EXPERIMENTAL EVALUATION OF HIGH ENERGY PUMP IMPROVEMENTS INCLUDING EFFECTS OF UPSTREAM PIPING

by

Donald P. Sloteman

Manager, Pump Research and Development

Timothy L. Wotring

Product Engineering Manager, Horizontal Utility Pumps

Ingersoll-Dresser Pump Company

Phillipsburg, New Jersey

Patrick March

Manager of Hydraulic Engineering

Dan McBee

Senior Mechanical Engineer

Tennessee Valley Authority Engineering Laboratory

Norris, Tennessee

and

Larry Moody

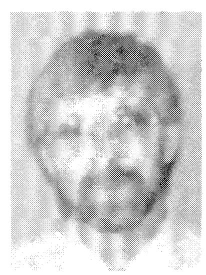
Director and Manager of Hydraulic Engineering

Tennessee Valley Authority Cumberland Steam Plant

Cumberland, Tennessee

Donald P. Sloteman is Manager, Pump R&D, for Ingersoll-Dresser Pump Company's Advanced Technology Department (a part of the Corporate Technology Department). He is responsible for executing programs and projects addressing technology and new product development needs for the various business units that make up IDP. Mr. Sloteman joined Ingersoll-Rand's Corporate Research Center as a project engineer. He has held positions of Senior Project Engineer and Project Manager previous to his present position. Besides his project management activity, he is also manager of IDP's Alexey J. Stepanoff, Memorial Hydraulic Laboratory in Phillipsburg, New Jersey. Mr. Sloteman received his B.S. (Mechanical Engineering) from Drexel University and his M.S. (Engineering, Executive Engineering) from the University of Pennsylvania.

Timothy L. Wotring is Product Engineering Manager for Ingersoll-Dresser Pump Company's Engineered Pump Business Unit. He is responsible for the design of horizontal utility pump and other large, nonutility related horizontal, multistage pumps. He joined Ingersoll-Rand's Pump Group (1975) as a design engineer. He has held positions of increasing responsiblity since that time. He received his B.S. (Engineering Mechanics) from Penn State University and is a licensed Professional Engineer in the State of Pennsylvania.



Patrick March is Manager of Hydraulic Engineering and Director of the Tennessee Valley Authority's Laboratory in Norris, Tennessee. Mr. March previously was Senior Mechanical Engineer with the Engineering Laboratory and Project Team Leader for teams of engineers responsible for providing engineering services and conducting multiple interdisciplinary projects in the areas of performance monitoring, cavitation erosion, flow measurement, instrumentation systems, and vibration analysis. He has also served as a Lead Engineer with the Alden Research Laboratory, an adjunct faculty member with the Worcester Polytechnic Institute and a Research Engineer with Bolt, Beranek and Newman. Mr. March received his B.S. and M.S. degrees (Mechanical Engineering) from the Massachusetts Institute of Technology.

ABSTRACT

The suction-impeller (single-suction) of a 33,500 hp, four-stage barrel pump used in boiler feed service at a fossil fuel steam plant experienced operating life ranging from 6.0 to 18 months. Cavitation erosion on the suction surface and in the hub fillet area caused this unacceptably short life. This pump design is unique in that its suction-stage operates at an inlet tip velocity of 281 ft/sec. This tip speed, coupled with a stage pressure rise of about 1160 psi places this machine well into the category of "high energy" pumps, and signals the need for special design attention to elements of its hydraulic design in order to provide suitable performance, reliability and operability.

In a joint program with the user and the pump manufacturer, the problem was investigated, a test program conducted, redesign of the impeller completed and replacement impellers commissioned in the field. Elements of the investigation and test program included the impeller design itself, suction bay distortions and effects of distorted flow generated by the upstream piping. The redesign was accomplished through the use of sophisticated computational techniques and experimental flow visualization. The final design, referred to as a biased-wedge, was manufactured using a precision, lost wax casting process, with a special stereolithographic technique used to translate the final blade design into usable patterns. One of the final set of impellers was tested using flow visualization.

The test program confirmed that the final design was free of cavitation bubbles at the baseload condition and significantly reduced vapor activity and hence damage potential at min-flow. The effects of flow distortions (on cavitation activity) presented to the impeller by the suction bay (so prevalent with the conventional design) was effectively countered by the biased-wedge design. Finally, the flow distortions (both introduction of swirl and uneven mass distribution) at the pump suction flange were modeled and tested with both conventional and biased-wedge impellers. In both cases, these distortions had no effect on the formation or collapse of cavitation vapor in the impeller eye. These conclusions are valid for only this single-suction type of configuration. The impellers have been commissioned in the field units and are currently accumulating operational hours.

INTRODUCTION—HIGH ENERGY PUMPS

Pumps that are considered "high energy" come in a variety of sizes and configurations. A common thread is their behavior when exposed to various internal and external stimuli. The responses can be classified under three general categories, each covering specific areas of concern:

- Mechanical
  - forces on internal components
  - vibration
  - structural integrity
- Fluid
  - unsteady pressures (i.e., pulsation)
  - cavitation formation
  - smooth flow delivery
- Materials
  - corrosion resistance
  - cavitation erosion resistance
  - sensitivity to erosion due to velocity

A comprehensive effort to define specific characteristics of high energy pumps has been developed since the late 1980s by Cooper, et al. [1, 2, 3, 4]. He has characterized high energy pumps in terms of the following critical design parameters; inlet tip speed, stage pressure rise and stage specific speed (N_s = $N\,Q^{.5}/H^{.75}$, where N is rpm, Q is flow in gpm and H is the total head rise in ft of water). These critical design parameters, coupled with information on the pumped fluid, led to quantifying energy levels of pumps in the following areas:

- Internal forces on diffuser vanes and volute cutwaters [4]
- Fluctuating forces on bearing and seal components [2]
- Cavitation formation (coupled with damage potential and introduction of pressure pulsation activity in the form of oscillating cavitation) [3].

In all cases, these definitions were not intended to impose limitations on pump operation, but to indicate the state-of-the-art of conventionally designed pumping machinery and encourage development of technology that would push that state-of-the-art. The focus here is on cavitation and its adverse effect on impeller life.

The cavitation erosion problem in high energy pumps requires more consideration than would be given to more common pumps, where the three percent head drop criteria is most important. Palgrave and Cooper [5] contributed to understanding the differences between the $NPSH_R$ required to suppress the three percent head drop (at constant flow and speed), the $NPSH_d$ required to provide the desired cavitation life for the impeller and the $NPSH_i$ required to suppress the formation of all cavitation bubbles in the suction impeller. These NPSH values have been listed in increasing order of magnitude (meaning that the $NPSH_R$ for three percent performance may be five to six times less than the $NPSH_i$ to suppress all bubble activity).

The field problem considered here involves a four-stage barrel pump, feeding half of the flow capacity to a boiler feed system that is part of a 1200 MW fossil fueled power plant. It is a single-suction design. The particulars of the critical design parameters follow:

- Stage head rise—2965 ft (rated)
- Stage pressure rise—1160 psi @ 330°F
- Inlet tip speed—281 ft/sec
- Stage specific speed—1600

These values qualify this machine as a high energy pump and as such, careful attention must be given to all aspects of the hydraulic and mechanical design in order to minimize the adverse responses to the areas listed above. In terms of cavitation life, the use of the cavitation life prediction equation developed by Gulich [6] and applied by Cooper in his definition of high energy pumps [3] was used to calculate a life of 7,380 hr for this suction impeller.

In a joint program between the user and the manufacturer, the nature of the impeller life problem was clearly defined, so that the ensuing program would cover of the elements contributing to the short impeller life. A summary of these elements follow:

- High impeller inlet tip speed—first order effect on cavitation formation
- High energy level—design modifications must not adversely impact performance in other areas.
- Side suction inlet bay—introduces significant flow distortions to impeller eye.
- Upstream piping—generates flow distortions at pump suction flange.
- Operating mode—baseload to min-flow cycling creates different flow conditions.

Any test program and resulting design improvement would have to consider these elements and assure that no single element is left out, since this would compromise the desired improvement. The use of alternate metallurgy was not considered. While harder materials with better erosion resistance than CA6NM material exist, the tradeoff of acceptable mechanical properties for the increased resistance was not desirable. Additionally, even if harder materials did extend the life of the impeller, the root cause, cavitation vapor in the inlet of the impeller, would still exist. The presence of these large amounts of vapor in a high energy pump is undesirable and merits its own effort to alleviate the problem.

The program had as a goal, to produce a five year impeller for this pump, operating in a cycling mode. It began with an evaluation of the field problem that documented the five elements listed above. While a specific analysis and experimental program was being formulated and conducted an intermediate fix consisting of blade profiling and contouring of a sand cast impellers was instituted. This fix was placed in the field for evaluation and provided valuable feedback for the development program. Sophisticated computational flow analysis programs were used to analyze existing and potential design candidates. Flow visualization with a fullscale, part speed model of the suction stage (using hardware manufactured from production tooling) was used to evaluate the five elements and provide feedback to the design team. Once a suitable design was finalized, precision investment castings were made, using tooling generated directly from CAD files that had been developed by the hydraulic design program. This assured exact conformance between the final part and the desired design. One of the production castings was tested in the flow visualization set up to confirm the expected result. The new impellers are currently being integrated into the four pumps at the power station.

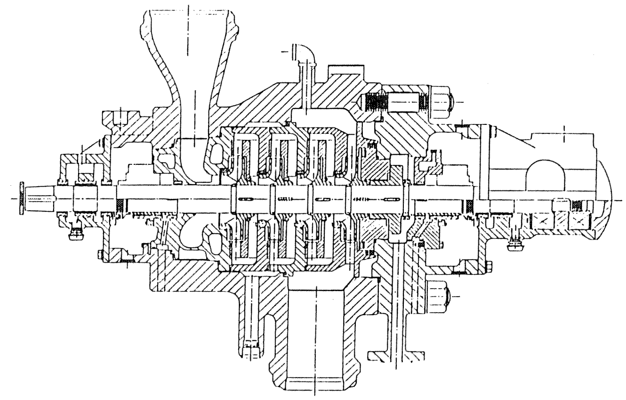


Figure 1. High Energy Boiler Feed Pump Cross-Section. This pump is typical of the 33,500 hp 12 × 18 CA that operates at a rated speed of 5,600 rpm. This results in an inlet tip speed on the single suction first stage of 280 ft/sec. A booster pump provides the NPSH to the pump and is equivalent to a nondimensional cavitation number of 0.40.

BACKGROUND—THE 12 × 18 CA BOILER FEED PUMP

The trend toward constructing larger and larger, supercritical steam plants, which began in the 1960s to satisfy increasing needs for electric power, required the application of multistage boiler feedwater pumps in ever larger sizes. Optimization of these feedpumps for overall power plant efficiency, produced a dramatic increase in rotating speed (to generate the higher pressures needed) and, hence, higher inlet tip speeds on the suction stage. Similar increases in the NPSH available were not forthcoming, so that these new pumps, with either boost pumps or elevated deaerator vessels to supply inlet pressure, operated with significant cavitation activity on the suction impeller. Numerous studies (referenced later) have demonstrated that the price for these higher tip speeds is increased erosion potential from cavitation and the possibility of system instabilities due to the presence of large volumes of cavitation (in a compressible vapor state) in the feedwater system. Aggravating this condition was the added requirement of these pumps to cycle between baseload and min-flow to accommodate fluctuating electric power demand. This pattern of activity led to the uncovering of numerous installations where equipment designed with 1960s technology was performing poorly in the 1980s, due to the increased demands placed on them.

The subject pump design (actually four pumps installed) was commissioned in 1972, installed as part of a two unit coal fired power station at Cumberland, Tennessee (a part of the Tennessee Valley Authority power generation system), serves as a halfcapacity feed pump. A crosssection of the pump is found in Figure 1. Each unit is rated at 1200 MW. Each pump is a four-stage machine rated at 33,500 hp. It delivers 11,000 gpm at 11,860 ft of head with a rated speed of 5,800 rpm. The first stage is a single suction design with an inlet tip speed of 281 ft/sec. Booster pumps are used to supply the feed water to these pumps. At design flow, the dimensionless cavitation number, τ (defined as $NSPH_R/U_e^2/2g$), is 0.40.

This combination of low NPSH available ($NPSH_A$) and high inlet tip speed resulted in a suction impeller life of about six months for the initial design. A photo of cavitation damage from an impeller from this pump is seen in Figure 2. No damage was observed on the pressure side (the hidden side of the blade). Correction of inlet blade angles improved the life to 12 to 18 months. Periodic cycling of the pumps from full flow to half flow in response to power demand also had an adverse affect on impeller life. This wide range of flow operation contributes to the presence of three of the well known cavitation damage mechanisms; sheet cavitation, vortex cavitation, and hubfilled cavitation. The utility also suspected that upstream piping was contributing to the cavitation problem by introducing flow distortions (swirl in the pipe and uneven flow distribution) at the pump suction flange (and carrying on into the eye of the single-stage impeller). TVA had encountered a cavitation erosion problem on a double-suction first stage impeller elsewhere in thier system that had been traced to inlet piping distortion. This problem was traced to uneven flow entering each side of the double-suction impeller.

Figure 2. Cavitation Damage on 12 × 18 CA Suction Impeller. The damage accumulated on this impeller occurred over a period of about 18 months operation. It is considered at the end of its operational life. No pressure (or hidden side) damage was observed.

The critical nature of the problem, aggravated by the high tip speed of the impeller, prompted a special development effort to provide a solution to this problem. Concurrently, the manufacturer had under development a new impeller blade technology, referred to as biased-wedge blading. This design technology held the promise of delivering radical reductions in cavitation activity within the suction stages of impellers. The blending of the technology with all aspects of the field problem is the subject of this study.

CAVITATION IN HIGH ENERGY PUMPS

Effects of cavitation in high energy pumps can be classified into two areas. First is the erosion process that shortens the life of suction impellers. The second is the introduction of a compressible volume (the cavitation vapor itself) into the essentially incompressible pump system. This compressible volume can lead to unsteadiness in the system that results in pressure pulsation, varying flow delivery and piping or pump vibration. This cavitation caused instability (sometimes referred to as oscillating cavitation) that has been observed in a variety of high energy pumps [2, 3, 7]. Clearly, both effects are undesirable in a high energy pumping system, and the ultimate solution to both problems is the removal of the vapor (or cavitation) from the system. For the application described here, the erosion problem emerged as the dominant area of concern. Poor running characteristics (manifested as vibration) over the flow range were not identified as a primary area of interest.

Work published by Cooper, et al., regarding definitions of high energy pumps [1, 2, 3], has been discussed previously. In summary, the combination of impeller stage pressure rise ΔP), specific speed Ns and inlet tip speed (U_e) determines how critical the design is in terms of cavitation behavior, generation of hydraulically induced forces, and how these forces affect the pump operation and longevity. What emerges from these various measures is that the 12 × 18 CA barrel pump considered herein qualifies as a high energy machine, meaning that special engineering attention must be given to elements of the hydraulic design in order to assure suitable life and operation in the field.

The hydraulic component considered here is the suction impeller stage of the 12 × 18 CA, which had experienced shorter than desirable life, due to cavitation erosion. The factors contributing to the cavitation life of a suction stage are documented in many sources. The most comprehensive of these are found in Gulich [6] and Cooper and Antunes [1]. The aspects of cavitation and the accompanying erosion process in high energy pump can be divided into four major areas.

The Fluid

The fluid properties that affect the formation of cavitation are the vapor pressure and density of the liquid and the thermodynamic properties of the vapor. These properties determine the amount of vapor and potential size of the individual bubbles that break off of the main cavity. The external influences that affect the cavitation process are the local pressure and the temperature of the pumped fluid. The other fluid related factors that influence cavitation are the amount and size of nucleation sites for cavitation bubble formation (could be particles or small air bubbles).

Hydraulic Design

The pump design factor that has the most influence on cavitation behavior is the impeller inlet tip speed (found by multiplying the shaft angular velocity, ω, by the radius to the impeller eye, r_e). This velocity is essentially the same as the velocity of the entering pumped fluid relative to the impeller. The kinematic relationship between the approach fluid and the rotating impeller is shown in Figure 3. This diagram identifies the relationships of the inlet tip velocity, relative velocity and the throughflow, axial component.

The geometry of the impeller blading is a second factor that influences the cavitation behavior of the impeller. The blade angle shown in Figure 3, is important, because it determines the ability of the incoming flow to enter the impeller blade system smoothly and without disruption. The difference between the incoming flow relative velocity and the blade angle, is referred to as an incidence angle. Too large an incidence angle results in flow separation and recirculation. An incidence angle near zero is ideal, since it permits the flow to enter the impeller blades smoothly, and without the generation of separation zones, or eddies (at the center of that is a pressure lower than the surroundings and a source for cavitation vapor).

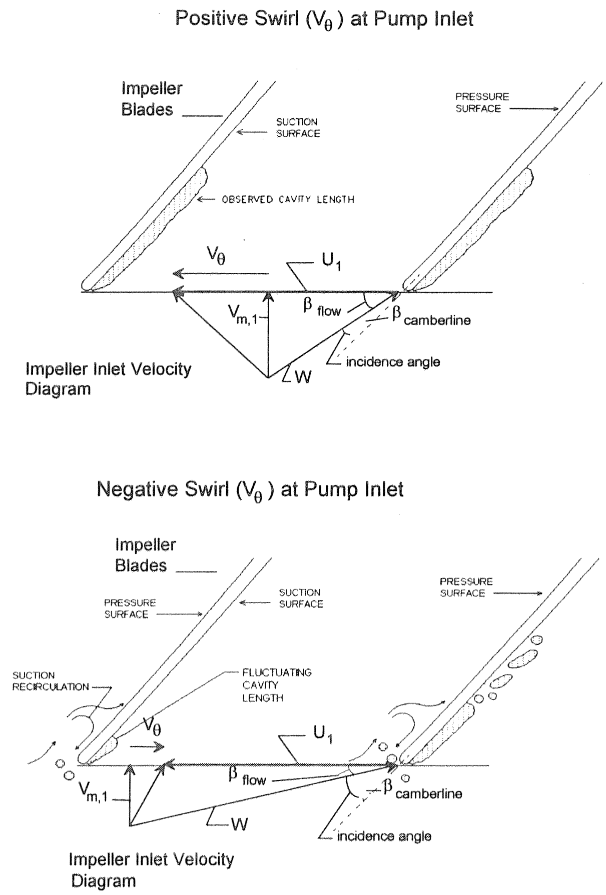


Figure 3. Effect of Inlet Flow Variations on Impeller Blade Behavior. The presence of positive and negative swirl, caused by the side-suction inlet bay that is a part of every barrel pump, leads to significant variation in the angle of attack relative to the impeller blade. This circumferential variation leads to differences in pressure distribution on the blade and hence cavitation activity.

The third factor is related to the impeller inlet blade angle distribution. The importance of designing the correct inlet blade angle distribution and its effect on cavitation formation and damage cannot be minimized. Schiavello and Prescott have documented the increase in cavitation life by correcting impeller inlet blade angles to improve life of impellers suffering erosion damage in the field [8, 9]. It is this blade development that is actually turning the flow and adding the energy that can be measured by the increase in pressure at the pump discharge. The blading close to the inlet plays a major role in the development of cavitation on the impeller. Blading that turns the flow too abruptly lowers the pressure on the visible (or suction) side of the impeller (i.e., visible when looking into the pump inlet) and promotes the formation of cavitation. Blades that don't turn the flow enough in the inlet region end up being too long and can hurt efficiency performance by introducing higher than desirable friction losses in the impeller passage itself.

Finally, for barrel type pumps that is the focus of this paper, the inlet suction bay design plays a significant role in cavitation formation. The purpose of this type of inlet is to take the incoming flow and guide it from a radial to the axial direction needed to enter the impeller. In doing this, the flow must split around the shaft, imparting a tangential component to the velocity, which on one side is in the direction of rotation and on the other side is counter to rotation (commonly referred to as positive and negative preswirl respectively). An outline figure of this type of inlet is found in Figure 4 [10]. The problem of positive and negative swirl exists whether the pump is using a single-suction or double-suction first stage impeller. Compounding this swirl variation is an accompanying distortion in axial velocity caused by uneven distribution of the mass flow. An analysis of the typical inlet shown in Figure 4 by Dhaubhadel, et al. [10], yields plots of tangential and axial velocities that approach the impeller eye. These plots are found in Figure 5. The distortions can be indexed to the inlet geometry in Figure 4 by understanding that the 180 degree position on the x-axis is the part of the inlet directly in line with the incoming flow from the suction flange. The 0 and 360 degree position is directly opposite the suction flange and is essentially masked by the shaft itself, accounting for the lower axial velocity component and the zero tangential velocity component. The effects of these inlet distortions on the fluid velocity diagram are found in Figure 3. The variations in velocity have an impact on the cavitation formation on the blade surface, as has been reported by Schiavello [11] and Dernedde [12]. While this inlet is representative of the types of flow distortions presented to the impeller, attempts at reducting this variation via design have been made. The more ideal suction bay design is shown in Figure 6, and is taken from Stepanoff [16]. Placement of the baffle and a scheduling of the inlet areas help reduce the tangential and axial variation. Barrel pumps like the 12 × 18 CA utilize this approach to suction bay design although constraints of the barrel dimension minimize its improvement.

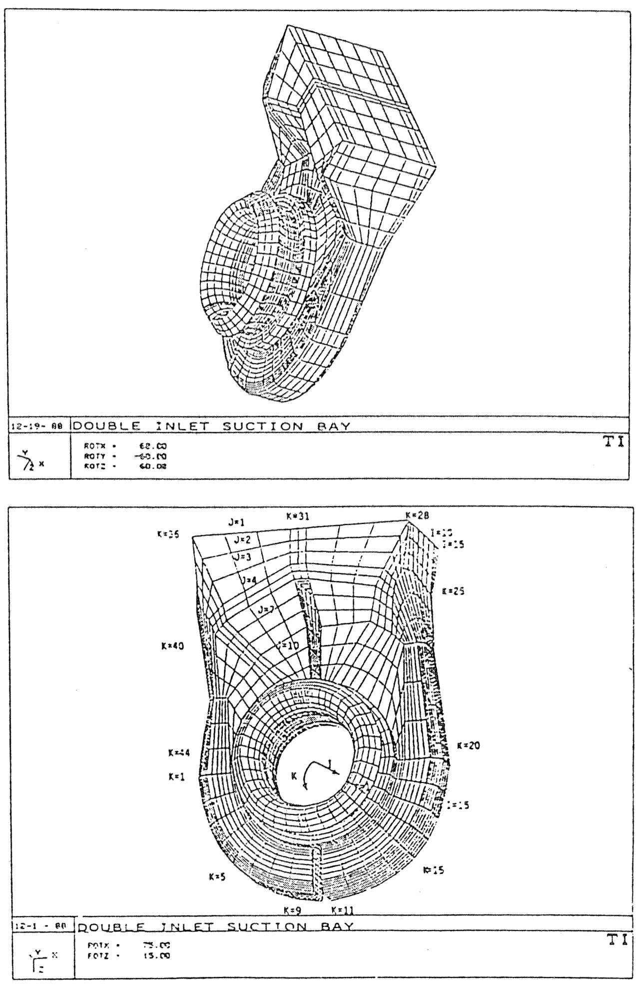


Figure 4. Example of a Side Suction Inlet as Used in a Barrel Pump. This is a model of a side suction inlet approach (a double-suction type design was used for this analysis) that was analyzed using a 3-D viscous computational fluid dynamics solver. It is included to show the types of circumferential distortions that are generated by the suction approaches of all barrel pumps [10].

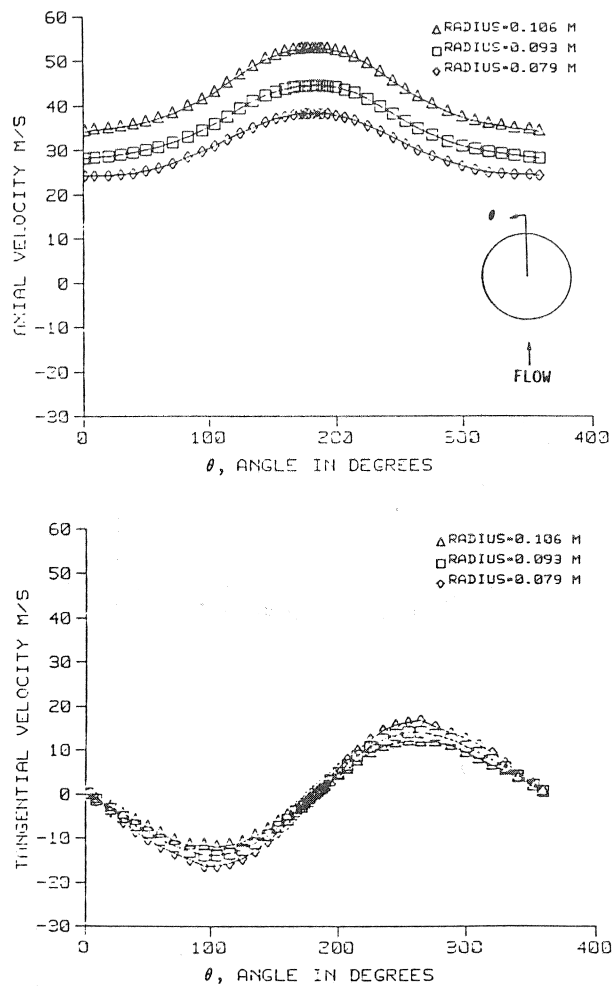


Figure 5. Representative Circumferential Velocity Distributions Found in Analyzed Inlet. The local tangential component (along with the axial component) define the inlet flow field seen by the suction impeller. For the analyzed inlet, a circumferential variation is seen in tangential velocity caused by the splitting of the flow about the shaft. The upper plot shows the circumferential variation in axial velocity, which implies that the regions away from the inlet pipe (that is at 180 degrees) are starved for flow [10].

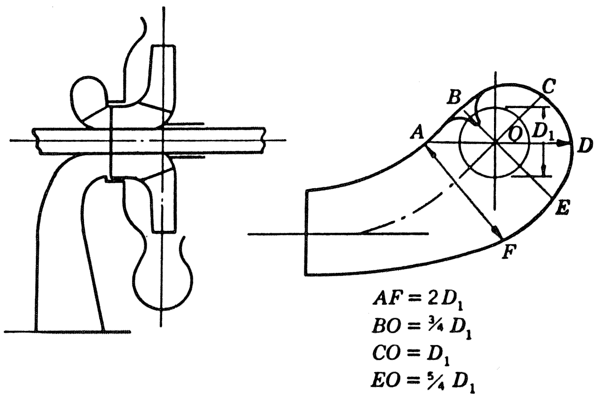


Figure 6. Classic Stepanoff Type Suction Inlet Bay. This design, shown here for single-suction impellers is similar in design philosophy to that found in the 12 × 18 CA. The scheduling of the inlet area is an attempt to better distribute the axial flow circumferentially. The placement of the baffle decreases the extent of negative swirl [10].

Impeller Metallurgy

If damaging cavitation is present in the suction stage of a high energy pump, the last "line of defense" against premature erosion is in the resistance of the impeller material itself. The erosion process is complicated, but it can be simply described as a combination of mechanical failure of the impeller material, coupled with a corrosion component of wear. The mechanical component results from the forces caused by pressure waves generated by the collapse of the low mass cavitation bubble and the nearly instantaneous return to the liquid phase. Impellers subjected to the constant pounding from these collapse generated pressure waves can suffer fatigue of the material from cycling compressive stresses. Accompanying the nearly instantaneous phase change is a sudden, localized increase in temperature in the fluid. This local temperature increase may promote a corrosion process that in turn weakens the base material near the surface and makes it more susceptible to the compressive stresses that result in erosion.

Key material parameters that affect erosion resistance are the modulus of elasticity (the lower the better), surface hardness (the higher the better), and tensile strength (the higher the better). Another characteristic of the material that is important in evaluating erosion activity within an impeller is the ability of the surface to work harden (achieved during the initial cavitation attack period). This is typical of the martensitic type stainless steels found in most high energy pumps. The cavitation resistance properties and the mechanical and thermal characteristics of the material must be balanced to achieve a suitable compromise between life and mechanical integrity of the impeller.

System Considerations

Factors external to the pump have a major impact on the potential for cavitation erosion. Pumps that are cycled between min-flow and flows in excess of best efficiency point (bep) flow create conditions at the impeller in excess of what "fixed geometry" machines can effectively tolerate. At flows beyond bep, cavitation appears on the hidden side of the impeller blade and can be characterized as a sheet type of cavitation. At flows around bep, the cavitation appears on the visible side of the impeller as a sheet, but with small bubbles breaking off in the region of the cavity sheet closure (i.e., the interface between the vapor phase and the liquid phase). For flows below bep, where suction recirculation is present, a mix of cavitation described as "vortexing" and some sheet type cavitation occurs. These vortex related bubbles are formed in the center of the separation eddies, which are created from the excessive angle of incidence and turning of the flow by the inlet portion of the blades. At very low flows, when the pump is in deep recirculation, any hint of a sheet type cavity disappears and the inlet is dominated by individual bubbles generated by the vortexing. Clearly, how the pump is operated will determine the nature of the flow and cavitation activity within the suction stage.

The characteristics of the fluid can change, depending on how the pump is operated. For example, in a fossil fueled steam plant, the boiler feed pump may be used for startup or test purposes, where instead of feeding the boiler with hot water, colder water may be pumped, at reasonably high energy levels. The cold water may have more air bubbles present, which presents more nucleation sites for cavitation inception. Additionally, the thermodynamic characteristics of cold water result in more higher vapor volume per lb of water vaporized, meaning a more energetic phase change, which yields a condition more prone to cavitation damage.

A third system factor is the design of the piping, upstream of the pump suction flange. Studies have shown how multiple elbows, configured out of plane, lead to the generation of significant swirling in the piping and even uneven flow distributions that are manifested as higher flow velocities in one region of the pipe *vs* another [13]. Piping layout is generally not designed out of consideration to the pump's needs, but is subject to the needs of the plant layout. How the distorted pipe flows affects cavitation in high energy pumps in a barrel configuration, with either single or double suction type, has not previously been studied. This distortion superimposed on the distortions inherent in the rightangle inlets on barrel pumps may have some additional effect on cavitation life. The phenomenon is significant enough to have produced cavitation problems in the field attributed to it.

Finally, the flow within the pumping system, if it is destabilized by nonpump related activity, can oscillate and, hence, vary the inlet pressure to the pump suction. The "pulsing" of the inlet pressure and flow can bring about different cavitation patterns within the impeller for what appears to be the same "average through flow" rate. This may result in damage patterns out of character from what one would expect from a single flow and pressure operating condition.

Estimating Impeller Cavitation Life

Attempts to estimate the cavitation life of a high energy pump impeller at the design stage should consider all of the factors just discussed. Early efforts led to simplification of the process, and a focus on the key elements that determine cavitation formation and the energy associated with the damage mechanism. Florjancic correlated the effect of inlet tip speed (ωr_e) and cavity length on material removal [14], and Dernedde and Stech improved on this by adding an exponent to the cavity length term [12]. This simple relationship follows:

$$\Delta m \sim U_e^a \, L^b \, T \qquad (1)$$

where Δm is the loss of material, U_e is the inlet tip speed, L is the cavity length, and T is the time of operation. In Equation (1) the exponent 'a' was stated to be between 3.0 to 4.0, and 'b' varied from 2.0 to 4.0. While this relationship was developed for typical boiler feed water applications, and it captured probably two of the most important factors relating to the damage process, it did not include the effects of impeller design and external influences such as fluid condition and system operation.

A more comprehensive attempt at modelling the damage process in centrifugal impellers was undertaken by Cooper [1], which incorporated many of the factors discussed in the previous section. It relied on first principles in many instances to define the impact of these factors on the cavitation damage process. The relationship defines the NPSH required to achieve a desired erosion rate ($NPSH_d$). This rate, taken with the design thickness of the blade, defines the life of the impeller. The relationship and a brief description of the terms follow:

$$NPSH_d/NPSH_R = (1/\tau)\ \{\ C_{p,l} - k_f\ MDPR\ UR\ /\ [\ (B/g)\ U_e^5\ \rho_L\ ]\ \} \tag{2}$$

where τ is the dimensionless cavitation number for three percent head loss, $C_{p,l}$ is a pressure reduction coefficient that defines the degree of pressure reduction for a specific impeller blade shape, k_f is a coefficient that describes the impeller geometry, MDPR is the mean depth of penetration rate, UR is the ultimate resilience of the matertial (tensile strength^2/modulus of elasticity), B/g is the fluid vaporization constant, U_e is the inlet tip speed and ρL is the density of the pumped fluid. This model, while complicated and as yet undeveloped, combined the key elements of the damage process and could enable impeller designers to evaluate and optimize designs for cavitation life in the design stage, before committing resources to test or field trials.

The final model to be discussed comes from Gulich [6] and bridges the simple two variable model and the more complicated approach that was based on first principals. Gulich resorted to correlation of vast amounts of field data on existing boiler feed impellers. In correlating the life and operating conditions of these impellers, Gulich included factors that account for many of the key elements contributing to the damage process. The equation defines an erosion rate for a set of operating conditions, referenced to a set of given conditions:

$$E_R = C_L\left(\frac{L_{cav}}{L_{cav,R}}\right)^{X_2}\left(\frac{(P_o - P_{sat})^3 F_{cor}}{R_m^2\ F_{mat}}\right)\left(\frac{\alpha_R}{\alpha}\right)^{0.36}\left(\frac{a_R}{a}\right)\left(\frac{\rho_R}{\rho}\right) \tag{3}$$

In this equation, the erosion rate, E_R, depends on the density of the liquid, ρ, the speed of sound of the liquid, a measure of its compressibility, a, the gas content α, the NPSH, P_o-P_{sat}, material properties for corrosion and strength, R_m, F_{mat}, F_{cor}, the length of cavity, on either suction or pressure side, L_{cav}, and a correlation factor, C_L. A reference cavity length, $L_{cav,R}$, is used to correlate the test experience with field data. Due to the many correlation factors used in this approach and the accuracy of the data used, error in predicting the erosion rate is expected. Also, the user of the model must have knowledge of the cavity length for that particular design, a fact difficult to determine if no model data is available (either in form of flow visualization, soft coating erosion, or actual erosion patterns).

Cooper, et al., described an improvement on this correlation, by using a generic cavity length function [3]. This function was developed from observation of flow visualization tests for a number of conventionally designed high energy impellers. It recognized the fact that for impellers with designs that reasonably match the incoming flow (i.e., the incidence angle is very small at bep flow) the inception of cavitation on the blade occurs at a cavitation number τ_i, (where $\tau = NPSH/U_e^2/2g$) equal to 1 + $\tau_{3\%}$. It also recognized that at the $NPSH_{3\%}$ condition, the length of the cavity typically reached a point on the suction side of the blade that forms the throat (or minimum area of the passage between the adjacent blade and the observed blade) of the impeller inlet. By observing the cavity lengths between these two boundary conditions, a function was developed that fairly accurately defined the $L_{cav}/L_{cav,R}$ term needed for Equation (3):

$$L_{cav} = 0 \text{ for } \tau \le \tau_{inception}$$

$$L_{cav} = \frac{\pi D_e}{Z}\left(1 - \frac{\tau - \tau_{3\%}}{\tau_{inception} - \tau_{3\%}}\right)^{0.25} \text{ for } \tau \le \tau_{inception} \tag{4}$$

This work on predicting cavitation erosion served to identify design problems associated with improving the life of high energy pump suction impellers, but did nothing to push the state-of-the-art in terms of design. The relationships just described instead focused the attention of users and manufacturers on minimizing cavity length on impellers, using existing design philosophy, as the means to improve cavitation life.

Rethinking Impeller Blade Design

The cavity length approach to understanding cavitation erosion damage led to numerous cases of redesigning damage prone impellers with improved inlet blade angles. Schiavello and Prescott [8] describe several cases where revised inlet blade angles improved life. This improvement occurred due to reduced incidence angles that resulted from the redesign. This indicated that the impellers were probably erroneously designed to begin with and these design changes only served to attain the state-of-the-art in terms of life. The optimization of the blade angle, while improving the average life, could not address the problem found in barrel pumps of circumferentially distorted inlet flows approaching the impeller blading and the resultant variability of cavity lengths. As Schiavello points out [11], this variability generates not only differences in cavity lengths, but also pressure pulsation and vibration as each impeller blade sees a different approach condition with each rotation. Accommodating this variation with a fixed geometry machine would require something more than a conventional approach to blade design.

Through the 1980s, attempts were made to pursue "nontraditional" design of impeller blading. These efforts took the form of profiling the inlet blade in a way that rapidly thickened and then thinned the blade thickness. Dernedde and Stech [12], Makay [15] and Cooper, et al., [2] have published results incorporating this nontraditional approach to inlet blade design. In cases where flow visualization was available, [12, 2], significant reduction in inception NPSH was achieved. Attendant reduction in cavitation erosion was also reported [15]. Concern with this type of profiling centered on the quality of manufacture and the adverse impact on three percent NPSH performance. Incorrect application of this type of profiling, or variability caused by casting variation (or worse, application of weld overlay) can increase the three percent NPSH value and, hence, reduce the margin between available NPSH and the three percent requirement. The result could be catastrophic if the pump is subjected to a severe suction transient, and the stages subsequently vapor lock because of the inability of the first stage to maintain pumping capacity.

Redesigning the 12 × 18 CA Suction Impeller

The baseline 12 × 18 CA impeller hydraulic design produced in the 1980s accurately reflected conventional design technology. As such, it actually fit quite well into the cavitation damage life correlations described earlier. Based on observations of the damaged zones on the suction side of the impeller and from flow visualization tests made on fullsize models at reduced speeds, a nondimensionalized cavity length of 0.27 was determined (based on $L_{cav}/[\pi D_e/Z_{blades}]$). This fits the generalized cavity length model described in Equation 4. Most of the pump operation was at baseload; however, it was determined that low flow operation was responsible for the severe hub fillet damage observed on the field units. Application of the EPRI life Equation 3 led to a predicted life of 7,380 hr, or approximately one year. This corresponded well with the 12 to 18 month life span of these impellers at Cumberland. The short life was not due to poor blade design, but to the extremely high inlet tip speed described earlier. Another factor to be considered was the known distortions to the incoming flow due to the upstream piping layout. It was possible that swirl and nonuniform flow at the pump flange was contributing to the variations at the impeller blading that made the extremes of approach condition (and hence, incidence angle) more severe. This, in turn, could cause longer cavity lengths than normal. The effect would be considered second order, since the cavity length model, without the effect of distorted flow, was predicting typical life for this design.

An attempt was made to use the profiling techniques described earlier on a 12 × 18 CA suction impeller. This yielded some improvement over the baseline design. Less suction surface erosion was observed, and significantly less hub fillet damage occurred. However, the user's goal of a minimum of five years life was not likely to be achieved by this approach.

This problem lent itself to application of new design technology for designing impeller blading for minimal pressure reduction and, hence, improved resistance to cavitation formation. This design approach, which integrates several design features is known as biased-wedge blading, and was described by Cooper, et al., in 1991 [3]. In concept, it generates a design that alters a typical pressure distribution for a conventional impeller, by eliminating the local drop in static pressure on the blade surface that is responsible for formation of cavitation on the blades suction (i.e., visible) surface. The computational support for this design approach, along with examples of application, have been widely published [2, 3, 12]. The design, if properly executed, should eliminate cavitation formation at baseload condition, minimize vapor activity at min-flow, and eliminate the potential for hub fillet damage. It was also expected that the unique shape of the blade at the inlet would tolerate a large variation of incidence angle (inherent in the side suction inlet bays found in barrel pumps), while not adversely affecting the $NPSH_{3\%}$ value. This would preserve the NPSH margin required to handle suction transients without vapor locking the rotor. A pictorial representation of this blading concept is found in Figure 7, along with descriptions of specific features.

Currently, the design of biased-wedge impellers is best accomplished via interaction between experimental and analytical means. Analytical techniques used to design sophisticated impeller blading using computational techniques is becoming a powerful tool and is useful in arriving at a near final design [3]. However, the computational capabilities necessary to deal with the two phase flow present in cavitating impellers is not yet developed. The use of flow visualization and hand implemented modifications are still necessary and will be described in the next section. However, with the experience of the 12 × 18 CA, added to previous impellers designed using the same techniques [2, 3] a reasonable database has been accumulated that will allow similar cavitation avoidance features to be designed into other impellers, without resorting to individual experimental programs.

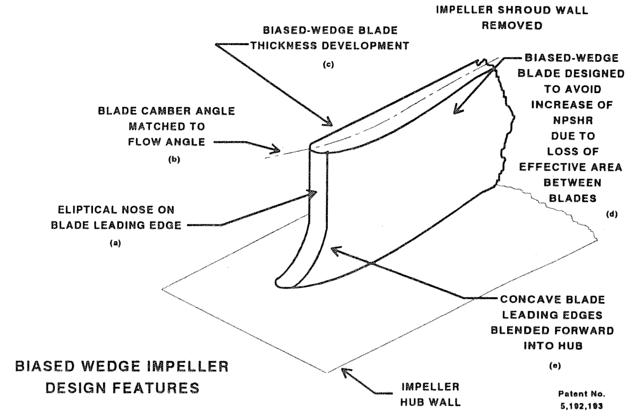


Figure 7. Biased-wedge Impeller Design Approach. The improved impeller design must incorporate these key features to achieve bubble free operation over a wide flow range, and to accommodate flow distortions arising from the suction inlet bay.

Manufacturing Biased-Wedge Impellers

Reproduction of high energy impeller designs is a constant source of concern. Besides the basic integrity of the casting and material, the impeller must accurately translate the intended design to the final part. Also, the resulting blading must maintain circumferential uniformity to prevent the introduction of hydraulic unbalance and variation of inlet blade angles. Flow visualization tests on sand cast impellers reveals differences in individual blades in more dramatic fashion than numerical inspection techniques. The pump industry has long recognized the deficiencies in sand casting high energy pump impellers, and has moved to higher precision casting processes.

Dimensional accuracy and repeatability are essential in the application of biased-wedge impeller blading. Experiment has shown that small changes in impeller blade thickness distributions of 0.010 to 0.020 in can destroy the potential benefit in cavitation reduction and compromise $NPSH_{3\%}$ performance significantly. Ultimately, the impeller could be machined using five-axis milling techniques (five-axis milling being necessary due to the complexity of Francis type blading found on high energy pump impellers), with the impeller shroud being attached after the machining process. More exotic forms of impeller machining can be considered (with the final impeller being machined from a single billet of material), but the cost required *vs* the value added from the process is excessive. A more cost effective approach to the manufacture of these impellers was undertaken.

Commercially, the manufacturer has used the precision investment casting process (lost wax method) to build the biased-wedge impeller. The use of precision dies to construct the positive wax pattern of the impeller is superior to sand casting methods where the cores in the mold were subject to a variety of operations that could ruin the dimensional consistency of the final part.

The critical aspect of any casting processes is in the translation of the blade shape to a pattern. In the process described by Stepanoff [16], a description of a blade is defined by a conformal grid and given to a pattern maker. The pattern maker generates a wooden master blade, which may or may not be totally faithful to the engineer's desired shape. Whether the master blade passes inspection or not; typically, some compromise has been made between the desired hydraulic shape and the final product. Today, most sophisticated impeller blade shapes are defined on CAD systems, and usually are the product of computations from hydraulic design programs.

The availability of these blade shapes in a computer generated three-dimensional wire frame or solid model lends itself to new prototyping techniques. The blade shape can be machined into a corebox, used to construct the soluble waxes used in the lost wax casting process. Or in the case of the 12 × 18 CA discussed here, the blade shape data were used to construct a polymer model of the blade using stereolithography. Special attention is paid to the jigging and support structure of the blade to prevent warping and distortion during the cure cycle. The stereolithographic process is accurate, quick, and produces a blade shape that is exactly what the hydraulic designer specified. This blade can then be used to form the corebox, to produce the soluble waxes needed for installation in the die. The positive wax pattern is made from this die and is then invested, burned out, and consequently lost. Good blade to blade uniformity from lost wax, precision casting of high energy pump impellers has been proven by excellent uniformity of cavitation patterns formed on each blade (the cavitation patterns being indicative of the interaction of incoming flow with each blade shape). Visual obseration and video documentation confirm this.

INLET PIPING

Providing good inlet flow to the pump suction is considered essential for good pump operation. Elbows or similar flow disruptions in close proximity to the pump has been shown to promote a variety of mechanical and hydraulic problems. TVA had previously investigated a cavitation problem on a double suction first stage impeller of a boiler feed pump, where one side of the impeller had experienced cavitation erosion and the other side had not. The cause had been traced to uneven mass flow entering the suction flange, which favored one side over the other with flow. Accordingly, the question was raised regarding the effects of flow distortions, i.e., introduction of swirl and uneven flow distributions in the pipe, on the single suction 12 × 18 CA installed at Cumberland.

Published data have documented variation in cavitation activity on suction impellers (observed via flow visualization testing) when exposed to distorted inlet flows [11, 12]. The possibility existed that superimposing distortions at the pump flange on the distortions caused by the suction bay could further aggravate this variation in cavitation activity. Conversely, the effect of accelerating the flow from the pump flange into the suction bay, and then guiding the flow into the impeller eye might "wash out" any adverse effects caused by the piping distortion.

In order to understand the effects of upstream piping on cavitation activity, the distortions produced by the upstream piping at Cumberland were introduced into the model test program intended to develop the biased-wedge impeller blading for the upgraded design. The controlled test environment would provide indication of whether the distortions at the pump flange could have adverse impact on the cavitation life of the impeller.

To accomplish this goal, the user commissioned a model test of the plant suction piping (carried out in their Engineering Laboratory). The piping at Cumberland connects the suction of the boiler feed pump to the booster pumps a level below. The feedwater flows through three inplane 90 degree elbows, through a gate valve, and into an out of plane 90 degree elbow that directs the feedwater vertically upward to the pump suction flange (Figure 8). The model constructed for the flow tests was made from 10 in pipe, which resulted in a model ratio of 1:1.8. Using cold water as the working fluid, the model piping was run with a Reynolds number of 480,000. The Reynolds number of 11 million for the full size left a Reynolds number ratio of 22:1. However, previous research (by Burton and Willison [17], Patterson and Abrahamsen [18] and Murakami and Shimizu [19]) had indicated that flow patterns in comparable situations typically become independent of Reynolds number for Reynolds numbers greater than about 100,000.

To measure the velocity profiles at the pump flange position, a one-component laser Doppler velocimeter (LDV) was used (in backscatter mode) to probe a transparent test section. The LDV approach is desirable because it is entirely nonevasive to the test flow. The axial and tangential velocity components were measured with this device and a survey of these velocity distributions in the pipe section along radials, 45 degrees apart was obtained. Visualization of the flow in the test section was done by injecting small amounts of air into the pipe and observing at the test section.

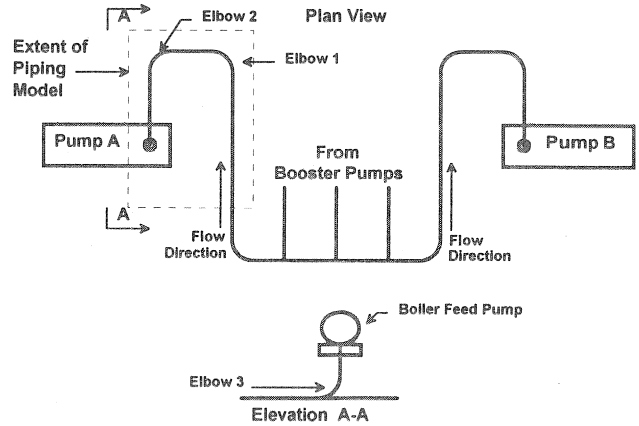


Figure 8. Suction Piping at TVA's Cumberland Station. This plan and elevation view of the suction piping approach for the 12 × 18 CA shows the combination of out of plane elbows and straight sections that cause swirl and distorted axial velocities at the pump suction flange. The extent of the model piping used to design the flow conditioners is shown.

A plot of the axial velocity contours at these positions in the intake piping is shown in Figure 9. The numbers are normalized to the mean axial velocity in the pipe. The top plot represents the condition right at the pump suction flange. The bottom is just upstream. Some mixing is occurring between these two sections because the axial velocity variation is becoming less severe. Swirl velocities are not plotted, but ranged from 5.6 to 22.0 degrees from the plane normal to the pipe centerline.

With these data, the user constructed a flow conditioner for use in the manufacturer's test program. This flow conditioner, constructed within a pipe suction and made up of swirl vanes and flow baffles, would impose swirl and distortions similar to those measured in the model piping. The conditioner was configured so it could be installed in the suction approach to the test pump. Before shipping the conditioner to the pump manufacturer, the TVA conducted a flow test to assure that the exit conditions were indeed similar to the model test.

EXPERIMENTAL VERIFICATION

Test Vehicle

At this stage in the development of biased-wedge impeller blading, the use of experimental verification was deemed necessary. This was done to assure proper functioning of this critical high energy impeller in terms of suppression of cavitation erosion and maintenance of suitable $NPSH_{3\%}$ performance. Accumulation of test cases will enable design and application of these impellers without confirmation testing. For this program, a fullscale model of the suction stage of the 12 × 18 CA was designed and built. Actual production tooling was used to fabricate the inlet suction bay, vaned diffuser and the test impellers. A cross section of this test rig is shown in Figure 10. In addition to achieving hydraulic similarity, the design incorporated a special, 360 degree viewing port designed and built into the

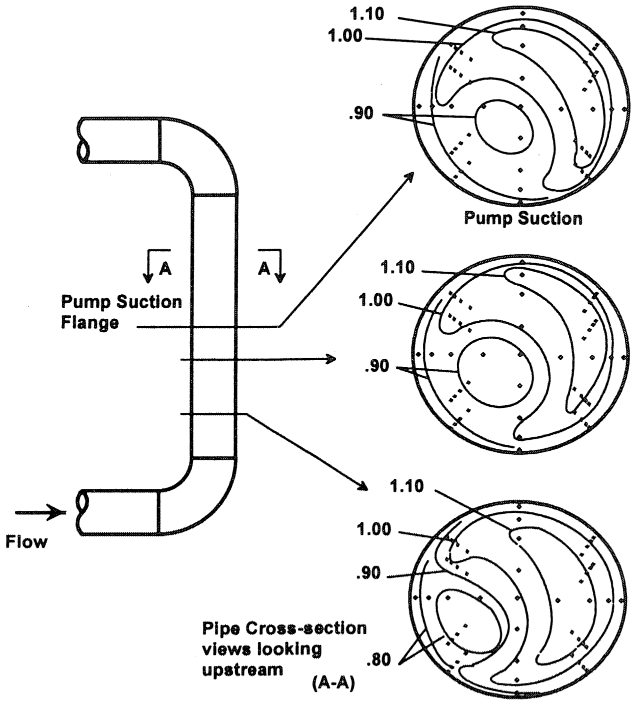


Figure 9. Representative Axial Velocity Variation at Pump Suction Flange. This set of plots show the variation in axial velocity at three locations, upstream of the pump flange. The data were obtained from model test of the fullsize piping. The variation in swirl angle varied from 5.6 to 22.0 degrees.

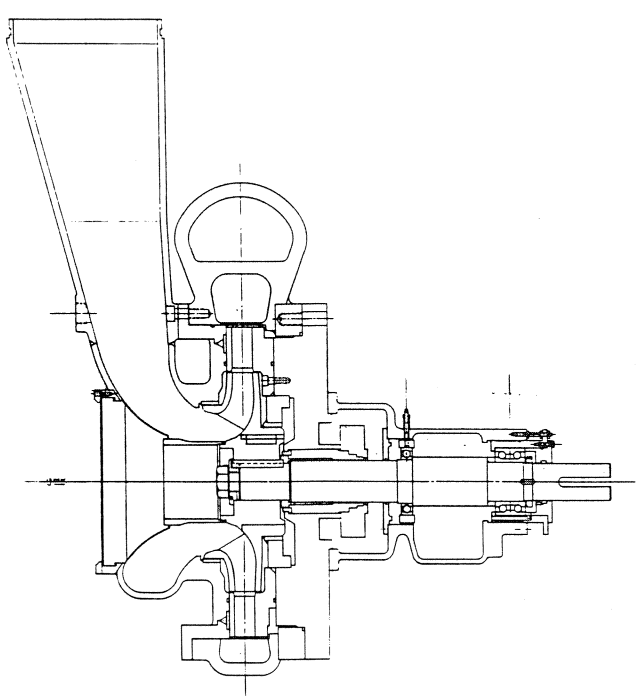


Figure 10. Test Rig Used for Flow Visualization Validation. This single stage pump is designed to incorporate full size, production type components for suction bay, impeller and diffuser. Unhindered visual access is provided to the impeller eye.

outboard side of the suction bay. The use of an overhung bearing cradle, from a commercial, end-suction process pump allowed for unhindered viewing capability.

The test rig was installed in a loop utilizing a 2000 gallon settling tank, deaeration and filtering capability, and inlet pressure control via an elevated pressure tank (maintained with either a vacuum or pressurized with air). Head breakdown in this closed loop was achieved via a special valve with cavitation resistant trim. Flow measurement was made via calibrated venturi. A photo of the test rig installed in this loop is found in Figure 11. The inlet pipe that connects the pump suction to the settling tank incorporates a honeycomb straightening section. The loop can be equipped with the flow conditioner, used to simulate the inlet piping distortions from the plant piping, by replacing one of the inlet sections of pipe with the section incorporating the flow conditioning vanes and baffles. Tests are run at a constant speed of 1500 rpm. Similarity with the plant conditions under cavitating conditions using the subscale test speed is maintained by operating the model at the same specific speed ($Ns = N\,Q^{.5} / H^{.75}$) and suction specific speed ($Nss = N\,Q^{.5} / NPSH^{.75}$) used in the plant (the variable speed of the drive turbine accounted for in the calculation of these conditions). Test work by Gulich [6] on the scale effects of cavitation has demonstrated that part speed tests yield excellent results as long as geometric and hydraulic similarity are preserved.

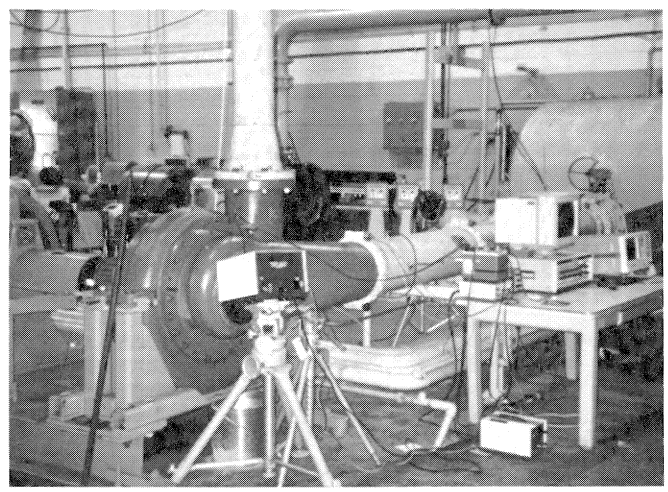


Figure 11. Test Rig Installation with Inlet Piping. The single stage, low speed test rig is installed in a closed loop system. The piping between the accumulator tank and the test pump is configured for either ideal, straight flow, or can incorporate a flow conditioner section that swirls and distorts the incoming flow at the pump flange.

Besides measuring the basic stage performance quantities (head, flow, power, NPSH), the flow visualization capability provides an opportunity to evaluate the cavitation formation and correlate this activity to damage patterns observed in the field. The visual record of cavitation activity in the test impeller reveals blade-to-blade inaccuracies, distortions in the flow field (from inlet design or possibly inlet piping configuration) causing circumferential variations in cavitation bubble activity, correlation of unsteady flow behavior with cavitation activity and finally, a means of observing and judging new impellers or modifications made to existing impellers. These modifications may take the form of manually applied leading edge contours, leading edge fairings, and development of blade thickness, which are important to biased-wedge designs. The modifications may be prompted from the analytical design studies or from ideas derived from the experimenters based on observation of previous tests.

Test Program—Conventional Impeller

In conducting the test program for the 12 × 18 CA, a series of impellers was evaluated and modified before arriving at the final design. The test program began with the conventionally designed, baseline impeller. Performance and NPSH data were collected, which scaled with test data from the production pump. A key part of the test program, was the collection of flow visualization information on the size and character of the cavitation vapor activity within the impeller at scaled operating conditions. The correlation of the cavity extent observed on the model with damage reports from the field was positive. Also observed were variations in circumferential cavity behavior, with the most vapor occurring in the region of negative preswirl. This was due to the negative swirl introduced by the suction bay.

The average bubble activity in regions of positive and negative preswirl for baseload and min-flow conditions is documented in the photos, Figures 12 and 13. The black lines marked on the suction surface of the blade are one-half inch apart. Since no damage to the pressure (or hidden side on these photos) side was reported, special techniques needed to view cavitation activity there were not used. The photos were taken of the same impeller

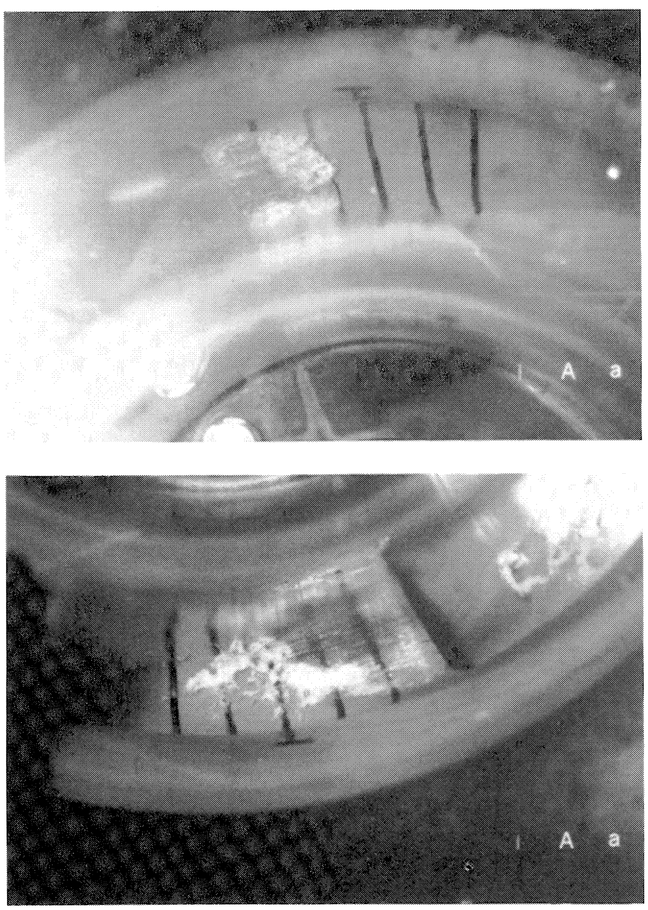


Figure 12. Cavitation on Conventional Impeller, Baseload Condition (106 percent of BEP Flow). These two photos, of the same impeller blade, show differences in cavitation bubble length attributable to distortions caused by the suction inlet bay. The top photo is taken in a region of positive preswirl, the bottom photo is from a region of negative preswirl.

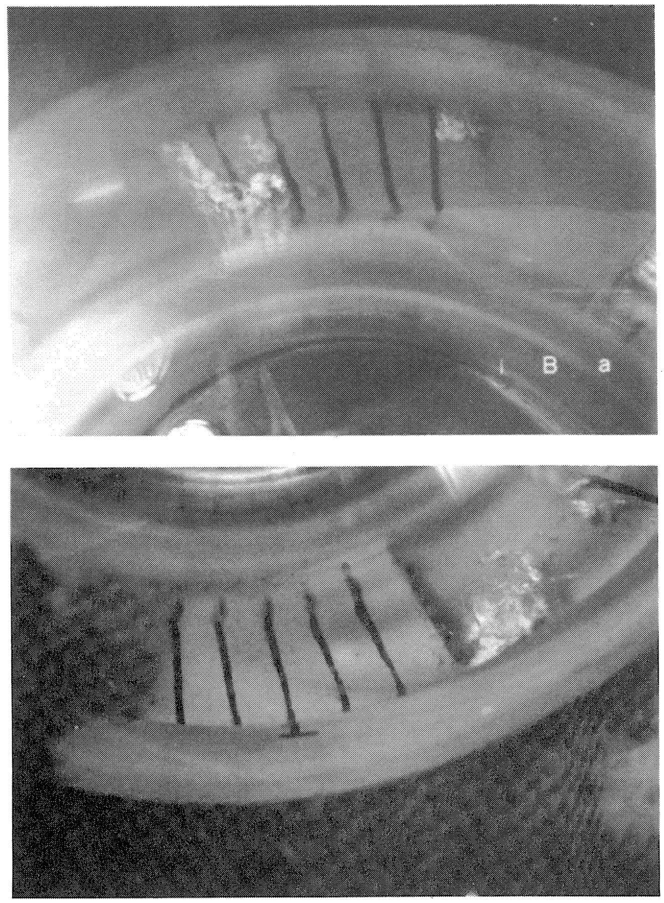


Figure 13. Cavitation on Conventional Impeller, Min-flow Condition (50 Percent of BEP Flow). These two photos are taken in the same locations as those in Figure 14. Distortions are not as easily detected at low flow due to effects of suction recirculation and much less incoming flow velocity.

blade, using a time delay strobe system to index the light flash to the reference blade position.

The baseline impeller was manufactured using a sand casting process, and some variability in cavitation activity on different blades was observed. The reference blade was chosen because it produced the average cavity length among all of the blades. At baseload flow (106 percent of bep flow at rated rpm) and inlet cavitation number τ (0.40) (Figure 12), there is noticeable variation between the top and bottom pictures in terms of cavity length, demonstrating the effect of presenting distorted flow to the impeller. The cavity length at this flow condition is relatively stable, with variations of about ± 14 percent of the average length. At min-flow (50 percent of bep flow at the reduced drive turbine speed and a cavitation number of 0.60), the effect of distortion is less obvious (Figure 13). The lower inlet velocities in the suction bay promote better mixing as the flow separates around the pump shaft, which results in better flow distribution and less tangential variation. The effects of suction recirculation probably influence the incoming flow at this condition as much as the distortions from the suction bay. The unsteadiness during recirculation, from the separated flow in the impeller, has an impact on the pressure distributions on the blade, and hence causes large variation in the cavity behavior.

It is common experimental technique to measure the cavity lengths on the impeller blade suction surface (and pressure surface if possible) for various flow and inlet pressure conditions. A map of such a test is shown in Figure 14. The plot is nondimensionalized by fraction of bep flow and the cavitation number, τ. The cavity length is also nondimensionalized using the impeller pitch at the eye diameter ($\pi D_e/Z_{blades}$). Nondimensionalizing the data in this fashion is desirable, because it can be compared with other pump designs and is more suitable for interpretation by analysts who are working on the design team [2, 3]. The plot is made for the reference blade, with the observations of cavity length made in a region of positive preswirl. Observations made in the negative swirl region would result in longer cavity lengths.

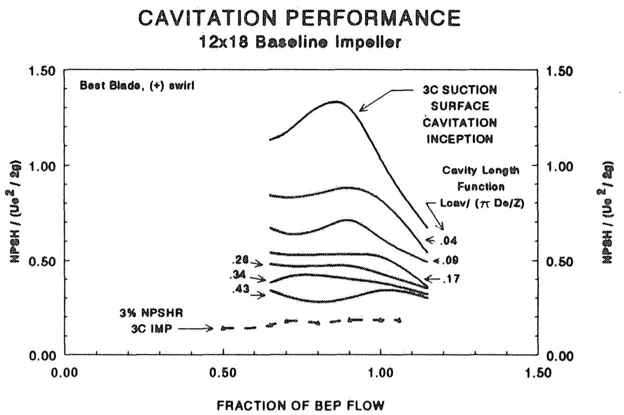


Figure 14. Cavity Length Map for Conventional Impeller. This mapping of cavity length, in nondimensional terms, is made for the reference blade in a region of positive swirl. Cavity length observations in regions of negative swirl would indicate longer cavity length for a given flow and cavitation number.

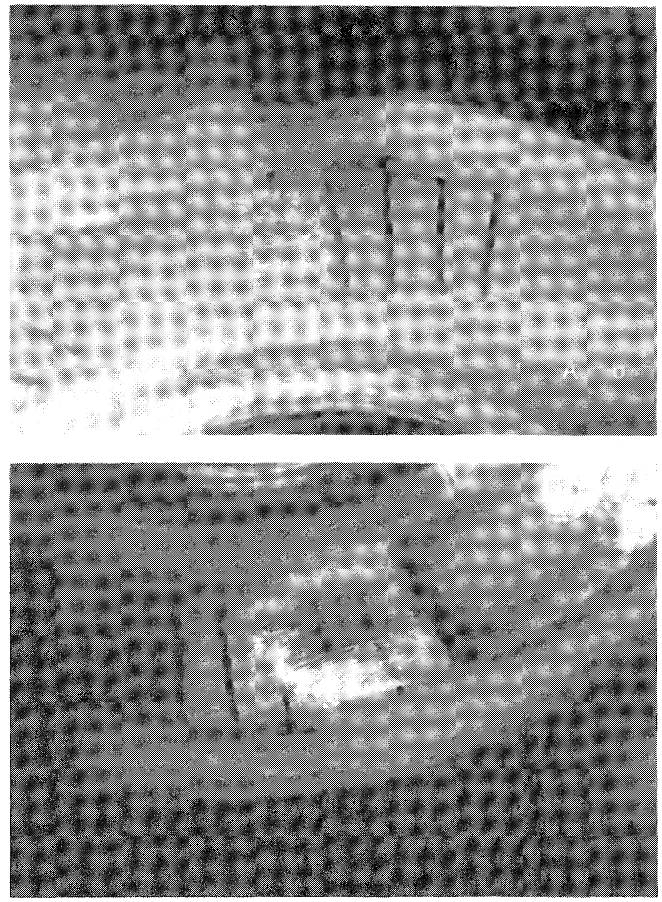


Figure 15. Cavitation Bubble Photos for Conventional Impeller (Baseload Condition) with Inlet Flow Conditioner in Place. These photos should be compared with Figure 12 (no flow conditioner). No discernible difference between bubble behavior is apparent.

The presence of flow distortions caused by the suction bay have been observed as circumferential cavity length variations (Figure 12) with undisturbed inlet piping flow. The inlet configuration used for this flow visualization test consisted of a straight pipe approach, with honeycomb flow straightener. Installation of the flow conditioner, meant to introduce swirl and flow disruptions at the pump flange similar to that caused by plant piping, would provide a means of evaluating the effect of imposing the pipe distortion onto the suction bay distortions. The photographs on Figure 15 are taken at baseload condition, with the swirl generator in place and should be compared to Figure 12. The result indicates no measurable difference in cavity length, and no discernible change in cavity stability. From this type of comparison, it can be concluded that the known velocity distortion at the pump flange at Cumberland plays no significant role in the cavitation erosion problem. This observation must be qualified by the fact that the manufacturer is using a single-suction first stage, with a significant acceleration from the inlet flange to the impeller eye. Distortions of the mass flow in a suction pipe may significantly affect the flow split in a double-suction type inlet bay. This flow split will result in one side of the impeller seeing different inlet flow conditions than the other, and so promote different cavitation activity and erosion potential.

The Biased-Wedge Impeller

A combination of experimental observation, computational analysis and fluid dynamical reasoning were used to arrive at the final blade configuration for the 12 × 18 CA suction impeller. Intermediate suction impeller designs were analyzed and tested with blade shape modifications. Correlation of these test results with analysis led to refinements in the computational technique, new designs analyzed, with final hardware being made and tested. The final shape was transferred via stereolithography from a three-dimensional wireframe representation, to a surfaced three-dimensional solid model, to a plastic blade pattern, to a soluble pattern and finally to a wax positive of the new impeller. The impeller pattern was then invested, the wax burned out, and the casting made. The test program concluded with a model test of a production impeller for performance and cavitation documentation.

The reduction in cavitation activity on the impeller was dramatic. Photos are presented (Figures 16 and 17) of the suction surface of the final impeller taken at identical positions and operating conditions (baseload and min-flow) as in Figures 12 and 13. In addition, the flow conditioner used to simulate the inlet piping flow was also installed (Figure 15 compares baseload condition with flow conditioner for the conventional impeller). The large reduction in vapor volumes at the min-flow condition (Figure 17) also resulted in a notable reduction in vibration and roughness in the test pump. The other notable feature is the tolerance of the blade design to the inlet generated flow distortions. These distortions produced the varying cavity activity observed on the conventional impeller (Figure 12). These distortions did not change with the different impeller,

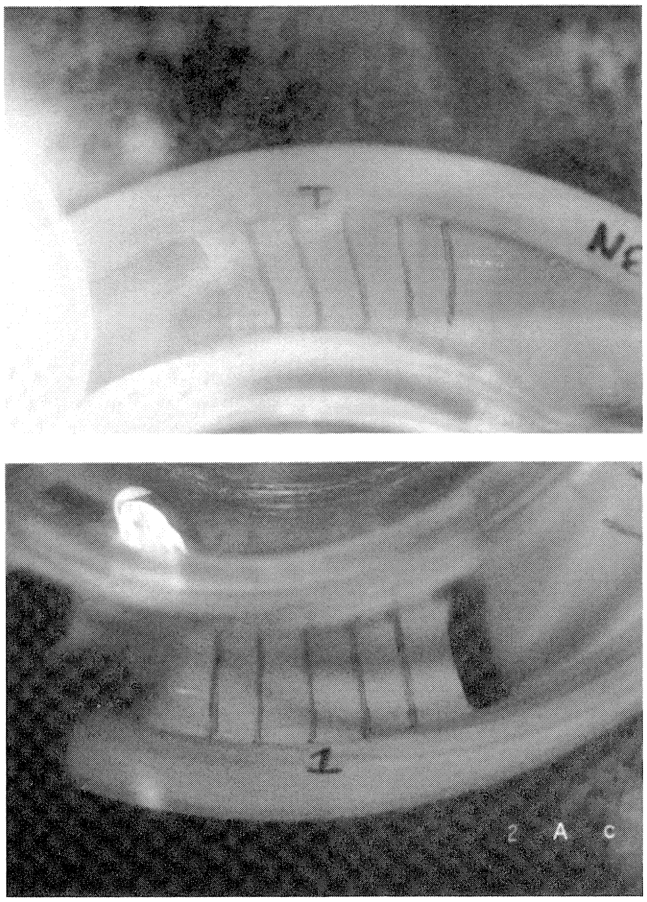


Figure 16. Absence of Cavitation on Biased-Wedge Impeller, Baseload Condition (106 Percent of BEP Flow). These two photographs taken in regions of positive swirl (top) and negative swirl (bottom), indicate the improved pressure distribution and the tolerance to varying inlet flow distortions and swirl conditions at the inlet.

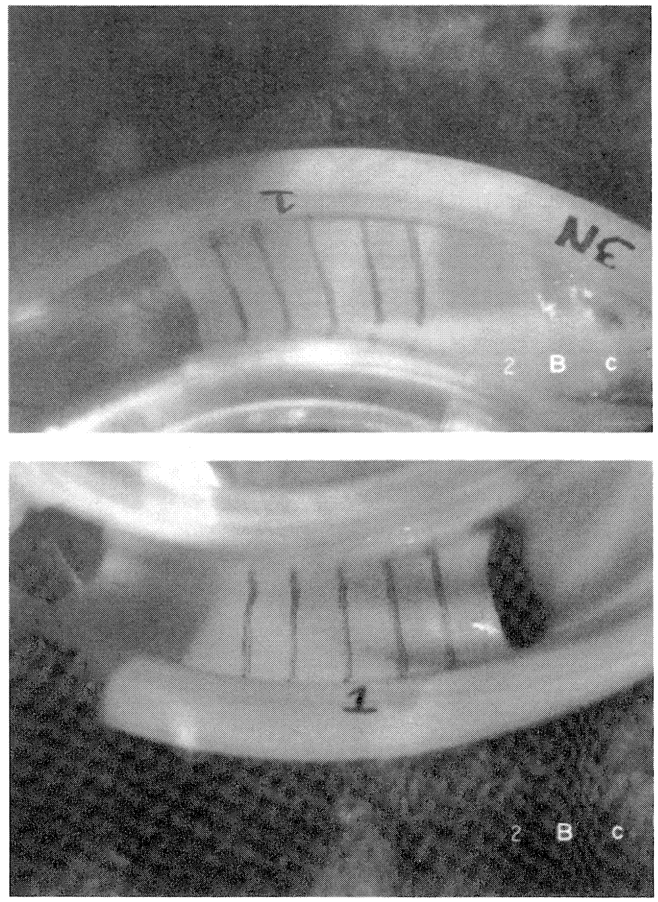


Figure 17. Biased-Wedge Impeller Operating at Min-flow Condition (50 Percent of BEP). Reduced cavitation activity, even under recirculating conditions, is observed with the improved blade shape. This reduction in vapor activity not only improves life, but improves operability as well.

since the same piping and suction configuration were used. It is apparent that the biased-wedge blade design was robust enough to accommodate the circumferential flow variation (and hence incidence angle) without suffering local pressure reductions below the fluid vapor pressure.

A comparison of the inception $NPSH_i$, as measured by the dimensionless, cavitation number τ, is shown in Figure 18. Just like the map in Figure 14, the observations were made on the same blade, as it passed through the region of positive swirl. The suction surface inception for the improved impeller is plotted back to 80 percent of bep flow. Below this flow, the onset of recirculation affects the bubble formation on the impeller blade, and characterizing the cavitation in terms of suction surface inception or cavity length is impractical. The testing also indicated a small increase in $NPSH_R$ over the baseline impeller. The increase is negligible and poses no threat to the operational integrity of the machine. This successfully ended the test program for the biased-wedge impeller. Installation in the four pumps at Cumberland followed.

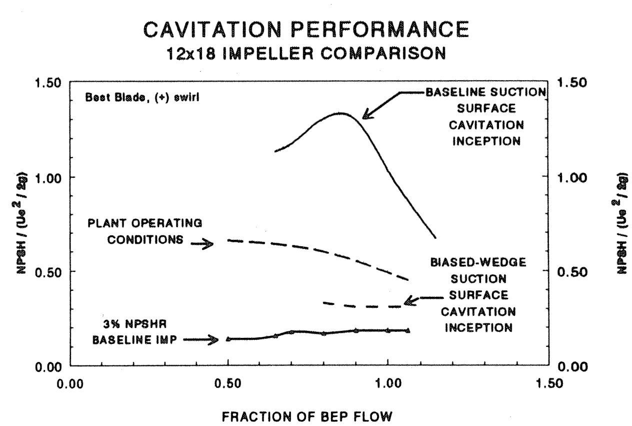


Figure 18. Cavitation for Biased-Wedge Impeller. This map is similar to the Figure 14 (for the conventional design). The reduction in inception cavitation number below the available NPSH means bubble free operation over a wide flow range. The impeller is also less affected by inlet distortions.

CONCLUSIONS

The authors have described how one pump user and manufacturer cooperated on a major development program that is extending the service life of a high energy, boiler feed pump suction stage. This stage, by any measure, qualifies as a high energy pump, and as such, must have careful attention given to all elements of its hydraulic and mechanical design. Complicating the impeller design was the generation of flow distortions at the impeller eye, inherent in the suction bay itself and possibly the inlet piping configuration. Cavitation damage had limited life to about 18 months in the final iteration of the conventional impeller design. Due to the high inlet tip speed of this single-suction impeller (281 ft/sec), little could be done to further optimize the conventional blade shape. A more radical approach was called for to improve the life expectancy of this impeller design.

Previous experience with profiling blade shapes to reduce cavitation activity had shown promise in related applications. For this case, an integrated design approach was used, combining computational analysis of the flows and pressures with experimental test and observation on a full scale, part speed test rig. The outcome was an optimized design that incorporated all of the features inherent in the biased-wedge approach to blade design. The biased-wedge design was essentially bubble free over most of the desired operating range. This reduction in vapor activity was accomplished even with the flow distortions produced by the suction inlet bay. Flow distortions from the inlet piping (modelled with specially constructed flow conditioners by the user) were found not to influence the cavitation in this single-suction configuration. Resorting to alternate metallurgy (with the possibility of compromised mechanical integrity) was not necessary, since practically all of the vapor had been removed via the hydraulic design. Improved low flow operating characteristics were expected because of this reduction in cavitation as well.

Utilization of new prototyping technology facilitated the transfer of hydraulic design of the biased-wedge impeller to final part. Stereolithography was used to generate a blade pattern and a lost wax, precision investment casting process to produce the final impellers. Suitable dimensional accuracy was achieved, as demonstrated in the flow visualization testing. The critical nature of the biased-wedge shape required these special techniques to be used in order to preserve accuracy of design and avoid compromising performance in the field.

ACKNOWLEGEMENTS

The authors wish to thank the management of the Tennessee Valley Authority and Ingersoll-Dresser Pump Company for their support in completing this program and for permission to publish this paper.

REFERENCES

1. Cooper, P. and Antunes, F. F., "Cavitation in Boiler Feed Pumps," Symposium Proceedings on: Power Plant Feed Pumps—The State of the Art, EPRI CS-3158, Cherry Hill, New Jersey, Session 2, pp. 24-29 (1982).
2. 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, Texas A&M University, College Station, Texas, pp. 3-19 (1991).
3. Cooper, P., Sloteman, D. P., and Graf, E., "Design of High-Energy Pump Impellers to Avoid Cavitation Instabilities and Damage," Proceedings of EPRI Symposium on: Power Plant Pumps, Tampa, Florida (1991).
4. Cooper, P., Wotring, T. L., Makay, E., and Corsi, L., "Minimum Continuous Stable Flow in Feed Pumps," Symposium Proceedings, Power Plant Pumps, EPRI CS-5857, New Orleans, Louisiana, pp. 2-97—2-132 (1988).
5. Palgrave, R., and Cooper, P., "Visual Studies of Cavitation in Pumping Machinery." *Proceedings of the Third International Pump Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 61-68 (1986).
6. Gulich, J. F., "Guidelines for Prevention of Cavitation in Centrifugal Feed Pumps," EPRI GS-6398 (1989).
7. 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, Texas A&M University, College Station, Texas, pp. 21-27 (1991).
8. Schiavello, B. and Prescott, M., "Field Cases Due to Various Cavitation Damage Mechanisms: Analysis and Solutions," Proceedings of EPRI Symposium on: Power Plant Pumps, Tampa, Florida (1991).
9. Schiavello, B., "Tutorial 3, Cavitation and Recirculation Field Problems," *Proceedings of the Ninth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1992).
10. Dhaubhadel, M. N., Akay, H. U., Singh, P. J., Spring, H., and Prescott, M., "Three-Dimensional Finite Element Analysis of Flow Through a Double Suction Pump Bay," Pumping Machinery—1989, ASME, FED *81*, pp. 237-245 (1989).
11. Arisawa, T. and Schiavello, B., "Visual Study of Cavitation—An Engineering Tool to Improve Pump Reliability." Proceedings of EPRI first International Conference on: Improved Coal-Fired Power Plants, Palo Alto, California (1986).
12. Dernedde, R., and Stech, P. R., "Design of Feed Pump Hydraulic Components," Symposium Proceedings: Power Plant Feed Pumps—The State of the Art, EPRI CS-3158, Cherry Hill, New Jersey, pp. 11-23 (1982).
13. Akashi, K., Watanabe, H., and Koga, K., "Flow Rate Measurement in Pipe Line with Many Bends," Technical Review, Mitsubishi Heavy Industries, Ltd. (1978).
14. Florjancic, D., "Mindestzulaufhoehen fuer speisepumpen (Minimum Suction Condition for Feed Water Pumps)," VGB Kraftwerkstechnik 60, Heft 12 (1980).
15. Makay, E. and Barret, J. A. "Ten Ways to Improve High-Energy Pump Performance," Power Magazine, pp 37-48 (1988).
16. Stepanoff, A. J., *Centrifugal and Axial Flow Pumps: 2nd Edition*. New York, New York: John Wiley & Sons (1967).
17. Burton, C. and Willison, R. "Application of Model Studies to Industrial Gas-Flow Systems," ASME Paper No. 59-A-280 (1959).
18. Patterson, R. and Abrahamsen, R., "Flow Modeling of Furnaces and Ducts," ASME Journal of Engineering for Power (1962).
19. Murakami, M. and Shimizu, Y., "Hydraulic Losses and Flow Patterns in Pipes with Two Bends Combined," Bulletin of Japanese Society of Mechanical Engineers, *20* (1977).