DEMONSTRATION OF CAVITATION LIFE EXTENSION FOR SUCTION-STAGE IMPELLERS IN HIGH ENERGY PUMPS

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ABSTRACT

Cavitation occurring in high-energy pump impellers often results in material erosion and system instabilities. Excessive erosion rates compromise life and increase ownership costs through higher maintenance and reduced availability. Instabilities in the pumping system resulting from the presence of unsteady volumes of vapor result in pressure pulsations and unsteady flow delivery. This behavior is a legacy of the increase in pump size and energy level that were required to support larger and more critical processes. The increased requirements resulted in "scaled-up" impeller designs that were not suited to deal with the resulting cavitation behavior. During the late 1980s to 1990s numerous development efforts were undertaken to correct these design issues. There has been little follow-up of these efforts so that the degree of success is largely undocumented. This paper revisits two such redesign efforts for boiler feed pumps that presented two separate challenges. Significant increases in cavitation life are documented by using design approaches developed during the early 1990s. Also considered is the growing use of computational fluid dynamics for predicting cavitating behavior of critical suction impellers.

INTRODUCTION

The increased capacity of many new fossil fueled steam plants in the 1960s and 1970s required the application of larger boiler feed pumps. This created many feed pump-related design challenges. While many of the mechanical design elements were successfully addressed, the design evolution of suction impellers required to operate at the design net positive suction head available (NPSHA) did not keep pace.

The NPSHA is a measure of the total pressure above vapor pressure measured at a reference location near the pump. Usually this location is the suction flange. The measure is used to correlate pump behavior characteristics with cavitation. The actual NPSHA at the impeller eye is the total pressure above the vapor pressure at the flange, less any losses incurred between the suction flange and the impeller. If this value drops below the level of the vapor pressure of the fluid being pumped a change of phase from the liquid to gas occurs. This is cavitation. This is most likely to happen within the blade row of the suction impeller due to the higher velocities associated with the impeller.

A photo of cavitation occurring on a blade of a suction impeller is shown in Figure 1. In general, the higher the tip speed (radius multiplied by angular velocity) the larger the potential for reduction in pressure on the blade surface. This is why pumping machinery having suction impellers with high inlet tip speeds would require higher levels of NSPH to keep the local pressures above the vapor pressure of the fluid. The cavitation, once formed (refer to Figure 1), will return to the liquid phase when exposed to a pressure region above the vapor pressure of the fluid. For the cavitation vapor cavity shown in Figure 1, its length is limited by the increasing pressure field along the impeller blades. At the trailing edge of the cavity, a white, opaque region exists that is composed of smaller vapor bubbles that are shed from the main cavity. While different types of cavitation vapor formations will be discussed later, in general, the return of the smaller bubbles to the liquid phase is accompanied by a large pressure pulse (due to the volume of low-density gas being filled quickly by the high density liquid.) This change of phase can cause system instabilities (usually at relatively low frequencies). A more common problem is the potential for erosion in the region exposed to the large pressure pulse. This pressure pulse literally fatigues the material; causing failure in compression and pitting that is a common sight on many pump impellers (and other fluid components exposed to the collapse pulse). This adverse behavior is aggravated in higher energy machines that generate high pressures and operate at high speeds.

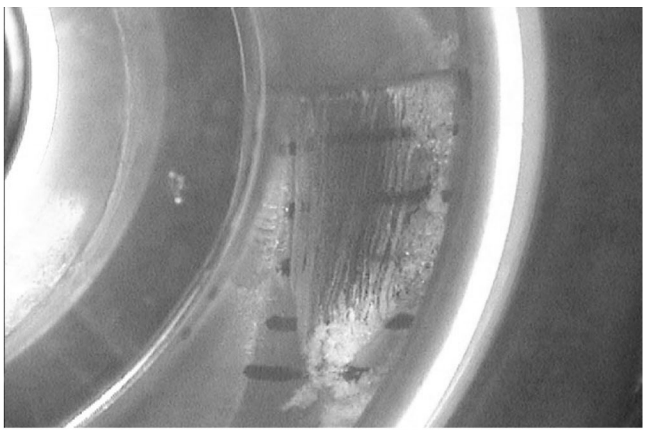

Figure 1. Typical Sheet Cavitation Pattern on High-Energy Feed Pump Impeller Blade.

A further problem arises if the pump is operated at flow rates less than the design flow. This causes an increase in the angle-of-attack between the oncoming flow and the blade that exacerbates the velocity related reduction in pressure within the blade row. More vapor volume and longer vapor cavity lengths are present with increased potential for damage or system instabilities. At much reduced flow rates (even with accompanying speed reductions if the pump is turbine driven) stall and separation of flow within the impeller occur that dramatically alter the flow velocities and can actually reverse the flow in the impeller passage. This brings with it increased potential for cavitation formation that will occur "off-the-blade." This type of cavitation will return to liquid phase in different regions of the impeller, greatly expanding the regions affected by cavitation erosion.

Higher NPSHA is one means of attenuating the adverse behavior that accompanies cavitation occurring at either high-flow or low-flow. Providing higher NPSH can be achieved by building a higher elevation feed tank (like a deaerator in a feed water system). It can also be achieved by adding a larger booster pump that operates at lower rotational speed (and hence can operate with a lower NPSHA) and boost the pressure to a level that is sufficient for the suction stage to perform as desired. Both approaches require costly system expenditures. For the large feed water systems designed in the 1970s and 1980s, the resultant systems were apparently designed to minimize NPSHA. In addition to providing marginally acceptable NPSH, large central station feed water pumps were called upon to cycle between the design plant load and a significantly lower load during off-peak periods (e.g., at night). This aggravated the off-the-blade cavitation patterns and damage potential.

This "marginal NPSH," coupled with cycling, resulted in suction impellers of that era operating with significant cavitation vapor (even though no significant head reduction was occurring in the stage). This operation often translated into high erosion rates and reduced impeller life. A result was failed impellers and in some cases system instabilities caused by the excessive cavitation vapor in the pump. Falling back on metallurgical upgrades to improve resistance to cavitation erosion was not always practical. Material development had reached a level where new material with increased cavitation resistance created compromises to mechanical properties (considered essential for some of these critical service machines). Clearly, new hydraulic design technology was required to meet the aftermarket challenge posed by the replacement of these impellers every one to two years.

During the late 1980s and early 1990s sophisticated analysis, experiment, and field development programs were undertaken to address this problem. Pump manufacturers applied basic fluid dynamic analysis tools and flow visualization testing in an effort to understand the cavitation problem with the goal being to improve suction impeller design and extend cavitation life to at least 40,000 hours (or ideally beyond). In many cases, these practices produced upgraded impeller designs that were expected to meet user expectations of improved impeller life with no performance degradation. These programs usually resulted in designs that only reduced the length and volume of cavitation vapor occurring in the suction stage impeller. Little "postprogram" documentation is available in the literature that addresses the efficacy of these efforts.

This paper deals with two such programs, conducted in the 1990s. While the underlying technology content of these programs has been discussed in previous papers, only now is definitive field data available that document the results of this work.

Two high-energy multistage boiler feed pumps are considered. Both had suction impeller stages that required replacement after less than two years service. Both impellers were evaluated for the existing operating profile of the plant. Both received state-of-the-art biased-wedge blade designs, modifications to the hub and shroud profiles as needed (keeping the external structure compatible for retrofit), and were manufactured using conventional techniques (precision, investment cast) from conventional materials (martensitic stainless steel). The improvement to the blade shape significantly lowered or eliminated the volume of cavitation vapor under normal operating conditions. It also reduced the vapor present at off-design flow rates that occur during cycling of the plant. The design approach is a product of the experimental testing (flow visualization) and fluid analysis practices of the time. The increase in operating life has now been documented.

In addition, the development of computational fluid dynamics (CFD), an advanced analytical tool that enables the designer to actually predict and visualize the cavitation behavior (or absence of such behavior) in the impeller, will be discussed. The use of CFD has evolved since the early 1990s and now has the demonstrated capability of analyzing not only the basic flow in the pump but also the two-phase behavior of all the critical elements of the pump stage. This includes the suction inlet that feeds flow to the suction stage impeller. This paper will show CFD results including two-phase behavior for suction stage impellers (both conventional and biased-wedge) that confirm the efficacy of the new designs of the 1990s. The use of this tool will also lead to improved designs that eliminate other adverse fluid behavior that exists in high-energy pumps.

BACKGROUND

Defining the Problem

The surfacing of excessive cavitation erosion resulting in reduced life of high-energy suction stages, particularly feed pumps, began in the late 1970s. A research institute report (Makay and Szamody, 1978) from that period is one document that traced reduced availability of central station power plants to outages of feed pumps with cavitation damage being one of the main causes. The picture of cavitation shown in Figure 1 (a sheet type of cavitation contained largely on the suction surface of the impeller blade) is typical of the amount of vapor present at normal operating condition. The erosion in this case would occur at the cavity closure with a resulting damage region well back from the leading edge. A photo of damage on the suction (or visible surface) of the blade is shown in Figure 2. The nature of the sheet cavity observed in Figure 1 indicates that this pump is operating near its best efficiency point, typically near what is termed "shockless entry." This term means that the relative flow direction is lined up with the blade angle with no angle of incidence between the incoming flow and the blade angle.

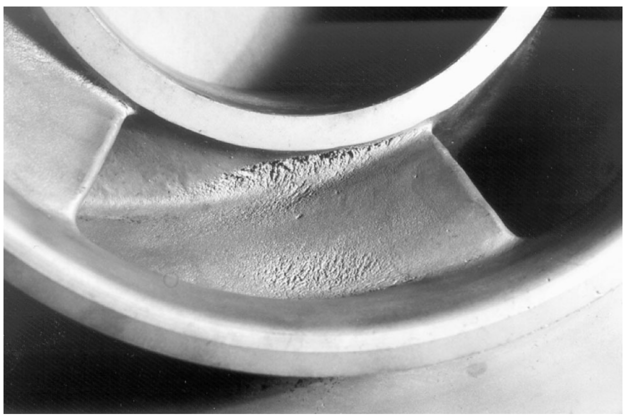

Figure 2. Cavitation Erosion on High-Energy Suction Stage Impeller. (Evidence of sheet cavitation and hub fillet cavitation as seen by the erosion patterns.)

The damage in the hub fillet (Figure 2) is a result of several fluid behaviors. It is a combination of low flow operation that causes separation off the leading edge due to a significant incidence angle between flow and blade (no longer "shockless entry"), the boundary layer along the hub surface and a stagnation point at or near the blade leading edge. This causes a horseshoe vortex fluid pattern with the streaming vortexes generating cavitation vapor that collapses in the hub fillet area downstream of the leading edge. Often this damage pattern is so deep that it penetrates the hub and attacks the shaft itself.

In some of the early discussion of the cavitation problem in literature, it was indicated that the amount of cavitation shown in Figure 1 is typical of a high-energy impeller under normal design conditions (Florjancic, 1980; Cooper and Antunes, 1982; Dernedde and Stech, 1982). For lower energy pumps, cavitation problems were defined by reductions in head generation, not erosion damage or system instabilities. These early high-energy pump investigations showed that significant damage rates (reducing life to less than two years operation) can occur at NPSHA one and a half to two times greater than the NPSH required to suppress 3 percent head drop of the stage ($NPSH_{3\%}$). In many fossil fired power plants of the time, this margin was typical.

Aggravating the reliability problems was the cycling of the power plants from base load to a minimum load when power demand was low. While the speed of the pumps (mostly turbine driven) dropped somewhat with the reduced plant output, the reduction in flow outpaced the speed change and often drove the operating point of the pump back to less than 50 percent of best efficiency point (BEP) flow rate. While operating at this condition, stall and separation in the impeller aggravate any cavitation and cavitation erosion and increases the levels of fluid induced vibration and loads. Operation in this low flow condition contributed to reduced availability of feed pumps (Makay and Szamody, 1978).

Recognizing the problems of low flow and cavitation activity in high-energy pumps in the 1980s began a long march toward designs, upgrades, and retrofits aimed at correcting the most severe problems. Specialists addressed many of the problems of high-energy pumps that included cavitation erosion as well as other adverse mechanical responses to challenging operating conditions (Makay and Barrett, 1988).

In 1989, a study of cavitation in high-energy feed pump impellers was published (Guelich, 1989). This study summarized the physics of cavitation formation and related it to the field problems associated with it. It addressed the damage mechanism that produced the erosion patterns seen in Figure 2. There was even discussion of identifying and quantifying cavitation erosion potential using an acoustic measurement approach. More importantly, it developed a sophisticated methodology for predicting impeller cavitation life as a function of the cavitation length (on either suction surface or pressure surface). While other authors had introduced the idea of cavity length and life, the Guelich (1989) work put together the fluid characteristics along with cavity length that made the model a practical means for determining the life of conventionally designed impellers. However, it remained for the designer to consider the multitude of design parameters available to him to produce a design that fit a desired operational profile. Tools like the cavitation life relationship contributed greatly to the design process.

Impeller Design Versus Suction Behavior Characteristics

The hydraulic design of high-energy pump impellers to obtain specific performance and behavioral characteristics requires the consideration of many parameters. A thorough discussion of these parameters and their effect on performance can be found in the Third Edition of the Pump Handbook (Karassik, et al., 2001). For the case of suction impellers, we will consider a subset of several characteristics: $NPSH_{3\%}$, $NPSH_{damage}$, and minimum flow. The behavioral characteristics are shown plotted on a flow (as a percentage of BEP) versus NPSH (defined as a cavitation number normalized to impeller tip speed velocity head) plot (Figure 3). While most pump original equipment manufacturers (OEMs) have developed their own calculation practices and prediction methods, the *Pump Handbook* (Karassik, et al., 2001) provides general methods for calculating these characteristics for conventional impeller geometry. These methods were collected from the literature and are discussed here, with those authors referenced.

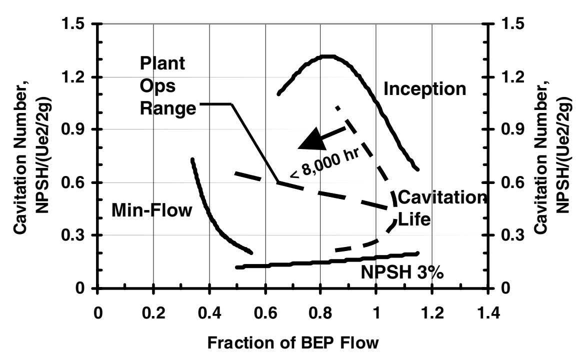

Figure 3. Typical Suction Behavior Characteristics for Centrifugal Pump Impeller. (This NPSH performance and impeller minimum flow characteristics were calculated in terms of flow coefficient and dimensionless NPSH. Design and operational parameters for the 12 × 18 CA suction stage impeller were used in these calculations. The plant operating range is also shown.)

$NPSH_{3\%}$ is the traditional measure of a stage suction performance. It is defined as the NPSH required to maintain 97 percent of the noncavitating head at a constant flow. For multistage pumps, the value is not indicative of a 3 percent drop of head for the entire pump. However, for our purposes, the $NPSH_{3\%}$ for the stage is important as it indicates the proximity the stage is to complete head breakdown, a condition that would be detrimental to the pump. A calculation method for conventionally designed impellers, by Gongwer (1941), is available to predict this $NPSH_{3\%}$ as a function of flow rate, inlet impeller diameter, and rotating speed (for zero inlet prewhirl):

$$Cavitation\ Number = NPSH_{3\%} / \left(U_e^2/2g\right) = \left(k_1 + k_2\right)\phi_e^2 + k_2 \quad (1)$$

where:

$$\phi_e = \left(Q / A_e\right) / \Omega r_e$$
$$and: \quad (2)$$
$$k_1 = 1.69\ and\ k_2 = .102$$

The resulting 3 percent NPSH line represents a lower boundary for performance. All high-energy pumps should be applied above this line by some margin. That margin is determined by the type of process and system sensitivity to suction transients.

$NPSH_{damage}$ is the NPSH required to provide a defined amount of life of the impeller. In Figure 3 the dashed line is shown for a calculated 8000 hours of life (approximately one year). Other values for different life expectancy, even infinite life, can be displayed. Life can be calculated from the penetration rate and an allowable thickness:

$$Life = \tau_{allowable} / MDPR \quad (3)$$

The penetration rate (MDPR) is a complex calculation. It requires knowledge of the fluid conditions, impeller metal properties, operating conditions, and the physical length of the cavity (refer to Figure 1) (Guelich, 1989). This is the relationship from Guelich as found in the *Pump Handbook* (Karassik, et al., 2001):

$$MDPR = \left[C \times \left(L_{cav}/10\right)^n \times \left(\tau_A - \phi_e^2\right) \times U_e^6 \times \rho_L^3 \times A\right] / \left[8 \times F_{mat} \times TS^2\right] \quad (4)$$

This relationship calculates an erosion rate for either:

- Suction surface sheet cavity: C = 8.28 E-06, n = 2.83
- Pressure surface sheet cavity: C = 396 E-06, n = 2.6

The cavitation number τ is defined as:

$$\tau = NPSH / \left(U_e^2 / 2g\right) \quad (5)$$

The cavity length (L_{cav}) can be determined from flow visualization, soft coating tests, and inspection of an actual damaged impeller or from a cavity length prediction method. An example of such a predictive scheme is described in the *Pump Handbook* (Karassik, et al., 2001). With these relationships, a prediction of cavitation life can be made for any design and operating condition (with the exception of the impeller operating in a suction recirculation condition). On Figure 3, the region within the dashed line represents operating conditions that would produce less than 8000 hour life. This region presents another boundary to be considered by the impeller designer. If the operating conditions of the plant fall within this boundary, less than the desired life would be expected. This specific relationship was not available to the designer of suction impeller during the "scale-up" of designs to meet the increased capacity of feed pumps.

Another characteristic that must be considered is the minimum flow of the impeller. The minimum flow is related to the effect that suction recirculation (resulting from stall and separation inside the impeller that occurs at flows well below the design flow rate) has on the mechanical integrity of the pump. Suction recirculation can cause increased vibration, noise, pressure pulsations, and surging flow delivery. The suction stage is a contributor to this behavior and if poorly designed can adversely affect the whole pump due to its exposure to low NPSH and possible cavitation behavior. Establishing a definitive minimum flow is difficult. Much depends on the structure of the surrounding components and their ability to withstand higher internal forces and vibration. Some definitions can be purely fluid based, as when flow begins to recirculate upstream, counter to the through-flow, per Fraser (1981). It can also be based on an experience basis for a particular class of pumps (Heald and Palgrave, 1985). The minimum flow calculation can be refined using several parameters and that can cover an array of design and operational variables (Gopalakrishnan, 1988). Here Gopalakrishnan (1988) uses the Fraser relationship and modifies it with a series of factors that vary from unity and zero. These factors account for NPSHA/$NPSH_{3\%}$ (K_3), flow rate and rotating speed (K_1), fluid properties (K_2), operational profile (K_4), and mechanical design margins—accounting for heavy-duty versus light-duty pump construction (K_5).

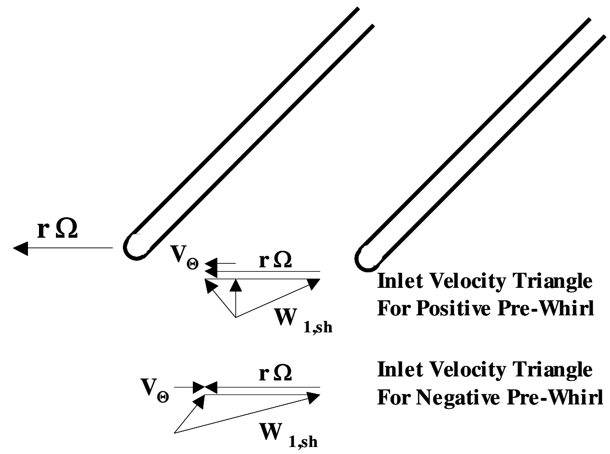

Figure 4. Comparison of Impeller Inlet Velocity Diagrams. (Suction impellers that are fed by right-angle inlets will see approach flows that have positive and negative prewhirl [or prerotation]. The result is a varying relative velocity [$W_{1,sh}$] that in turn influences the cavitation behavior of the stage. Also, variation of the though-flow velocity (not shown) due to the inlet will also influence cavitation behavior.)

Gopalakrishnan's minimum suction flow as shown in the *Pump Handbook* (Karassik, et al., 2001) follows:

$$Q_{min} = \pi \Omega r_e^3 \left(1 - D_S^2 / D_e^2\right) \times \tan\beta_1 \left[1 - .2091\left(\beta_1 - 9.5\right)^{0.4}\right] K_1 K_2 K_3 K_4 K_5 \quad (6)$$

Also included in the basic Fraser formulation is the inlet impeller blade angle $\beta 1$ and critical inlet diameters of the impeller. This relationship forms a boundary, as seen in Figure 3, which shows that at flow and NPSH combinations to the left of the line, vibration, pressure pulsations, and forces that are detrimental to the life of the pump will be encountered.

Also shown on Figure 3 is a line defining the suction surface, cavitation inception (a similar line at high-flow would describe the pressure, or hidden-surface, inception characteristic). This line in addition to the $NPSH_{damage}$ and $NPSH_{3\%}$ shows the range of "cavitation activity" occurring within the pump impeller. It is a range of which the impeller designer must be aware. The impeller designer must consider the application requirements (e.g., flow range and NPSH), the detailed design parameters of the impeller design, and the proximity of the resultant "boundary lines" to his application.

Another design influence is not directly shown on Figure 3. This influence is the effect the suction inlet that feeds the stage has on the flow entering the impeller. All conventional between-bearing pumps must feed flow to the impeller through a right-angle turn. This turn will distort the flow approaching the impeller to some degree. This distortion is a circumferential variation of mass flow and tangential velocity (also known as prewhirl or prerotation). This prewhirl is either in the direction of rotation of the impeller or counter to the direction of rotation. It can also vary in magnitude. A simplified example of the difference positive or negative prewhirl can make on the inlet velocity diagram (this diagram describes the kinematic relationship between the approach velocity relative to the rotating blade row) is shown in Figure 4.

The impact that these circumferential distortions (both in mass flow that affects V_e and tangential velocity that affects the velocity relative to the blade, $W_{1,sh}$) have on suction related behavior can be assessed by looking at the Gongwer (1941) relationship for $NPSH_{3\%}$. This relationship is rewritten from Equation (1) to more clearly show the influence of the components of the inlet velocity diagram.

$$\tau_{3\%} = NPSH_{3\%} / U_e^2 / 2g = k_1\left(V_e^2 / 2g\right) + k_2\left(W_{1,sh}^2 / 2g\right) \qquad (7)$$

While this relationship is written for the 3 percent NPSH requirement, it can also be used to understand the relationship of other suction related behavior such as cavitation inception, cavity length, or $NPSH_{damage}$. The importance of the degree of inlet distortion and the resulting variation of cavitation behavior within the impeller should be considered when designing or redesigning suction impellers.

Examining the input parameters for each of these behavior characteristics shows that they are interrelated. Changing one parameter, for example inlet diameter D_e, affects each behavior characteristic to some degree and alters the region of acceptable behavior on Figure 3. Many redesign efforts have focused on "readjusting" an original design to shift the behavior characteristics to better fit the application. However, in recent years, new design approaches have evolved that dramatically alter the traditional behavior characteristics. One such design approach is suction impeller blade design that significantly reduces the NPSH required to prevent cavitation formation. If an impeller that originally operated with cavitation severe enough to reduce life can be replaced with a design that eliminates the formation of the cavitation completely (and not significantly alter the other behavior characteristics) then the suction impeller would last the life of the pump. This paper outlines the results of such design efforts and demonstrates how collective and independent technology development evolved a design approach that has provided significant operational improvements for users who operate high-energy pumps.

Design Evolution

The trail of documented suction-impeller redesign begins in the late 1980s. Many cavitation research and development programs were undertaken by pump OEMs and a major research institute in the early 1980s. The technology from these programs fueled the redesign programs that appeared in the literature starting in the late 1980s.

Palgrave and Cooper (1986) were typical of the documented research and development (R&D) programs that made extensive use of flow visualization to characterize the different types of cavitation that exist in suction impellers. They also described how leading edge configuration can alter the inception of cavitation on the blade surface. In 1986, Aisawa and Schiavello (1986) described how flow visualization of several impeller designs was used to optimize a redesign of a suction stage impeller with the goal to improve its life. The examination of different inlet diameters (D_e) and blade angles led to a preferred design for their application. This work also investigated the effects that distorted flow from a right angle inlet had on cavitation formation. Numerous field fixes made to eroded suction impellers were performed without direct recourse to experimental or flow visualization studies. Makay and Barrett (1988) documented their approach that had been applied to cavitation damaged, suction impellers. Makay termed the design modification an "antistall hump," which consisted of applying weld to a region of the blade that had been damaged by sheet cavitation and then blending the weld material into the existing suction surface. This had the obvious advantage of filling in the damaged region of the impeller but also recontoured the suction surface of the blade and made it less susceptible to cavitation formation in the first place. However, without detailed analysis or testing it was later found that too much thickness or poor shaping could result in worse $NPSH_{3\%}$ performance (Cooper, et al., 1991a).

Improved suction impeller design efforts were not limited to boiler feed pumps. Two papers that dealt with large crude oil pipeline pumps appeared in the early 1990s. While cavitation damage in hydrocarbons is not as significant a problem as in feed pumps, the presence of cavitation vapor and its inherent unsteadiness can be problematic for pipeline operation. Bolleter, et al. (1991), showed that pipeline pumps could be designed with reduced cavity lengths using flow visualization. In addition, they applied the cavitation life equation published by Guelich (1989) using properties for crude oil. They further showed that modifications to the suction inlet could be made that eliminated the presence of a vortex cavitation pattern that streamed off the suction inlet stop piece. This vortex cavitation apparently added to the instability of the pump. In the same year, Cooper, et al. (1991a), documented a cavitation bubble-free pipeline impeller design. In order to demonstrate this, an improved impeller that used what would be later termed "biased-wedge" design was tested in an identical inlet against a conventionally designed impeller of equivalent performance. The conventional impeller was designed for 40,000 hours of cavitation life with an operating range from 50 percent to 135 percent of BEP flow. The biased-wedge impeller was found to be cavitation-free at the rated NPSH from 80 percent to 120 percent of BEP flow with significantly less cavitation vapor being present over the remainder of the range. The almost complete absence of vapor also reduced or eliminated cavitation-induced unsteadiness in the flow. This paper also demonstrated the benefit of performing single-phase, quasi-three-dimensional flow analysis to better understand the potential for two-phase activity in the impeller. This inviscid method of calculating velocity and pressure proved valuable in determining the final blade shape that was so effective at preventing cavitation formation.

Feed pump cavitation continued to be a topic during 1991. Schiavello and Prescott (1991) described improvements in life for impellers that were redesigned to better match incoming flow. These design changes also matched the current and/or future operating profile of the plant's feed water system. It demonstrated that often a suction impeller operates at different plant conditions than originally planned. Also in 1991, Cooper, et al. (1991b), published an extensive flow visualization study and redesign of a suction impeller used in the world's largest boiler feed pump (20 × 25 CA, four-stage, 63,000 hp). This redesign used biased-wedge style blade design. Cavitation inception occurred at NPSH levels corresponding to that available in the plant, thus reducing the presence of vapor at that flow as well as at reduced flow conditions. Also eliminated were cavitation related low-frequency instabilities (pressure pulsations and vibration) on the model pump. Another design feature that evolved from this program was the incorporation of a blade extension from the hub surface to the mean line of the blade. This shape was shown to discourage the formation of the horseshoe vortex flow pattern that generates cavitation along the hub fillet region of the impeller. This feature was incorporated into a patent (Cooper and Sloteman, 1993) that claimed the features of the biased-wedge design approach. A schematic showing this design approach is in Figure 5.

The biased-wedge concept was applied to a redesign of a suction stage for a 12 × 18 CA boiler feed pump (with a suction stage inlet tip speed of 281 ft/sec). This joint effort between the utility, the utility-sponsored agency, and the OEM was reported in Sloteman, et al. (1995). This work included detailed flow visualization studies

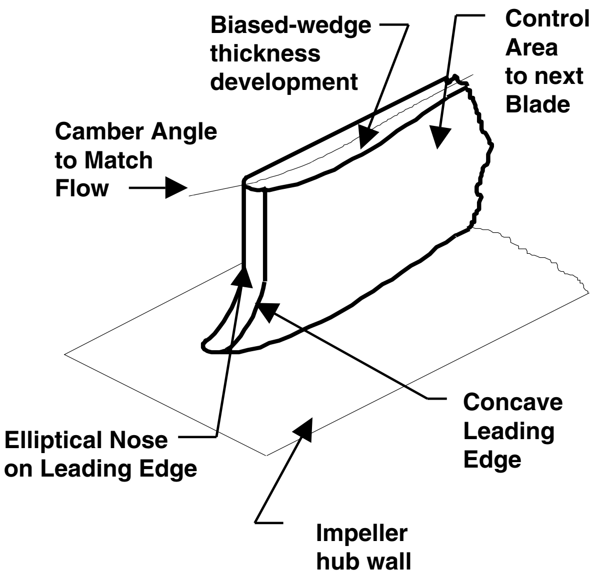

Figure 5. Improved Impeller Design Approach.

and investigation into effects of upstream piping on incoming flow to the suction inlet. The subsequent field results from this effort are discussed later in this paper.

Another study involving impeller leading edge contours and other design features by Hergt, et al. (1996), showed how inception NPSH levels could be altered. It also contained additional information regarding suction impeller design such as blade number, blade angle, and geometry effects on suction performance. Visser, et al. (1998), presented a redesign effort of a high-energy feed pump for a nuclear power plant. This experimental program used flow visualization to optimize the leading edge shape. Also, an adjustment of the suction inlet design was executed to further reduce the average cavitation length and through the use of the Guelich (1989) life equation predicted an increase of cavitation life. (A typical barrel type boiler feed pump is shown in Figure 6.)

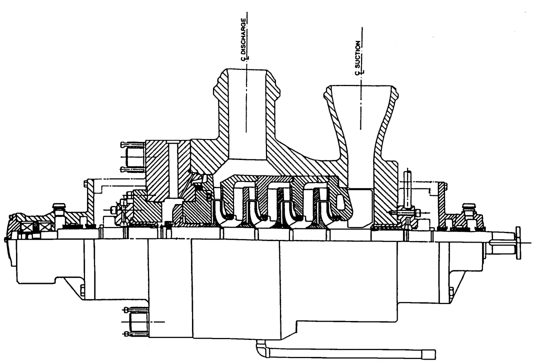

Figure 6. Typical Barrel Type Boiler Feed Pump. (This pump is typical of the construction used on the high-energy pumps described in this paper. Vaned diffusers [shown here] or volute type collectors can be used. Typically, a different first stage impeller design [sometimes a double-suction configuration] is used to provide acceptable suction performance levels.)

The references cited in the previous sections contain a wealth of information regarding cavitation and design of suction stage impellers for high-energy pumps. Almost all the technology has been the result of field observation, experimental programs, and fundamental, single-phase flow analysis (that ignores the viscous properties of the pumped fluid). Discussed later are new analysis tools that provide more accurate and rapid prediction of fluid behavior (both single and multiphase) for a given geometry. Missing from these references are examinations of the success of the redesign efforts. The measure of success should include improved availability of the plant (via extension of impeller life) with no adverse operational characteristics due to the redesign. What follows are field results from two redesign efforts, for two high-energy pumps, each with very different suction conditions.

IMPROVED CAVITATION LIFE FOR SUCTION IMPELLERS

12 × 18 CA Feed Pump—Suction Stage Redesign

The 12 × 18 CA is a half-capacity feed pump design that is steam turbine driven, consists of four-diffuser stages, and uses a single-suction first stage impeller. The pump is rated at 33,500 hp. Four pumps are installed at a fossil fuel power plant in Cumberland, Tennessee. The plant contains two 1300 MW units with two feed pumps supplying feed water to each unit.

Each unit uses three booster pumps to supply water to the feed pumps. The plant was commissioned in 1972. Table 1 summarizes the key design performance and actual operating conditions of the plant and suction stage impeller.

Table 1. Performance Characteristics, 12 × 18 CA Boiler Feed Stage.

Design	Pump rpm	5,800
	Suction flow, gpm	11,000
	Stage head, ft	2,965
	Suction Temp, °F	330
	Stage pressure rise, psi	1,160
	Inlet tip speed, ft/sec	281
	Stage specific speed	1,600
	Cavitation Number	.40
Maximum Capacity (Approximate)	Plant Load, MW	1,300
	Pump rpm	5,750
	Suction Temp, °F	325
	Suction flow, gpm	11,560
	Fraction of BEP flow	106%
	Inlet tip speed, ft/sec	279
	Cavitation Number	.45
Minimum Capacity (Current Min-Capacity 1,100MW)	Plant Load, MW	864
	Pump rpm	4,700
	Suction Temp, °F	311
	Suction flow, gpm	4,457
	Fraction of BEP flow	50%
	Inlet tip speed, ft/sec	228
	Cavitation Number	.66

As many fossil fueled power plants of its time, this plant was often called upon to cycle load daily between its maximum and minimum capacity. This range of loads (summarized in Table 1) forced pump operation between 106 percent and 50 percent of the BEP flow (however, an 18 percent reduction in pump speed occurred when operating at minimum capacity). Today, with a higher minimum plant load, the minimum pump flow is 80 percent of BEP flow. During the first 20 years of operation, suction impeller life of less than 18 months was typical. Typical damage patterns are shown in Figure 7. Suction surface damage from sheet cavitation and hub fillet damage from the horseshoe vortex is visible. Little pressure surface damage was reported indicating that at the maximum flow, the blade angle was well matched to the incoming flow. In addition, the design did not suffer from severe suction recirculation related cavitation damage (also appearing as damaged on the pressure side).

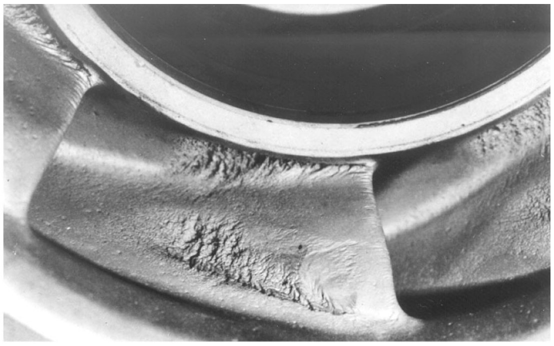

Figure 7. Cavitation Damage on Original 12 × 18 CA Suction Impeller.

Over the initial 18 years of plant operation, several impeller blade design and metallurgical changes were made to the impeller. These improvements brought the life into the 18 month range. It was not until an experimental program undertaken cooperatively between the utility and the OEM, focusing on applying new blade shape technology, that the cavitation life was extended to desirable levels.

The underlying concept of the blade design, the biased-wedge, had been analyzed using the best available computational tools and experimentally proven earlier (Cooper, et al., 1991b). Flow visualization testing was executed for this specific problem using a full-scale model of the suction stage (inlet, impeller, and diffuser), operating at reduced speeds. The exact operating conditions were modeled (per Table 1) using the pump scaling laws as found in the *Pump Handbook* (Karassik, et al., 2001). A detailed description of the experimental program was reported by Sloteman, et al. (1995). A sample of the flow visualization results is reproduced here for the maximum plant load condition (Figures 8 and 9).

The difference between the top view and bottom is the circumferential position of the impeller blade in the inlet (lines on the blade surface represent 0.5 inch spacing). With flow entering from the right of the pictures, positive prerotation exists in the inlet for the top view and negative prerotation for the bottom view (refer to Figure 4). Differences in cavity length are apparent. Applying Guelich's (1989) life equation to these two extremes results in a calculated life of almost 32,000 hours if the machine operated with uniform positive prerotation and only 6500 hours with negative prerotation. If an average cavity length of 1.38 inches is assumed the calculated life would be 12,864 hours. Given that life of this design ranges from 12,000 to 16,000 operating hours the calculated life seems to be of the right order of magnitude. It also indicates that damage from part load or low flow operation is not as severe as the higher speed, maximum load condition (with the exception of the hub fillet type damage).

The redesigned suction stage was subjected to the identical test program as the baseline. The photos in Figure 8 are taken at the identical operating conditions as those in Figure 9. The existence of positive and negative prerotation had no effect on cavitation formation. The new biased-wedge blade design could operate back to 80 percent of BEP flow without cavitation vapor. Below 80 percent evidence of suction recirculation generated vapor cavities (existing in the free stream) appeared. (The photos in Figure 8 and 9 are from a flow visualization test [Sloteman, et al., 1995] showing cavitation occurring on the suction impeller at two positions in the suction inlet. Positive prerotation is present in the top photo, negative prerotation dominates in the bottom photo. The impeller being tested in these photos is operating at reduced speed, but at the same flow coefficient and cavitation number as at the plant.)

The only noticeable difference in performance was a small increase in the $NPSH_{3\%}$. The conventionally designed impeller required a $\tau_{3\%}$ of .20 at BEP flow. The new impeller required .23. Using the cavitation number, τ of .41, resulted in a change in R value (defined as $\tau_{available}/\tau_{3\%}$) from 2.0 to 1.8. This ratio provides a measure of the ability of the suction stage to withstand suction transient events.

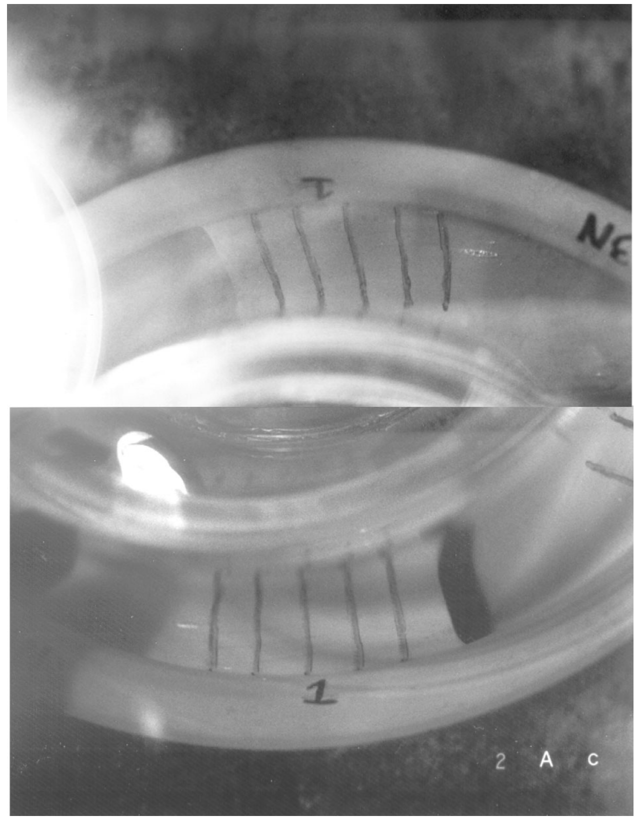

Figure 8. Photos Showing Elimination of Cavitation Vapor Using Biased-Wedge Blade Design.

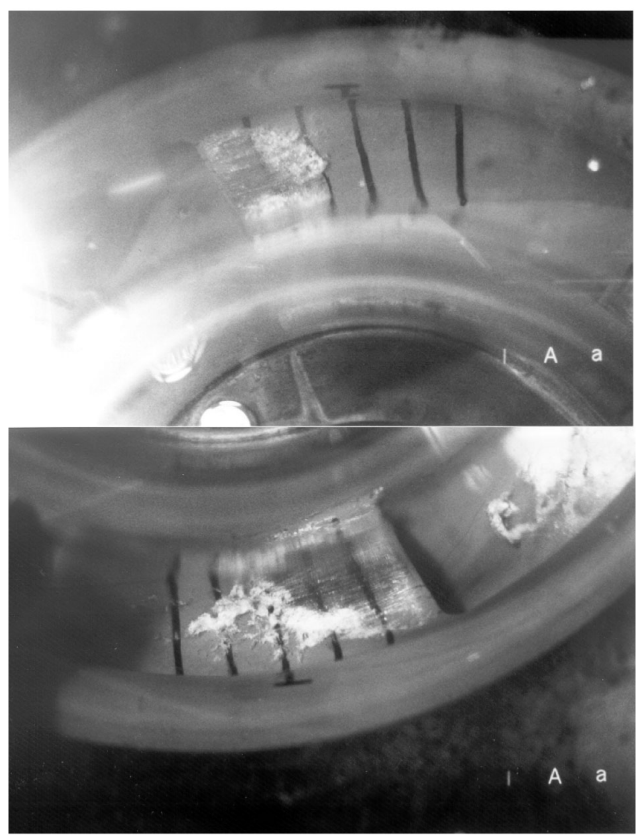

Figure 9. Cavitation Formations Occurring in Original 12 × 18 CA Impeller Design.

Installation in the field occurred in 1994 and, upon first inspection in 1998 (following 42 months operation), the impeller showed no signs of significant cavitation erosion (Figure 10). Operation continues though its second 42 month operational cycle. A seven year life is expected. All feed pumps in the plant have the new impeller installed.

Figure 10. Improved Impeller after Four Years' Service.

14 BFI Four Stage Feed Pump—
Double-Suction First Stage Redesign

The 14 BFI is a steam turbine driven, full-capacity feed pump. It consists of a first stage, double suction impeller working in a dual volute collector. The three following intermediate stages utilize vaned diffusers. The pump, rated at 11,000 hp, is installed at a fossil fuel power station in Baldwin, Louisiana. The plant was commissioned in 1970 and today cycles from approximately 250 MW to 75 MW on a daily basis. Table 2 summarizes the key design performance and actual operating conditions of the plant and suction stage impeller.

The suction piping on this pump is routed from the unit's deaerator. The 17 feet of suction piping proceeding the pump includes two 90 degree elbows, two 45 degree bends, a suction isolation valve, a suction strainer and a reducer. This configuration does not coincide with the recommended practices of at least six to 10 pipe diameters of straight run pipe preceding the pump.

The 14 BFI has been plagued its entire life by first stage impeller cavitation issues. On record, there have been at least 16 first stage impeller replacements from 1971 to 1997. Pump maintenance inspections usually revealed one or more "missing inlet vanes" after six to 18 months of operation. For this reason, the first stage impeller has historically been considered the "weakest link" in terms of feed pump reliability. The pictures in Figure 11 show extensive damage after less than one year of operation. Visible is severe pressure side damage with additional erosion evident on the suction surface and at the hub fillet.

Table 2. Performance Characteristics, 14 BFI Boiler Feed Stage.

Design-Rated	Pump rpm	5,700
	Suction flow, gpm	5,800
	Stage head, ft	1,460
	Suction Temp, °F	367
	Stage pressure rise, psi	556
	Inlet tip speed, ft/sec	230
	Stage specific speed	1,300
	Cavitation Number	.20
Base Loaded	Plant Load, MW	305
	Pump rpm	5,364
	Suction Temp, °F	355
	Suction flow, gpm	4,858
	Fraction of BEP flow	83%
	Inlet tip speed, ft/sec	205
	Cavitation Number	.18
Low Load	Plant Load, MW	75
	Pump rpm	4,050
	Suction Temp, °F	281
	Suction flow, gpm	1,116
	Fraction of BEP flow	25%
	Inlet tip speed, ft/sec	155
	Cavitation Number	.31

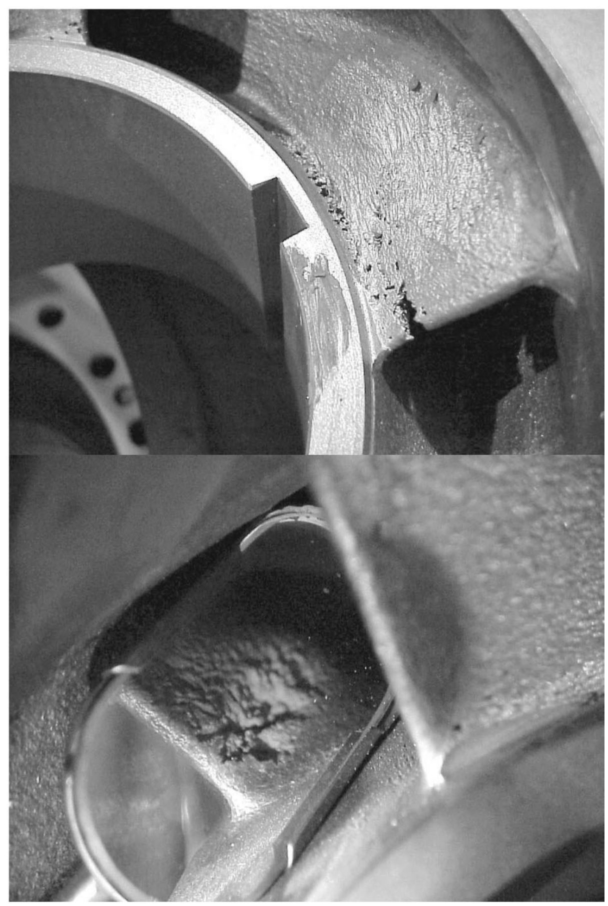

Figure 11. Suction and Pressure Surface Cavitation Erosion. (This pair of photos show slight cavitation erosion on suction surface (top photo) and severe pressure surface as viewed in a mirror (bottom photo) accumulated in less than one year's service.)

In order to understand the interaction of the impeller behavior with the plant operating conditions, a map similar to Figure 3 is constructed using the information from Table 2. Here, the minimum flow, 3 percent NPSH and the plant operating condition are plotted in nondimensional form. The calculation of expected cavitation life is not practical since the pump is operating to the left of the minimum flow line. The life Equation (4) is predicated on sheet type cavitation and not on the off-blade type cavitation associated with suction recirculation. In addition, the pump operates with an R-value ($\tau_{available}/\tau_{3\%}$) of 1.6 and to reduce this value might compromise the ability to handle suction transient conditions.

The designer, when confronted with these facts, searches for the best available design approach to attenuate the excessive damage rate while not negatively impacting performance. For example, reducing the inlet diameter (D_e) would push the minimum flow line to the left and attenuate the adverse effects of recirculation. But from Equation (1) we see that this will also increase the $NPSH_{3\%}$ value and reduce the R-value, thus affecting the pumps ability to handle suction transients.

An alternative was arrived at in the form of the biased-wedge blade design. Since no direct experimental program was associated with this specific impeller, results of earlier efforts were used. Testing that was described by Cooper, et al. (1991b), concluded that, for impellers with similar nondimensional performance and configuration to the 14 BFI, low flow suction recirculation caused cavitation vapor was reduced through the use of this blade shape. Photos from a model impeller operating at a cavitation number of .40 and Q/Q_{bep} of 35 percent are shown in Figures 12 and 13. Even though low-flow cavitation like this is unsteady, the experimenters observed that the biased-wedge style blade shape produces noticeably less vapor in the free-stream and near the vane surfaces. It was their conclusion that this would translate to less erosion potential. In addition, the forward sweep of the blade length at the hub (intended to mitigate the horseshoe vortex pattern that causes hub-fillet damage) was seen as a feature to consider due to the fillet damage.

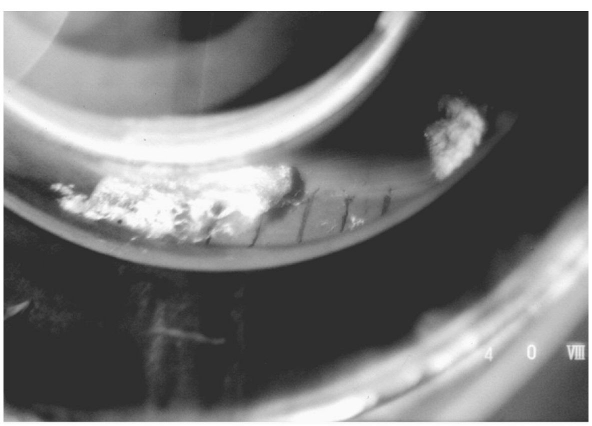

Figure 12. Cavitation Vapor Activity on Conventionally Designed Impeller at 35 Percent BEP Flow Rate. (This shows vapor cavities visible upstream and between blades. This is due to the interaction of the high-velocity backflow emanating from the impeller and the incoming flow. The resulting cavity collapse can occur on the pressure surface causing severe pitting.)

A suction impeller redesign was initiated in 1999 to address the pump's chronic cavitation issues. The goal of this effort was to reduce the cavitation on the inlet blades while maintaining the pump performance (suction and total dynamic head). The engineering approach for this process included evaluating the historical impeller designs with damage (Figure 11), evaluating the plant operating conditions (Figure 14), incorporating the best available technology for impeller design (Figure 5), and manufacturing techniques. The impeller was produced as an investment

Figure 13. Cavitation Vapor on Biased-Wedge Impeller Blade Operating at 35 Percent BEP Flow Rate. (This shows the impeller is similar in performance to the impeller shown in Figure 12, and uses biased-wedge shaped blades.)

casting with a new bias-wedge blade shape. The impeller material was upgraded to a proprietary derivative of CA15 (12 percent chrome, martensitic stainless steel), which has a small improvement in cavitation resistance as compared to the previous17-4PH material. Another change was an optimized meridional area distribution (hub-shroud profile) for smooth transition of area from the inlet to the exit. The predicted performance of the final design was checked and compared with models of similar geometry as well as evaluated using hydraulic calculations similar to those discussed earlier to assure satisfactory performance.

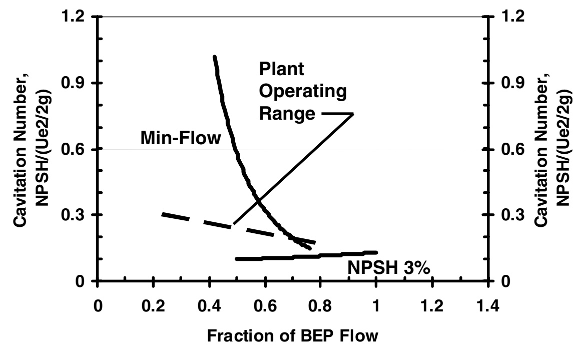

Figure 14. Suction Behavior for 14 BFI Suction Stage. (This map shows calculated performance for $NPSH_{3\%}$ and minimum flow. The plant operating range is also shown. The pump operates at an R-value of 1.6. In addition, the pump runs below the calculated minimum flow for all plant operating conditions.)

The redesigned impeller after two years of operation is shown in Figure 15. Only some slight suction surface and hub fillet erosion is visible. Some roughness was reported on the pressure side. This was in keeping with the expectation that suction recirculation would still be occurring but that the damage potential of the vapor would be reduced with the new blade shape. The significant result was that life was more than doubled for this pump operating over the same operational map and similar impeller material.

COMPUTATIONAL DESIGN TOOLS AND THEIR FUTURE APPLICATION

CFD for Performance Prediction

The 1990s saw a revolution in the way complex fluid flows could be modeled and analyzed. As computer hardware increased processing speed and decreased in cost, solving the Navier-Stokes equations, the fundamental relationships that define fluid behavior became practical. The Navier-Stokes equations are used to define

Figure 15. Improved Suction Impeller for Fossil Fuel Plant in Louisiana.

the viscous, three-dimensional behavior of any flow. The turbulence characteristics were included through mathematical modeling of the Reynolds stress terms. Rotating passages are analyzed with inclusion of Coriolis and centrifugal terms. The result was a computational fluid dynamics analysis tool that is similar to the familiar finite element analysis (FEA) for analysis of structures (although requiring a more complex solver). Cooper and Graf (1994) described early analysis of pump components using a commercially available solver that was described as a finite volume code. It used a generalized turbulent flow model (needed to avoid complex and calculation intensive detailed modeling of boundary layer behavior) to simulate turbulence and separation characteristics. Reasonable performance prediction was achieved and even more important, critical flow field details were revealed in terms of velocity distributions, pressure field maps, and separation zones within the passage (either rotating or stationary).

Guelich, et al. (1997), and Cugal and Bache (1997) described work comparing actual pump stage performance to CFD-based performance prediction using similar analytic methods. Basic CFD analysis was now influencing design decisions for critical service pumps. However all this analysis remained single-phase in nature. While pressure distributions could reveal regions that may fall below vapor pressure, there was no way to accurately model the interaction of the vapor cavity (low density), with the attendant evaporation and condensation process occurring at its interface, with the surrounding flows.

CFD for Multiphase Flow Calculations

While single-phase CFD has become a commonly used tool, only recently has CFD been integrated with the ability to perform analysis that includes the vapor phase. Typical of early work was that of Hirschi, et al. (1997). Here the authors developed an approach that can be referred to as "interface tracking." This consisted of performing an analysis of the passage or blade shape using CFD and solving for the velocity and pressure. If the pressure was below the vapor pressure, a cavity of some volume was assumed to form. Repeated iterations of solutions using a cavity volume whose interface tracked the vapor pressure would produce a final cavity shape. The solution views the cavity as a blockage to the flow with the noncavitating flow passing around it.

A program that incorporated CFD, two-phase analysis, and experimental flow visualization was conducted by Dijkers, et al. (2000). Here, a combination of CFD using a potential flow code (a three-dimensional solver without viscous effects) and a Reynolds averaged Navier-Stokes code (RANS) was used to solve the flow field for a centrifugal impeller and identify at what pressure the local pressure on the blade dropped below vapor pressure. These results were confirmed by flow visualization and a design was generated that achieved the minimum $NPSH_{inception}$ very near the duty flow.

Dupont (2001) furthered the interface tracking method of predicting cavity length and volume by analyzing several styles of impellers (radial to semiaxial) and actually predicting not only cavity length but also the $NPSH_{3\%}$ value. The pursuit of the interface tracking method was justified by the economy of the calculation procedure as compared to commercially available CFD codes where the two-phase behavior was predicted by a phase change model that was embedded in the code and was a part of the calculation for each elemental volume. However, with the increase in computational capability of reasonably priced work stations, the use of RANS codes with the two-phase behavior integrally modeled has become practical.

Visser (2001) used a commercially available RANS code to calculate incipient NPSH as well as the $NPSH_{3\%}$ characteristic from 65 percent to 118 percent of BEP flow rate. Good correlation was achieved near the design point flow, with reduced accuracy at off-design. DuPont and Okamura (2002) reported on work that compared three commercial CFD codes using different cavitation models. In this comparison, the authors concluded that one RANS code was, overall, the most appropriate code for predicting cavitating flows in impellers. The version of the RANS code used a constant enthalpy vaporization (CEV) model for cavitating behavior. This approach establishes the vapor mass fraction as defined by the local pressure and enthalpy of the flow. Little empiricism was required as the fluid equilibrium phase diagram contained sufficient information. While this approach was shown to work reasonably well for impellers, in some flow conditions the computational stability became an issue and convergence problems resulted.

A more recent approach, applied to the same RANS solver, used what is termed a volume of fluid (VOF) approach to model the two-phase behavior. The VOF uses a truncated version of the Rayleigh-Plesset equation to model the rate of vapor generation. This term is included in the volume fraction equation of the solver and is directly coupled to the mass and momentum terms of the RANS solver. The model also accounts for the presence of noncondensable gas in the fluid (e.g., air). Thermal equilibrium between the two phases is also assumed. The VOF approach results in a more complex set of equations to solve. However, it has been found to be more robust especially in areas of high acceleration.

Use of CFD for Predicting Cavitating Behavior

The technology applied to the two impeller redesign efforts described here, like all the past efforts, relied on field experience, experimental test, flow visualization, design experience, and some degree of fluid analysis using computational tools of limited accuracy. With increasing accessibility to sophisticated CFD solvers that embrace two-phase flow behavior, the ability to screen and evaluate candidate impeller blade designs for suction performance and cavitation formation is here. To support this conclusion the authors benchmarked previous designs with known behavior (using flow visualization and performance NPSH testing) against the two-phase CFD solver. This benchmarking was done for not only the conventional blade designs representative of pre-1990 practice, but for new blade shapes represented by the biased-wedge approach.

A complete benchmarking study requires investigation of not only the impeller, but also the suction inlet that feeds the impeller. The authors have found that side-suction inlets can be analyzed using CFD and velocity distributions identified for even the most complex designs. They have seen (Figure 8) that an impeller blade sees a continuously variable circumferential variation of prewhirl and mass flow with each rotation and correspondingly different cavitation lengths. The inlet effect on the impeller can be modeled either by solving the complete stage model or by performing multiple two-phase CFD solutions where each solution uses an inlet boundary condition defined by a specific region of the inlet. The complete benchmarking analysis is beyond the scope of this paper; however, results for impeller blades used in the 12 × 18 CA suction impeller redesign are discussed here.

The two key features used for benchmarking the accuracy of the two-phase CFD solution are the calculated cavity size and shape and the NPSH versus head characteristic at constant flow (used to identify $NPSH_{3\%}$). This information is acquired from CFD solutions for a given flow rate for multiple inlet pressures. The cavity length characteristic feeds the cavitation life Equation (4) and the $NPSH_{3\%}$ is used to establish the "R" value in terms of the application's ability to operate near the 3 percent head falloff condition.

Figure 16 is an example of the benchmarking process. A cavity length is observed from flow visualization testing and a calculated cavity length is shown on the right.

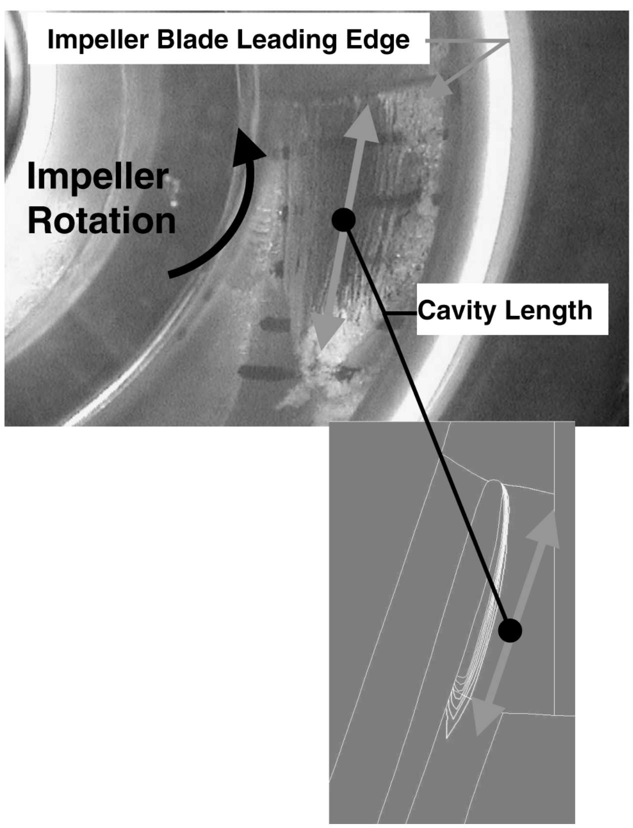

Figure 16. Correlation of Suction Surface Cavity Prediction with Flow Visualization. (The photo of the impeller blade on the left is operating near its design flow rate with a substantial suction surface sheet cavity. The plot at the right shows the vapor volume fraction calculated using CFD. The correlation of flow visualization with computational results supports the use of two-phase-CFD as a tool for designing improved impellers for use in critical applications.)

While the visual correlation is important, even more important is the prediction of head falloff with decreased cavitation number, τ. Figure 17 is an example of the conventional 12 × 18 CA NPSH versus head characteristic at rated flow. Embedded with Figure 17 are views of the suction blade surface showing the cavity development as NPSH is reduced (NPSH being shown as the nondimensional cavitation number "τ"). The square symbols are from test data; the diamonds are from the CFD solution. Good correlation between test and computation was found.

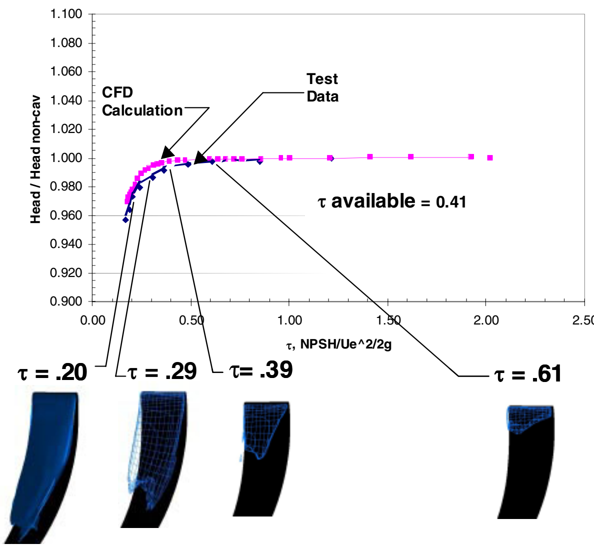

Figure 17. Suction Performance of Impeller at Rated Design Flow. (The comparison of predicted $NPSH_{3\%}$ performance with test data is good. Also shown are the sheet cavities calculated using two-phase CFD with the boundary defined by the cavity representing > 10 percent vapor content. The extent of the vapor at τ = .20 is considerable with the length extending back to the throat formed with the adjacent blade. The analysis uses zero inlet prewhirl, and is calculated for the low speed model in cold water.)

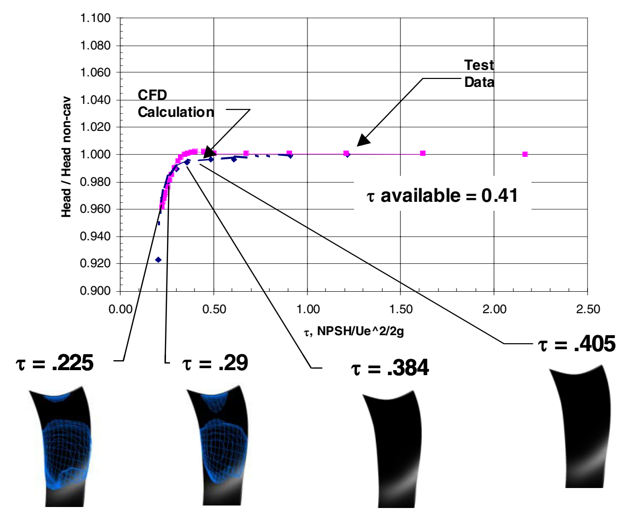

Figure 18. Suction Performance of Impeller with Biased-Wedge Blade at Rated Design Flow. (The comparison of predicted $NPSH_{3\%}$ performance with test data is good. Also shown are the sheet cavities calculated using two-phase CFD with the boundary defined by the cavity representing > 10 percent vapor content. The extent of the vapor at τ = .20 is considerable with the length extending back to the throat formed with the adjacent blade. The analysis uses zero inlet prewhirl, and is calculated for the low speed model in cold water.)

Another NPSH versus head characteristic is shown for the biased-wedge based 12 × 18 CA redesign. Figure 18 shows similar data as Figure 17. The absence of predicted cavitation well below the $\tau_{available}$ of .41 is apparent (and was confirmed via flow visualization tests).

Again good correlation of the performance NPSH characteristic is achieved. The unique cavity shape of the biased-wedge design at low NPSH is seen on the suction surface blades that are a part of Figure 18.

The authors have found that application of the biased-wedge shape to other impeller "platforms" can provide improved operation over the intended flow range (Cooper, et al., 1991a and 1991b) in terms of cavitation inception and suction performance. There is also ample evidence to conclude that proper two-phase CFD analysis of these impeller blade shapes will provide reliability where suction conditions are critical to the pump operation with a high degree of confidence.

CONCLUSIONS

Suction impeller design technology, in terms of cavitation behavior, did not keep pace with the trend to larger, super-critical, fossil fired steam plants that occurred from late 1960 through the early 1980s. As a result, numerous incidents of short impeller life due to cavitation damage and system instabilities due to cavitation were reported. Many efforts by OEMs, third-party consultants, and utility-sponsored agencies were directed at correcting this problem. Redesign programs were conducted and reported throughout the literature. However, little post redesign documentation of results is available.

This paper focuses on two such redesign efforts initiated in the 1990s. These applications represented two extremes of application, one having the benefit of a booster pump but operating at a very high inlet tip speed, and the other using a deaerator system, operating at a lower ratio of $NPSH_{available}$ to $NPSH_{3\%}$ One redesign effort had the benefit of an extensive experimental program. Both used a new impeller blade shape called a "biased-wedge." This blade design approach evolved from extensive experimental flow visualization studies and fluid analysis using the tools available at that time. The results of these efforts more than doubled the life of these impellers and significantly improved the availability of the feed pumps, thus reducing the costs associated with short life.

Today, advanced computational fluid dynamic analysis tools enable accurate prediction of two-phase behavior for impeller blade shapes. This tool has been used to validate the cavitation behavior for one of the impeller designs described in this paper. The availability and accuracy of this tool enable designers to evaluate and optimize alternate designs without resorting to expensive and time-consuming model testing. Pump applications where suction conditions can adversely impact impeller life, reliability, and system stability will benefit from application of the design technology and analysis capability described here.

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