE) BLADE-SS	INNUMEROUS MINICRATERS	SMALL RADIAL CLEARANCE (GAP B)	Q < Qsl	CAVITATION DUE IMPELLER- DIFFUSER (VOLUTE) INTERACTION	CIDI
FILLET BETWEEN BLADE AND SHROUDS (IMP AND/OR DIFF)	DEEP CRATERS	SMALL OR ZERO FILLET RADIUS	ANY (LOW MORE LIKELY)	CORNER VORTEX CAVITATION (EXIT)	CVCE

Damage Patterns

• *Damage Location.* The initial step for looking at the damage pattern is the location of the damaged zone. Two key areas should be carefully inspected, i.e., impeller suction (including suction casing, guide vanes, etc.) and impeller discharge (impeller and/or diffuser/volute). Within each of these zones the damage can be present at various spots as listed in Table 2, depending on the prevailing cavitation mode(s).

• *Damage Aspect and Extension.* The physical appearance of the damaged zone is also very indicative of the nature of the cavitation damaging mechanism. The aspect and the extension (by both the area and also the depth) of the damage is especially important, in that it reflects the intensity of the source and also the residual life of the component. The most characteristic aspects of damage pattern, which have been described previously in relation to the corresponding cavitation mode, are listed in Table 2.

Geometry Peculiarity

In some failure cases, the overall picture of the damage pattern can present additional distinct characters like peculiar geometrical differentiations. Typical situations are: variation from one blade to the other (nonperiodicity), circular asymmetry around the shaft axis, planar asymmetry for double suction impeller, different severity at tip and hub (impeller outlet, diffuser/volute inlet), and localized zones (corners). In all these cases, the geometry of the pump (impeller, suction casing, diffuser/volute) and/or suction

piping has peculiar geometrical features, which are primary factors in causing or enhancing the failure. Some, but not all direct peculiar geometrical factors are listed in Table 2.

Operating Capacity

The quantitative determination (even if not accurate) of the pump operating capacity range is extremely important to draw clear final conclusions about the cavitation mode. In fact, there are damage patterns that occur in the same locations with aspects not too different (in case of extensive or nearly terminal damage), but are caused by different flow mechanisms, depending on the operating capacity. This situation is frequently found when the damage is located on the impeller blade pressure (hidden) side, which can be caused by sheet cavitation on blade pressure side ($Q > Q_{sl}$) or vortex cavitation, due to suction recirculation ($Q < Q_{rs} < Q_{sl}$). In a few field troubles of pumps, which were operating in a wide range of capacity above and below BEP and presented damage on blade pressure side, the author has successfully proven that blade sheet cavitation was the cause of the damage, while other analyses erroneously blamed suction recirculation. The illustrating and decisive step in the author's analysis was the quantitative correlation of the operating range with the shockless capacity and the suction recirculation onset capacity. The author's quantitative analysis excluded suction recirculation as a probable factor, because part flow operations were demonstrated to be above the recirculation onset point, while high flow pressure side cavitation appeared the most likely source of damage. On the opposite sides the erroneous analysis approach was focused only on a part of the data (operations below BEP), without any quantitative approach.

The interrelationship between the pump operating range and the failure mode that is outlined in Table 2 should be always kept in mind for effective diagnosis of the damage source.

Failure Mode

The various cavitation modes that usually produce metal damage have been described above and are listed as damage failure modes in Table 2. The diagnosis of the failure mode, starting from symptoms and also using quantitative data, is the conclusive step of the failure analysis, as the basis of the troubleshooting strategy. In Table 2, the failure mode is identified with the corresponding flow mechanism and a short code which refers to both the physical nature of the cause and also the location of the damaged zone.

Unsteady Flow Failures

There are field troubles when the pump is operating at part flows, and is subjected to various type of distress, usually high vibrations along with other aspects of concern, which can be related generally to highly unsteady internal flows. The following flow mechanisms can be sources of potentially strong flow unsteadiness at part capacity:

- Suction recirculation
- Rotating stall at impeller inlet
- Surge

The most frequent symptoms are vibrations and pressure pulsations, which can become too high and not tolerable even for short operating time. However, the hydraulic-mechanical response of the system is as crucial about the failure as the nature and intensity of the above exciting flow mechanism(s). Moreover, other part flow pump instabilities, also occurring at part capacity, might be primary or contributory sources to the failure.

The troubleshooting methodology for these types of failures is not discussed herein. The pump operator can find useful generic insights in the literature [5, 16, 21, 24, 25, 26, 28, 29] and should discuss these types of failures case by case with pump hydraulic design experts.

SOLUTION STRATEGY

The fourth step is to decide on an effective solution strategy which must include three major aspects: a) technical change(s) as suggested by the diagnosis, b) time factor as dictated by both the severity of the trouble and also the life expectancy, and c) field implementation cost (material, installation, operation, potential loss of production). Commonly there are three levels of intervention:

- Urgent but temporary fix.
- Medium-long term solution (quasi permanent).
- Very ultimate solution (fully permanent).

The solution strategy, which may involve only one level or a sequence of levels of intervention, should be jointly discussed and agreed on by the pump designer (or the troubleshooter) and the pump user, in order to evaluate the impact of the solution on the pump performance/reliability, cost of fix, and plant availability. Consistently with the diagnosis procedure outlined herein, the various potential solutions suggested by both the field experience and the present knowledge are presented hereinafter with reference to metal damage.

Damage Failures

The main scope of the solution strategy is essentially focused on the damage rate with increasing requirements of its reduction, control and possibly elimination. The second objective is a low degree of monitoring and inspection to give the field pump operator a high confidence and strong sense of security. The third objective is to minimize the impact on the production schedule and plant availability (for installation and especially future operations). The fourth objective is to maintain the pump performance (head, efficiency) and avoid restrictions for the operating range. The fifth objective is to minimize the installation cost and, especially, the pump maintenance cost.

The various solution approaches for damage failures are listed in Table 3.

Table 3. Solution Strategy for Damage Failures (Low or Marginal Cost).

FAILURE MODE		URGENT BUT TEMPORARY FIX			MEDIUM LONG TERM SOLUTION ACCEPTABLE LIFE TARGET WITH MARGINAL COST (INSTALL AND OPERATE)		
FLOW MECHANISM	NAME	PUMP GEOMETRY	MATERIAL	SYSTEM	PUMP REDESIGN		SYSTEM MODIF
					GEOMETRY	MATERIAL	
SHEET CAVITATION (PS)	SCPS	BLADE LEADING EDGE INDUCER	WELDING (SPECIAL CASES)	FLOW STRAIGHTENERS VANED ELBOWS (MITRA TYPE OR SPECIAL DESIGNS)	IMPELLER (INDUCER)	UPGRADE	NO
SHEET CAVITATION (SS)	SCSS		UPGRADE			A) HIGH CAVITATION RESISTANCE MATERIALS	NO
VORTEX CAVITATION (SUCT RECIRC)	VCSR	VARIOUS THROTTLING INLET DEVICES INDUCER INLET GUIDE VANES	A) HIGH CAVITATION RESISTANCE MATERIALS				NO
CORNER VORTEX CAVITATION (INLET)	CVCI		B) HYDROLOY (IQ11)			B) HYDROLOY (IQ11)	NO
FLOW DISTORTION CAVITATION	FDC	BAFFLES GUIDE VANES IN SUCTION CASING SPECIAL INSERT AT IMPELLER EYE			IMPELLER AND/OR SUCTION CASING		FLOW STRAIGHT VANED ELBOWS
FLOW IMBALANCE (SHEET CAVIT AND/OR VORTEX CAVIT)	FIC						
CAVITATION DUE IMPELLER-DIFFUSER (VOLUTE) INTERACTION	IDIC	LARGE GAP IMP-DIFF (VOL) BLADE (IMP DIFF) MODIFICATION	LOCAL REPAIR WITH SPECIAL ALLOY DEPENDING ON BASE METAL		IMPELLER AND/OR DIFFUSER BLADE (VOLUTE CUTWATER)		NO
CORNER VORTEX CAVITATION (EXIT IMPELLER OUTLET DIFFUSER OR VOLUME INLET)	CVCE	LARGE FILLET RADIUS BLADE (IMP DIFF) MODIFICATIONS					NO

Urgent but Temporary Fix

When the damage has progressed in extent and in depth to the point of threatening the pump integrity, then it is necessary to do some quick fixes for temporary relief. Typical cases are: perforation of the blade(s) (impeller, inducer, diffuser/ volute) sufficiently long and nearly close to blade breakage, loss of large part of blade(s) with potential vibration for imbalance, blade cracks, impeller hub perforation, and shaft damage. Moreover, a temporary fix might be requested by the plant operator because of

production needs or plant outage schedule. Further, usually a simple temporary fix is associated with relatively low installation cost, and so might be chosen by the user because of economic considerations.

Pump-System Geometry Modification. With reference to all the failure modes that are induced by cavitation at the impeller inlet (Table 2), the following criteria should be applied for selecting and properly sizing the pump (system) fix:

· Change and move the impeller shockless capacity more close to the base load pump capacity (Q_{rated}/Q_{sl} modified = 1.0 ± 0.2).

· Increase the NPSH margin by lowering or shifting, even marginally, the $NPSH_R$ curve.

· Reduce any sharp change of flow direction upstream of the impeller eye (suction casing, suction piping).

· Eliminate sharp corners between blade surfaces and impeller shroud (tip/hub).

· Restrict the pump operating range in a more narrow window in relation with the modified shockless capacity (Qrated- modi-fied/Q_{sl}-modified = 1.0 ± 0.2) and, eventually, the modified suction recirculation capacity (Qrated-modified/ Q_{rs}-modified > 0.8 to 1.0).

Then the most appropriate modification for the pump geometry depends on the specific cavitation failure mode expected for the actual pump capacity with the longest period of service.

Sheet Cavitation on Pressure Side (SPCS). The impeller shockless capacity should be increased (Figure 1). This can be done in several cases by cutting back appropriately the blade leading edge. Also, blade grinding on the pressure side (when accessible) is effective. Moreover, the $NPSH_R$ at high flow is reduced by these modifications for common impeller designs. However, the risk of cavitation and/or suction recirculation at part flow is increased. Then a higher minimum continuous flow should be set along with these temporary modifications.

A second approach is to use an inducer designed for moderate suction specific speed (from 15,000 to 18,000 US units) and optimized for high flowrate, close to the basic pump duty. With the current inducer technology, an inducer can be designed and man-ufactured with an NC machine in two to three weeks. The use of inducer certainly allows quick/low cost effective solutions for high flow duties. With inducer tip speed Ut-ind up to 100 ft/s (N = 3600 rpm, D_t-ind = 6 1/2 in), the time length of this solution can vary from one year to three years, roughly depending on various cavitation intensity factors (Table 1) and minimum continuous flow. Inducers with tip speed up 125 ft/s (3600 rpm, D_t-ind = 8 in) can be still used as a temporary fix for one year at maximum, but it requires a careful evaluation of the operating range, especially the minimum continuous duty.

A third approach is to operate the pump below a maximum allowed capacity, then limiting production.

Sheet Cavitation on Suction Side (SCSS). The impeller shockless capacity should be reduced (Figure 1). This is obtained in the most simple way by reducing the net area at the impeller eye and so increasing the through flow (or meridional) velocity. Some inlet devices based on this approach of throttling the impeller eye include: stationary casing wear ring extended into the impeller eye (L-shape, nozzle-shape), stationary ring in the pump suction noz-zle, rotating obstruction like venturi inside the impeller eye. An additional effect of these devices, if well designed with appropri-ate contour, is to produce a lower "effective" peripheral velocity (Ueye-eff) and so reduce the cavitation damage rate (Table 1) [9, 17]. The original $NPSH_R$ curve is shifted toward lower capacity and marginally risen at part flow. But the optimum range for minimum cavitation erosion rate (V-shape in Figures 1, 2) is moved close to the actual operating range, thus reducing cavitation

damage (better incidence angle, lower "effective" $U_{eye-eff}$). These devices have a major drawback of preventing pump duties at high flowrate, because of sharp increase of $NPSH_R$ and also remarkable reduction of head and efficiency, if the throttle diameter, D_e, is too low in comparison to the "free" impeller eye diameter, $D_{eye-free}$. According to the author's experience, the penalty at high flow is marginal, if the throttle diameter complies with both the following restrictions (th = throttle):

$$(D_{th}/D_{eye-free}) > 0.85$$
$$(A_{th}/A_{eye-free}) > 0.75$$

where both the throttle area, A_{th}, and the "free" eye area, $A_{eye-free}$, include the blockage due to the impeller hub/shaft. However, it is recommended to set a limit for the maximum continuous flow roughly close to the BEP capacity at the impeller design diameter ($Q_{max-cont}/Q_{bep-des}$ = 1.0 ± 0.1).

A second approach is to use an inducer designed for relatively moderate suction specific speed (S = 16,000 to 20,000) and optimized for low flowrate close to the basic pump duty. The maximum limit value for inducer tip speed can be slightly higher for suction side cavitation: U_t-ind = 125 ft/s (3600 rpm, D_t-ind = 8 in) for relatively medium term solution, and U_t-ind = 150 ft/s (3600 rpm, D_t-ind = 10 in) for short term (one year maximum) solution, along with careful evaluation of the operation range (minimum and maximum continuous duty).

A third potential approach is to use an inlet guide vane of relatively simple shape for quick manufacturing by N.C. machin-ing. A twofold advantage can be achieved with guide vanes, generating an appropriate degree of positive preswirl (in the same direction of the impeller rotation) i.e., a) reduction of the shockless capacity and b) reduction of the $NPSH_R$ level at low flow. Also, the inlet relative velocity is reduced, which theoretically should lead to lower cavitation erosion rate.

A fourth approach is to operate the pump above a minimum allowed capacity, thus limiting the operating range.

Vortex Cavitation (Suction Recirculation) (VCSR). The suction recirculation onset capacity should be lowered. The quickest and most effective way is to reduce the shockless capacity. Then the same approaches discussed earlier (SCSS) are also instrumental for a quick intervention and a temporary relief of the metal damage produced by the vortex cavitation, due to suction recirculation. In summary:

· Impeller eye throttling devices

· Inducer for low capacity and moderate suction specific speed

· Inlet guide vanes

The same dimensioning criteria and the limitations/drawbacks outlined previously (SCSS) apply also to this case (VCSR).

A fourth approach is to raise the minimum continuous flow of the pump (Q_{mcf}) closer to or even above the suction recirculation capacity Q_{rs}, say $Q_{mcf}/Q_{rs} \approx 0.7$ to 1.0 (small size, low energy pumps) or $Q_{mcf}/Q_{rs} > 1.1$ (large size, high energy pumps).

A fifth intervention is to bypass part of the flowrate from the pump discharge to the suction by installing a flow recirculation line. In case this line already exists, the size of the line (tube diameter, valve) has to be increased. Then, the minimum contin-uous flow at the pump inlet (Q_{mcf}-inlet) should be increased in accordance with the above guidelines.

Corner Vortex Cavitation at Inlet (CVCI). For this failure mode, the most frequent location of the damage is at the corner between the suction side of the blade and the impeller hub surface. The primary reason for this highly localized cavitation is a high incidence flow angle at the blade root. While a secondary reason is the sharp corner, or even an acute angle, between the convex blade surface and the concave impeller hub.

Flow visualization studies show that an effective way to reduce the cavitation bubble length at the blade root is to use a cone-insert at the impeller hub [14]. This cone increases the local meridional velocity and reduces the incidence angle.

Modifications of the blade angle at the inlet can be used as a temporary relief, but with very marginal impact. One way is to grind the blade on the suction (visible) surface near the hub. Another way is to cut the blade leading edge from the hub to the tip along an appropriate line to optimize the incidence angle distribution. This type of blade modification can be decided only by the pump designer, with the necessary support of the impeller pattern drawing. An incorrect cut of the blade leading edge dramatically risks worsening the cavitation damage. In several cases, attempts have been made to reduce the sharpness of the corner, and also increase the cavitation resistance, by filling the corner with special materials (cobalt base alloy, nitriding, epoxy coating), which are considered to have strong resistance to cavitation damage. In practice, these methods have failed nearly systematically and in short time, especially for high speed and/or high energy pumps (feedwater), even to making the local damage worse. The main reason is that a very strong bond between the base metal and the filling material is needed to withstand the fatigue stresses from the pressure pulses of extremely high intensity, produced by the collapse of the cavitation bubbles.

Flow Distortion Cavitation (FDC). A flow distortion at the impeller eye means that the through-flow velocity (meridional component) is not uniform as assumed in designing the impeller. Thus, there are zones with high velocity along with pockets of low or zero velocity, generating a nonuniform repartition of the total capacity among all the blades. Moreover, the flow distortion may be characterized by irregular zones of positive swirling flow along with zones of negative swirling flow. A highly distorted flow can produce the simultaneous occurrence in the impeller of all the above four cavitation modes from one blade to another. Each blade also experiences, during one impeller rotation, the sequence of various cavitation regimes (pressure side, suction side, recirculation and corner vortex). It is evident that no impeller modification would be helpful.

For end suction pumps, the root of the flow distortion is exclusively related with the suction piping configuration, namely elbows and/or sudden area changes (especially enlargements) and/or asymmetric pipe components. For pumps with side inlets, the suction casing is another potential area for generating a flow distortion or enhancing any flow disturbance coming from the suction piping.

The adoption of the most effective quick fix depends on the available NPSH (apparent).

$NPSH_A$-apparent/$NPSH_R \geq 1.5$

The $NPSH_A$-apparent is the one easily obtained in the usual way from suction pressure data and fluid temperature, assuming uniform flow. On the contrary, the $NPSH_A$-actual at the impeller eye can be much lower locally, because of the flow distortion and is truly unknown. For suction piping improvements, it is recommended to use a flow straightener, especially in case of several elbows in series. An advanced design flow straightener is now available [30], which is easy to install and also produces a low head loss. Also, vaned elbows can be used with simple vane design (mitre-bend) [31]. A recently developed flow stabilizer with special shape of the vanes [32] can be adopted in case of a single elbow close to the pump suction flange. It is worthwhile to notice that all these devices, which certainly produce an additional head loss, and so reduce the $NPSH_A$-apparent, in effect improve the actual $NPSH_A$ at the impeller eye by eliminating or reducing the flow distortion. Then, the verification of these devices simply requires that the resulting $NPSH_A$-apparent is still above the $NPSH_R$ with the usual

margin suggested by the pump designer or the personal experience.

With regard to the suction casing, a quick improvement can be made by inserting baffle(s) [33], splitter(s), guide vanes, or modifying the shape of the casing ring and stop piece. However, it is recommended to refer to the pump designer for modification of the side suction casing.

$NPSH_A$-apparent/$NPSH_R < 1.5$

In this case, any additional loss (flow straightener, etc.) upstream of the impeller eye, which will reduce the $NPSH_A$-apparent, must be carefully considered even assuming a full benefit from total elimination of the flow distortion. Then an appropriate temporary action could be to narrow the pump operating capacity, by excluding pump capacities too far away from the shockless capacity, and so limiting both the high flows ($Q/Q_{sl} = 1.0 \pm 0.1$) and also the low flows ($Q/Q_{rs} = 1.0 \pm 0.2$). It is evident that these margins depend primarily on the peripheral velocity at the impeller eye, U_{eye}, and other factors which can be inferred from Table 1.

Flow Imbalance Cavitation (FIC). With pumps using a first stage double suction impeller, it may happen that the total capacity is not equally split between the inboard eye and the outboard eye of the impeller. In this situation, called "flow imbalance," one side of the impeller receives a higher fraction of the total flow, while the other side of the impeller might be partially or completely starved. This means that, with reference to Figure 1, each side of the impeller works at an off-design condition that usually is far away from the optimum one. Then highly intense cavitation can be caused, even if the total capacity is seemingly near to the optimum or shockless capacity with apparently ample $NPSH_A$-to-$NPSH_R$ margin. Moreover, the cavitation mechanism and location of damage is totally different at each side of the impeller. Depending on Q_{rated}/Q_{sl}, the damage pattern corresponds to the simultaneous occurrence of one of the following combinations of failure modes (Table 3): a) heavy SCPS and more of less light SCSS (plus eventually CVCI); b) heavy SCSS (plus eventually CVCI) and more or less light SCPS; c) heavy SCSS (plus eventually CVCI) and relatively heavy VCSR (suction recirculation); and d) heavy VCSR and relatively heavy SCSS (plus eventually CVCI).

In all the cases of flow imbalance a strong flow distortion is also present which makes the cavitation damage rate even stronger, even if the $NPSH_A$-to-$NPSH_R$ margin is apparently very large.

The root of flow imbalance is in the suction piping configuration and/or in the geometry of the suction casing. The various methods for temporary fix outlined above for flow distortions can be used also in case of flow imbalance. Evidently, the action of limiting the range of capacity is not effective here, since the situation can shift from one of the previous combinations of failure modes to another one, e.g., from event a) to event b) which can be equally damaging.

Other specific temporary actions are:

• Make the geometry of one side of the suction casing as much as possible specular like mirror image of the other side, eventually putting appropriate inserts (baffles, etc.) of same shape at symmetrical locations.

• Attenuate the effect of the dominating cavitation mechanism at the impeller side with the most severe damage by using a pertinent temporary fix, as suggested above, for that side of the impeller [14].

Cavitation Due to Impeller-Diffuser (Volute) Interaction (CIDI). The peak-to-peak amplitude of the pressure pulsations at the impeller discharge should be reduced. The most effective quick modification is to increase the radial clearance between the diffuser blade leading edge (or volute cutwater) (diameter D_3) and the impeller blade trailing edge (diameter D_2). The minimum required ratio, D_3/D_2, increases with the pump (stage) head, H, and also the peripheral velocity at the impeller outlet, U_2. As a generic guide-

line, with H > 300 ft and/or U_2 > 130 ft/s it should be: (D_3/D_2)min > 1.03 for vaned diffuser or $(\bar{D}_3/D_2)$ min > 1.06 for single volute. However, this minimum ratio is also related to the blade angle at the impeller exit, and to the combination of the number of vanes for the impeller and the diffuser. As first choice, it is preferable to trim the impeller (smaller D_2 and lower U_2). The next choice is to reduce D_2 and increase D_3. The third option is to cut back the diffuser vanes/volute cutwater (larger D_3). With an increase of two percent to five percent of the ratio D_3/D_2, in most cases, the peak-to-peak amplitude of pressure pulsations can be reduced by 50 percent to 80 percent, which should drastically reduce or even eliminate the cavitation interaction, and, so, the consequential damage. An excessive enlargement $(D_3/D_2$ above 1.08 for diffuser) of the diameter D_3 is unnecessary and dangerous, as there is risk to cause flow separation in the diffuser, due to short length between the inlet throat and the exit throat. Also, excessive enlargement of D_3/D_2 for a single volute may be detrimental and produce a droop in the head curve near shutoff.

A second, but less effective, method to reduce the pressure fluctuations is to cut the impeller blades or the diffuser vanes (volute cutwater), either with a taper angle from the hub to the tip along an oblique line suggested by the damage pattern, or with a V-shape. This modification can be combined with the enlargement of the radial clearance using an average hub-to-tip diameter D_3, but verifying the maximum allowed clearance at the point of the largest diameter D_3.

A different approach to reducing the instantaneous cavitation at the impeller exit is to raise the instantaneous minimum pressure above the vapor pressure. This can be done by raising the suction pressure, if possible, or temporarily limiting the operating range to duties with higher margins $NPSH_A$-to-$NPSH_R$, usually at lower capacity/speed.

Corner vortex cavitation at impeller exit (CVCE). It is necessary to attenuate the intensity of each corner vortex (impeller blade on pressure side, vaned diffuser on the suction side), which is primarily determined by the local blade angle (too steep in relation to the local relative flow angle), and the fillet radius (too small, resulting in a sharp corner).

A quick fix is to increase the local fillet radius by filling those corners which show local cavitation damage with the same material as the base metal (careful welding is required). Then the new corner should be rounded and polished, also giving a smooth transition with the blade and the shroud. Impeller blade overfiling (lower blade angle) might give an additional positive contribution (corner vortex at impeller outlet).

When the corner vortex cavitation is mostly occurring at the diffuser inlet, a local reduction of the blade angle is, in principle, helpful to reduce the incidence angle, and so, cavitation intensity. Then the blade can be ground on the suction side (the visible side, facing the impeller outlet), if the original blade thickness is large enough to maintain adequate blade mechanical resistance. The method of locally reducing the blade angle just at the leading edge by filling material over the blade suction side, should not be used, even if it is claimed in the literature. The main reason is that the blade surface would have a hump with a sharp change in the blade angle, generating local flow separation and cavitation. A temporary effective action is also to raise the minimum continuous flow, since this type of corner cavitation at the impeller outlet usually occurs at part capacity.

• *Operating Range Limits*. Restrictions about the range of operating capacity, such as limits on minimum and/or maximum continuous flow should be always considered as temporary actions, as previously outlined, case by case.

• *Material*. With regard to material, a quick temporary fix can consider welding, coating or upgrade.

Welding. If the scope of a quick repair is to reconstruct the eroded areas and restore the original geometry, or eventually make local geometrical changes, then the filling material is usually the same as the base metal, provided that this has adequate resistance to cavitation attack. The preferred method of repair is by welding, which should be conducted very carefully, to minimize blade geometry distortion. Then both the opportunity of the repair and the method of application should be evaluated with an expert metallurgist and the pump designer.

Weld overlay, using a more resistant material, has been used to protect areas known to be particularly susceptible to cavitation, but has produced poor results. In the author's experience, severe cavitation damage with inducer for chemical plant service [14] could not be relieved even temporarily by welding over the inducer blades (the substrate material was a Cr.Ni. 26-5 S.S. similar to ASTM A890 Grade 5A) an overlay of Stellite® Grade 6. This material, from published laboratory data, appeared to have a resistance to cavitation damage one order of magnitude higher than the blade base material. But the stellite coating did not resist too long, because of its brittleness or the welding process. Moreover, the hot welding process generated a small distortion of the blade geometry, which made the cavitation erosion rate even worse.

Coatings. Commonly used materials with relatively poor cavitation resistance including cast iron and bronze, are either non-weldable or weldable with considerable difficulty. Various coatings have been developed and highly promoted to repair this damage. Moreover, in several cases, the repair is intended to increase the local resistance to cavitation damage by applying special coatings. Some efforts to prolong life of cast iron impellers from circular water pumps by using a widely publicized proprietary cavitation resistant coating were spectacularly unsuccessful. This experience, which was brought to the author's attention by a chemical plant user, is typical, and is largely attributable to the fact that coatings which are applied by spray, brush, or trowel form only a tenuous mechanical bond to the substrate, and so they are easily destroyed in service. In another case, with a very large bronze propeller from a circulating water pump, which was repaired by the plant operator using a "ceramic metal" coating, the coating rapidly fails and is now routinely replaced every two years, even though the cavitation intensity is not very severe.

In conclusion, the above examples are typical of the poor experience with coatings in pump applications and emphasize the need for materials with inherently better cavitation resistance.

Upgrade. The use of more resistant materials, which can represent a quick but more than temporary fix will be discussed later.

Medium-Long Term Solution at Marginal Cost

A cavitation damage troubleshooting strategy for identifying a solution, which at the same time produces technical results acceptable for a medium-long time period and requires only a marginal cost, should be based on the following criteria:

• Look at an effective control rather than at the total elimination of the cavitation and/or the suction recirculation mechanism. Then the true target for a practical engineering approach is the intensity of these phenomena and not their presence. It is accepted to live with these phenomena, by making them less severe, and especially lowering the damage rate under a level which permits a target life of several years, usually specified by the pump operator, and also eliminates the statistical risk of a forced plant outage.

• Act on those components of the pump and/or system that have a dominating role in the failure mode and can be upgraded at marginal cost, including both the first cost of installation and the operating cost of maintenance/inspection. In this regard, the goal

is to replace only one or a few parts, avoiding major replacements for the pump and the system, which would imply high cost.

With the state-of-the-art know how of cavitation and suction recirculation phenomena, including new theoretical insights, laboratory data and field indications, it appears that a modern trouble-shooting strategy for medium-long term solution at marginal cost should be developed with the following options: A) advanced hydraulic design; B) superior cavitation resistance materials; C) sound inspection/predictive maintenance plan.

In all of the cases, step A, which gives the most resolutive contribution, is a necessary and highly recommended action. In many cases, step B is recommended as an additional and highly positive action, while in specific cases, it might appear as a single but adequate change. In general, step C should be simply considered a complementary action for enhancing step A and/or step B.

Advanced Hydraulic Design

Impeller

Expected Improvements from New Design. A first engineering problem is defining "cavitation intensity." A widely accepted definition for "cavitation intensity" has not yet found a way to reflect the physics of the phenomenon and also identify a direct measurable parameter. However, from an engineering standpoint, the truly significant approach is to quantify the "cavitation intensity" with a cavitation erosion index, which permits the correlation of the erosion rate with a quantitative parameter, firstly, using experimental data and, secondly, applying theoretical analysis.

According to an early empirical formula [9] the cavitation erosion (damage) rate, expressed as weight loss-to-operating time ratio, is proportional to the cavitation bubble length, Lc (exponent: 2 to 4) and the impeller eye peripheral speed, U_{eye} (exponent: 3 to 4), neglecting other factors. A more recent research [17] indicated that in the cavitation regime "blade attached (sheet) cavitation" the erosion rate, expressed as ER = MDP/T (MDP = mean depth penetration, inches or mm, and T = operating time, hr) is proportional to cavitation bubble length, (exponent 2.6 to 2.83) and the net suction pressure above vapor tension (exponent 3.0). Also, other factors (impeller material, fluid properties) are explicitly included in the correlation.

Another recent formula [17] correlates the erosion rate with the "cavitation noise level" (CNL). With similar condition of flowrate (i.e., at the same fraction of the shockless capacity) and suction pressure (i.e., at the same dimensionless cavitation coefficient: $NPSH_A/(U_{eye}^2/2g)$), the cavitation bubble length, and also the cavitation noise level, are strictly related with the impeller geometry for given fluid properties.

Then, one of these parameters (L_c, CNL) can be used as "cavitation erosion index," to compare one impeller design to another for troubleshooting reasons or to develop more advanced impeller design criteria for cavitation minimization and impeller life extension.

A second engineering decision is "what level of cavitation intensity is acceptable." Here, the basic criterion is to determine the level of the damage rate, which permits achieving a reasonable impeller life under the expected operating conditions. This implies that the impeller designer and the plant operator should agree about:

The impeller life term. This is given by the maximum cumulative damage that can be allowed by the pump user. One criterion recommended by Energy Power Research Institute (EPRI) for boiler feed pump is that the maximum mean depth penetration be 3/4 of the blade thickness [34].

The impeller life period. This should be at the same time actually achievable and economically acceptable. Typically, for boiler feed pumps and other high energy pumps, a target life of 40,000 hr or

five years of cumulative service, is currently accepted in the pump industry [9, 13, 14, 34, 35]. For large size single stage pumps for which the price of an impeller is high, a target life longer than five years might be desirable.

Several cavitation field troubles have been successfully solved in recent years [14, 15], simply by replacing old style impellers with advanced design impellers which were aimed at reducing the cavitation intensity and, consequently, the erosion rate. The key hydraulic design criteria that have produced advanced impeller geometry, include:

• The selection of the shockless capacity in relation to the basic load, and also the expected operating range, was carefully considered and decided after obtaining from the plant operator quantitative indications, even if probabilistic, about the expected operating mode of the plant in terms of load *vs* operating hours.

• The entire geometry of the blades from the hub to the tip at the impeller inlet (angle, thickness and camberline) was optimized to reduce the cavitation bubble length. The design objective was to avoid high peak of local velocity close to the blade leading edge at off design, as suggested from both basic fluid dynamics and also quasi-three-dimensional flow analysis [36, 37].

• The $NPSH_A$-to-$NPSH_R$ ratio was marginally increased by lowering the $NPSH_R$ in the entire capacity range, if needed, to reduce the cavitation bubble length. Thus, the suction specific speed was increased, in certain cases, without detrimental effects on damage rate and rangeability, which is possible and proven with advanced impeller design criteria [14, 15, 36, 37].

• The onset of the suction recirculation was moved to lower capacity to reduce or even eliminate the risk of vortex cavitation within the operating range. This was made possible by the application of advanced design criteria for minimizing suction recirculation [4, 15, 16, 23, 26].

The effectiveness of these new design criteria for the impeller has been fully proven in the laboratory, in many cases, by using the criterion of the cavitation bubble length, which can be measured with special test rig allowing cavitation visualization [11, 36]. A clear example is shown in Figure 5, which corresponds to an actual field case of four large boiler feed pumps [14, 36, 38]. Heavy damage due to corner cavitation had perforated the impeller hub and attacked the shaft in 14,000 hr for one pump and 6,000 hr for
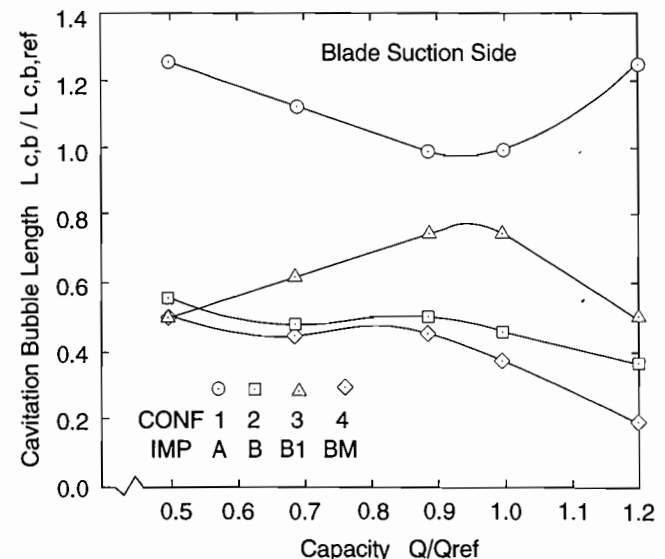

Figure 5. Cavitation Bubble Length at Plant $NPSH_A$ [14].

another. 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 (October 1986) in a time period of about six months (analysis-design-manufacturing-installation).

The cavitation bubble length (on blade suction side) $L_{c,b}$ from model tests is compared for the original impeller A (damaged impeller) and three new design variants (B, BM, B1) at various capacities and at plant $NPSH_A$. The cavity length for impeller A at BEP is made reference length of unity in the figure. For impeller B (Conf. 2) the bubble length $L_{c,b}$ was drastically reduced across the entire operating range. Moreover the suction recirculation point was also lowered (65 percent Q_{BEP}) with respect to impeller A (90 percent Q_{BEP}). Grinding the vane suction surface at inlet gave impeller BM, the best of the four.

The analysis of the shape of the cavitation bubble on the model furnished clues to the damage mechanism. On the original impeller A, the bubble pattern was triangular, with high cavitation at blade root and none at the tip, indicating a pronounced three-dimensional flow that caused high shear forces and intense local vortices. Vortex collapse then caused corner cavitation with high rate of damage. On impeller B, the cavitation-bubble sheet was rectangular (nearly equal length from hub to tip) pointing to a two-dimensional flow around the blade's leading edge. Thus, shear forces and local vortices at the blade root were expected to be drastically reduced, and so the corner cavitation damage practically eliminated.

Soft paint on the model blade, to obtain rapid damage evaluation, was damaged over a band parallel at the hub (no corner vortex). For the counter check, the blade was cut back at the hub, to produce a high positive incidence angle giving a triangular cavitation bubble sheet. The soft-paint erosion pattern was also triangular with worst erosion at the hub (corner cavitation). Various field inspections of impeller B at regular time intervals for all the four pump units, by means of borescope, and also direct visual checks (overhaul after 20,306 hr), have confirmed that the corner cavitation damage have been practically eliminated even after 30,000 hr (borescope inspection of July 1992).

It is then clear, that with the new design approach for the blade leading edge, the corner vortex cavitation at the impeller inlet (CVCI) can be fully controlled, and even completely eliminated as shown by laboratory research, and also confirmed by field data [14, 39]. It is worth noticing that statistical field indications point out that a very large fraction of the population of the existing high energy pumps (b.f. pumps, high speed pumps, water injection pumps) have the first stage impeller plagued with corner cavitation damage at the blade root-impeller hub.

It is even more important to underline that new design impellers permit reduction of the "blade attached (sheet) cavitation," and successfully solve field problems associated with such cavitation mechanism. In fact, the corresponding erosion rate can be lowered considerably, as shown in Figure 6 [14, 38, 39]. The theoretical erosion rate (ER) has been calculated by using the most recent correlation, based on cavitation bubble length [17]. For the impellers connected with the case mentioned above [36, 38], the predicted erosion rate for the blade suction side (SCSS) has been derived by using the visual cavity length $L_{c,b}$ from model tests and is shown in Figure 5. New designs, impeller B and BM, are estimated to have a rate at least one order of magnitude less than the original old design, impeller A.

Moreover, a newly published method [40] for predicting with probabilistic approach the expected impeller life, based on cavitation bubble length, has been used for the three new design impellers shown in Figure 5. Plant data have been used for the load spectrum along with other operating conditions. The calculations [38, 39] indicated that the probability of reaching the target
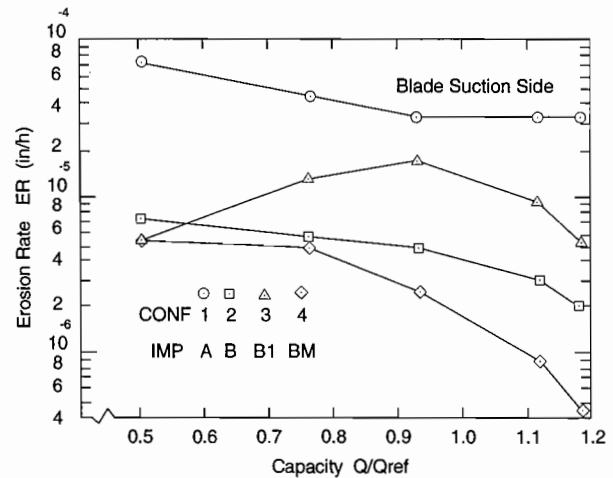

Figure 6. Erosion Rate on Blade Suction Side Vs Capacity [14].

impeller life of 40,000 hr with impeller B is about 65 percent. The theoretical probability raises to about 80 percent if a maximum erosion depth ED_{max} equal to the full blade thickness is allowed. Impeller BM has a theoretical probability of 90 percent of reaching 40,000 hr impeller life (ED_{max} = 75 percent of the blade thickness). However, at the first overhaul, impeller B was directly checked after 20,306 hr (September 1990), and had some cavitation erosion on the suction side of each blade (SCSS). The actual maximum erosion depth was lower than the predicted value, indicating that the probability of reaching the target life of 40,000 hr is close to 95 percent (ED_{max} = 75 percent of the blade thickness), while theory [40] has indicated a probability of 65 percent. From a recent inspection at 30,000 hr, the actual life expectancy probability is likely equal to 99 percent.

The key point is that with the present state-of-the-art, it is possible to control the "cavitation intensity" (cavity length) by impeller design. Also, the erosion rate can be predicted with some approximation, which is sufficient to estimate the impeller life expectancy, with a reasonably good probability of success.

Improvement Verification by Cavity Length

The assessment of the cavitation bubble length as index of the cavitation intensity is a key step of the solution strategy. The cavity length, L_c, can be estimated in the following way:

Actual damage length ($L_{c,ed}$ or $L_{c,el}$). The inspection of the erosion pattern in the impeller operating in the field permits to measure a cavity length as the distance from the blade leading edge of the point of maximum depth of damage ($L_{c,cd}$) or the point at the end of the eroded area ($L_{c,el}$) [14, 38, 39]. Each of these definitions, whichever is used, is very significant for assessing the relative improvement in terms of life extension with a modern design impeller (replacement), as compared to an old design impeller (source of trouble). A first field inspection of the new impeller is already indicative, if it is performed after an operating time of 10 percent to 20 percent of the total service time, which forced the replacement of the old design impeller.

Model visual cavity length ($L_{c,vis,m}$). In this case, cavitation visualization model tests with a special test rig are used [11, 36, 37]. The cavitation bubble length ($L_{c,vis,m}$) can be observed and recorded (videotape, manual sketches) by simulating various operating conditions (flowrate, speed, $NPSH_A$). Also, a soft paint technique (stencil ink) [37, 38] can be used to identify, with short endurance model tests, the "damage pattern" looking at the removal of the coating. A "damage cavity length" is obtained, which is even more significant of the visual bubble length, because it

indicates if the bubble collapse is occurring near the metal surface and truly producing damage [40]. However, the soft paint technique can give widely scattered results. The scale factors (speed and size) and the scaling criteria for meaningful cavitation similarity should be agreed between the designer and the user, following some published guidelines [34]. Cavitation visualization model tests are expensive and time consuming. They are recommended for special cases, when the pump has a critical economic impact on both the plant availability and also the maintenance cost.

Estimated visual cavity length ($L_{c,vis,est}$). The value of the cavitation bubble length can be interpolated from internal data base (proprietary to pump manufacturer), which includes many cavitation visualization tests performed for basic research and previous field troubles. In the data base, the visual cavity length, which has been obtained from many model tests with impellers using both old style design concepts and especially modern design approaches, is correlated with basic key design parameters and flow coefficient and cavitation coefficient. Internal appropriate plots with these correlations are used for estimating the value of cavity length for the specific design impeller (replacement) and even for impeller life prediction [14, 39] without the need of cavitation visualization model tests for the application which is being considered.

Predicted cavity length ($L_{c,cfd,pred}$). The cavitation bubble length can be, in principle, predicted using the modern tool of computational fluid dynamics (CFD). Computer programs are now available for analyzing the flow in the impeller with various levels of approximation [36, 37, 41, 42]. The value for the cavitation bubble length can be predicted ($L_{c,cfd,pred}$) at various flow conditions with reasonable accuracy, only for "blade attached (sheet) cavitation" on suction side (SCSS) and pressure side (SCPS), which corresponds to relatively steady flow in absence of flow separation (recirculation) [37]. To the author's knowledge, no experimental verification has been published for the computational method of predicting the cavity length with three dimensional flow analysis (a two-dimensional approach is by and large inaccurate). However, the method gives valid insights about the cavitation intensity reduction with a new design impeller as compared to the one (taken as baseline) causing the field trouble.

Improvement Verification by Cavity Noise Level

As mentioned before, according to recent research [17, 40] the cavitation noise level CNL in the high frequency range is also an indicator of the cavitation intensity and the consequent erosion rate. Then a troubleshooting methodology using the criterium of cavitation noise level can be successfully applied [20] as alternative to the approach based on cavity length. The positive and negative aspects, from scientific and practical standpoint, are not discussed herein. More field validation and less scattering of the data are needed before using the method for impeller life prediction on absolute basis (when data from baseline impeller is not available). However, the method can be powerful for field comparative evaluation between a replacement impeller (new design) and the old one causing the field trouble. Then it can be used for estimating the expected impeller life extension *vs* a baseline value.

Moreover, the CNL-method has the great advantage of easy periodic monitoring and damage control *vs* operating time with reference to a) field lower baseline level (at the field start of the new impeller), and b) field upper limit (at the field operating conditions of the old impeller before removal from the plant). On the contrary, the CNL criterion does not help characterize the specific cavitation failure mode and identify the location/aspects of the erosion pattern (visual observation is needed). Finally, a direct relation between the CNL parameter and the impeller geometry has not yet focused (as sufficiently done for cavity length). Therefore, this cavitation intensity criterion is not present-

ly (or only marginally) indicative for the optimization of the impeller geometry (a key design step in the solution strategy).

Application of Advanced Design Methods

Integrated hydraulic design methods are currently available which combine a) computational flow analysis, b) experimental flow visualization, c) basic fluid dynamic considerations and d) field experience with cavitation in high-energy pumps [36, 37, 38, 41]. The application of this design method has permitted reduction, as proven by experimental data, the cavitation intensity in a wide range of capacity not only in terms of damage but also pressure pulsations and noise level related with the cavity volume. These improvements have been obtained without penalty on the overall steady performance (head and efficiency) and are unrelated with the suction specific speed, which has been even increased in certain cases in comparison with old style impellers causing damage in narrow range of operations near BEP capacity.

Using this integrated design approach, a new impeller blade geometry has been developed, which includes several new features as listed below and shown in Figure 7 [41].

| *Feature*	*Benefit*
(a) Elliptical noses on blades	Local pressure-drop spike minimized
(b) Blade camber angle matched to analyzed flow	Cavity reduced or eliminated at BEP
(c) Biased-Wedge blade thickness development	Widens range of cavity-free flow rate
(d) Biased-Wedge blade designed to avoid increase of $NPSH_R$ due to loss of effective area between blades	Maintains $NPSH_R$ of conventional impeller designs
(e) Concave blade leading edge, blended forward	Removes hub-fillet cavitation and damage
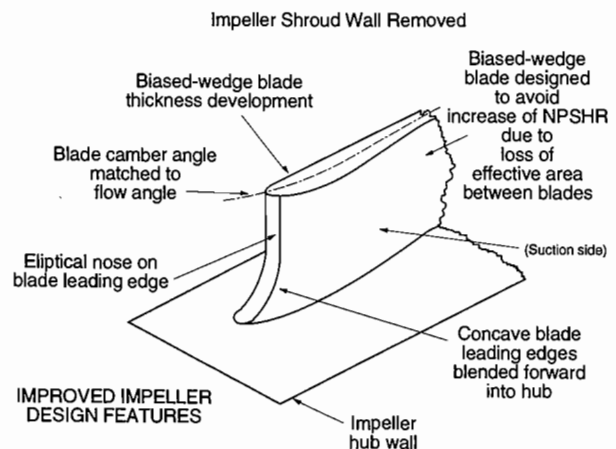
Figure 7. Design Features of Advanced Impeller [41] Patent Pending.

The combination of the above features permits minimizing/ eliminating blade attached (sheet) cavitation in a wide range of

duties (features a-b-c), and achieve, at the same time, relatively high suction specific speed (feature d) such as S = 11,000. Moreover, the corner cavitation is eliminated (feature e). However, it is important to underline that [41] "a small difference in a critical dimension or variation in shape for the above blade can result in a failed design and performance levels poorer than that obtained from conventional impellers." In fact, Sloteman, et al. [41], said "if misapplied, these features can cause degradation of performance, such as reduced $NPSH_A/NPSH$ three percent margin, increased pressure pulsation amplitudes, and separated flow along the hub streamlines at off-design."

A particular warning for the pump user concerns the application of 3-D-flow analysis computer codes to solve a cavitation problem. The computed flow field and corresponding cavitation pattern is very dependent on the solver algorithm used in the code and especially the boundary conditions. Several computer codes are available in the market, which in principle can be used by people with different background (designers, users, consultants, contractors, university experts). However, the use of these codes can be totally misleading, if the computational methodology for blade optimization is not fully supported by previous successful applications, which should be clearly proven by experimental results from laboratory research on models (cavitation visualization data) and especially field results with full speed/full scale pumps. Therefore, a strongly recommended criterion is to use thrust flow analysis methodology with proven field success. In this regard, the author is aware of only a few proven cases of integrated approach (computational plus experimental), which were produced exclusively by pump manufacturers involved in extensive investigation, combining numerical flow analysis along with experimental research about pump cavitation and suction recirculation [5, 18, 22, 23, 36, 37, 41, 43].

Suction Specific Speed

A basic general conclusion, which can be drawn from the entire body of the experimental indications (laboratory and field) and is also supported by the theoretical investigation about impeller geometry optimization, is that the suction specific speed should not be considered as a direct enlightening factor for developing an effective strategy of troubleshooting cavitation/suction recirculation field problems. Moreover, the suction specific speed is not a valid criterion for evaluating modern impeller designs optimized for low cavitation/suction recirculation intensity in a wide range of operations. In other words, approaches using maximum "reliable" or "allowed" values for suction specific speed have very limited significance, and even can be misleading for the purpose of troubleshooting and/or the evaluation of impeller design, in terms of reliability/rangeability. Such approaches largely ignore the results of the investigation in the last 20 years about cavitation/suction recirculation and impeller design optimization as well.

Inducer

The inducer is a special device purposely designed for very low cavitation requirements ($NPSH_R$) and high S values from 15,000 to 25,000 (or higher) for industrial application [43, 44].

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 N_s from 15, 000 to 20, 000 U.S.). Typically, the brake horsepower of the inducer is ranging from only five percent (at BEP capacity) up to 10 percent (at 50 percent BEP capacity) of the full pump horsepower.

Some published analysis and conclusions based on a statistical survey of centrifugal flow pumps [27] 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 horse-

power 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 [14, 15, 46]. 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 the 1930s (first inducer patent) to 1990s (cavitation visualization and acoustic measurements plus internal flow measurements by Laser Doppler Anemometry [47], has generated good insight about inducer design optimization. Moreover, a very large industrial population of inducers (a few thousand) 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 in and 14 in) plus high specific gravity (S.G. = 1.0 to 1.3), plus low flows duties (60 percent to 80 percent of BEP capacity), plus inadequate $NPSH_A/NPSH_R$ margins (from 1.15 to 1.30), plus inducers material with low resistance to cavitation attack. All these field cases were successfully fixed by using a new design inducer optimized for part capacity (inducer B) and a more resistant material (CA6NM, Duplex S.S.) [14, 15].

The key factors determining the inducer reliability as related to cavitation and/or suction recirculation are the same listed for the impeller in Table 1.

It should be noted that the preceding factors do not show any direct influence of the suction specific speed on the cavitation damage rate, and so on the inducer life. Then, in the case of inducers, the suction specific speed by itself is not a meaningful parameter for defining an effective troubleshooting strategy or evaluating a hydraulic design in terms of inducer reliability/rangeability.

Suction Casing

The solution strategy should include modifications of the geometry of the suction casing in presence of the failure modes of a) flow distortion cavitation (FDC), and b) flow imbalance cavitation (FIC), when these failures cannot be uniquely attributed to the suction piping geometry.

In case of FDC at the impeller inlet, the most direct indicator is the variation of the cavity length on a given blade at different angular positions (e.g., at 12/3/6/9 hour of a clock-face). The largest is the difference between $L_{c,max}$ and $L_{c,min}$, the strongest is the flow distortion [11]. Moreover, an additional character is that this difference increases with increasing capacity even at constant $NPSH_A$-to-$NPSH_R$ ratio. Clearly, a solution approach using the criterion of the cavity length will require a test rig, which has to be geometrically similar to the suction casing in the field (eventually at reduced size), for cavitation visualization model tests. The cavitation visualization test program should include the original design in the field and some variants, which are expected to diminish/eliminate the flow distortion. This solution method can be economically justified only for special cases.

Model air testing is an alternative experimental approach to evaluate the effectiveness of casing modifications on flow distortion. A wooden model of the suction casing only (half casing for double suction impeller is usually adopted), possibly with transparent window(s), is used for air testing. Eventually, a fully transparent model in acrylic resin can be selected. The geometry scale factor and the fluid dynamic similarity parameters (Reynolds and Mach numbers) should be decided case by case [33, 34, 48]. Some qualitative tests for flow visualization (nylon threads, lamp-

black, and oil coating technique) are recommended. However, the most indicative information about the degree and character of the flow distortion is obtained with quantitative measurements of the flow pattern at the location of the impeller eye, also referred as P-plane [33], by means of three-dimensional probes (five pressure taps). Then the complete map of total and static pressure is determined along with the velocity field (including all three velocity components). In addition, the inlet loss coefficient can be derived [33], which also helps to evaluate the change of $NPSH_A$ associated with each geometrical variant.

In certain situations, air model testing is quite adequate and even more enlightening than cavitation visualization model tests, if the main objective is to optimize the suction casing geometry through effective, but easy modifications. Air model testing can be performed at a fraction of the cost and especially the time, which would be required for model cavitation visualization test. Air testing is a useful approach when a modern design impeller, fully proven, is already available, which needs an undistorted flow pattern at inlet according to design assumptions.

A third experimental approach to evaluate the impact of suction casing on cavitation intensity is to use the method of cavitation noise level and, eventually, the measurements of suction pressure pulsations and also the vibration level [49]. All these parameters tend to increase with increasing flowrate in the presence of strong flow distortion at the impeller eye. Then, they can be used as nondirect indicators to judge the effectiveness of casing modifications, in comparison with the baseline trend given by the original design casing.

A computational fluid dynamic (CFD) approach using computer codes for fully three-dimensional flow analysis is presently principly feasible. However, no satisfactory validation of these approaches with experimental data is presently available. Moreover, the suction casing is truly geometrically complex and the numerical flow simulation is time consuming, because of the preparation of geometrical grid. A very fine mesh should be used to capture the local flow gradients that are usually very pronounced with flow distortion induced by suction casing [50]. The numerical flow analysis then is even more difficult and especially dependent on the assumptions about boundary conditions. At the present time, a CFD-approach for suction casing geometry optimization is still a qualitative and difficult tool.

With reference to the geometrical variants which can be effective in reducing/eliminating the flow distortion the list includes [11, 33, 48, 49, 50]: casing baffle (number and location of leading edge/trailing edge), casing wear ring contour, stop piece (angular location, orientation, shape) suction chamber shape (volute, semi-volute, semiconcentric), guide vanes (number, shape, location). The selection of one or more variants should be decided case by case by pump designers expert in hydraulics, after a careful evaluation of both the experimental data (model tests and/or field indications), and also the geometry of the suction casing (pattern drawing and actual pump) in conjunction with the suction piping configuration. The use of flow straighteners in the suction piping is a first recommendation, if elbows are present in the suction line as discussed above for quick fix solution.

In case of FIC, which can occur with pumps using a double suction impeller, the solution strategy should start from a careful analysis of the geometry of the suction casing (pattern and assembly drawing). Also the actual suction casing installed in the field should be closely inspected, looking for any geometrical difference (including roughness and any physical obstruction of the waterway) between the inboard and the outboard half. Moreover, the overall configuration of the suction piping should be closely analyzed and any geometrical feature (asymmetric components, elbows, etc) which is suspected to produce a flow distortion at the pump suction flange, should be eliminated or counterbalanced (flow straighteners, vaned elbows, etc.), as first step of the solution strategy.

Then, the second step is to correct the suction casing geometry focusing on the elimination of geometrical difference, i.e., the two halves (inboard and outboard) should have a specular geometry as much as possible. This may require redesigning some internal component [14, 39] and, in the extreme case, to completely reshape the entire flow passages (inboard and outboard) from the pump suction flange towards each impeller eye. The basic concept is to equalize the capacity between the inboard eye and the outboard eye of the impeller. Moreover, any flow distortion at each impeller eye should be reduced/eliminated, adopting one or more of geometry change(s) mentioned above (CFD).

The author is not aware of published experimental data, from laboratory and/or field, which can indicate in a clear way the occurrence of flow imbalance and its intensity, while the pump is in operation. It seems that a proven specific experimental quantitative indicator has not yet been identified. From fluid dynamic reasoning, it can be expected that flow imbalance could be detected by comparing the measurements of some significant physical parameter at the inboard side *vs* analogous data from the outboard side.

In the case of single stage pumps, the amplitude level and the frequency spectra of one or more of the following parameters should be measured at each side of the impeller: a) suction pressure pulsations (two pressure transducers flush mounted at symmetrical points close to the impeller eye, one in each half of the suction casing), b) cavitation noise level (two hydrophones at symmetrical points), and c) bearing vibrations, radial and especially axial, at the inboard and the outboard side of the pump. Also, failures associated with unexpected axial thrust problems with double suction impeller can be a symptom of flow imbalance. Other types of measurements (accelerometer on each side of the discharge casing wall, pressure taps near each impeller eye or at each side of the discharge casing) might be indicative, in case of strong flow imbalance. Also in the latter case, qualitative indications might be obtained by looking for differences in acoustic sound (directly detected with a stethoscope or more roughly a screwdriver in contact with the ear) from the inboard to the outboard side.

It is recommended to select one of the above quantitative methods when a flow imbalance is suspected to be the source of cavitation/recirculation failure with double suction impeller pumps (single stage). The measurements should be taken at various capacities and possibly at different suction pressure with the trouble pump, as baseline data. Then, the same type of measurements at similar operating conditions with modified suction casing and/or new suction piping can give a good idea of the effectiveness of the solution.

In the case of multistage pumps with double suction first stage impeller, a direct evaluation of the flow imbalance (in the original design and the modified one) is in reality not practical, although conceptually possible. One realistic way is to use one of the approaches suggested in the case of flow distortion with single suction impeller pump for comparing the adopted solution with the original pump configuration (baseline).

Vaned Diffuser/Volute

In the case where the failure mode has been identified as cavitation due to impeller-diffuser (volute) interaction (IDIC), a troubleshooting strategy that considers the hydraulic redesign of the vaned diffuser (volute) can be a very effective step for medium-long term solution.

The target for the new design is to reduce the peak-to-peak pressure pulsations in relation with the original configuration in the field. For this purpose, and in the light of experimental data

along with basic fluid dynamic reasoning, the most crucial design parameters for the diffuser are:

• *Impeller-diffuser (volute) radial clearance or "gap B."* In first approximation, the minimum allowable value for this parameter ($B = 1 - D_3/D_2$, D_2 = impeller blade outlet diameter, D_3 = diffuser blade inlet diameter) to prevent a strong flow interaction between the impeller and the diffuser is increasing with the head per stage H_s [34] assuming as baseline: H_s = 330 ft; $B \geq$ three percent vaned diffuser; and $B \geq$ six percent twin volute.

Other influences should be also considered, such as: combination of the number of vanes for the impeller and diffuser [51], specific speed, peripheral velocity at the impeller exit, impeller and diffuser design. The generic guidelines given above for the quick fix are also applicable in this case.

• *Vane number combination.* One selection criterium is based on the ratio (Z_{imp} = number of impeller vanes, Z_{dif} = number of diffuser vanes): $(m_1 \cdot Z_{imp})/(m_2 \cdot Z_{dif}) = 1$, with m_1 and m_2 = 1, 2, 3 .... The higher the values, m_1 and m_2, which satisfies the above relation, the lower is the amplitude of the pressure pulsations [34]. Other criteria may be applied, justified by the design/field experience of each pump manufacturer. Also, diffusers with a high number of vanes produce smaller peak for pressure pulsations.

• *Vane geometry at inlet (angle, camberline shape, thickness).* This should be optimized case by case, in order to determine the shockless flow (zero incidence) for the diffuser in relation with the basic duty and, also, the expected operating range. In principle, the design methodology used for optimizing the blade geometry at the impeller inlet in terms of smoothing local velocity gradients around the blade leading edge is also valid for the diffuser vanes.

• *Leading edge taper angle.* A leading edge with an angle *vs* the axial direction in a radial plane ("taper angle") should in principle attenuate the pressure pulsations. The sign of the taper angle ($D_{3tip} > D_{3hub}$ or $D_{3tip} < D_{3hub}$) should be decided case by case after analyzing the absolute flow angle at the impeller exit and, especially, the erosion pattern.

With reference to volute, the dominant parameters are similar to the ones for the diffuser, listed above. It should be expected that a double volute geometry produces smaller amplitude of the pressure pulsations than a single volute. Sometimes, a partially extended splitter may help to reduce pressure pulsations. The shape and the angle of the volute(s) cutwater is also important, which can be beneficially modified by an expert hydraulic designer with a partial change of the volute pattern core box. In certain cases, cavitation erosion has been found at the volute tongue for pump operating at high flowrates. The source of the problem was not pressure pulsations in strict sense, but rather a steady type cavitation (similar to "cavitation sheet") that was caused by the combination of three factors: a) mismatch between the absolute flow angle and the cutwater angle (high incidence angle at volute tongue), b) low pressure rise (head) produced by the impeller, and c) low suction pressure. In this case, an effective hydraulic design action is to modify the volute tongue geometry after a correct analysis of the incidence angle.

It should be emphasized that an effective solution strategy for the failure mode related to impeller-diffuser (volute) interaction (IDIC) should look at the impeller redesign as well. With focus on the impeller the key design parameters to lower the amplitude of pressure pulsations are:

• Gap B (see above)

• Vane number combination (see above)

• Vane geometry at the exit (angle, camberline shape, thickness). This should be optimized case by case with the basic

objective to reduce local velocity/pressure gradients and local vane loading in the expected range of duties.

• Blade trailing edge geometry includes taper angle and rake angle and other detailed geometrical features which can positively influence the fluid mechanism of vortex shedding leading to pressure pulsations.

As far as the solution quantitative methodology is concerned, a sound quantitative prediction of the amplitude of the pressure pulsations is not feasible with the present computational tools. Generic qualitative insights may be obtained with some kind of unsteady flow analysis. However, this approach is still at an early stage of research for pumps and appears more as a conceptual approach than an analysis tool. From an engineering standpoint, some useful design indications can be derived from a three-dimensional steady flow analysis of both the impeller and diffuser (CFD-approach). The scope would be to reduce the local vane loading at impeller exit and/or diffuser inlet in comparison with the original design giving the field trouble. Similar indications, but more sim-plistic, can be obtained by expert hydraulic designers applying less sophisticated methods based on streamline theory. It is important to emphasize that any engineering quantitative analysis needs the full detailed geometry or the impeller and diffuser (volute), which can be obtained only from corresponding pattern drawings or careful extensive measurements of the actual pump components.

A truly valid experimental indicator for evaluating the effectiveness of a new design variant, with respect to the original configuration (baseline), is given by the field measurements of the pressure pulsations (amplitude and frequency spectra). The pressure transducer(s) should be flush mounted close to the leading edge of the diffuser (volute) vanes. This may be feasible for single stage pumps, but impractical for the first stage of multistage pumps. In the latter case, only a periodic visual inspection can give a sound answer.

Obviously, cavitation visualization model tests can be planned in principle, but in practice they are rarely justified from an economical standpoint.

When the failure mode is corner vortex cavitation at the impeller exit (CVCE) (i.e., in the corner(s) between the blade surface and the shroud at the impeller outlet and/or diffuser/volute inlet), a major modification implying hydraulic redesign will concern the vaned diffuser (volute) and/or the impeller. Then the new design criteria should be aimed to reduce the corner secondary flows, which give localized vortices, and eventually local cavitation erosion, with or without flow separation. Also in this case, the hydraulic design objective is to smooth the local velocity/pressure gradients and reduce the local vane loading. Therefore, the above design indications for diffuser (volute) and impeller suggested for IDIC-failure mode apply to the CVCE-failure mode. An additional and specific design recommendation is to use large fillet radius between vane surface and shroud wall.

The author does not have suggestions about the solution quantitative methodology and the experimental indicator of the effectiveness of the design variant. The problem, which is not reported in literature as a frequent case, is very peculiar and has not been investigated independently from other cavitation modes. Thus specific indications and solution criteria are not available.

Superior Cavitation Resistance Material [52]

Need for Superior Material

The use of a material for the impeller/inducer (in most of the cases), and the vaned diffuser/volute (in particular cases) with a cavitation resistance remarkably higher than the material in the field, should also receive particular consideration for a medium-long term solution. The material upgrade can be combined with a

new hydraulic design in order to maximize the benefit of the solution in terms of impeller life and pump rangeability as well.

It is not in the scope of this discussion to discuss the criteria for selecting the traditional existing materials with good or high resistance to cavitation damage. In this regard, it is sufficient to refer to the ranking on the basis of the cavitation resistance of the most commonly used materials in the pump industry, as shown in various publications [53, 54].

It is important to understand the reasons for which a change of material can contribute in a substantial and exclusive way to the success of a medium-long term solution for cavitation/suction recirculation troubleshooting. In other words, it should be clear from the methodological point of view when and why a net specific benefit is obtained from the material upgrade. This should appear as an additional crucial factor by the nature, and also the amount of the impact, for drastic increase of the probability to achieve the expected life target, regardless of other positive changes (say new hydraulic design).

The cavitation damage problem is complex and often very difficult to handle exclusively from a design standpoint. There are various cavitation regimes, as described above. Depending on the operating conditions and design of an impeller, one cavitation mode can be dominant over the other. Moreover, for pumps operating with widely cycling load, several of these cavitation regimes can be encountered. Typically, the curve of the cavitation erosion rate *vs* capacity at constant speed/NPSH$_A$ has a peculiar V-shape, as shown in Figure 2. It is worth noting that the erosion rate increases by a factor of four from 100 percent to 120 percent of the shockless flow. If the impeller geometry is optimized to reduce erosion rate at high flow duties (SCPS) then cavitation damage gets worse at low flow (SCSS). Moreover, the mode of vortex cavitation from suction recirculation (VCSR), which occurs at part flows, is brought into the operating range of the pump. Vice versa, hydraulic optimization to minimize cavitation damage at part flows has negative impact at full load operations, and also would imply a design penalty on peak efficiency.

With new hydraulic design approaches, reductions in cavitation erosion rate exceeding an order of magnitude have been obtained (Figure 6). While this very significant improvement can be achieved at a given flowrate, or over a limited range, it cannot be obtained over the full range where many high energy pumps (e.g., boiler feed pumps) are often required to operate.

Note from Figure 6 that, at part load, the new design impeller B and BM have three to five times the erosion rate at shockless or near full load. Moreover, moving from one cavitation mode to another, the erosion rate can change by a factor of 10, as indicated by field evidence. Further, the prediction of the cavitation erosion rate [17] involves high uncertainty with a scatter bandwidth of one order of magnitude.

It is evident that in order to improve the pump rangeability and also minimize the risk of loss of plant availability, avoiding uneconomical design and operation safety margins, it should be desirable to reduce the cavitation erosion rate very drastically, say by one order of magnitude by improving at marginal cost one or more crucial factors beside the hydraulic design. One of these factors is certainly the material change that can become a truly beneficial action for medium-long term solution, technically and economically, if the cavitation resistance is enhanced by a factor of three to five times, at least.

Traditional High Resistance Materials

It has long been known that there is a relationship between hardness (or tensile strength) and cavitation resistance. Investigators [53] have demonstrated that damage rate, expressed as mean depth of penetration, varies inversely as the square of the Brinell hardness. This relationship was established for martensitic stain-

less steel (e.g., CA15, CA6NM) and has been extended to other materials. This principle of enhancing the cavitation resistance by looking at higher tensile strength/hardness material is currently the mostly applied criterium with cavitation troubleshooting strategies focused on material upgrading. However, the improvement in cavitation resistance through hardening a martensitic stainless steel is much less than needed, in order to obtain high life and wide rangeability with large confidence (high availability) in the actual field conditions (not in lab clean tests). Moreover, the high cavitation resistance obtained in this way has drawbacks of difficult castability, poor machinability, and weldability.

While material hardness is a key parameter affecting cavitation resistance, there is both field experience and some theoretical basis suggesting that other material properties are equally if not more important such as metallurgical structure and corrosion resistance of the alloy.

Cavitation erosion is known to be a high strain phenomenon. In fact, the macroscopic cavitation vapor pocket (usually called "cavitation bubble") breaks off at its trailing edge into many small vapor bubbles which are swept by the flow into regions of higher pressure where they collapse. A great many bubbles may form and collapse in a small area, producing many microjets of high kinetic energy and consequent impact pressure of 2000 to 3000 psi. This occurs in a time frame of microseconds and produces pressure pulses which impact upon the metal surface generating local strain/stresses even above the material tensile strength. As a consequence, nominally soft materials with the ability to work harden, such as austenitic stainless steels, can develop good cavitation resistance. It has been observed that the metallurgical mechanism which enhances the resistance to cavitation damage is a phase transformation of austenite into martensite induced by cavitation stresses. Typical rank of austenitic stainless steel is given by Larson [53].

Also, if corrosion is contributing to the metal damage, austenitic stainless steels are preferable to a martensitic one. Further, if repair by welding is planned for life extension, then austenitic stainless steels are recommended because of an easier weldability (with the same austenitic alloy overlay).

Among the existing alloys, Stellite 21® is a cobalt alloy with the highest cavitation resistance obtained through stress induced phase transformation. However, this alloy is very difficult to cast and nearly impossible to machine, while as welded overlay breaks off very rapidly.

Most Recent Superior Material (Hydroloy HQ913®)

Characteristics and Development

Recently, Simoneau [55] determined that high cavitation resistance is associated with minimal plastic deformation needed to transform the austenite phase to the martensite phase. Advanced microscopic investigation of the cavitation damage mechanism allowed the identification of a proper mixture of elements, which in austenitic stainless steel strongly enhances the high strain hardening process and, consequently, gives high cavitation resistance. The high strain hardening rate which is obtained allows the formation, on exposure to cavitation, of a smooth thin hardened surface layer. This works synergistically with a softer substrate to absorb cavitation energy in an efficient manner, producing a low erosion rate.

As a result of his investigations, Simoneau developed and patented a new cavitation resistant austenitic stainless alloy having cobalt and manganese substituted for nickel [56], which shows high work hardening rate similar to the cobalt base Stellite® alloys. This new alloy is known as Hydroloy HQ913®. Standard vibratory ultrasonic cavitation tests, ASTM G32-77, for material evaluation have indicated that this new experimental cobalt mod-

ified austenitic stainless is almost entirely transformed to martensite through superior strain hardening, induced by cavitation.

This results in cavitation resistance comparable to Stellite® 21 and eight to ten times better than type 308 stainless steel [55].

The original objective for developing this new alloy was to produce the material as a weld filler metal for the repair of cavitation damage in hydraulic turbines. Direct field experience, using both Stellite 21 and Hydroloy HQ913 for weld repairs in large hydraulic turbine runners, showed that the new stainless alloy outperforms Stellite at a fraction of the cost [57]. However, welding for repair or overlay requires accessibility which is possible only for very large impellers normally used in large hydraulic turbines and also high capacity pumps.

The existing standard materials for high energy pumps (boiler feed pumps), and also high speed pumps, are martensitic stainless steels typically CA-15 and CA6NM. As an upgrade, for enhanced resistance cavitation, some pump manufacturers offer 17-4PH. The relative cavitation resistance of these alloys is shown in Figure 8 [55]. Cavitation damage to impellers used in these pumps is not easily repaired by welding for various reasons. Often the damage is not accessible. Also, postweld heat treatment of martensitic stainless impellers to recover adequate mechanical properties after welding is not possible with finish machined castings as dimensional tolerances could not be maintained. On the other hand, attempts in the past by various manufacturers to extend life by a) surface hardening, b) cladding and c) weld coatings (Stellite or other) have proven entirely unsatisfactory. There continues to be a need for a castable and machinable material, with adequate mechanical properties, having cavitation resistance far better than that of conventional materials.
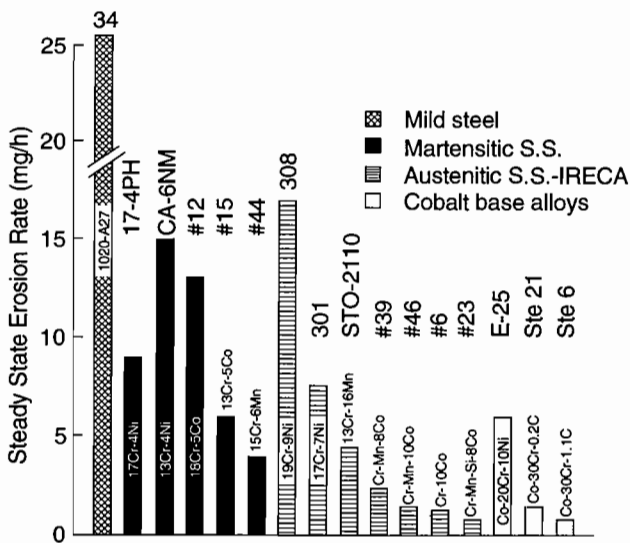

Figure 8. Comparison of Cavitation Resistance of Ferritic, Martensitic and Austenitic Stainless Steels and Cobalt Base Alloys. IRECA is Hydroloy [[5].

In Figure 8, the cavitation resistance of Hydroloy HQ913® is compared to that of CA-15 and other alloys commonly used for high energy pump impellers. The laboratory data shown in Figure 8 [55], which were originally published for cobalt manganese modified austenitic stainless, and since confirmed by other researchers [58], indicated that major improvements in cavitation resistance, approaching one order of magnitude, were possible. As previously discussed, this is the extent of improvement needed to complement the optimization of the hydraulic designs. Consequently, following suggestions from the author, materials research

efforts described by McCall, et al. [52], focused on the Hydroloy HQ913® composition as the most likely candidate to develop as a casting alloy. A joint primary objective of the foundry and a pump manufacturer was to produce boiler feed pump impellers in a cast version of the alloy, meeting all the requisite mechanical and quality requirements of this critical application, which called for field confirmation of the laboratory indications.

With the present foundry experience, techniques have been refined for producing hydroloy impeller castings in a variety of designs including single and double suction impellers of different sizes. A data base is being established covering the mechanical properties of Hydroloy HQ913® castings in the solution annealed condition. Typical values are given below:

	Hydroloy HQ913	CA6NM
Tensile strength	125 ksi	120 ksi
Yield strength	65 ksi	90 ksi
% Elongation	35	25
% Reduction	40	40

These properties are considered adequate for boiler feed pump impellers and are clearly comparable to those for standard materials typically used in this application, as shown above.

Castings can be upgraded by welding with appropriate procedures. Minor repairs to finish machined impellers have been made without problems [57]. Current experience from four different shops indicate that weldability is a very positive feature of this material.

Machining of impellers has not proven to be a problem, once the correct tooling was established, and appropriate machine tool feeds and speeds determined. Hydroloy castings do have a marked tendency to work harden and, therefore, do not machine as quickly as the materials they are intended to replace. However, the extra time for machining represents a marginal added cost.

First Field Applications

Initial field applications of the new cast alloy include several impellers for high energy boiler feed pumps in both Europe and North America. All are first stage impellers replacing either CA-15, CA6NM, or 17-4PH in applications where severe cavitation damage caused unacceptably short life in those standard materials.

The first application [52] in the pump industry of a hydroloy impeller dates from March 1991. It was installed as the first stage impeller in a high energy large boiler feed pump in Europe. The power station has four generating units of 660 MW each. Each generating unit uses a full capacity boiler feed pump with six stages, which absorbs up to 37800 hp at full load and maximum speed of 5200 rpm. All four pumps with an old style design impeller (A in Figure 6) had heavy cavitation damage [14, 36, 38]. A new design impeller (B in Figure 6) was installed in all four boiler feed pumps. Periodic inspections by borescope revealed that the impellers of three generating units showed blade suction erosion rate (SCSS) close to predictions and zero erosion on the blade pressure side (SCPS). The fourth impeller showed cavitation erosion about two times higher than predictions on the blade suction side, and also significant damage, unexpected, on the blade pressure side.

The operating conditions were similar with all four pumps operating at basic and full load. However, it was found that the peculiar pump was operating in a generation unit using a new and different water treatment with higher gas content. This produces a higher density of cavitation nuclei, which likely increases the cavitation intensity measured as the number of cavitation pressure pulses per second due to bubble collapse. However, the ultimate effect on damage rate is unknown.

A direct inspection of the impellers from two generating units after about 20,000 hr of cumulative service clearly indicated that the cavitation attack in the peculiar generating unit using special water treatment was much more severe than for the other units and unrelated to the impeller design. This appeared to suggest that the life target of 40,000 hr would be reached by three units, but not the remaining unit with the special water characteristics that were much desirable for the plant operator, in order to solve other process problems (rate of deposits and corrosion in the boiler). This field case was judged by the author as a suitably severe application for the initial application of hydroloy. An impeller of same design cast in Hydroloy HQ913® was placed in service in March 1991.

The first borescope inspection, which was conducted in March 1992 after 6960 hr of operation, showed [52] that the cavitation erosion rate on the blade suction side is significantly less for the hydroloy impeller, as compared to the CA6NM impeller (after 8797 hr). Borescope pictures suggest that the density of craters is nearly three times lower for the hydroloy impellers, although this indication should be considered quite preliminary. Clearly, the extension and depth of the erosion signs for the impeller made in hydroloy appears to be much lower. Small surface cavities may be superficial heteregeneous inclusions rather than cavitation erosion. Also, borecope inspection of the pressure side of the vane after 6960 hr indicated no sign of cavitation erosion with the impeller made of Hydroloy HQ913®. On the contrary, with the impeller made in CA6NM the cavitation damage was very evident after 8797 hr. The pump with a hydroloy impeller has been operating at basic and full load with the same time patterns for the key operating parameters, including the new method for water treatment, which led to the unexpected and unexplained damage rate of the impeller made of CA6NM.

At the present time, the impeller made of hydroloy has already accumulated about 12,000 hr of continuous operation at the most severe conditions of mechanical and thermal loads. Although it is too early to draw definitive conclusions from this first application, it seems that the field indications are quite encouraging and consistent with laboratory observations about the cavitation resistance and the mechanical properties of Hydroloy HQ913®.

The hydroloy impellers have also been supplied to several utilities in the United States, all as replacements for standard materials, which suffer early damage as the first stage impeller in boiler feed pumps. These pumps are typically operated at part load or other off-design conditions. The material change was combined with new design impeller in one case with the goal to eliminate the corner cavitation and also reach a target life of at least 60,000 hr (as suggested by the analysis) in a boiler feed pump that is expected to operate with swinging load. This impeller was planned to start operations in early 1993. In a second case, the material upgrade was combined with limited quick modifications of the impeller patterns. One impeller (out of three) had started running in late 1991. In a third case of relatively mild cavitation (current impeller life between 20,000 and 30,000 hr), the use of Hydroloy HQ913® alone without a change of the impeller design was considered sufficient for reaching a life expectancy of 40,000 hr. This impeller has initiated field operations in spring 1992.

Clearly, many other pump applications such as condensate extraction, cooling water, water injection, and others could also benefit from cavitation troubleshooting in terms of life extension from such advanced material. At the present time more consolidated field indications are needed to confirm the early field and laboratory data. Specific corrosion resistance data are required for a full evaluation. However, it appears from Figure 8 that a field response with hydroloy even five times lower than laboratory indications should be still very remarkable. Therefore, material upgrading with this material is certainly a strong option, at least for short-medium term cavitation troubleshooting solution (material change alone) and medium-long term solution, if combined with new design impeller/inducer or vaned diffuser, as appropriate.

Sound Inspection/Predictive Maintenance Plan [15]

According to statistics, metal damage is the most frequent truly harmful effect of cavitation and suction recirculation. Audible noise can be present, but not necessarily, as damaging pressure pulses are in the high range above 2.0 or 3.0 kHz. Pump users who intend to develop a maintenance strategy for preventing unexpected failures or, eventually, get a quick diagnosis from the pump designer, should make visual inspections and record the data at regular time intervals. The pump areas to be inspected for metal damage have been mentioned.

In many cases, it is sufficient to take close proximity photographs using a mirror for the back side of the vane. A borescope, however, could be used with a borescope window in the pump casing for a high energy pump operating at critical service (off-design duty, no standby pump). The inspection interval can vary from a few months (critical pumps) to one year or more (small pumps, light duty). A few inspections at close intervals, however, are recommended for the following situations:

• after the first startup;

• with a new operating mode, especially if the new duty is far from the pump bep capacity (at design diameter, not a trimmed one);

• after long trouble-free service (roughly 40,000 hrs of actual operation).

In particular cases, special hydrophones can be used. But the installation, and especially the analysis, of the noise signal (amplitude and frequency) require direct assistance of the pump designer. Also, vibration signature analysis can tell the pump designer whether critical recirculation or other severe unsteady flows are occurring due to reasons (e.g., bearing and seal life) other than metal erosion in the impeller. It is important to record pump operating parameters (speed, capacity or head, suction pressure, and fluid temperature) on a periodic basis with statistical significance for characterizing the operation mode (basic duty or variable load) in relation to the service time. The evaluation of both the field operating data and visual inspection documents should be done jointly by the pump users and the original pump manufacturer. The manufacturer has full records of both test data (labs, shop, field) and design information (calculations and drawings). In critical situations, the pump supplier can assist the user with effective diagnosis and solution, or provide technical assistance of experts from the corporate engineering staff.

Cavitation or recirculation problems, if detected at an early stage ("incipient failure"), can be solved effectively and quickly with the assistance of the pump designer without any loss of production. In fact, the damage is cumulative in nature, thus requiring time for reaching the last stage of rapid and uncontrollable deterioration. Then, the user ought to establish a predictive maintenance program with the assistance of the pump supplier before reaching the stage of trial-and-error troubleshooting [15].

The cost of solving cavitation/recirculation related problems can be as low as a fraction of the pump price (replacement of the impeller/inducer), plus limited engineering analysis/redesign at an early detection of incipient failure. But the cost can be up to hundreds of thousands of dollars due to loss of production and expensive engineering services (the last stage of failure).

The pump supplier can be helpful in suggesting quick fixes (geometric modifications, material upgrades, appropriate coating applications) or other actions. Thereafter, the pump supplier can identify a solution which meets both the technical requirements and the cost objectives of the pump operator.

With the present know-how of these phenomena, the most common approach, or at least the approach that gives the greatest reduction of the most harmful effects, is to install a new design impeller/inducer. And, eventually, the supplier and user can select new materials with superior cavitation resistance, which are now moving from the lab to the field applications [52, 55, 58].

Very Ultimate Solution (Fully Permanent)

An ultimate solution is intended to vastly reduce and fully eliminate the cavitation/suction recirculation damage in the operating range. Then the cost of the solution is not an objective, but rather an unavoidable price.

In certain field cases, a medium-long term solution strategy, including the technical options listed above, is not fully satisfactory for the pump user from a technical standpoint. The author's experience indicates that this can happen particularly but not exclusively in the following circumstances:

· The impeller (inducer) life as expected by the user is very high, corresponding to many years (ten or even more).

· The pump operating range which is expected by the user to be practically free from damage, is very wide from low capacities (30 percent to 40 percent of Q_{BEP}) up to high flowrates (120 percent to 130 percent of Q_{BEP}).

Moreover, one or more of the following aggravating factors is strongly contributing to the cavitation/recirculation damage:

· Peripheral velocity at the impeller eye U_e > 100 ft/s (rough limit).

· Inadequate NPSH$_A$-to-NPSH$_R$ ratio in the operating range (margin: 1.10 or below, roughly).

· High specific gravity and cold fluid (e.g., water).

The rank of the intensity cavitation factors (Table 1) clearly suggests as ultimate solution the following step(s): a) Reduce the pump rotational speed. This approach produces twofold positive effects by reducing U_{eye} and also increasing the ratio NPSH$_A$-to-NPSH$_R$ (smaller or zero cavity length). The pump must be replaced and a new motor or gear box is required; b) Increase the suction pressure (NPSH$_A$). The only effect in this case is to reduce or eliminate the various cavitation modes. A booster pump (installation or replacement) is the most viable way, as the increase of the static suction head (pressure and water level in deaerator, pump elevation) gives only marginal gain.

It is recommended to always verify that the aforementioned ultimate solutions also reduce undesired effects of suction recirculation on pump reliability and rangeability beside the failure mode of vortex cavitation (VCSR). In fact, option a), which implies a different pump geometry, does effect the intensity of suction recirculation (lower speed), but may not significantly change as needed the onset capacity in relation with the range of operating capacities. On the other hand, option b), which maintains the pump geometry, does not affect the onset and the intensity of the suction recirculation.

It is always opportune to consider the benefits of combining the above solutions, a) or b), with the options for medium-long term solutions discussed above, particularly a new impeller design and/ or a superior cavitation resistance material.

The preceding ultimate solutions impose major system modifications, which are costly and may require a long outage of the unit.

CONCLUSIONS

The most frequent cavitation modes which cause pump field troubles, mostly characterized by metal damage and life reduction, have been discussed. The various basic flow mechanisms (sheet cavitation, suction recirculation vortex, corner vortex, flow distortion, flow imbalance, impeller-diffuser/volute interaction) have been interrelated with six characteristic and frequent cavitation modes, the pump geometry, and the operating conditions.

Field experience and recent literature have been distilled and presented as a systematic troubleshooting methodology. This is organized in the three main logical stages of: failure analysis, diagnosis, and solution strategy.

At the level of failure analysis, the key objective should be to focus the problem by identifying the failure mode(s). For this purpose, some qualitative criteria have been given in order to screen truly indicative symptoms from irrelevant field observations. Moreover, special consideration must be given to a quantitative engineering analysis, as described, in order to rank the intensity of the most crucial parameter, related to: the pump design (geometry and material), the system configuration, and the operating conditions.

With reference to diagnosis, it has been shown how to isolate the physical flow mechanism(s) as the source of the failure mode(s) and also establish the actual severity of the field problem in view of an effective solution.

An effective solution strategy should always include three major aspects: technical options, time factor (urgency and life expectancy), and field implementation cost. The sequence of levels of intervention should be jointly discussed and agreed between the pump designer (or the troubleshooter) and the pump user, in order to evaluate the impact of the solution on the pump performance/reliability, cost of fix, and plant availability.

The urgent but temporary fixes, which have been indicated for each failure mode, include changes of: geometry of the pump and eventually the system, operating range limits, and pump material.

A troubleshooting strategy for medium-long term solution at marginal cost can be very successful by using the state-of-the-art know how of cavitation and suction recirculation phenomena. With this approach the main objective should be to reduce the intensity of the cavitation/suction recirculation under a level which is acceptable in terms of life expectancy with the intended plant operating mode, basic or swinging. This is made possible after remarkable progress in recent years in the areas of: impeller hydraulic design, life expectancy prediction, and field monitoring. New impeller design criteria, proven in the laboratory and also in the field, have permitted dramatic reduction of the cavitation erosion rate and also lower the suction recirculation effects. Moreover, a new material has been developed which has a superior cavitation resistance, according to laboratory data and very preliminary field indications, combined with the desired features of acceptable castability and machinability, plus easy weldability.

Ultimate solutions, intended for long impeller life target and/or extremely wide operating range, require major system changes (replacement of pump-motor, booster pump) having high impact on cost and plant outage.

The damage caused by cavitation and suction recirculation is cumulative in nature. If the problem is detected at an early stage ("incipient failure"), it can be solved effectively and quickly with the assistance of the pump designer without any production loss. The user should establish a sound inspection/predictive maintenance program with the assistance of the pump supplier.

A basic conclusion, which is supported by recent experimental indications (laboratory and field) along with new impeller design criteria for cavitation/suction recirculation minimization, is that current maximum values allowed by some pump specifications for the suction specific speed should be thoroughly revised and relaxed. Also, inducers with relatively high suction specific speed can be used effectively and reliably.

APPENDIX

Determination of the Shockless Capacity

The shockless capacity, Q_{sl} corresponds at the flow conditions of zero incidence angle at the impeller tip, i.e.,

$$i_{tip} = (\beta_{1b} - \beta_{1f})_{tip} = 0$$

where β_{1b} = blade angle at inlet (deg)

β_{1f} = flow angle at inlet (deg)

The blade angle is the angle between the tangent to the blade camberline at inlet and the tangential direction (as the velocity U). This angle is usually known only by the pump manufacturer from the design data and/or the impeller pattern drawing (proprietary). It can be directly measured (average value between pressure side and suction side of the blade) from cast impeller using special templates or three-dimensional coordinate measurement machine.

If it is assumed that in the impeller eye the flow is uniform (no distortion), one dimensional (no curvature effect), and fully axial (zero swirl component), then the shockless capacity is given by the following relation

$$Qsl = R^{3}_{eye} [1-(R_{h}/R_{eye})^{2}]f_{bb} \tan \beta_{1b\text{-}tip}$$

where = angular velocity

R_{eye} = impeller eye radius

R_{h} = impeller hub radius

f_{bb} = blade blockage factor

The blade blockage can be assumed between 1.05 and 1.2, depending on a blade geometry (inlet angle, thickness, number) and impeller eye diameter.

In case the shockless capacity can not be directly calculated, then a reasonable assumption is:

$$Qsl/Q_{BEP\text{-}des} = 1.1 \text{ to } 1.3$$

NOMENCLATURE

CNL	cavitation noise level
D	diameter
ER	erosion rate
H	total dynamic head
L_c	cavity length
$NPSH_A$	net positive suction head available
$NPSH_R$	net positive suction head required (three percent head drop)
N_s	specific speed (US unit)
Q	capacity
S	suction specific speed (US units)
U	peripheral speed
Z	number of vanes

Subscripts

BEP	best efficiency point
cfd	computational fluid dynamic
d	damage
des	design
dif	diffuser
est	estimated
eye	impeller eye
i	incipient
imp	impeller
m	model
mcf	minimum continuous flow
pred	predicted
rs	suction recirculation
sl	shockless
vis	visual

REFERENCES

1. Minami, S., Kawaguchi, K., and Homma, T., "Experimental Study on Cavitation in Centrifugal Pump Impeller," Bull. JSME, *3* (9), pp. 19-29 (September 1960).

2. Murakami, M. and Heya. N., "Swirling Flow in Suction Pipe of Centrifugal Pumps (1st - 3rd Report)," Bull. JSME, *9* (34), pp. 328-352 (September 1966).

3. Grianko, L.P. and Papira, A.N., "Determining Flow Parameters at the Impeller Inlet of a Multistage Centrifugal Pump during Hydraulic Stagnation," Lopastnie Nasosi (Bladed Pumps), Mashinostroenie - Leningrad Dept., pp. 50-66 (in Russian) (1975).

4. Janigro, A. and Schiavello, B., "Prerotation in Centrifugal Pumps. Design Criteria," Off-Design Performance of Pumps, LS 1978-3, Von Karman Institute, Rhode-Saint-Genese, Belgium (1978).

5. Schiavello, B. and Sen, M. "On the Prediction of the Reverse Flow Onset at the Centrifugal Pump Inlet," Performance Prediction of Centrifugal Pumps and Compressors, ASME 22nd Annual Fluids Engineering Conference, New Orleans Louisiana, pp. 261-272 (1980).

6. Fraser, W.H., "Recirculation in Centrifugal Pumps," Materials of Construction of Fluid Machinery and Their Relationship to Design and Performance, ASME Winter Annual Meeting, Washington D.C., U.S.A., pp. 65-86 (1961).

7. Gongwer, C.A., "A Theory of Cavitation Flow in Centrifugal-Pump Impellers," ASME Trans., *63* (1), pp. 29-40 (1941).

8. Grist, E., "Net Positive Suction Head Requirements for Avoidance of Unacceptable Cavitation-Erosion in Centrifugal Pumps," Cavitation, Fluid Machinery Group, Heriot-Watt University, Edinburgh, Scotland, ImechE Conference Publication CP13-1974, London, Paper C163-74 (1974).

9. Dernedde, R., and Stech, P.R., "Design of Feed Pump Hydraulic Components," Symposium Proc: Power Plant Feed Pumps - The State of the Art, EPRI CS-3158, Cherry Hill, New Jersey, pp. 11-23 (1982).

10. Florjancic, D., "Development and Design Requirements for Modern Boiler Feed Pumps," Symposium Proc: Power Plant Feed Pumps - The State of the Art, EPRI CS-3158 Cherry Hill, New Jersey, pp. 74-99 (1982).

11. Schiavello, B., "Visual Study of Cavitation-An Engineering Tool to Improve Pump Reliability," Proc. of EPRI 1st Intern. Conf. on: Improved Coal-Fired Power Plants, Palo Alto, California (1986).

12. McNulty, P.J. and Pearsall, I.S., "Cavitation Inception in Pumps," Trans. ASME, Journal of Fluids Engineering, *104*, pp. 99-104 (1982).

13. Cooper, P. and Antunes, F.F., "Cavitation in Boiler Feed Pumps," Symposium Proc. Power Plant Feed Pumps-The State of the Art, EPRI CS-3158, Cherry Hill, New Jersey, pp. 24-29 (1982).

14. Schiavello, B., "Cavitation and Recirculation Field Problems," Tutorial presented at Ninth International Pump Symposium, Turbomachinery Laboratory, Department of

Mechanical Engineering, Texas A&M University, College Station, Texas, unpublished (March 1992).

15. Schiavello, B., "Troubleshoot Centrifugal Pumps," Chemical Engineering Progress, pp. 35-42 (November 1992).

16. Schiavello, B., "Critical Flow Engendering Appearance of Recirculation at Centrifugal Pump Impeller Inlet: Determinant Phenomena, Detection Methods, Forecasting Criteria," La Houille Blanche, 2/3, pp. 139-158 (in French) (July 1982).

17. Gulich, J. and Pace, S., "Quantitative Prediction of Cavitation Erosion in Centrifugal Pumps," Proc. of the 13th IAHR Symposium on: Progress in Technology, Montreal, Canada (1986).

18. Okamura, T. and Miyashiro, H., "Cavitation in Centrifugal Pumps Operating at Low Capacities," Polyphase Flow in Turbomachinery, ASME Winter Annual Meeting, San Francisco, California, pp. 243-251, (1978).

19. Lalive, J., "Über den Wirkungsgrad von Zentrifugalpumpen," Schweizerische Bauzeitung, 19 (1936).

20. Florjancic, S.S., Clother, A.D., and Lopez, F.S., "A Case History: Improved Hydraulic Design Lowers Cavitation Erosion and Vibrations of a Water Transport Pump," Proceedings of the Tenth International Pump Users Symposium, The Turbomachinery Laboratory, The Texas A&M University System, College Station, Texas (March 1993).

21. Sloteman, D.P., Cooper, P., and Dussourd, J. L., "Control of Backflow at the Inlets of Centrifugal Pumps and Inducers," Proceedings of the First International Pump Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 9-22 (1984).

22. Okamura, T. and Matsunaga Y., "Intensity of Cavitation Erosion in Centrifugal Pumps Operating at Low Flow Rate," Cavitation in Hydraulic Structures and Turbomachinery, R.E.A. Arndt, Ed., ASME, pp. 63-70 (June 1985).

23. Schiavello, B. and Sen M., "Experimental Investigation on Centrifugal Pump Design Criteria for Minimum Reverse Flow Delivery," Improvements in Turbomachines and Related Techniques, H. Worthington European Technical Award 1979, V, Hoepli, Milan, pp. 191-239 (1981).

24. Sen, M., Breugelmans, F.A.E., and Schiavello, B., "Reverse Flow, Prerotation and Unsteady Flow in Centrifugal Pumps," Fluid Mechanics Silver Jubilee Conference, NEL, East Kilbride, Glasgow, U.K., Paper No. 3-1 (1979).

25. Breugelmans, F.A.E. and Sen, M., "Prerotation and Fluid Recirculation in the Suction Pipe of Centrifugal Pumps, " Proceedings of the Eleventh Annual Turbomachinery Symposium, Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M, University, College Station, Texas, pp. 165-180 (1982).

26. Schiavello, B., "The Impact of Fluid Dynamics Design on Minimum Flow," Proceedings of the Fifth International Pump Users Symposium, Panel Session on Specifying Minimum Flow, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 186-192 (1988).

27. Hallam, J.L., "Centrifugal Pumps: Which Suction Specific Speeds Are Acceptable?," Hydrocarbon Processing (April 1982).

28. Massey, I. C., "The Suction Instability Problem in Rotodynamic Pumps," Intern. Conf. on Pump and Turbine Design and Development, National Engineering Laboratory, East Kilbride, Glasgow, U.K., Paper 4-1 (1976).

29. Kasztejna, P.J., Heald C.C., and Cooper P., "Experimental Study of the Influence of Backflow Control on Pump Hydraulic-Mechanical Interaction," Proceedings of the Second International Pump Symposium, Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 33-40 (1985).

30. Akashi, K., Watanabe, H., and Koga, K." Flow Rate Measurements in Pipe Line with Many Bends," Mitsubushi Heavy Industries Technical Review, June 1978 (U.S. Patent N. 3, 840, 051 (1974).

31. Miller, D.S., "Internal Flow Symstems," BHRA Fluid Engineering, Cranfield, U.K., pp. 154-155 (1978).

32. Jones, J.L., Kosla, L.A., Cheng, D.Y., and Eley S.R., "Flow Stabilizers Solve Many Plant Piping Systems Problems That Result from Flow Separation in Elbows," EPRI Conf. on Plant Improvement, Miami, Florida, (1992).

33. Silvaggio, J.A., Jr., and Spring, H., "Air Model Testing to Determine Entrance Flow Fields," Proceedings of the First International Pump Symposium, Turbomachinery Laboratories, Texas A&M University, College Station, Texas, pp. 57-62 (1984).

34. Stech, P.R., "Feedwater Pump Technical Specifications Guidelines" EPRI-Draft, Palo Alto, California (1987).

35. Vlaming, D.J., "A Method for Estimating the Net Positive Suction Head Required by Centrifugal Pumps," ASME Paper No. 81 - WA/FE-32 (1981).

36. Schiavello, B., Arisawa, T., and Marenco, G., "Flow Visualization-A Design Tool to Improve Pump Cavitation Performance," Proc. of the 2nd Intern. Symposium on Transport Phenomena, Dynamics and Design of Rotating Machinery, Honolulu, Hawaii, pp. 484-449 (1988).

37. Cooper, P., Sloteman, D.P., Graf, E., and Vlaming, D.J., "Elimination of Cavitation-Related Instabilities and Damage in High-Energy Pump Impellers," Proceedings of the Eighth International Pump Users Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 3-19 (1991).

38. Schiavello, B., Marenco, G., and Arisawa, T., "Improvement of Cavitation Performance and Impeller Life in High Energy Boiler Feed Pumps," Proc. of the 14th IAHR Symposium on "Progress Within Large and High-Specific Energy Units," Trondheim, Norway, Paper J3 (1988).

39. Schiavello, B. and Prescott, M., "Field Cases Due to Various Cavitation Damage Mechanisms: Analysis and Solutions," Proc. of EPRI Symposium on Power Plant Pumps, Tampa, Florida (1991).

40. Gulich, J. and Rosch, A., "Cavitation Erosion in Centrifugal Pumps," Sulzer Technical Review 1, pp. 28-32 (1988).

41. Sloteman, D.P., Cooper, P., and Graf, E., "Design of High-Energy Pump Impellers to Avoid Cavitation Instabilities and Damage," Proc. of EPRI Symposium on Power Plant Pumps, Tampa, Florida, (1991).

42. Spring, H., "Affordable Quasi-Three Dimensional Inverse Design Method for PUmp Impellers," Proceedings of the Ninth International Pump Users Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 97-110 (1992).

43. Hergt, P., "Design Approach for Feed Pump Suction Impellers," Proc. of EPRI Symposium on: Power Plant Pumps, Tampa, Florida (1991).

44. Cooper, P., "Analysis of Single and Two-Phase Flows in Turbopump Inducers, " Trans. ASME, Journal of Engineering for Power, 89, pp. 577-588 (1967).

45. Janigro, A., and Ferrini, F., "Inducers: Optimizing Pump Performance in Industrial Applications," Pump World, 2 (1) (1976).

46. Palmateer, L., and Grout, P., "Pump Suction Specific Speeds Above 20,000," Hydrocarbon Processing (August 1984).

47. Boccazzi, A., et al., "Cavitation Behavior and Internal Flow Measurements by Laser Doppler Anemometry in a Low Solidity Inducer," Proc. of the XXth Hydraulics Open Days - "Hydraulic Machinery," Lyon, France (1989).

48. Pilarczyk, K. and Rusak, V., "Application of Air Model Testing in the Study of Inlet Flow in Pumps," Cavitation in Fluid Machinery, ASME Winter Annual Meeting, Chicago, Illinois, pp. 91-108 (1965).

49. Bolleter, U., Schwarz, D., Carney, B., and Gordon E.A., "Solution to Cavitation Induced Vibration Problems in Crude Oil Pipeline Pumps," Proceedings of the Eighth International Pump Users Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 21-27 (1991).

50. Enomoto, T., "Experimental Study on Prerotation Induced by Suction Volute Casings," Proc. of 12th IAHR Symposium on: Hydraulic Machinery in the Energy Related Industries, Stirling, U.K., Paper 6.1 (1984).

51. Makay, E., "Pump Troubleshooting: Diagnosis Methods and Case Studies," Short Course No. 2 - EPRI Symposium on: Power Plant Pumps, Tampa, Florida, (1991).

52. McCaul, C., Schiavello, B., and Dawson, R., "A New Highly Cavitation Resistant Casting Alloy and Its Application in Pumps," NACE-Corrosion '93, New Orleans, Louisiana (March 7-12, 1993).

53. Larson, J., "Pump Materials Selection," Short Course presented at the Third International Pump Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1986).

54. Hanson, C.M. and Hanson, I.L.H., "Cavitation Erosion," Friction Lubrication and Wear Technology, ASM Handbook, 18, pp. 214-220 (1992).

55. Simoneau, R. "A New Class of High Strain Hardening Austenitic Stainless Steels to Fight Cavitation Erosion," Proc. of the 13th IAHR Symposium on: Progress in Technology, Montreal, Canada, Paper No. 83 (1986).

56. Simoneau, R., "Co Containing Austenitic Stainless Steel with High Cavitation Erosion Resistance," U.S. Patent No. 4, 588, 440, (May 13, 1986).

57. Simoneau, R. and Mossoba, Y., "Field Experience with Ultra-High Cavitation Resistance Alloys in Francis Turbines," Proc. of the 14th IAHR Symposium on: Progress Within Large and High Specific Energy Units, Trondheim, Norway, Paper K1 (1988).

58. March, R., "Evaluating the Relative Resistance of Materials to Cavitation Erosion: A Comparison of Cavitating Jet Results and Vibratory Results," Proc. ASME Cavitation and Multi Phase Flow Forum, pp. 79-83 (1987).

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