THE USE OF BASIC HYDRAULICS TO SOLVE FIELD PROBLEMS: THREE CASES

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CASE 1: COOLING WATER PUMP CASE STUDY–CAVITATION PERFORMANCE IMPROVEMENT

by Lev Nelik and Jeffrey Freeman

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INTRODUCTION

Single stage double suction horizontally split pumps are widely used in cooling tower water applications. Their operational record varies depending on installation (sump and piping detail), mode of operation, speed, specific and suction specific speed, metallurgy, and other parameters. Pump noise and cavitation damage are the most frequently uncounted problems with these types of pumps. Even though much has been published on pumps cavitation, the upstream conditions of the cooling water pumps are such that many of the classical cavitation remedies are difficult to apply.

For example, there are established rules for minimum flow, such as a classic paper by Fraser [1]. But, in reality, the demand for water to cool the heat exchangers varies through the year, as the ambient temperature changes. The user is forced to choose to either buy many pumps and operate in parallel, switching on and off per demand, or to limit the investment to few pumps (typically two or three, of which one is usually a spare), and be forced to operate below the minimum flow. Another restriction is that the cooling water feeds the pump from the open sump at atmospheric pressure plus some submergence, which means that the $NPSH_A$ is around 30 ft, when adjusting for pipe losses. The classic recommendation of $NPSH_A$ margin over $NPSH_R$ is practically impossible to maintain in many cases. Furthermore, by the nature of its design, cooling towers supply a highly aerated water to pumps, along with modified chemistry due to water treating agents, both dissolved and entrained air, and chemistry increases the vapor pressure, which results in lower $NPSH_A$ than calculated for pure water [2], some times as much as 5.0 to 6.0 ft, although the exact number has not been experimentally proven.

Obviously, a pump design that would allow a wider operating flow region would significantly ease the restriction on the user, and be economically attractive. The case presented in this tutorial is an example of a joint effort between the pump manufacturer and a user to come up with the design solution, which would optimize the above mentioned requirements, reduce noise and cavitation damage, and improve overall reliability of these pumps.

INSTALLATION

The overall view of the cooling water tower is shown in Figure 1 with three pumps. Depending on load, one or two pumps are operating, with the third a spare. The total flow requirements vary between 16,000 to 25,000 gpm, based on the process heat exchangers load, ambient temperature, system minimum settling flows, and other considerations. This translates into the requirements for each individual pump to operate between 8,000 to 17,000 gpm. These double suction pumps are running at 1180 rpm, have 23.25 in impeller OD. Specific speed Ns=2260, and suction specific speed is Nss=10,050 (flows used in calculation of Ns and Nss are per impeller eye, i.e., half total flow).

A detail of an individual pump is shown in Figure 2. The sump was designed to ensure convenient pump flooding by just opening the butterfly valve, since the water level in the sump is above the impeller centerline. At first, the pump piping was suspected as causing a problem. The incoming sump velocities are very low (under 1.0 ft/sec), and no visual vortices were noted. The 36 in elbow and the bell were removed and the pump tested, indicating no appreciable difference in noise. The butterfly valve was installed in horizontal and then vertical stem orientation, both in the field and in the manufacturer's test facility. In the field, no noticeable change in noise was observed; in the test facility, the horizontal stem orientation was less noisy than the vertical (as expected), but the difference believed to be not significant enough to be a cause of the noise and damage. The original pump impellers were of cast iron, which is known to have poor cavitation resistance properties. From the hydraulic design standpoint, the impeller eye size was too big for the range of Ns and Nss, as applied to cooling water pumps. As shown on Figure 3, and can be calculated from Fraser's data [1], the onset of the suction recirculation starts at around 13,000 gpm. At flows lower than that, the cavitation damage becomes quick, especially in light of poor metallurgy. The pump performance with operating flow range is shown on Figure 4.

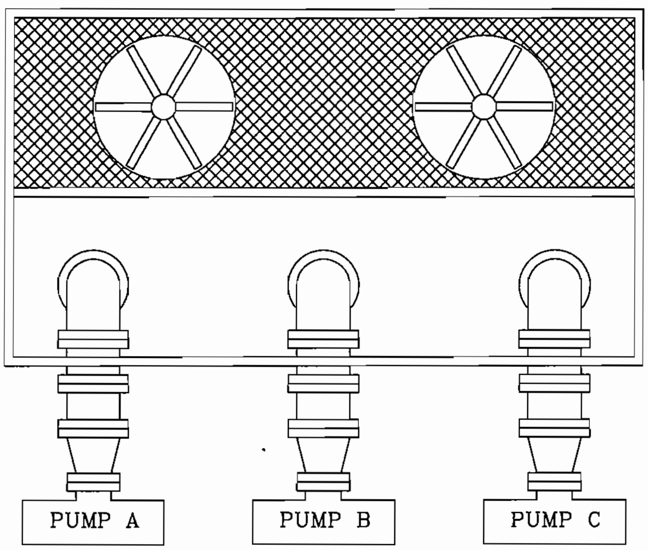


Figure 1.

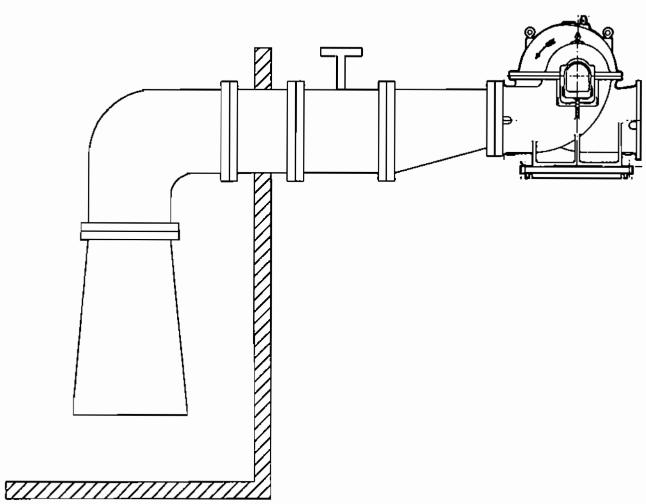


Figure 2.

After about three months of operation, pump "C" impeller was replaced by 316ss metallurgy, as it was desirable to determine the effect of superior metallurgy on cavitation life.

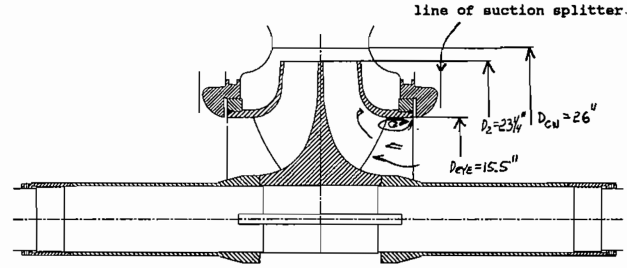


Figure 3. Existing Pump.

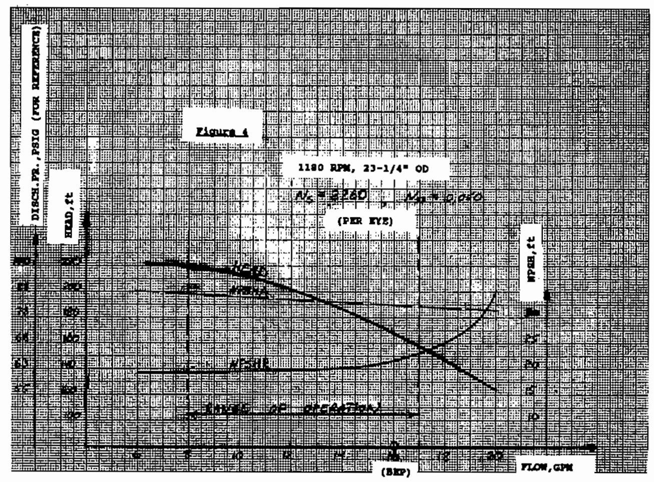
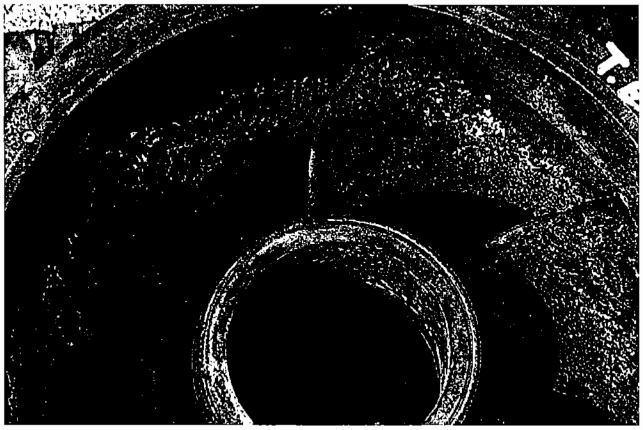


Figure 4. Pump Performance Curve.

A comparison is shown in Figure 5 between the cavitation damage to the impeller after approximately six months operation. Pump "B" (iron impeller) shows a significant damage on the suction side of the blades, at the shroud region. Some damage (not visible on the picture) was found also on the pressure side, which suggested a combination of low flow cavitation (suction side effect), and recirculation (pressure side effect). Since pumps operated over the wide range of flows during the course of the year, it was impossible to relate the damage pattern to the operating flow. However, a noise data was taken at different flows, and results plotted on Figure 6 (triangles). A well defined region of flow recirculation onset is at about 13,500 gpm, which is close to a theoretical prediction by Fraser's method.

In order to reduce recirculation, as well as to better match flow angle to the blade angle, a special bullring (Figure 7) was attached to the existing casing rings. Even though the axial length available to fit this ring (between the impeller shroud face and the suction splitter, as shown on Figure 7) was limited, this special "bellshape" curvature of this ring should further enhance flow distribution into the eye. The noise data were again taken over the range of flows (Figure 6, circles), showing improved noise characteristic at low flow, no substantial change at the midrange, and noisier operation at higher flows.

The following modification was aimed at improving flow distribution around the impeller inlet. The original suction splitter is only 15 degrees from the vertical line, and was shaped poorly, as shown on Figure 8 (left). Much better designs are given [3], and these guidelines were followed, as shown on Figure 8 (right)–much better contouring, and spitter centerline moved to 55 degrees from the vertical. In order to accommodate such modification in the field, the old suction splitter was ground off, and a new one was attached to the casing ring (Figure 9). The noise data for this modification is shown on Figure 6 (squares). Note that the whole noise curve is shifted downwards by about 3 dBA, and gets even more significantly better at the very low flow.



Cast Iron (PumPB): Extensive Damage at Impeller Inlets (Starting at the Shroud Area, SuctionSide; Some Damage on Pressure Side). (12 months operation)

316SS (Pump C): Very Little Damage (Minor Pitting at Blade Inlets, Shroud Area).
Figure 5. Comparison of Cavitation Damage of Existing Design Impeller: Cast Iron vs 316SS.

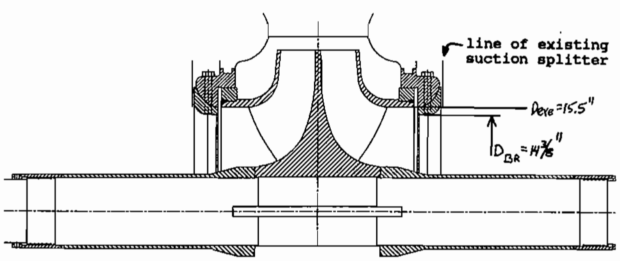


Figure 6. Bullring Modification of Casing Ring (Attempt to Reduce Eye Recirculation)

Finally, a combination of a new suction splitter and a bullring was tried (Figure 10). The noise signature became the lowest at low flows (up to 12,000 gpm), and then reached the nonmodified configuration at 14,000 gpm, and further increased above 14,000

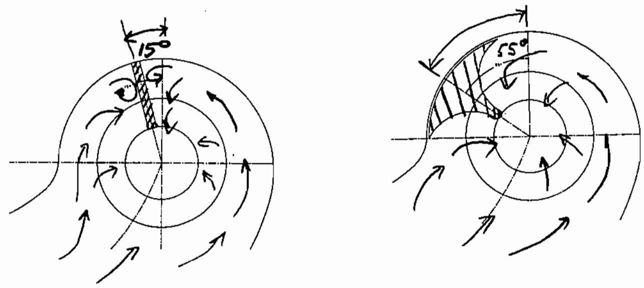


Figure 7. Comparison Betweenthe Old (Left) and New (Right) Splitter Design for Better Flow Feed into the Impeller Eye.

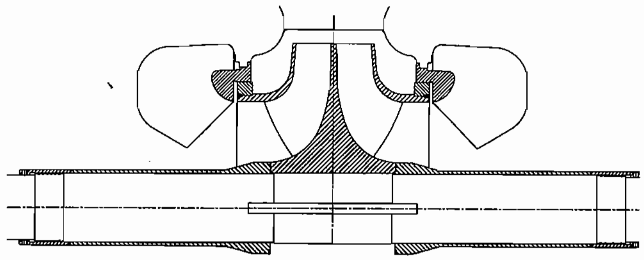


Figure 8. New Splitter: Repositioned, and having Better Contour Shape (Ref. Figure 7).

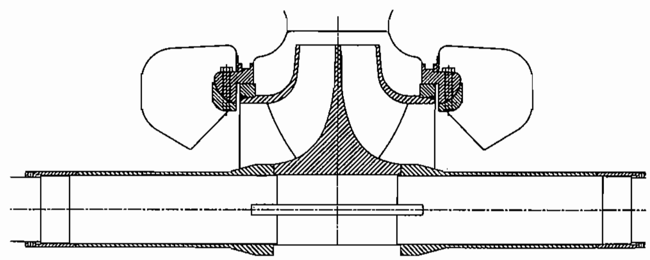


Figure 9. Bullring with Splitter Modification.

gpm. At that point, it was decided to operate the pump at 9,000 gpm for several months, and then open it up and inspect for damage, comparing to other pumps.

After three months, the pump "A" (with above modifications) was opened up and inspected (Figure 11). The damage pattern was different than before; it spread more evenly over a wider area of the suction side of the blades, but the depth of damage was less than originally experienced. It was apparent that the incremental improvement by the modification of the splitter and a bullring was not enough to achieve the desired impeller life of five years.

At that point, it was decided to design a new impeller, based on the information learned in previous testing. This new design impeller has smaller eye (14.28 in), and features a special profiled "P-blades" [4], as shown on Figure 12. Due to smaller eye, the Nss value is smaller, Nss=9500. From the discussion above, it is clear that if this impeller was made in cast iron, a direct estimate of the design improvement would be possible. However, the additional cost and time could not be justified at that time, and the impeller was made from duplex stainless steel CD4MCu, which is believed to have somewhat better cavitation characteristics as compared to even 316ss. Keep in mind that the old design impeller made of 316ss (pump "C"), experienced little damage after six months of operation, and is estimated to will have last for the total of about three years, i.e., somewhat below the customer target of five years.

The comparison is shown in Figure 13 in noise levels between the old impeller design (pump "C") and the new one (pumps "A" and "B"). The noise was lowered for a new design - both for "A"

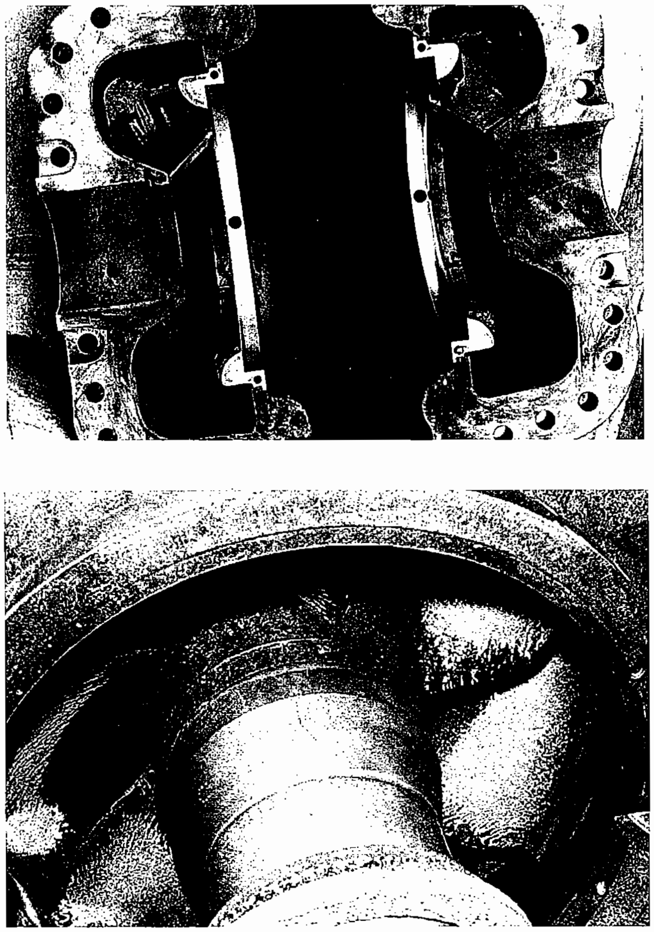


Figure 10. Special Bullrings and a Profiled New Splitter Shown Installed in the Casing, (top); on the bottom are Impeller Inlets, Suction Side, after Operating with above Combined Modifications for (3) Months.
Note the Damage Region now is in the Middle of the Blade, as Compared to original Damage Pattern (no Modifications) near Shroud.

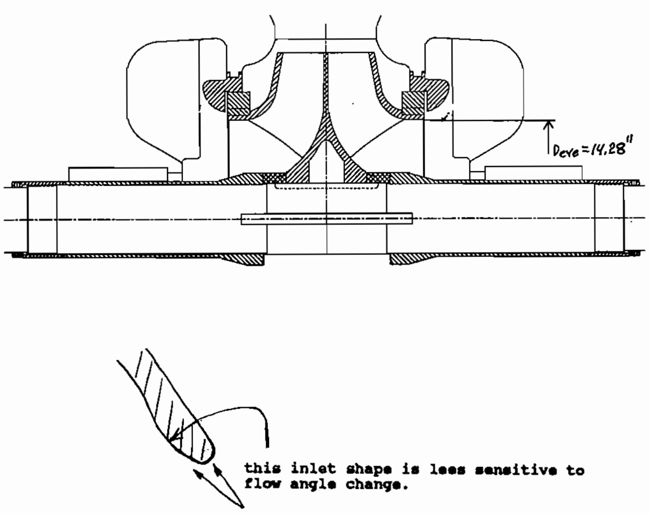


Figure 11. New Impeller Design (Smaller Eye and Special "P-Vlade" Inlets)

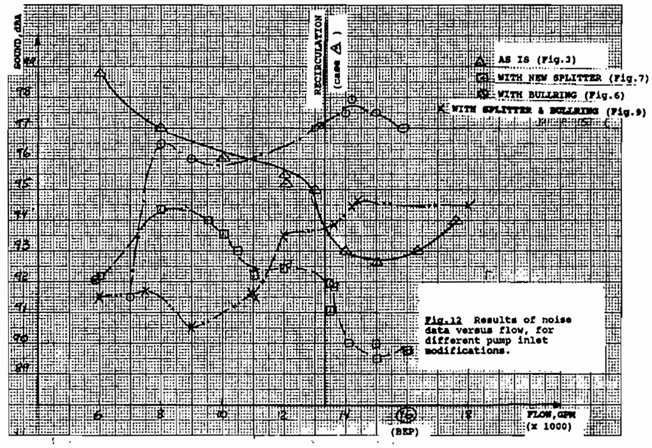


Figure 12. Results of Noise Data vs *Flow, for Different Pump Inlet Modifications.*

and "B" pumps. Pump "B" had a new impeller with smaller eye and special "P-blades" [4], but the casing still had the original blunt suction splitter (Figure 8, left), while pump "A" had also a contoured splitter (Figure 8, right). The combination of the new impeller design and contoured splitter resulted in lowest noise level. The smaller eye impeller requires higher $NPSH_R$ though, which is the reason for the noise curve to begin to rise at the lower flow than the original impeller. This was not a problem in this case due to sufficient NPSH margin available, and the desired operating flow below 12,000 gpm. The contoured splitter offsets this disadvantage somewhat, as can be seen on Figure 13 (pump "A").

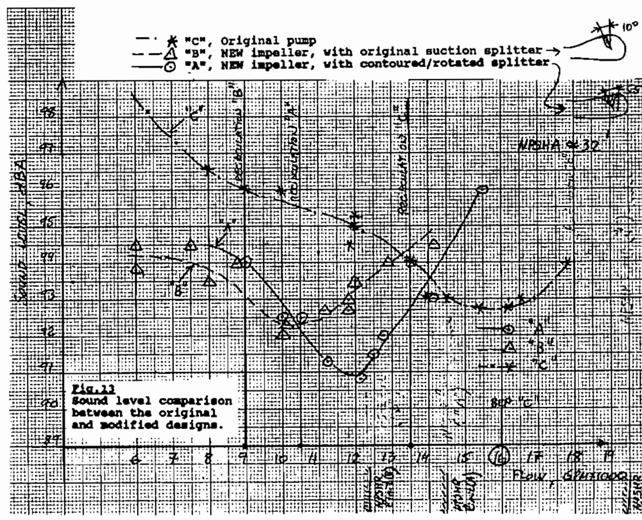


Figure 13. Sound Level Comparison between the Original and Modified Designs.

SUMMARY AND CONCLUSION

Operation of cooling water pumps is unique in terms of severity of conditions: lack of ample $NPSH_A$, varying water chemistry from one installation to another, presence of dissolved and entrained air and its effect on vapor pressure, variety of sump and connecting piping geometry, and internals metallurgy.

Bullring and suction splitter have little effect on cavitation damage life improvement, even though it has an effect on reducing noise and vibration level.

Special impeller design have beneficial effect on noise reduction and cavitation life extension, although the quantifiable data on life extension are still lacking.

Impeller metallurgy have a predominant effect on cavitation resistance. Cast iron impellers have poor cavitation resistance (as low as three months), and 316ss metallurgy can significantly extend impeller life (four years). CD4MCu is believed somewhat superior to 316ss, at relatively small (10 percent) incremental cost. A combination of optimum impeller design, good profiling and positioning of suction splitter and metallurgy, together, could best improve pump cavitation resistance, and extend life to more than five years.

Some data relating, Ns, Nss, eye size and cavitation life is available, but not conclusive enough. More case studies of different installations and a database compilation could lead to some sort of empirical charts and correlations to predict cavitation resistance life, as a function of the design, metallurgy and system variables.

ACKNOWLEDGMENTS

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CAVITATION DAMAGE AS RELATED TO OPERATING POINT

by Joe Silvaggio

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Since joining Imo Delaval in 1968, he has held several positions in the Research and Advance Product Development Department and the Product Engineering Department. He has also worked as a member of the Compressor Development Group and as a project engineer in the Compressor Department.

Mr. Silvaggio is a member of the Engineering Honor Fraternities of Sigma Tau and Pi Tau Sigma. He is also an active member of ASME and has held several offices in the Trenton, New Jersey, section. At present, he is on two ASME Performance Test Code Committees. He holds both B.S.M.E. and M.S.M.E. degrees from the University of Pennsylvania. He has written and has been a coauthor on numerous technical publications. He is a member of the International Pump Users Symosium Advisory Committee.

ABSTRACT

A field cavitation damage problem is described that occurred as a result of running a single stage, double suction pump for extended periods of time at flows beyond the design point. The pump is large (60 in inlet flange and 54 in discharge flange) and is used to circulate cooling water. The pump takes suction from an open channel and discharges into headers that provide cooling water to a condenser which discharges to a natural draft cooling tower. When the pump was run in the field, the required discharge head was substantially lower than the design head. This caused the pump to run at a higher flow where the NPSH requirement was higher than the available NPSH provided by the open channel. Because the NPSH available was lower than the NPSH required at this new, higher flow, the pump cavitated and caused cavitation damage to the impeller. A suggested field modification to the impeller diameter is discussed. This modification will reduce the discharge flow and reduce the required NPSH to a level which is below the available NPSH and alleviate the cavitation damage to the impeller

INTRODUCTION

Large single stage, double suction circulating motor driven water pumps, as shown in Figure 14, were supplied to provide cooling water to power plant condensers. These pumps are relatively large motor driven pumps, having a suction flange diameter of 60 in and a discharge flange diameter of 54 in. They have horizontally split cases with a double volute discharge. The pump has two ring oil lubricated split sleeve radial bearings and a double acting tilting pad thrust bearing. The pumps were originally specified to deliver a head of 120 ft at a flow of 120,000 gpm with a required NPSH of 31 ft. The performance curve is shown in Figure 15.

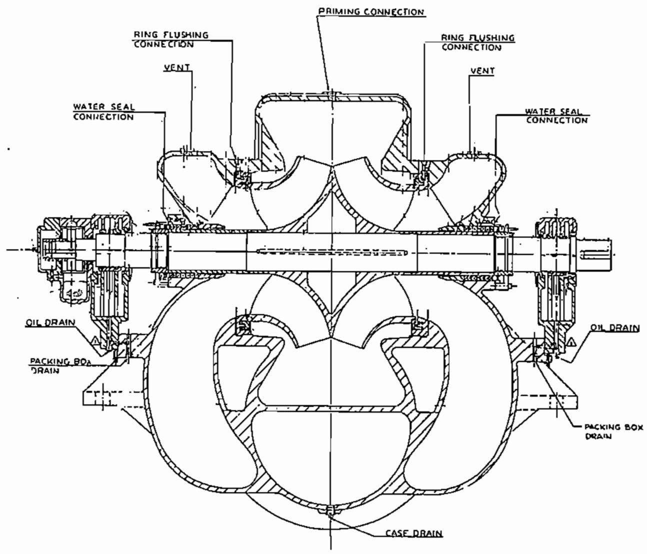

Figure 14. Cross Section View of a W 60/54 (60" Inlet, 54" Discharge) Single Stage, Double Suction Circulating Water Pump.

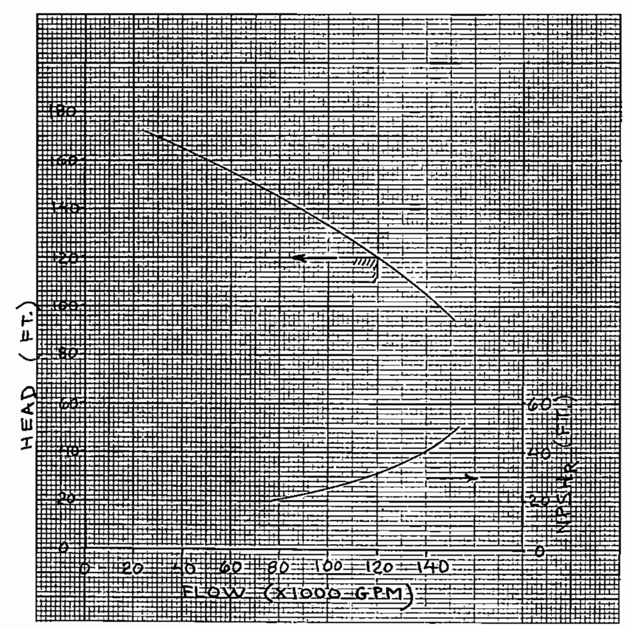

Figure 15. Performance Curve of the W 64/54 Circulating Water Pump.

After the pumps were installed at the power plant, the impellers were reported to have cavitation damage. Further investigation showed that the pumps were being operated at a higher flow than the design flow. This was because the head required at the field installation was less than that originally specified.

BACKGROUND

Cavitation is a phenomenon that occurs in a liquid when the local pressure drops below the vapor pressure and vapor bubbles are formed. These vapor bubbles can be carried along with the flow field. If an adverse pressure gradient of increasing pressure is encountered by the flow field, the bubbles will implode when the pressure increases above the vapor pressure. If the implosions occur near a metal surface, and if the hydrodynamic cavitation intensity exceeds the cavitation resistance of the impeller material, cavitation damage (a type of erosion) can occur to the metal.

Cavitation damage initially has an appearance of a 'sand-blasted' surface. As cavitation damage advances, the pits get deeper and in very advanced cases, the pits can even penetrate through metal surfaces.

In a pump stage, the lowest pressure is near the leading edge of the impeller vane in the eye area. As the flow enters the impeller blade area, energy is imparted to the liquid being pumped by the action of the impeller vanes. The flow is accelerated around the leading edge of the impeller vane and along the blade length. As the flow is accelerated, local areas of lower pressure will occur. Depending on the pressure level in these areas, vapor bubbles can be formed. As the flow then proceeds along the blade, the flow enters areas of higher pressure where vapor bubble implosion will occur. If the implosions occur near the blade surface, cavitation damage will occur.

Consider a typical pump performance curve as shown in Figure 16. This figure contains five curves: a Head-Capacity curve, an Efficiency curve and three NPSH curves. The three NPSH curves are: 1) the Performance-NPSH curve, which is the amount of NPSH required to maintain hydraulic performance, 2) the NPSH curve, which is the amount of NPSH required to limit cavitation damage, and 3) the NPSH curve, which is the amount of NPSH required to prevent vapor bubble formation entirely.

Usually, the NPSH curves are developed for various percent of head loss at a certain flow utilizing test data taken by varying the pump suction pressure. If a pump is operated on the NPSH curve, there is a very high possibility that vapor bubbles will be formed and that cavitation damage will occur. Three types of cavitation zones are also shown in Figure 16. Zone I is the minimum flowrate area where backflow due to suction recirculation is well established. Cavitation damage appears on the pressure side or the non-visible side of the blade. Vapor bubbles are swept off of the suction side of the blade by the backflow and recirculated toward the pressure side of the blade where they implode and cause cavitation damage.

Zone II is the area where blade flow mismatching occurs and causes cavitation damage on the suction side or the visible side of the blade. At lower than design flow, the angle of attack of the flow velocity vector on blade leading edge will cause the flow to separate from the blade on the suction side. Under low pressure conditions, vapor bubbles will be formed. This is illustrated in Figure 17. As the bubbles travel downstream, they will implode and cause cavitation damage on the suction side of the blade. The higher the amount of incidence, the further downstream the cavitation damage will occur.

Zone III is the area that incurs runout cavitation damage that appears on the pressure side, or nonvisible side, of the blade. At higher than design flow, the angle of attack of the flow velocity vector on the blade leading edge will cause the flow to separate on the pressure side of the blade, as illustrated in Figure 17. Under

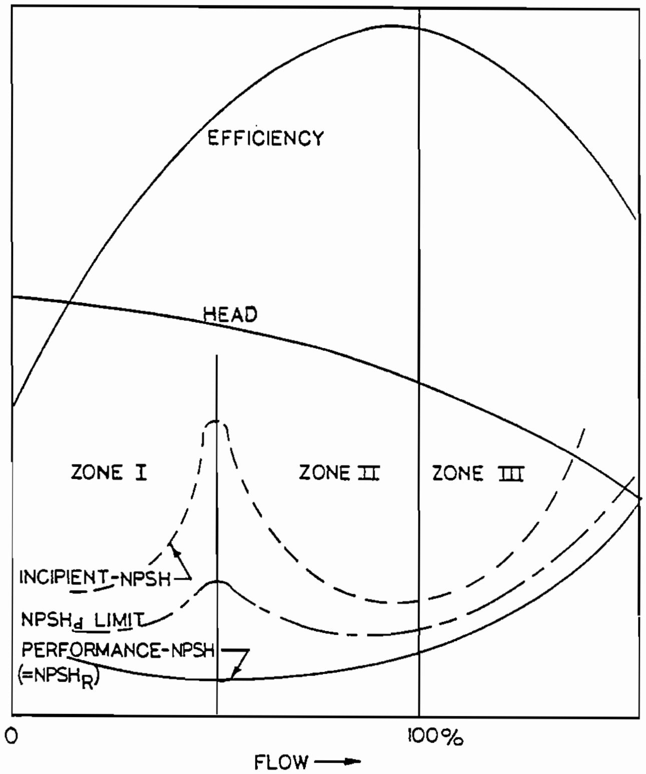

Figure 16. Typical Pump Performance Curve, Showing NPSH Required, a) to Maintain Hydraulic Performance or Pump Head, b) to Limit Cavitation Damage, and c) to Prevent Vapor Bubble Formation, Incipient NPSH. (Source: Cavitation Damage In Boiler Feed Pumps by Paul Cooper and Fred Antunes, Symposium on Power Plant Feed Pumps, June 1982).

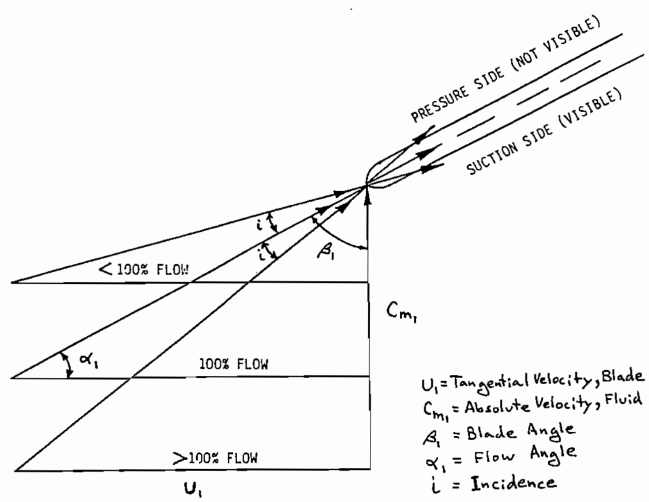

Figure 17 The Relation of Inlet Flow Velocity Vectors and Cavitation Assuming No Prerotation of the Inlet Flow.

low pressure conditions, vapor bubbles will be formed and carried downstream into higher pressure areas. When they implode on the pressure side of the blade, cavitation damage will occur. The cavitation damage will occur further downstream for higher values of incidence.

This simplified background information on cavitation, by utilizing basic hydraulics, provides insight into determining the root cause of cavitation damage to impellers.

DISCUSSION

The performance curve of the circulating water pump is shown in Figure 2. This figure shows that the pump has a design head of 120 ft at a design flow of 120,000 gpm. At these conditions, the available NPSH was 38 ft. The required NPSH at this design condition is 31 ft. At this operating condition, there was no indication of cavitation damage.

When the pump was put in service at the power plant, the impellers were reported to have cavitation damage. Investigation of the pump operation and field tests showed that the pumps were being run at a head of 105 ft and at a flow of 140,000 gpm. At this operating point, the required NPSH is 40 ft and exceeds the available NPSH by 2.0 ft. As a result of operating the pump at runout conditions, where there is not enough NPSH available, the pumps cavitated, causing cavitation damage to the impeller.

The obvious solution was to run the pump at the design flow, where the available NPSH was ample and would have prevented cavitation damage. The simplest way to accomplish this would be to trim the impeller outside diameter. By trimming the impeller outside diameter, the pump will deliver less flow at the field head requirement of 105 ft. The trimmed impeller would deliver 120,000 gpm at a head of 105 ft. At this condition, the required NPSH would be less than the available NPSH and would provide cavitation free pump operation.

Where feasible, this solution is the least costly method of solving the cavitation damage problem. However, with this solution, the user must be satisfied with the original design flow and not the additional flow that is now being provided by the pumps. If the additional flow would be required, a more complicated and costly solution would be to redesign the impellers for the higher flow conditions using present day design techniques (which would be the subject for another paper).

CONCLUSION

A field cavitation problem was investigated and found to be caused by running the pump beyond its design flow. Although the pump was specified to operate at a certain head and flow, it was operated at a higher flow and lower head. This caused the pump to run in a zone of runout cavitation where the required NPSH was higher than the available NPSH. This caused cavitation damage to the impeller. A solution to this problem, by trimming the impeller outside diameter, is presented. The smaller impeller diameter will deliver the design flow at the lower head requirement and provide cavitation free operation.

CASE 3: USE OF BASIC PUMP HYDRAULICS TO SOLVE PUMP MAINTENANCE PROBLEMS

by John Joseph

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Mr. Joseph received his B.S. degree (Mechanical Engineering) from the University of Texas at El Paso (1972) He is a member of the planning committee for the Rotating Equipment Repair Users Council, and a member of the International Pump Users Symposium Advisory Committee.

ABSTRACT

Most pump problems are rooted in basic errors that are made in the design, application, installation, maintenance, and/or operation of pumping systems. When the final, "correct," solution to the system problem is determined, most engineers look back and say "I knew that." The solution is obvious when all the facts are known, and a proper root cause failure analysis is performed.

The analysis of the problem should evaluate each of the nine primary areas that can possibly contribute to symptoms. These nine areas include, but should not be limited to:

- Proper mechanical design of the pump case, bearing housing, pump shaft, and impeller.
- Proper design and installation of the pump baseplate.
- Proper hydraulic fit of the pump to the "REAL" process requirements and conditions.
- Pipe strain and nozzle loads at the lowest level possible at the operating conditions.
- Proper alignment targets and procedures to achieve less than 0.1 degrees misalignment between the coupling and either the pump or driver at operating conditions. Transients should not exceed 0.5 degrees misalignment.
- Proper repair quality criteria.
- Proper operating criteria for all phases of operations.
- Proper design and installation of auxiliary systems.
- Proper arrangement of warm up piping to allow thorough and uniform heat soak of the pump case.

The 201-JA/JB pumps on the Pipe Stills at the Amoco Petroleum Products Refinery in Texas City, Texas are discussed. Repeat mechanical failures of the bearings led to several attempts at fixing perceived problems and then finally to a root cause failure analysis and recommended solutions.

BACKGROUND

The pumps in question were purchased in 1981 for a depropanizer feed service. These pumps experienced bearing failures on the initial startup and during subsequent startups after repairs. For some reason, after a series of bearing replacements during each of these cycles, the pump would start up and run smooth for an extended period of time. This pump design is a single stage, double suction, between bearing, API design with sleeve journal bearings and a rolling element thrust bearing set. The bearing failures were due to loss of metal in the sleeve bearings and were accompanied with vibration. Limited maintenance history was documented on these pumps, since the bearings could be easily replaced in the field, and therefore, historical maintenance costs were not available. The current Maintenance Management system allowed capturing of the most recent cycles of failures and repairs. When this last cycle of repeat bearing failures began, the cost of maintenance were monitored. The 1994 event totaled $117,000 in materials and labor.

DISCUSSION

The 201-JA/JB pump set is in depropanizer feed service drawing off an elevated drum. The original data sheets showed process requirements of 1194 ft of head at a normal flowrate of 1040 gpm and a rated flowrate of 1260 gpm. The specific gravity of the stream was 0.628 and the pumping temperature is 150°F, with

$NPSH_R$ equaling $NPSH_A$ at 10.5 ft. In actuality, the $NPSH_A$ is 100+ ft, since the suction drum is elevated and the vapor space has a natural gas blanket of 90 psig.

During the 1994 series of bearing failures, much of the initial trouble shooting emphasis was placed on the quality of repairs performed on the pump. The suitability of the bearings and the quality of lubrication were also suspect. Multiple bearing replacements and closer attention to repair details yielded the same results of bearings failing on startup. The pump was returned to the shop for repeated tear downs and checks to determine if any internal misalignments were contributing to the problems. Field adjustments were made to the bearing housing to center the shaft, although no direct evidence existed that case alignment was incorrect. The pump to driver alignment was also checked and found to be satisfactory.

The approach taken to eliminate the problem seemed random and based on the superficial symptoms rather than an ordered pattern of analysis. After much money was spent without a solution, a root cause failure analysis was requested and conducted to bring order to the problem solving effort. Data were collected on mechanical repair specifications and results. The process folks were asked to review and document the actual conditions in the field during these events. The vibration signatures were thoroughly reviewed to determine the information that the vibration presented. Pipe strain was checked to determine its possible contribution. The hydraulic fit of the pump to the process requirements was reviewed.

The picture started putting itself together when the vibration signature study indicated high levels of vibration at 4× running speed. The impeller used had four vanes and this type of vibration could indicate uneven flow distribution exiting the OD of the impeller caused from low flow conditions. From test to test, the vibration levels and bearing temperatures varied quite a bit. The only parameter that was changing with vibration was the flow through the pump. Higher flow lowered the vibration and lower flow raised the vibration.

During one disassembly, it was noted that the piping also had a substantial offset when unbolted. Looking at the piping supports revealed that wood wedges had been placed under support columns where the concrete base had crumbled sometime in the past. Some benefit in the reduction of 1× vibration was gained through proper alignment and support of the piping.

The best efficiency (bep) for this pump and impeller model was determined to be 1690 gpm. The normal flow design point for the pump on the data sheet was 1040 gpm when purchased. The actual operating flow for this pump service during these episodes averaged about 700 gpm, or 41 percent of the bep flow. The suction specific speed of this pump impeller combination is 12,700, which indicates limited tolerance to operation to the left of the bep flowrate.

ROOT CAUSE

The root cause of the bearing failures was a determination that the pump is operated at significantly lower flowrates than the OEM originally intended. The low flowrate of 700 gpm (41 percent of bep flow) created high 4× vibration in excess of the bearings load carrying capability. Operations personnel, in fact, wanted to decrease flowrates to lower levels, since lower flowrates better suit their process needs to improve yields on this unit. Several flow tests were performed to confirm the flow sensitivity of this pumping system. The tests supported the belief that high vibration was a direct function of flow reduction. The high suction specific speed of 12,700 can indicate a likelihood of impeller recirculation with low flowrates. The recirculation problem can only add to the intensity of vibration at the lower flowrates.

Proper flowrates for this pump are in the area of 75 to 115 percent of bep flowrate [5].

RECOMMENDATIONS

- Install a minimum flow bypass to allow operation of the pump in the safe range. This will generate an economic penalty by consuming energy dollars on a continuous basis to attempt to save maintenance dollars.
- Operate the process to achieve flowrates closer to 75 percent of bep, 1270 gpm. The process is capable of this, but an economic penalty is suffered, due to less than optimum product splits.
- Retrofit the existing pump case with gap rings and an impeller suitable to the current and projected flowrates. The economic investment to perform this retrofit was higher than expected and didn't yield the level of efficiency increase expected.
- Replace the existing 201-JA/JB pumps with pumps specified to the current requirements for the process.

CONCLUSIONS

The hardest task of all is to convince those with the money, (the Operations Line Organization), that the least expensive long term solution involves spending dollars on new pumps. With that in mind, the group evaluating the alternatives looked at multiple options to hold costs down. The group evaluated six new pumps including the pump installation and operating costs against the retrofit case, the process modification case and the minimum flow bypass case. The group chose the best financial and process fit of the new pump offerings. Further savings can be gained by utilizing the existing baseplate and installing an adapter plate to fit the new pump. For additional savings, the existing motor can also be used with little efficiency penalty. One last item offered to tempt operations into approval of spending the funds is to break with tradition and recommend only one of the original large pumps be replaced and, therefore, reduce costs in half again.

Yearly energy savings with this approach are \$16,300 over the existing pump operation at 700 gpm. The total payback period based on energy savings will be 1.7 yr. Maintenance costs were not considered in the benefits, since they were not uniform with time nor were they accurately known over an extended time frame. The group feels exercising the option of installing one properly sized pump in the place of one of the existing pumps will save money and make this maintenance problem go away.

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