CFD ANALYSIS OF A DOUBLE SUCTION COOLING WATER PUMP

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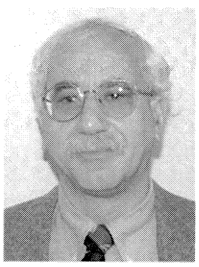

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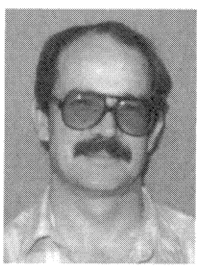

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ABSTRACT

In 1991, four large double suction cooling pumps were installed at a chemical plant in Texas. Although these pumps met performance specifications on the test stand, they proved to be noisy when installed. Sound power levels greater than 93 dbA were observed. Pumping applications involving cooling water have been especially difficult to solve over the years, due to the presence of dissolved air inherent in a cooling tower sump. Water that contains large amounts of dissolved air changes the apparent required net positive suction head (NPSH). In such applications, traditional correction techniques failed because the entire system was not analyzed, and the source of the noise generation could not be pinpointed. This paper deals with the steps involved with obtaining a solution to this problem.

A computational fluid dynamics (CFD) analysis was performed on the pump and impeller. Initial investigation revealed that the impeller was not the source of the problem. The casing inlet was then analyzed. The model indicated the presence of separation in the suction nozzle, causing a flow distortion at the impeller eye. A unique guide ring was developed to minimize the flow separation. The CFD analysis was revised to reflect the new geometry, and a significant improvement in the flow was predicted. A prototype guide ring was manufactured and installed in one of the four pumps. Pressure profiles were obtained by experiment to validate the analyses. The results are discussed.

INTRODUCTION

Noise is a chronic problem with many double suction cooling water pump installations. Noise has proven itself to be a very difficult problem to analyze and cure, especially since the degree of noise generated depends upon the pump design, the system it is installed in, and the fluid it is pumping.

Single stage double suction horizontally split pumps are particularly susceptible to noise problems because of the way they are designed. The design of the pump starts with a suction chamber that wraps around a portion of the discharge volute. Ideally this chamber should uniformly guide the liquid into the impeller eye. But as the flow enters the suction chamber, it splits at the discharge volute and undergoes a series of turns as it approaches the impeller (Figure 1). This is analogous to flow through a series of elbows. Consequently, a nonuniform velocity/pressure distribution is imposed on the impeller inlet. Cooling tower water service is a very aggressive application and often turns normally well behaved pumps into "bad actors."

This paper deals with a unique noise problem and the techniques employed to solve it. Four large double suction cooling pumps

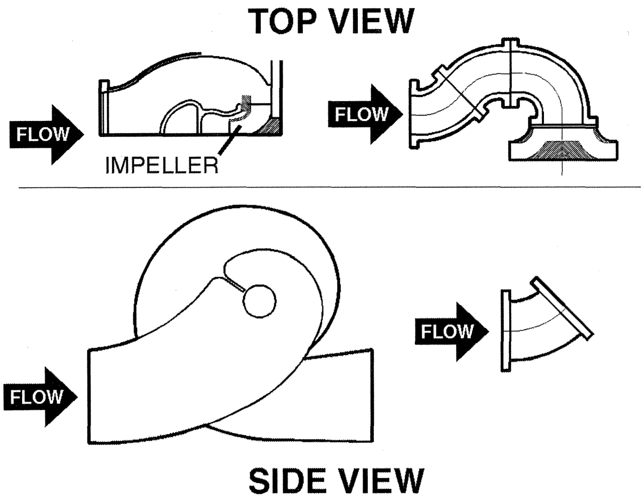

Figure 1. Double Suction Inlet Analogy.

were installed at the Dow Chemical plant in Freeport, Texas. The pumps in question are four 30 × 30 × 38 double suction cooling tower pumps operating at 36,000 gpm and 710 rpm operation. The pumps were factory tested and met all contract requirements. The testing was carried out under ideal conditions, so noise problems were never discovered. However, the pumps were noisy upon startup in the field. Sound power levels greater than 93 dbA were observed at a distance of approximately three feet from the pump casing.

The subject cooling water pumps were operating under duress as indicated by noise and other signs. The impeller was removed after 12 to 18 months of service and some cavitation damage was evident. The casing top half was also inspected and a small area of pitting was observed near the stop piece.

BACKGROUND

The degree of noise generated by a pump depends upon the pump design, the system it is installed in, and the fluid it is pumping. To understand how design affects noise, one must first look at what is happening internally within the pump. A centrifugal pump is designed to meet a particular head-capacity range. Within this range, the pump obtains its maximum efficiency at a given (design) capacity, commonly known as the best efficiency point (BEP). The hydraulic passages in a well designed pump are optimized for operation at this capacity. Consequently, the pump will operate with the lowest noise, vibration, and pressure pulsation levels at the design head-capacity. At capacities other than the design capacity, the impeller angles no longer approximate the fluid angles. Operation at lower or higher capacities than BEP causes flow separation to occur at the impeller vanes (Figures 2 and 3) and at casing throat (Figure 4). Raising or lowering the capacity away from the BEP causes the vibration, noise and pressure pulsation levels to increase (Figure 5).

An interesting phenomenon usually, but not always, occurs at lower capacities than the BEP. The slope of the noise/pressure pulsations curve changes abruptly (Figure 5). The capacity at which this occurs is the point of recirculation or a flow reversal occurring within the impeller. This occurs when the centrifugal pressure balances the dynamic head (Fraser, 1981). Noise, vibration, and pressure pulsations are usually at a maximum in the recirculation region. The point of recirculation can occur at capacities greater than BEP for impellers that are designed with high suction specific speeds (Fraser, 1982, Figures 6 and 7).

For a typical pump the noise/pressure pulsation curve will also tend to rise at capacities higher than BEP (Figure 5). As the capacity increases, the margin between the NPSH available and the NPSH required gets smaller. NPSH testing demonstrates that noise

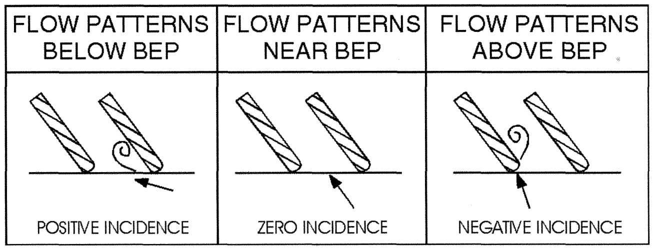

Figure 2. Typical Inlet Flow Incidence.

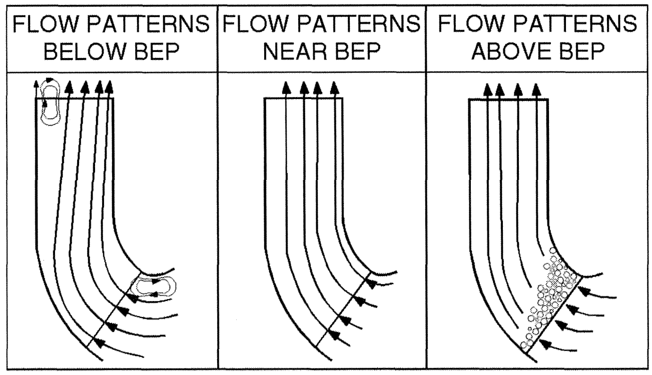

Figure 3. Typical Flow Patterns within Impeller.

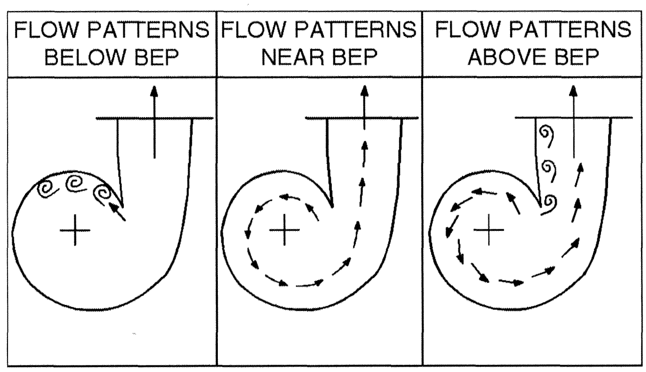

Figure 4. Typical Flow Patterns within Casing.

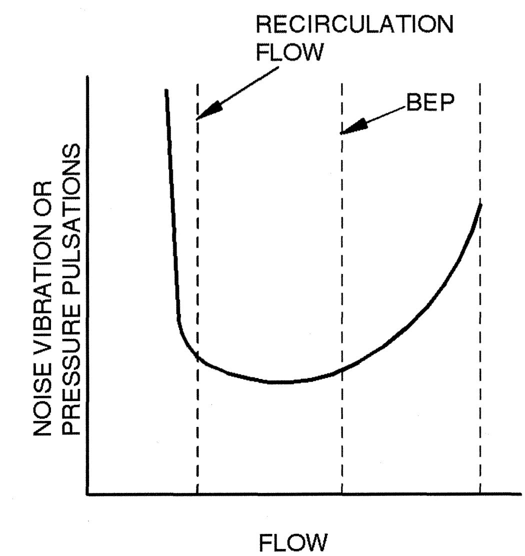

Figure 5. Typical Noise, Vibration, and Pressure Pulsation Versus Capacity.

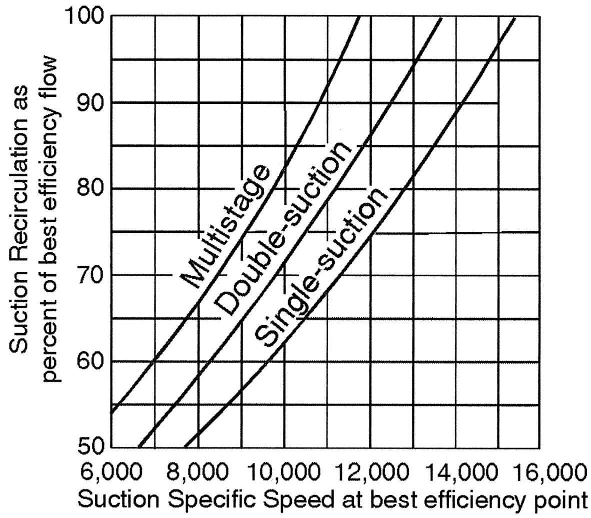

Figure 6. Recirculation Flow for Specific Speeds from 500 to 2,500.

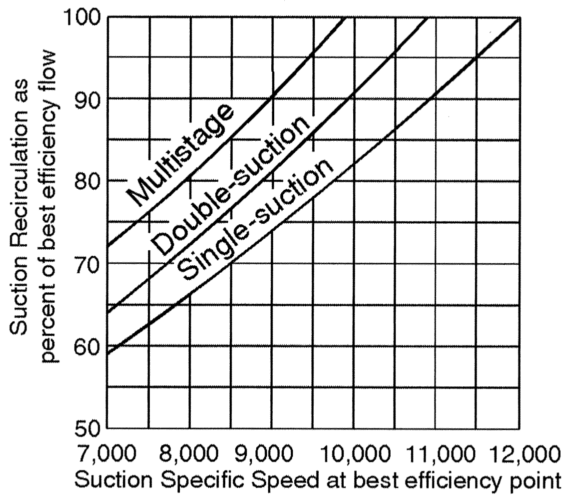

Figure 7. Recirculation Flow for Specific Speeds from 2,500 to 10,000.

and pressure pulsations start to rise before any noticeable head drop due to cavitation occurs (McNulty and Pearsall, 1982, Figure 8). The peak values of noise have been documented to occur even when the $NPSH_A$ is many times (2 to 15) the $NPSH_R$ (McNulty and Pearsall, 1982; Vlaming, 1981).

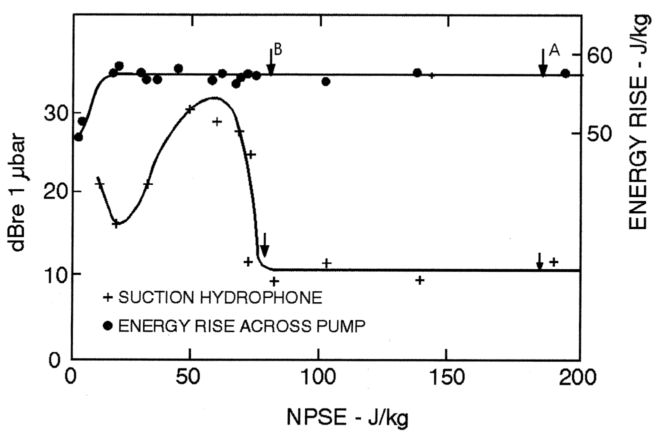

Figure 8. Typical High Frequency Noise.

Pump design features affecting noise/pressure pulsation are:

- Cutwater clearance—influenced by the impeller to casing distance, and by the shape and number of impeller vanes passing the casing tongue (Figure 9).
- Shape of the suction collector passage. If the flow makes an abrupt change in direction as it enters the eye of an impeller, separation occurs (Figure 10).
- Nonuniform supply to the impeller (Figure 1). Double suction pump casings, because of the nature of their design, impose an asymmetrical fluid velocity distribution entering the impeller eye.
- Cavitation at the wear rings (Figure 10).
- Position and contour of double suction inlet stop piece (splitter) (Nelik and Freeman, 1996, Figure 11).
- Rotor imbalance can cause a low frequency rumble. Also, as the pump vibrates, bearing and seals can wear, which also leads to increasing noise problems.

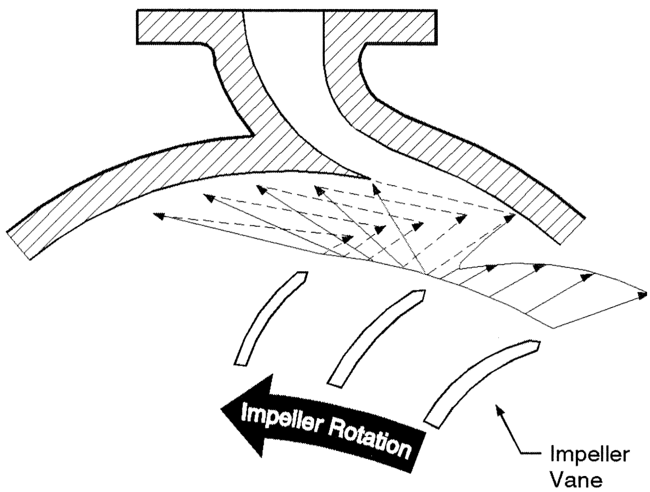

Figure 9. Absolute Velocity Distribution at Impeller Exit.

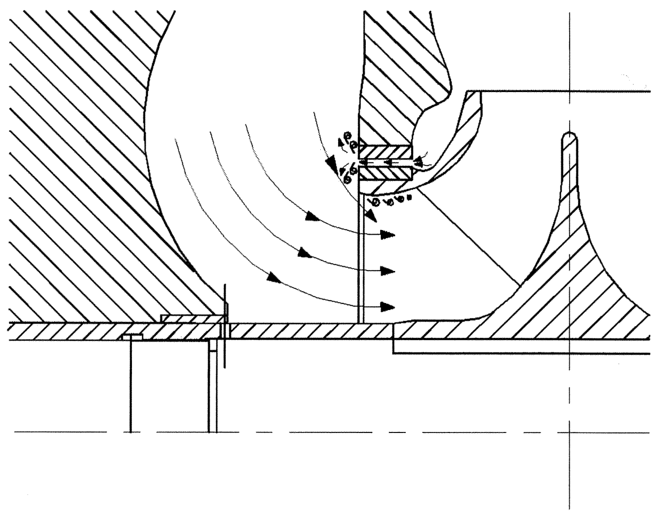

Figure 10. Flow Separation at Impeller Eye and Cavitation at Wear Ring.

The system in which the pump operates is often a very significant factor in a chronic noise situation. Noise excited by the installation is affected by many factors that include, but are not limited to, the following:

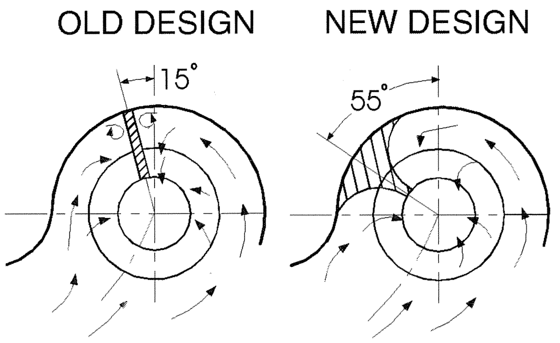

Figure 11. Comparison of Stop Piece Design in a Horizontally Split Double Suction Pump.

- Mechanical
  - Baseplate resonance
  - Poor foundation design
  - Piping support
- Piping/Sump
- Prerotation of the fluid entering the pump, caused by poor intake or poor inlet piping design, produces a low frequency rumble.
- Valves cavitating cause high frequency noise.
- Velocity of the fluid within the pipes can excite resonances and low frequency noise (pipe organ effects).
- Horizontal elbow on DS pump suction. Nonuniform velocity/pressure distribution from side to side of double suction (Figure 12). In very extreme cases, cavitation damage has been observed on one side of impeller and recirculation damage on the other. Similar problems occur when valves are incorrectly placed near the pump suction (Sulzer, 1989, Figure 13).
- Driver/Electrical noise
  - High pitched noise caused by the electromagnetic fields exciting alternating forces on the motor's rotor and stator
  - Variable frequency drives can also produce a high frequency noise caused by harmonics.
  - Noise emanating from fan cooled motors

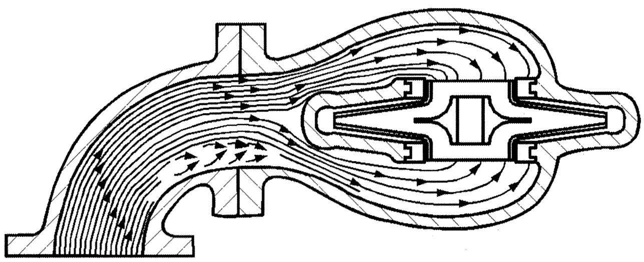

Figure 12. Effect of Elbow Directly on Suction of a Double Suction Pump.

Noise and its associated damage is a function of the thermodynamic properties of the fluid pumped (Stepanoff, 1965). Cold water (30°C) is the liquid that exhibits the most noise when cavitation occurs (Knapp, et al., 1970). Unfortunately, this is the same temperature range found in most cooling water sumps.

Noise is also dependent upon the air content of the pumped fluid. Water collected at the base of cooling towers contains relatively large amounts of entrained air. Naturally, the amount of air released depends upon the design and placement of the sump, the approach velocity of the liquid in sump, the degree of vorticity in the sump, the quantity of liquid in the sump, and the temperature of the liquid in the sump. Much of the time this air tends to diminish the effect of cavitation, lowering the noise and any subsequent damage. But sometimes too much air has a detrimental effect on the $NPSH_R$ (Knapp, et al., 1970). It is not uncommon to see an increased $NPSH_R$ of 2 to 3 ft with increasing air content (Sulzer, 1989, Figure 14). Pumps operating with low margins of $NPSH_A$ to $NPSH_R$ (typically less than 2 ft) and highly aerated liquids tend to cavitate.

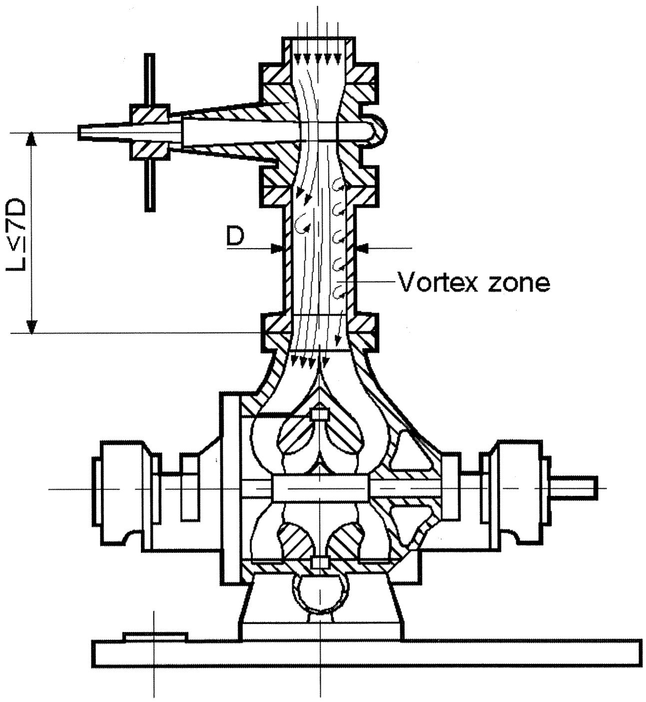

Figure 13. Effect of a Badly Fitted Valve in the Pump Inlet Line.

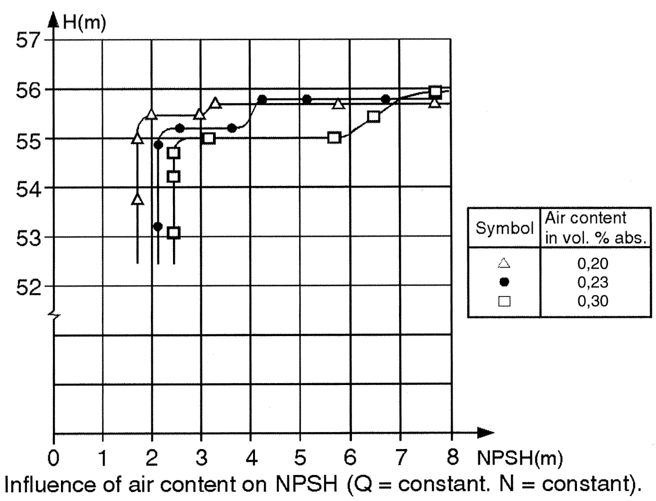

Figure 14. Influence of Air on NPSH.

Finding the root cause of cooling water pump noise problems usually leads to its cure. The most common cures involve either or both of the following:

- Impeller redesign/replacement in order to optimize the impeller's blade incidence, recirculation, or cavitation characteristics.
- The addition of bull rings or other similar devices has been effective in reducing the suction recirculation effects, provided that they do not adversely affect the pump's NPSH required.

If noise problems are caused by the system or installation design, cures can get to be very difficult to initiate. It is usually more desirable to change impeller material than system piping. Because of the aggressive nature of cooling tower water, fixing only one problem may not lead to an over all cure for the noise problem.

The only way to understand and cure the root causes of the problem is to use the best analytical tools available.

METHOD OF ANALYSIS

Experienced engineers often offer solutions to noise problems, based on simple techniques that have worked in the past. These solutions are sometimes effective but only partly understood. The combined effects of suction passage geometry, impeller inlet design, system piping design, and the state or condition of the system fluid make it very difficult to comprehend the real contributors to a noise problem. For the pump described by this paper, a couple of these "traditional" techniques had been applied, but with little success. The situation evolved such that all factors had to be understood as best as possible. The operating pump was examined in the field and the variables external to the pump were not seen as potential contributors to the noise problem. The approach then became an investigation of what was happening inside the pump.

In order to establish a good understanding of the fluid behavior within the pump, a computational fluid dynamics (CFD) model was developed. CFD codes are evolving into useful engineering tools that can predict fluid behavior within almost any geometry. Even if the fluid condition and system effects are not understood, a CFD model can provide a best case scenario, whereby the pump casing design and impeller design can be evaluated. If the analysis indicates that the pump will perform poorly under ideal conditions, then one would expect it to perform even worse with real life conditions. A less than ideal fluid or a less than perfect inlet design is sure to lead to problems. With little control over the condition of the fluid in most cases, the pump design should be optimized to perform to the best possible level to minimize its own contribution to problems in a difficult service. CFD analysis was the best available tool to evaluate the current pump design and predict the effect of modifications to the design.

MODELLING

The modelling began by first creating a three dimensional CAD model of the suction inlet portion of the pump casing based on the original pattern drawings. The CAD model was then imported into the CFD package and a mesh was created within the fluid space. Note that the discharge volute of the pump was not modelled. According to measurements taken in the field, the source of noise was confined to the suction. Therefore, the discharge region was not regarded as a trouble area. The impeller was modelled by starting with digitized data taken from an original blade pattern using a coordinate measuring machine. Surfaces were then created in CAD from the digitized data and additional surfaces for the hub and shroud were added based on drawings. The CFD package created the mesh in the fluid space between vanes.

The software used to create and solve the models was TASCflow®. The suction inlet model required approximately 230,000 nodes to create a fine mesh of hexahedral (brick) elements. A standard k-ϵ turbulence model was used, with wall boundary conditions applied at the casing surfaces. The solution of this model took roughly 20 hours to complete on a mid-range workstation.

The working fluid of the model was water with nominal room temperature properties. Three flowrates were examined based on the typical operating range of the pump:

- The best efficiency point (BEP)
- 50 percent BEP
- 120 percent BEP

The boundary conditions were selected to best represent the conditions surrounding the pump in the field. The field installation was relatively simple (Figure 15). A short length of pipe and a diffuser connected the suction flange to a wall of an open sump. A positive head exists at the suction centerline. Since the entrance velocities were relatively low and there was nothing unusual in the suction piping, the boundary condition at the suction flange was specified as a uniform total pressure. The magnitude of the static pressure throughout the model would be relative. The results could later be adjusted by any known or assumed static pressure at any given point. The boundary conditions on the downstream (impeller) end of the model would later come from a study of the impeller by itself.

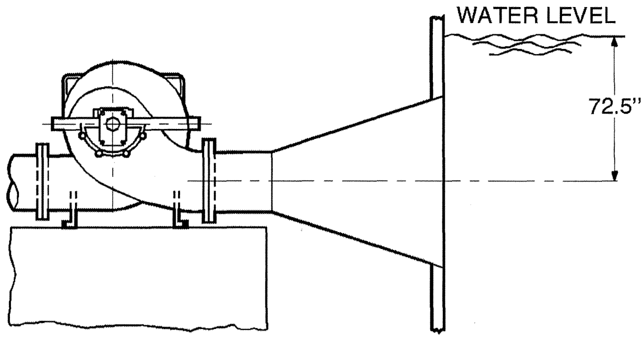

Figure 15. Pump Installation.

The casing (suction inlet), wearing ring, and impeller were intended to be modelled together, but as a preliminary study the impeller was modelled separately to isolate its influence on the pumped fluid.

INITIAL RESULTS

Pressure plots and flow vectors were used to examine the flow between the blades, hub, and shroud of the stand-alone impeller model. Multiple cuts or view planes were made in an effort to identify any anomalies within the flow. Neither backflow nor separation of flow was identified within impeller passages. Recirculation would have been identified by backflow near the blade inlets if it were occurring; none was observed. Overall, the analysis of the impeller showed nothing unusual.

Since the impeller was well behaved in the initial analysis, it was ruled out as a problem source. The remainder of the analysis focused on the suction inlet, with the volume between impeller hub and shroud included in the model to provide downstream effects. An angular momentum term was derived from the impeller analysis for the flow entering the impeller eye. Hence, this would be used as a downstream boundary condition in the inlet analysis.

CASING RESULTS

The results of the casing analysis were interpreted graphically by creating plots on several key planes that pass through the model. These plots display static pressure, total pressure, velocity vectors within a plane, and magnitude of velocity components perpendicular to a plane. There are two major planes of interest. The first is perpendicular to the shaft and just outside the impeller eye. The second is parallel to the shaft and passes through its center, while lying at a 51 degree angle from the horizontal centerline of the impeller (Figure 16). The first reveals information about the flow as it enters the impeller. The second displays the profile of the wearing ring, impeller, and a nearly central portion of the suction volute, and reveals information about how the flow approaches the impeller.

The magnitudes of velocity components entering straight into the impeller eye, just upstream of the impeller are shown in Figure 17. In an ideal pump, this type of plot should display one uniform

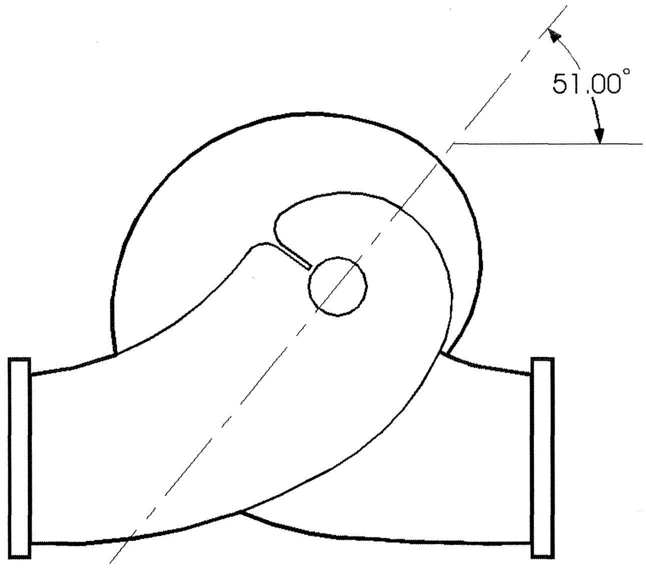

Figure 16. Section Cut Used for Results Plot.

flow field. This analysis, however, indicates variation in inlet velocity in both radial and circumferential directions. Radial variations imply that flow entering between impeller blades will have a different speed near the hub than near the shroud. Circumferential variations imply that at any given instant one impeller blade will be loaded differently than the next. Note from the lower region of the plot that there are some negative velocities. Some of the flow is actually exiting the impeller in this plane. This indicates that a certain amount of backflow is present. Similar plots (not shown) were created for the 50 percent and 120 percent BEP flow conditions. The flow variations were much less severe at the lower flow, but worsened at the higher flows.

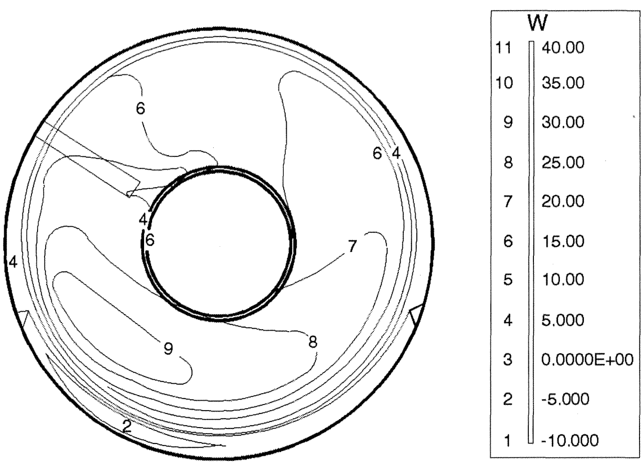

Figure 17. Original Geometry, Magnitude of Flow into Plane (Ft/S).

The static pressure variations in the same plane can be seen in Figure 18. Here also, there are both radial and circumferential variations. One particular point of interest is the localized pressure zone found near the splitter. This location corresponds to a localized area of pitting (cavitation damage) found on the top half of the casing. This discovery lent some credibility to the accuracy of the analysis prior to any experimental verification.

Further upstream of the impeller, the velocity distribution in the suction passages is depicted in Figure 19. Due to the location of the cut plane, the right side of the figure includes the flow traveling from the suction nozzle, while the left side of the figure includes

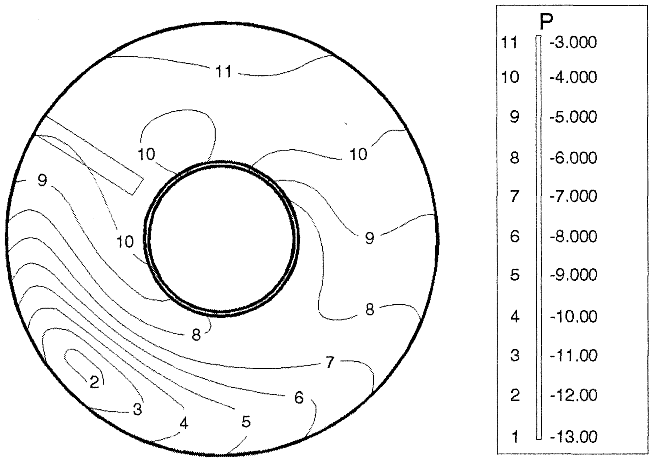

Figure 18. Static Pressure (PSI) near Impeller Eye.

only flow that passes around the casing wearing ring and enters the impeller from the opposite side. The inner wall of the suction nozzle has a deep valley that lies just upstream of the wearing ring. This profile is created by the wall of the discharge volute that wraps around the impeller and passes through the suction passage. The valley is formed by the blending of the impeller housing and the discharge volute. Some of the flow entering the pump tends to follow the contour of this valley, but then must climb back out on its way over the wearing ring. This redirection of flow causes some vortexing in the valley region.

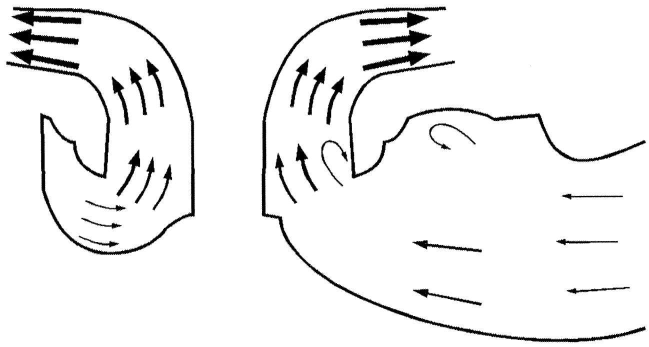

Figure 19. Velocity Distribution in the Suction Inlet.

For the portion of flow that follows this contour and does continue over the wearing ring there is yet another vortex within the impeller. The inside profile of the wearing ring creates a sharp discontinuity in the flow boundary. As a result, the fluid momentum carries the high velocity fluid over the discontinuity, but viscous forces cause a small portion of it to slow down and turn back into a somewhat stagnant region. Thus, the vortex is formed. The net effect is clearly visible in Figure 20, which is a plot of static pressures (psi relative to suction pressure). An isolated area of low pressure can be identified in the eye of the impeller. This is undoubtedly a region of cavitation when the NPSH available is low.

Thus far, the CFD analysis of the pump casing had revealed that the suction inlet design is less than optimal. The most significant conclusion at this stage of the project was that cavitation was likely to occur irrespectively of the impeller design. In other words, noise could not be eliminated by redesigning the impeller and any efforts to do so would be foolish. The impeller was not the root cause of the noise problem.

PROPOSED SOLUTION

Once the major trouble spots were identified, the next step was to find a way to alter the flow to significantly reduce cavitation.

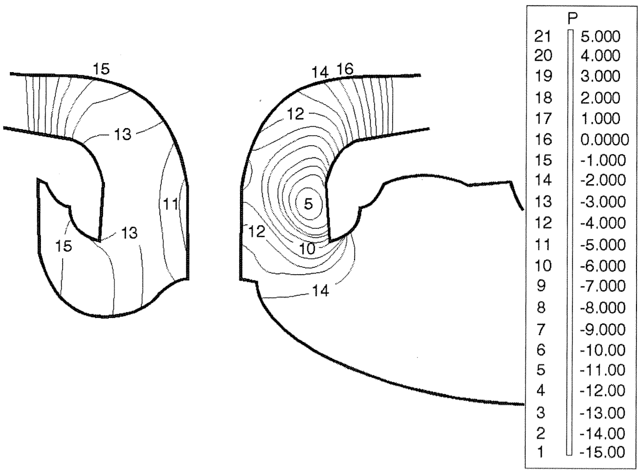

Figure 20. Static Pressure in the Suction Inlet.

The solution was based on two factors:

- A desired ability to upgrade the pump in the field
- The location of a majority of flow disturbances near the casing wearing ring

The casing wearing ring is a replaceable part in this double suction pump design, so this became the component that would be altered to smooth the flow.

The radius on the inside of the ring was increased significantly to avoid separation of flow as the fluid accelerates into the impeller eye. In addition, large tabs or ears were added on a portion of the ring circumference facing the suction nozzle. The desired effect was to create a bridge over the valley formed by the curvature of the discharge volute (Figure 21).

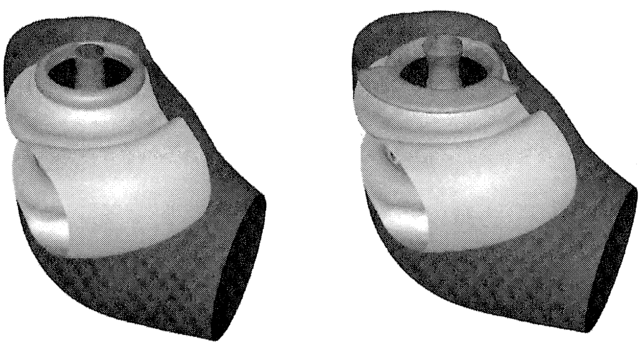

Figure 21. New Ring Design Compared with Original Ring Design.

With the original ring installed, some of the flow in the suction passage would follow the contour of the discharge volute and climb up and over the profile of the ring, creating a vortex just outside the ring. With a new ring design, this same portion of flow would be diverted by the "ears" on the ring, causing it to either flow smoothly across the top surface of the ring or travel beneath the "ear" and around the perimeter of the ring to the opposite side, where it could enter the impeller more slowly.

FINAL ANALYSIS

A new wearing ring model was created in CAD according to the revised design. The CFD model was updated accordingly. Then the analysis was rerun using the same boundary conditions as before. The results were examined in the same areas as in the initial run.

The velocities normal to the plane of the impeller eye can be seen in Figure 22. The circumferential and radial variations are much less prominent than they were with the original wearing ring (Figure 17). There is no longer a region of negative velocities, thus no flow exiting the eye. Overall, the flow is much more uniform in velocity. The revised static pressure distribution can be seen in Figure 23. As in the velocity plot, the variations still exist but are much less severe.

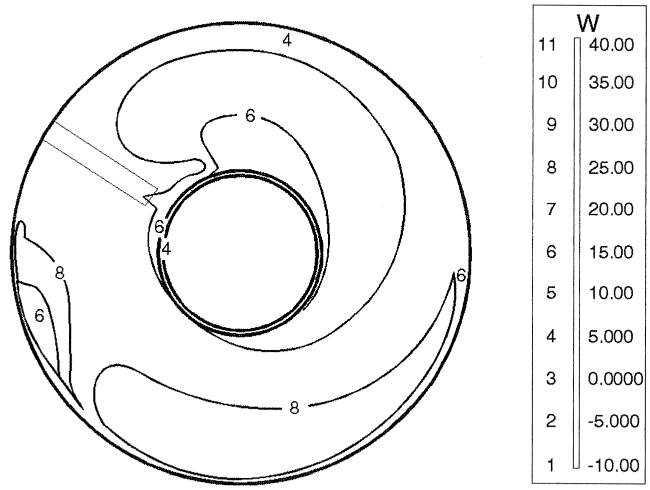

Figure 22. New Ring Installed, Magnitude of Flow into Plane (Ft/S).

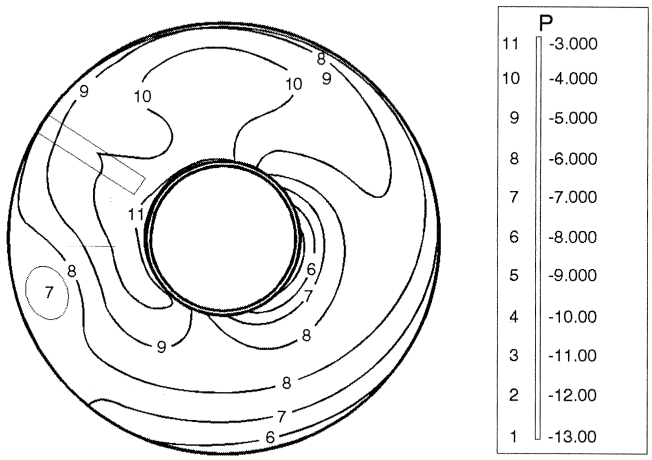

Figure 23. New Ring Installed, Static Pressure (PSI) near Impeller Eye.

Improvements can also be seen in the inlet velocity distribution as depicted in Figure 24. The extension on the new ring appears to serve its intended purpose. The entrance flow moves smoothly along the outer surface of the ring and accelerates much more smoothly into the impeller. The vortex patterns are no longer visible in the valley created by the discharge volute nor are they visible inside the impeller. The magnitude of velocity is more nearly equal from one side of the impeller to the other as compared to the case with the original wearing ring. A close comparison of color vector plots reveals that the acceleration into the eye is less than with the original rings. This is a direct result of providing a large radius turn into the eye. These effects can also be seen in the static pressure plot (Figure 25). Compared to the pressure distribution in the original geometry (Figure 20), the gradients are much less severe. The isolated low pressure region has collapsed into nothing more than a gradient near the inner surface of the wearing ring.

All of these analysis results point toward a better pump design. The entrance flow into the impeller is much better behaved.

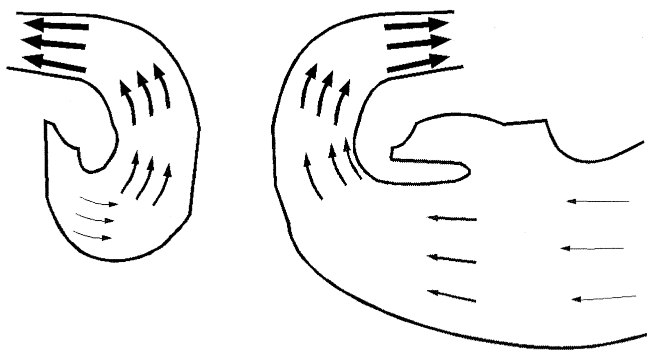

Figure 24. New Ring Installed, Velocity Distribution in the Suction Inlet.

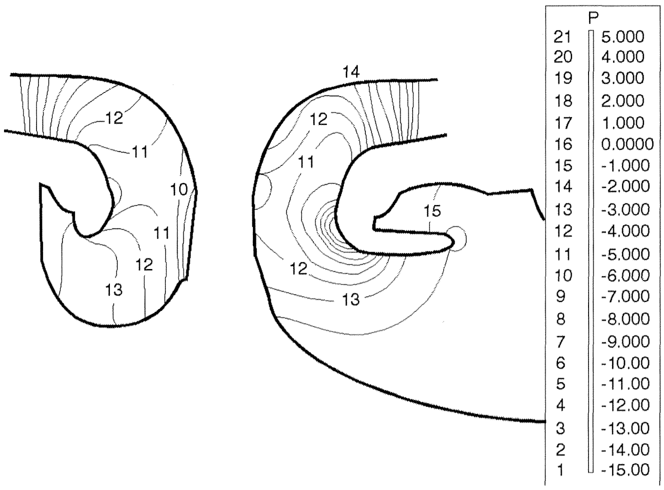

Figure 25. New Ring Installed, Static Pressure (PSI) in the Suction Inlet.

Recirculation losses upstream of the impeller were virtually eliminated. Recirculation in the impeller eye has disappeared. The magnitude and direction of flow entering the impeller is more uniform in the circumferential and radial directions. The final CFD analysis has predicted a greatly improved inlet flow to the impeller and one would expect better pump performance.

VALIDATION

In order to prove that the analysis was correct in that the new ring could improve the noise situation, the next logical step was to create such a ring and test it. A new bronze casing wearing ring was cast and machined to replace the original. A series of tests was conducted in the manufacturing facility using a spare pump identical to the other four in service at the chemical plant. Testing at the manufacturer's location allowed for precise control and measurement of flowrate, while avoiding any interference with the chemical plant operation. The pump was run first with the original rings and then with the new rings to compare the performance. The measurements in each case included:

- Head.
- dbA and overall noise levels.
- Pressure pulsations in the suction volute.

The data were taken at five different flowrates, including the best efficiency point for the reduced diameter impeller as was running in the field.

Testing was conducted with the pump operating above an open sump. The resulting NPSH available was too low to allow for the additional losses of a suction valve, hence NPSH testing could not be performed. Power limitations required that the test speed be reduced to 97 percent of the rated speed, but the flows were not factored because of the need to directly correspond with the analysis flow. The flowrate was measured using a venturi. Power measurements were not taken.

Sound power levels were taken directly at the pump casing in a somewhat unique way. Rather than maintaining a fixed distance from the pump, the probe of the sound power meter was placed in direct contact with the casing at a marked location on the suction inlet. This technique was employed in an effort to minimize variations caused by meter placement and background noise. The pressure pulsations were determined by placing a very sensitive pressure transducer in a tapped hole at the top of the suction volute. Due to time constraints and the limited availability of approximately 1300 kW to run this large pump, no data repeatability studies were performed.

A comparison of test results with and without the new ring can be seen in Figures 26, 27, and 28. The head curve (Figure 26) is seen to have a negligible change, while the recorded sound levels (Figure 27), the most direct indicator of noise, are significantly less. With the new rings installed, a minimum reduction of two to three decibels is observed across the flow range. At 20,000 gpm there is a reduction of 6 dbA. The true noise reduction may actually be even greater in light of any sources of error in measurements. The recorded values were unfiltered. Consequently, the effects of background noise were not removed. In addition to the noise of the motor and gear used in the power train of the test equipment, there was also a great deal of noise emanating from a cavitating discharge valve. Any or all of these factors may disguise the true noise improvement.

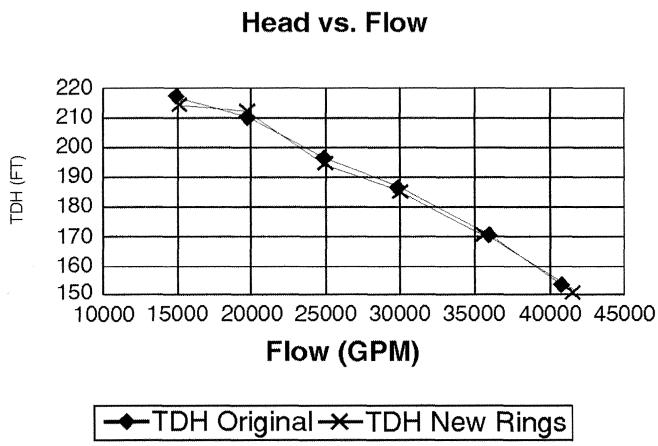

Figure 26. Head Versus Flow with Original and New Wearing Ring Design.

Pressure pulsations are also significantly less, as shown in Figure 28. The trend in pulsations also reveals something about the change in inlet flow. Whereas the original configuration caused a sudden rise in pulsations when throttling back between 25,000 and 20,000 gpm, the new rings delay this effect until flow is reduced below 20,000 gpm. The sudden change in pulsations indicates recirculation. Therefore, the recirculation point has been pushed back to a lower flow.

In addition to the conventional test data described above, efforts were made to verify the analysis results by means of measurements within the suction flow field. A static pressure probe and pitot (total-static) probe were used to record pressures and velocities at specific locations within a plane that lay just outside the wearing ring. The measurements were taken by inserting the probe in each of two tapped openings on the suction inlet. Both openings were machined in the top half of the casing for easy access. The location of the holes was determined prior to the test setup based on the output of the CFD analysis.

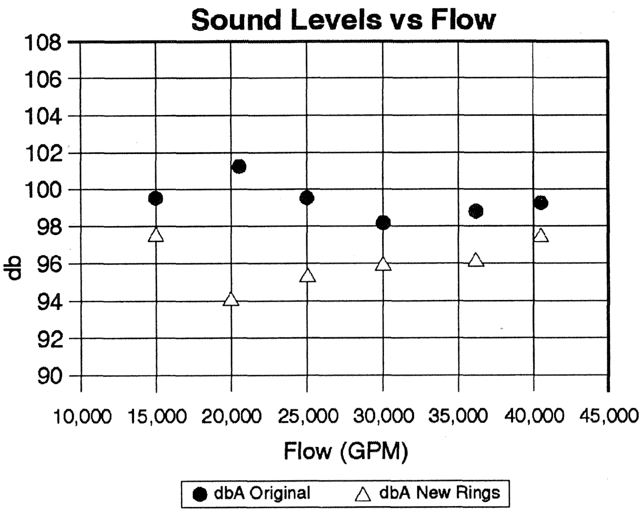

Figure 27. Sound Levels Versus Flow for Original and New Ring Design.

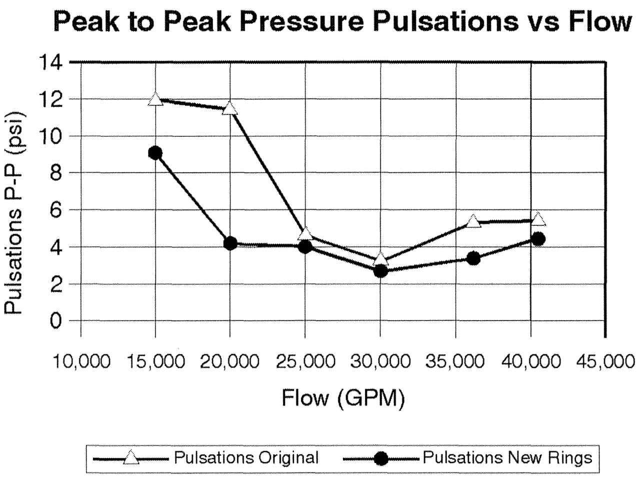

Figure 28. Pulsations Versus Flow for Original and New Ring Design.

The static pressure distribution in the plane of the probe measurement is shown in Figure 29. Lines were added to the plot to show the probe locations. The path of each probe was predetermined to intercept an area of maximum change, while avoiding obstacles such as the shaft or the splitter. The analysis results were later reviewed to interpolate data at specific points, corresponding to locations where the experimental data were determined.

The magnitude of the static pressure measurements corresponded quite nicely with the analytical data. Comparisons are shown in Figures 30 and 31.

The velocity data are shown in Figure 32. The points in each plot indicate the magnitude of the components of velocity in a series of planes perpendicular to the probe axis. Each datum point represents a different depth relative to the insertion point of the probe in the casing. Most of the experimental points lie reasonably close to their analytical counterpart. There are one or two datum points that do not correlate well, but these deviations may be explained by an occasional faulty reading from a plugged probe tap. In general, the experimental data agree quite well with the analytical predictions at these locations.

SUMMARY

The computational fluid dynamics model proved to be useful in revealing the patterns of flow within the double suction pump. The impeller model served to illustrate a well behaved impeller design, thus limiting the scope of the investigation to the suction inlet waterways. The suction inlet model provided a means to identify key problem areas. The presence of irregular flows entering the impeller eye reinforces the notion that the impeller is not a primary source of noise. A perfect impeller would still have problems under

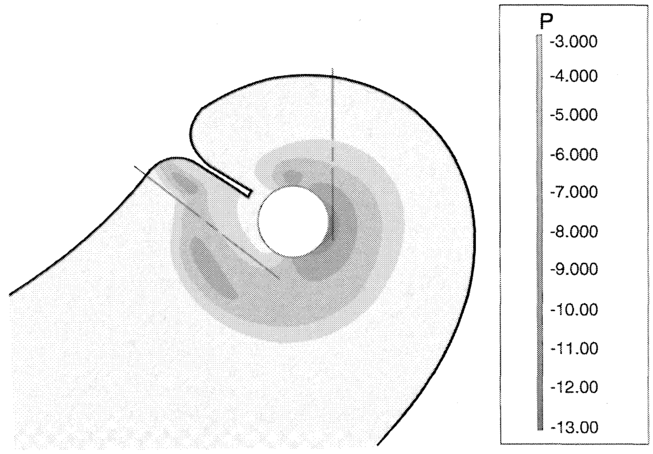

Figure 29. Static Pressure Distribution in Plane of Probe Measurements.

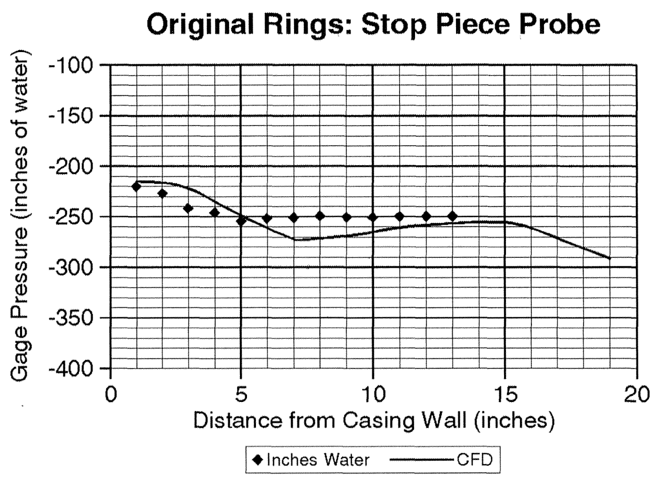

Figure 30. Static Pressure Measurements, Probe Near Stop Piece.

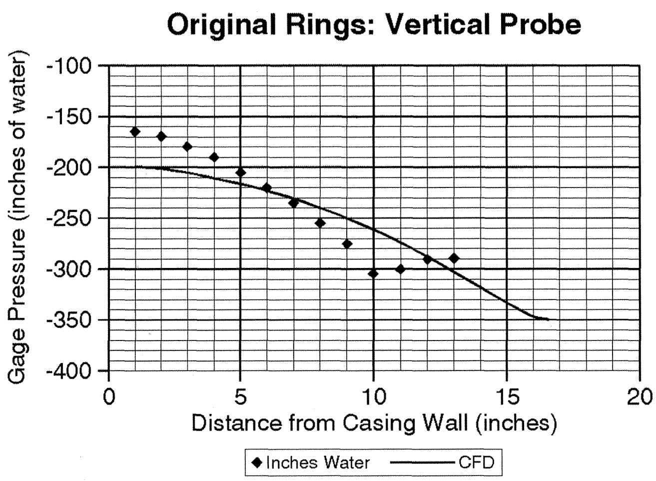

Figure 31. Static Pressure Measurements, Vertical Probe.

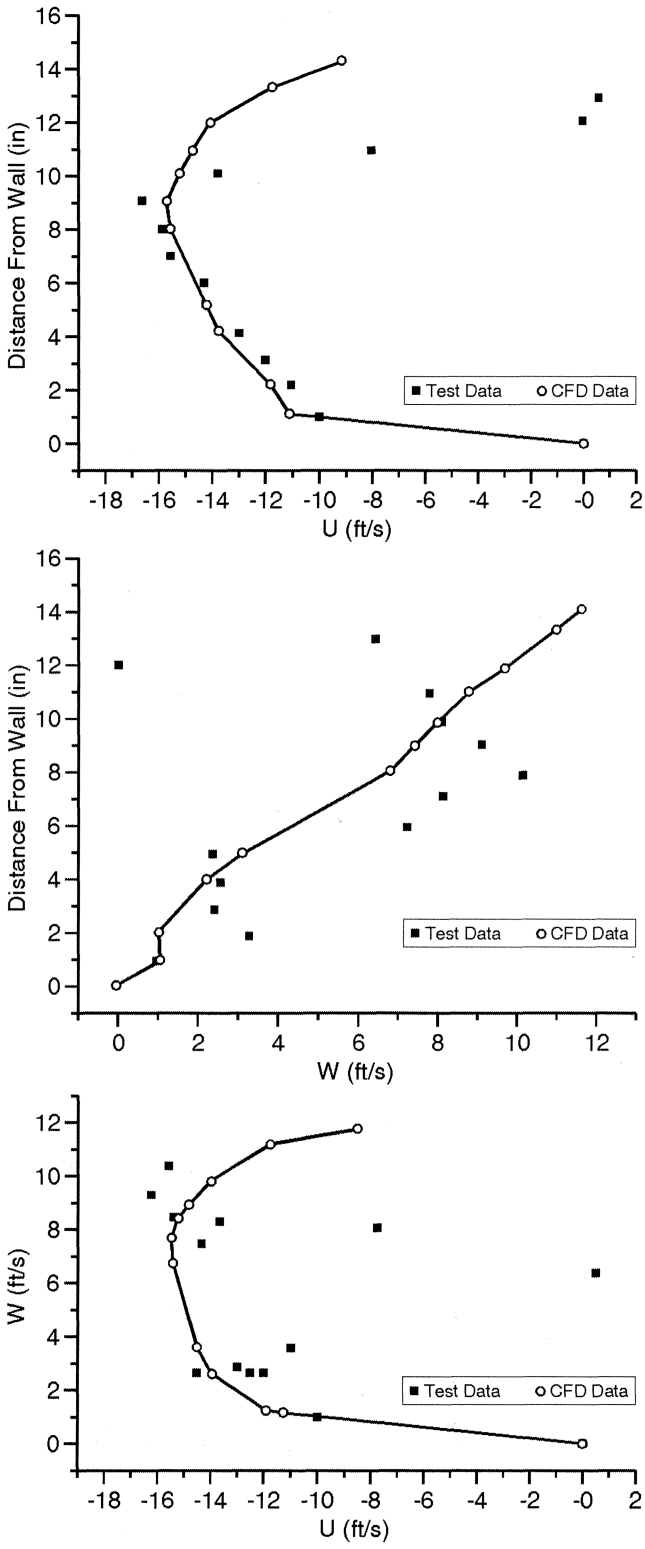

Figure 32. Velocities Measured Along Vertical Probe.

such circumstances. The pattern of flow entering the eye of the impeller must be uniform in order for the pump design to function as intended.

The proximity of the problem areas to the casing wearing ring, and the need for a field replaceable part, lead to the development of a potential solution in the form of a new wearing ring. A revised CFD analysis predicted that the new ring would improve the characteristics of the flow entering the impeller. Lab experiments confirmed that the pump does in fact operate with less noise using the new ring.

The flow improvements, whether in theory or in the lab, should translate to the field where noise problems are exacerbated by dissolved air and other system related factors. Field testing is planned for the near future but has not yet been conducted, due to the ongoing demand for plant cooling water. When a plant shutdown can be utilized to fit the new rings into one of the installed cooling water pumps, sound level measurements and pressure pulsations will be taken to quantify the improvement. An exact correlation with lab experiments will be difficult, due to a lack of accurate flow measurement in the field, but a noticeable improvement is anticipated.

The pressure probe experiments demonstrated the accuracy of the CFD model, thereby substantiating its value as a tool for analysis, design, and even troubleshooting. The CFD analysis was successful in locating the problem areas within the double suction pump and in predicting the success of the new wearing ring.

CONCLUSION

The primary cause of noise in the $30 \times 30 \times 38$ horizontally split case double suction pump was caused by flow separation occurring in the suction chamber. Classical techniques would not have lead to a solution of the problem.

CFD was used successfully to:

- Identify the problem area within the suction passage (volute suction nozzle).
- Rule out impeller recirculation as a problem.
- Predict an improvement with a new ring.

Testing was used to validate the results of the CFD analysis and the improvement obtained with the new wearing ring. There was an overall drop of 3 db out of 96 db (dbA) in noise and a drop of pulsation from 2 psi to 5 psi (peak-to-peak) without effecting performance. Improved suction characteristics were obtained at low capacity.

CFD proved to be a useful tool in solving a difficult field problem.

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