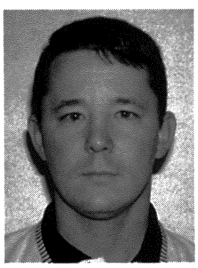
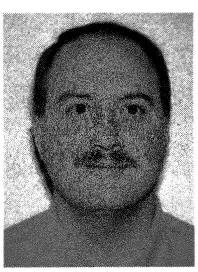
A CASE HISTORY-IMPROVEMENTS TO AN EXISTING COOLING TOWER SUMP AND HORIZONTAL SPLIT CASE PUMPS

by
Paul J. Wallen
Maintenance Engineer
Farmland Industries, Inc.
Lawrence, Kansas
and
Gregory S. Towsley
Mechanical Engineer
JCI Industries, Inc.
Lee's Summit, Missouri



Paul J. Wallen is currently a Maintenance Engineer at Farmland Industries, Inc., in Lawrence, Kansas. He previously worked as a Maintenance Engineer at Huntsman Petrochemical Corporation for three years, and Texaco Chemical Company for three years as a Project Engineer.

Mr. Wallen received a B.S. degree (Mechanical Engineering) from The University of Kansas (1991).



Gregory S. Towsley provides technical support to all areas of JCI Industries, Inc., located in Lee's Summit, Missouri. JCI Industries, Inc. is a value-added distributor of pumps, process equipment, and accessories, and provides after-sales services that include the repair of pumps and electric motors. Prior to the 11 years with JCI Industries, Inc., he was an Application Engineer for Goulds Pumps, Inc. for two years.

Mr. Towsley received a B.S. degree (Mechanical Engineering) from The University of Kansas (1986). He is currently pursuing courses leading to an M.S. degree (Engineering Management) at The University of Kansas.

ABSTRACT

In adding a second urea plant in 1974 at a Midwestern nitrogen plant, higher cooling water requirements necessitated the construction of an additional cooling tower. A 130 million Btu/hour cooling tower was installed with two 18×20-22H horizontal split case pumps rated for 13,000 gpm at 115 ft total dynamic head.

From nearly the beginning of the continuous operation of these two pumps, severe vortices were visible in the cooling tower sump, as well as high amounts of noise and vibration. Maintenance costs on these units were higher than expected due to cavitation erosion on the impellers and replacement every 2.5 years. Of course, with the accelerated wear of the pumps, the required capacity and the efficiency were affected. A change in the material of construction of the impeller was made in an effort to increase the mean time between repairs.

With the inherent design of this pumping system, it is not economically feasible to correct all the elements that prevent these units from operating at preferred industry standards. This paper discusses the investigation in the analysis of the problems, concentrating on the review of the cooling tower sump, the improvements made utilizing published recommendations, and the results of the alterations. Other methods of increasing the pump life and efficiency of the units are discussed. The effects of these additional changes are also discussed.

INTRODUCTION

The design of an additional urea plant at an existing ammonia production facility for the Cooperative Farm Chemical Association (now Farmland Industries, Inc.) began in 1973. The additional plant required a new cooling tower with two 18×20-22H horizontal split case pumps rated for 13,000 gpm at 115 feet total dynamic head (TDH) each. Naturally, with the requirement of a cooling tower, a basin was designed to retain the cooling water.

During operation of the pumps, severe vortices in the basin, near the pump suction inlet, were clearly visible. As is typically observed when vortices are present, unusually large amounts of noise and vibration were present at the pump. Based on physical observation, cavitation is obviously a problem, although the net positive suction head (NPSH) available is greater than the NPSH required. However, as discussed in Hydraulic Institute (1998), the recommended NPSH margin ratio for this type of pump and service is not being met. Due to reductions in performance and routine maintenance inspections, the rotating elements of the pumps were typically repaired every 2.5 to 3 years. These repairs included the replacement of bearings, the impellers, and wear rings.

As requirements increased for cooling water in the production of Farmland's product, attempts had been made over the years to improve the performance of the pump and reduce the cost of maintenance. Minimal changes were made due to budget restrictions.

Farmland made contact with a local rotating equipment sales and service organization to again attempt to improve performance of the pumping system and reduce maintenance costs. As the NPSH margin ratio could not be improved without major capital expenditures, initial discussions targeted two objectives:

- Improve pump suction conditions, focusing on basin improvement
- Improve pump performance

In order to meet the objectives, the history of the pumping system would require investigation, as many of the personnel involved with the equipment over the years were no longer available. The original and current conditions of service and installation were reviewed. Possible corrections to the sump and mechanical improvements to the pump were researched. The best alternatives were selected and implemented based on the least amount of time and costs required. The results of the modifications to the sump and pumps were then reviewed to determine if they aided in meeting the objectives stated.

HISTORY

The pumps' technical characteristics are shown in Table 1. In addition, the original condition of service and impeller diameter for each pump is shown in Table 2. These original conditions required only one pump in operation to meet the new plant's cooling water requirements. In 1988, additional cooling water was required for increased plant production. Two pumps operating in parallel met the new service condition requirements with a larger impeller diameter, while remaining under the motor design horsepower.

Table 1. Pump Technical Characteristics.

Speed	N_{ss}	Impeller Eye Dia.	Eye Peripheral Speed (U_{eye})
1180 RPM	7838	19.18 in.	98.83 fps

Table 2. Pump Conditions of Service and Impeller Diameter.

Conditions	Capacity	TDH	Eff.	NPSHR	IMP. DIA.
Original (1973)	13,000 gpm	115 ft.	82%	28 ft	19.5"
Revised (1988)	9,000 gpm	160 ft.	76%	18.5 ft.	20.0"

The original operating point, as shown in Table 2, was at approximately 108 percent of the best efficiency point (BEP) for its impeller trim. The new system condition point, for the larger impeller diameter and parallel operation, is at approximately 72 percent of BEP for its impeller trim. As single pump operation does not occur due to process requirements, no flow conditions are available for this situation.

As previously stated, the reoccurring high maintenance costs became the driving factor for review of the pumps and the cooling water pumping system. Due to pump failure, reduced pump performance, or scheduled down time, the impeller, wear rings, shaft sleeves, bearings, and packing were replaced every 2.5 to 4 years. The typical material costs were approximately $16,000, with labor costs at approximately $1300 per repair. Typically, the damage found to the pumps included cavitation damage (Figure 1) and discharge leakage flow (Figure 2).

In an effort to reduce the damage to the impellers due to what was perceived as cavitation and vortices, material changes of the impeller were made to extend impeller life. The initial pumps were provided with iron impellers. Per Hydraulic Institute (1994), iron is not a recommended material of construction for impellers, due to the aerated condition of cooling water. As recommended, bronze impellers were installed in 1985. A ceramic metal coating was also applied to the bronze impellers. This can be seen in Figures 1 and 2. In 1988, with the increase of the impeller diameter, new impellers were purchased in 316 stainless steel material. This material is also recommended by Hydraulic Institute (1994) for use in cooling water services. During the review of materials for impeller metallurgy enhancement, a harder, more wear resistant material was desired. Other materials than cast 316 stainless steel (ASTM A743 CF8M), such as a 13 percent chromium iron (ASTM A487 CA6NM), were considered. However, these nonstandard, wear resistant materials for split case pumps were not available from the original equipment manufacturer (OEM) to meet the delivery requirements of the owner.

Figure 1. Cavitation Effects on a Bronze Impeller with Ceramic Metal Coating.

Figure 2. Discharge Leakage Effects on a Bronze Impeller's Wear Ring.

It must be noted that an alternate impeller design for this pump was not available from the OEM as standard. An alternate impeller design may have allowed for higher flow characteristics closer to BEP, or improved hydraulic flow patterns within the pump.

The suction specific speed of the pump impeller, approximately 7840, is acceptable for this impeller design. However, the suction energy, as described by Budris (1998), falls in the high to very high suction energy range. This results in high vibration, suction pressure pulsation, and severe cavitation.

Even with the change of operating conditions in the pumps and materials of construction, damage was still occurring to the 316SS impellers (Figure 3). Of course, the continued use of metal wear rings required typical clearances as recommended in the petroleum industry. This, in turn, reduced pump performance.

SUMP DESIGN CONSIDERATIONS

Introduction

The construction of the new urea plant and cooling tower basin took place within the company's existing property. It must be noted that ample space was available for the construction of the basin and location of the pump units. Figure 4 (Plan View) and Figure 5 (Elevation View) show the original layout of the pump suction inlet area of the cooling tower basin. Figure 6 shows a view of the pump unit and cooling tower basin installation. Figure 7 is a view of the pump suction inlet piping.

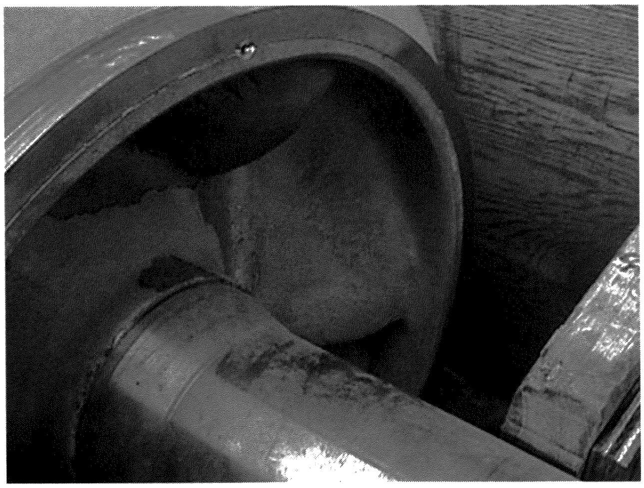


Figure 3. Cavitation Effects on a 316SS Impeller with Ceramic Metal Coating.

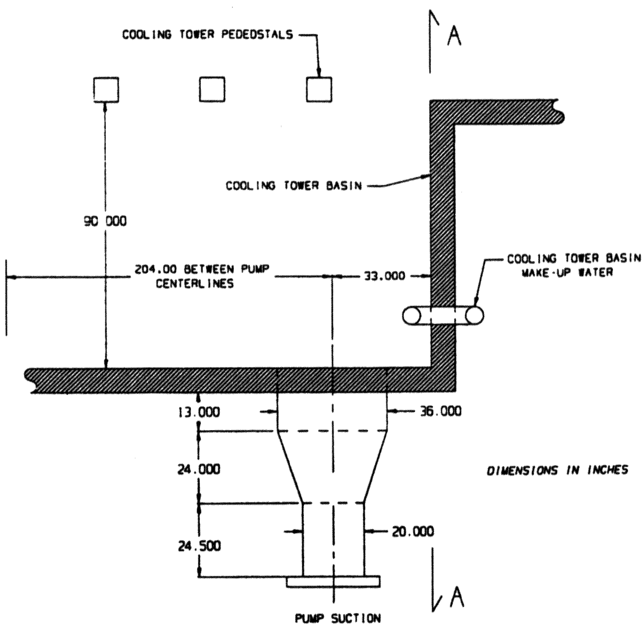


Figure 4. Original Cooling Tower Basin—Plan View.

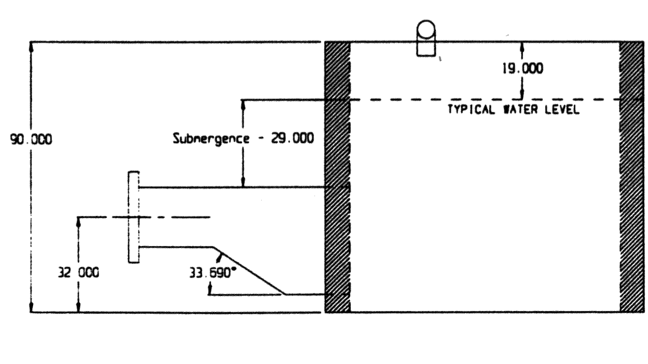


Figure 5. Original Cooling Tower Basin—Elevation View.

Since the construction of the cooling tower basin, water level vortices have been present. Both pump units always had two vortices located near the suction piping inlet, as is depicted in

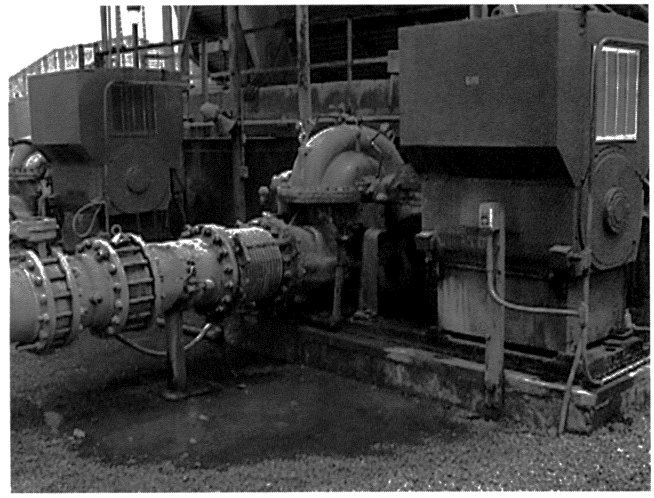


Figure 6. Pump and Cooling Tower Basin Installation.

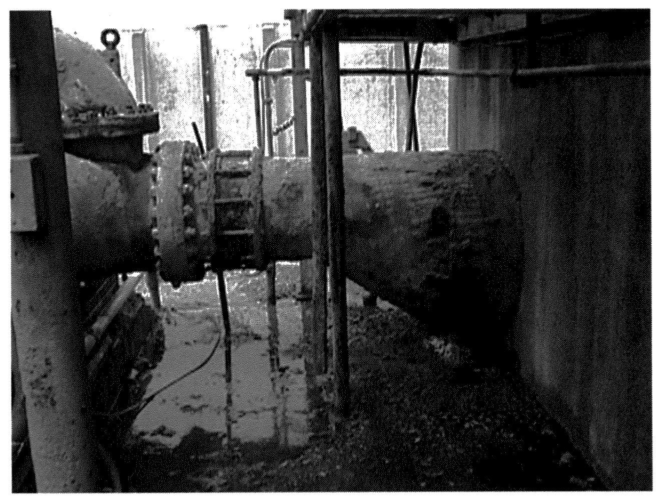


Figure 7. View of Pump Suction Inlet Piping.

Figure 8. Based on Hydraulic Institute's (1994) Vortex Classification System, as shown in Figure 9, the vortices were a Class 4, due to the air/vapor bubbles observed in the core. Occasionally, Class 5 vortices were also observed. As this chart depicts, large amounts of air and any top-water floatables would be immediately pulled into the vortex, and eventually into the pump.

From Ingersoll-Dresser Pump Company (1988) and Hydraulic Institute (1994), sufficient submergence and proportional design of pump basins to distribute even inlet velocities are necessary to reduce the likelihood of vortices, excessive noise and vibration, and to ensure that the performance of the pump is maintained. Each of these references discusses, in detail, recommended designs and arrangements based on model testing and field experience.

Existing Design and Dimensions

Utilizing the Hydraulic Institute's *American National Standard for Centrifugal Pumps* and Ingersoll-Dresser Pump Company's *Cameron Hydraulic Data*, the existing cooling tower basin design and dimensions were compared with those published in these references. Figure 10 provides typical dimensions for horizontal pumps in dry pit sumps. Figure 11 provides information on recommended sump dimensions based on flow. Figures 12 and 13 are used to apply the recommended dimensions from Figure 11.

The actual and recommended dimensions are shown in Table 3. These are based on the revised capacity of 9000 gpm and inlet diameter (D) of 36 inches.

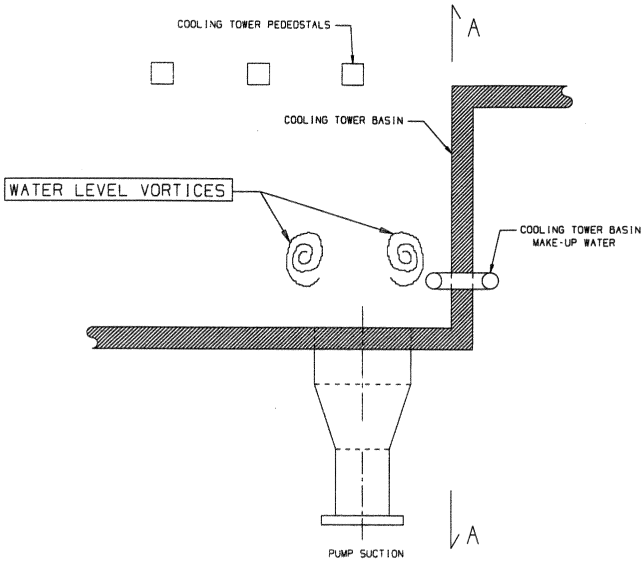


Figure 8. Location of Water Level Vortices.

Table 3. Actual and Recommended Sump Dimensions.

	Recommended	Actual
Submergence (S)	36 inches[1] 46 inches[2]	19.00 inches
Pump Cell Width (W)	58 inches	36 inches[3] 204 inches[4]
Reducing Angle	10 degrees	33.69 degrees

Table 3 notes:
1. Ingersoll-Dresser Pump Company (1988).
2. Hydraulic Institute (1994).
3. Dimension to wall.
4. Dimension to centerline of second pump.

Sump Corrections to the Existing Basin

Based alone on those dimensions in Table 3, major modifications to the cooling tower basin and pump installations would be required. However, reconstruction of the basin, moving the pump units, and modifying the piping are cost prohibitive.

Utilizing Figure 10 and Figure 14, modifications were made to the sump to reduce or eliminate vortices. Four modifications were made to the cooling tower basin at the pump suction inlets as described below and shown in Figures 15 and 16.

• *Make up water*—The cooling water basin make up water was located approximately 24 inches from the suction inlet of one pump. It also emptied into the basin significantly above the water level. Although cooling water is inherently aerated, the turbulence caused by the original arrangement created additional bubbles and a larger vortex. To improve the piping arrangement, the make up water pipe was extended down below the water level.

• *Perforated baffle*—The installation of a perforated baffle in front of the suction inlet allows for "straightening" of the flow and equalizes the distribution of the velocity into the pump suction inlet. In lieu of a "perforated baffle," stainless steel mesh screen, with 5/8-inch openings, was utilized. The existing sump design limited the location of the screen from the suction inlet, as recommended by Ingersoll-Dresser Pump Company (1988) and Hydraulic Institute (1994). The installed location was the best to allow complete access to the screens.

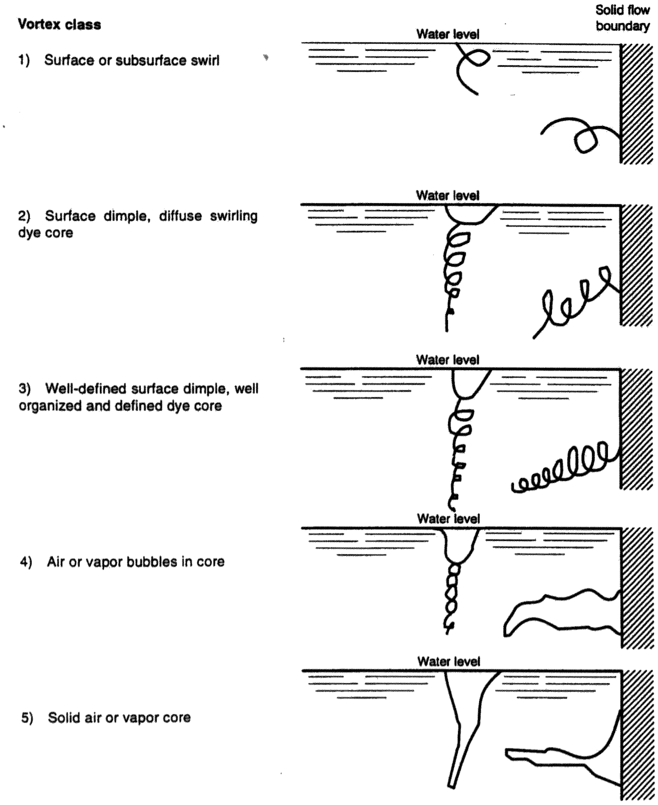


Figure 9. Vortex Classification System. (Courtesy of the Hydraulic Institute)

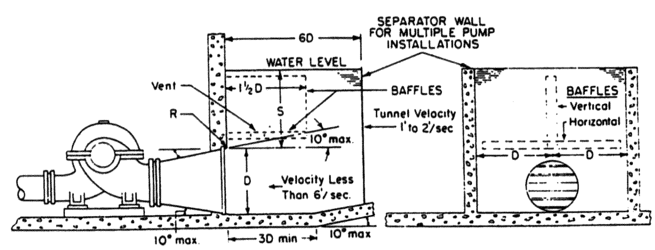


Figure 10. Inlet Basin Recommendations—Elevation. (Reprinted with permission from Cameron Hydraulic Data, Seventeenth Edition, Second Printing. Copyright Ingersoll-Dresser Pump Company, Liberty Corner, New Jersey.)

• *"False wall"*—As the pump suction inlets are not located near separation or cell walls as recommended, it was necessary to fabricate and install a "false wall" to assist in improving the pattern of the inlet flow (Figure 17). This false wall was also fabricated of stainless steel. This wall was located approximately 64 inches from the side basin wall to reduce the velocity of the water entering the suction inlet to the recommended 1 ft/sec (or less) velocity. This wall also supports the screens, as well as the horizontal baffle as described below.

• *Horizontal baffle over the pump suction inlet*—Typical submergence over the pump suction inlet was approximately 29 inches. Based on the Ingersoll-Dresser Pump Company (1988) and Hydraulic Institute (1994), the submergence should be one foot for each foot per second velocity at the suction inlet diameter. This corresponds approximately to 49 inches of submergence for the original capacity and 34 inches of submergence for the revised capacity. As this level of submergence is not possible in this cooling tower water basin, a horizontal baffle was installed. Ingersoll-Dresser Pump Company (1988) states that the submergence required by the pump can be

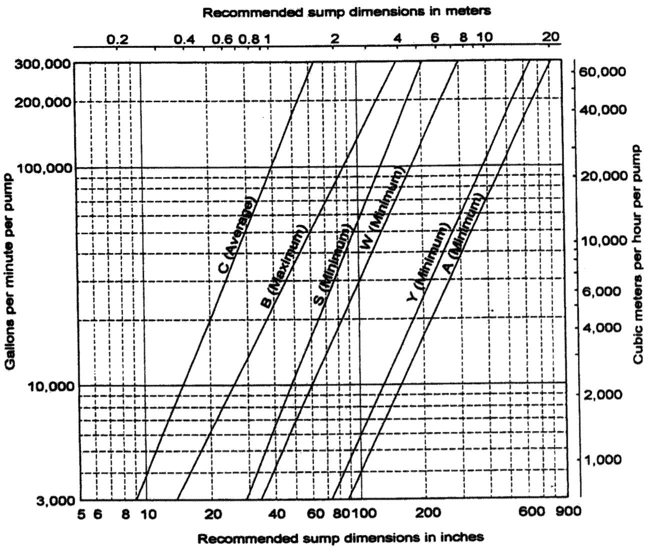


Figure 11. Sump Dimensions versus Flow. (Courtesy of the Hydraulic Institute, 1994)

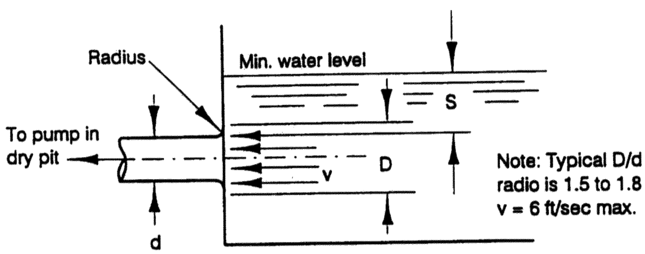


Figure 12. Elevation View—Horizontal Intake—Circular Section. (Courtesy of the Hydraulic Institute. 1994)

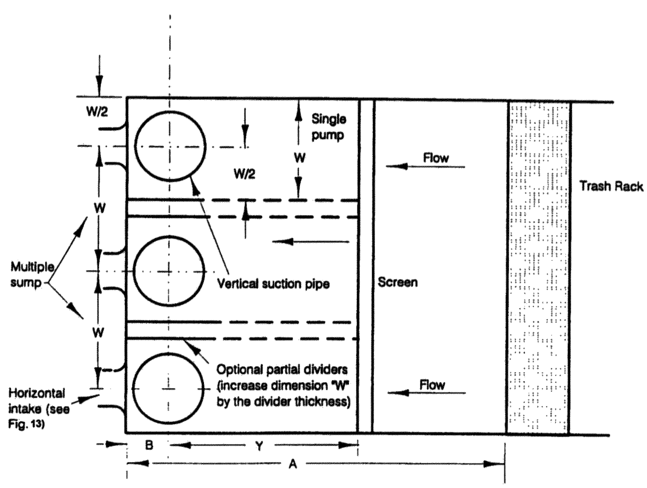


Figure 13. Sump Dimensions Plan View. (Courtesy of the Hydraulic Institute, 1994)

reduced by half with the installation of a horizontal baffle. Existing valves installed on the suction inlet wall required that the horizontal baffle be installed as near to the wall as possible, without interfering with the valves. The horizontal baffle was extended to the screens. It must be noted that no reference could be found to recommend the height above suction inlet for the horizontal baffle. The horizontal baffle was also constructed of stainless steel material. To also assist in reducing the possibility of vortex formation, the level of the water in the cooling tower basin was maximized.

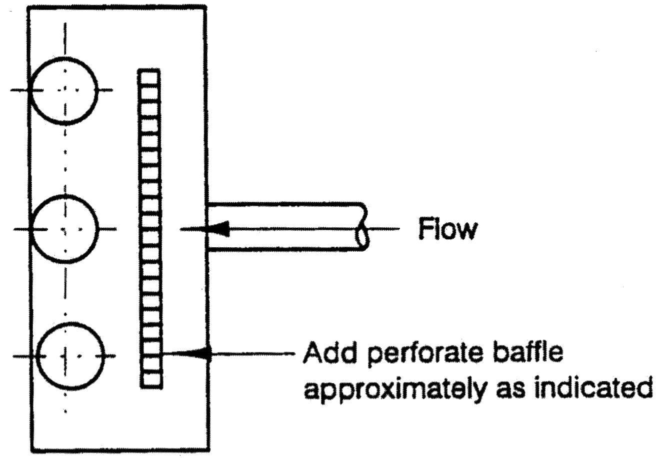


Figure 14. Correction of Existing Sumps. (Courtesy of the Hydraulic Institute, 1994)

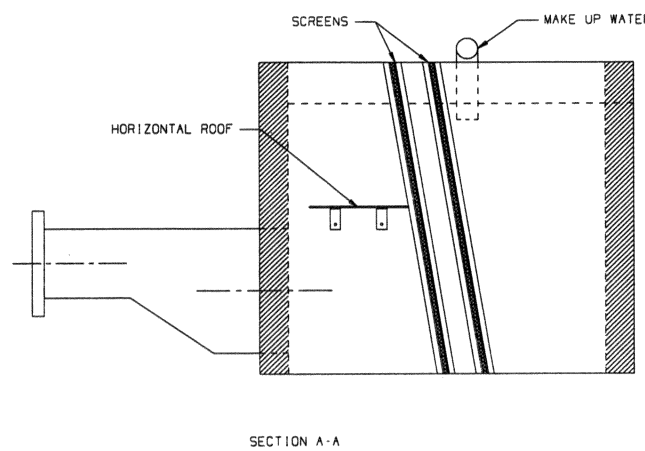


Figure 15. Elevation View of Cooling Tower Basin with Modifications.

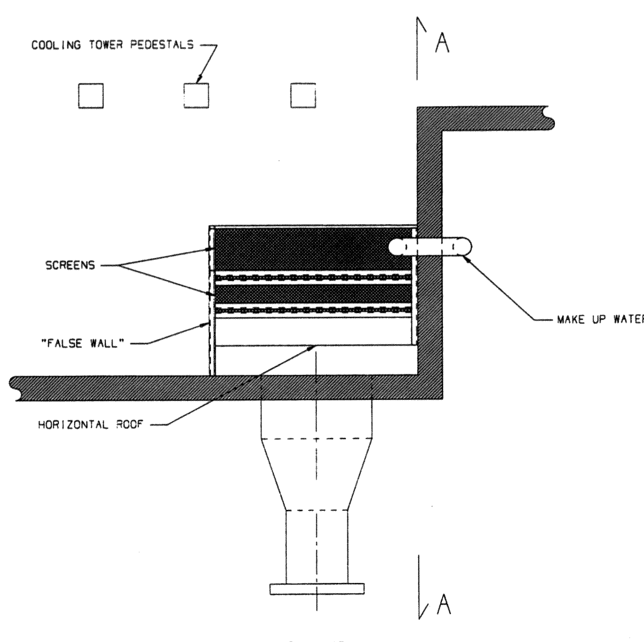


Figure 16. Plan View of Cooling Tower Basin with Modifications.

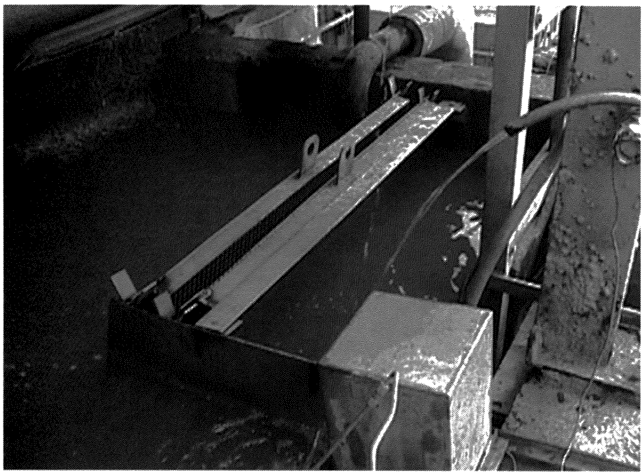


Figure 17. View of "False Wall" and Screens.

PERFORMANCE CONSIDERATIONS

Introduction

The performance that is expected from a user's equipment can be considered as a function of its output capacity, pressure, power, noise, etc. If the output is not meeting the requirements of the system, then an evaluation of the situation should be considered. In addition to the output, the cost of maintenance on the equipment is a factor in the equipment's performance. If maintenance costs are high, then the equipment is not performing to the expectations of the user.

As previously stated, the high maintenance costs were the incentive in investigating possible improvements to the cooling tower sump and the pumps. Maintenance was initiated by a reduction in the performance output of the pumps. The original design of the cooling water system did not incorporate a flow monitoring device at the pump. Typically, the discharge pressure and motor amperage were utilized in monitoring the pumps' performance. As vortices and other undesired hydraulic conditions were present, vibration monitoring was not performed. The vibration was so violent, accurate readings were not possible.

Effects of the Sump Design on Pump Performance

A cooling water pump's performance can be affected by many factors. Numerous articles have been written about cooling water pumps and the effects of cavitation, NPSH available, entrained air, submergence, dissolved air, pump design, and suction piping. Many of these factors are evident in this installation, but may never be eliminated due to costs, inherent pump design, or environmental conditions. In investigating and improving this particular installation, the focus was the improvement of the cooling water basin and the pump materials of construction, to extend the mean time between repairs.

As can be seen in Figure 5 and Figure 7, a standard eccentric reducer is installed in the horizontal position. With the eccentric reducer positioned in this fashion, there are no pockets for air to become trapped. It is therefore unlikely that cavitation is occurring due to air entrapped in the suction piping being drawn into the pump with the incoming water flow.

Pump performance is affected not only over time due to cavitation erosion, recirculation, or normal wear, but during normal operation of the pump. As stated in Karassik (1986), air bubbles caused by vortices can lower the efficiency of the pump, as well as cause the impeller to vibrate. It was also observed in this installation, prior to the sump modifications being made, that the vortices that were present, magnified by entrained air, reduced the suction pressure at the pump. As could be expected over time, damage to impeller vanes, as well as increased wear ring clearances, reduced the effectiveness of the pump in operation.

Material and Mechanical Improvements

• *Impeller*—Hydraulic Institute (1994) provides a guide to applying pumps in various services, including cooling tower water. At a minimum, bronze or stainless steel materials are recommended for the impeller, due to the aerated condition of the water. The pumps were originally provided with cast iron impellers. The effects of the design of this cooling water basin and system, as well as the operating conditions, caused extreme damage to the cast iron impellers within 2.5 years.

As stated previously, the impeller material of construction was later changed to bronze. Although severe damage occurred to the bronze impellers, the extent was not to that of the cast iron material. A ceramic coating was later applied to the bronze impeller, but as can be viewed in the previous Figures 1 and 2, the coating eventually deteriorated as well. Although the ceramic coating may have extended the life of the bronze material, it is not known how the coating initially affected the impeller eye velocities, or the balance of the rotating element as the material gradually deteriorated. The bronze impeller shown in the figures of this paper was replaced prior to a complete failure, four years after installation.

The material of the impeller was later changed to 316 stainless material, with a ceramic coating. The damage to the stainless steel material was considerably less, as seen in the previous Figure 3 and Figure 18. The damage on the suction side of the vanes has not penetrated the vanes. The 316 stainless steel impeller shown in the figures was replaced after 5.5 years of being in service. There is noticeably less wear than with the bronze impeller. The additional life of the impeller assisted in reducing the operation and maintenance costs of the pumps. In addition, the amount of wear found can now be used to estimate the optimum time to replace the impeller, while meeting system requirements and keeping operation and maintenance costs minimized.

Figure 18. Damage to 316 Stainless Steel Impeller.

• *Wear rings*—Again, the cast iron wear rings that were originally installed were later changed to bronze. The previous Figure 2 shows the effects of discharge leakage flow on the bronze wear rings at the impeller tips. This type of damage cannot be seen in Figures 3 and 18, showing the 316 stainless steel impeller wear rings with bronze case wear rings.

A change to a polyetheretherketone (PEEK) material for the wear rings was made in December 1997, with the cooling water basin modifications. The characteristics of toughness, strength, chemical resistance, low coefficient of friction, and wear resistance make PEEK materials ideal for the substitution of metals in wear rings. The installation of PEEK case wear rings and 316 stainless steel impeller wear rings allowed for the clearance to be reduced to

0.014-inch, compared with the 316 stainless steel and bronze wear ring clearance of 0.034-inch.

• *Balance of rotating element*—Previous repairs of these horizontal split case pumps do not show documentation of the rotating element being balanced. As part of the repair completed on these pumps, balance of the entire rotating element was completed by the local service organization. Although the International Standards Organization (ISO) recommends a balance quality grade of G6.3 for pump impellers, the rotating element was balanced to the G2.5 balance quality grade. This assisted in the reduction of vibration in the pump unit.

RESULTS

Pressure

• *Suction*—Initial gauge readings on the suction side of the pump showed a pressure of six inches of water. With the modifications made to the cooling tower basin, the gauge readings showed an increase of suction pressure to 19 inches of water. It is assumed that the difference between the gauge readings and the actual water level above the gauge is due to the inlet friction losses and turbulence. This will assist in improving pump performance.

• *Discharge*—With the modifications made to the cooling tower basin and to the impeller wear ring clearances, the discharge pressure increased approximately 5 psig with the pump in a "new" condition. The tighter wear ring clearances reduced discharge leakage flow from a calculated 108 gpm to 34 gpm.

• *Vibration*—Vibration monitoring was not a routine practice prior to the modifications to the sump and the pumps. The minimal vibration data previously taken show numerous mechanical and hydraulic problem sources that have been corrected. This vibration, as described by the owner, actually caused high vibration in the catwalk above the unit (Figure 6 and Figure 7), as well as other associated piping and equipment. The current vibration readings are well within all recommended vibration specifications. The owner reports that the current vibration is half what was previously present. However, the vibration readings and graphs indicate cavitation and vane pass frequency. It must be noted that vibration readings were also taken at the point where the suction piping is reduced to 20 inches. Cavitation and vane pass frequency can also be observed with these readings.

• *Visual*—As can be seen in Figure 19, the vortices, as indicated in Figure 8, are no longer present. A smooth, stable flow into the pump suction inlet can be observed. The nonexistence of the vortices is a tremendous improvement over the ever-present, violent vortices that previously existed.

• *Noise*—Although elimination of all noise was not achieved with the cooling tower basin modifications and pump improvements, a noticeable reduction presently exists. In closing the discharge valve to approximately one-eighth open, no noticeable change in the level was detected. Due to the inherent design of the specific pump, as well as hydraulic conditions that cannot be improved, it is not expected that the noise will be totally eliminated.

Expectations

• *Life*—As the cooling tower basin modifications and pump improvements were only recently made, the life of the pump impeller and the reduction in mean time between planned maintenance can only be anticipated to improve. An ability to obtain vibration readings on a scheduled basis, as well as pressure and motor amperage readings, will provide a baseline to track the conditions of the pump units.

• *Costs*—Based on the improved life expectations, overall operating and maintenance costs are expected to be decreased. As the 316 stainless steel impeller had a typical life of 5.5 years, those costs are expected to be distributed over a longer period. Only continued improvements made to the pump and system will enhance these savings further.

Figure 19. View of Suction Inlet Area.

CONCLUSIONS

Many factors affect the reliability and performance of horizontal split case pumps in cooling tower water services. Although all these factors may not be correctable, those that are correctable should be investigated with the financial limitations available. With respect to the pumping system discussed in this paper, it is accepted that it is a substandard installation. The authors chose to initially analyze two areas, cooling tower sump modifications and pump materials, to attempt improvements and reliability. With limited capital available for improvements, the objective was equipment reliability improvement. Ideally, intake modelling would be the best resource to determine the causes and solutions to the system problems.

Utilizing the many references available, including Ingersoll-Dresser Pump Company (1988) and Hydraulic Institute (1994), modifications proved to be successful within the desired expectations initially established. It should be required by all end users that the engineers of their pumping systems utilize these references, as well as other published standards, to design successful installations. It must also be noted that these references are guidelines only, and that each application and service should be reviewed individually. Users of these references must be aware of the publishers' disclaimers to the information provided in the guidelines. Through the efforts of members of The Hydraulic Institute, a new separate standard designated as "HI 9.8-1998 Pump Intake Design" will be available in December 1998.

The particular installation presented in this paper will continue to be monitored and other improvements investigated. A scheduled inspection of the pump units is planned for the Fall of 1998. The findings from this inspection may be provided at a later date as an addendum to this technical paper. The additional physical enhancements to the pump have increased the pump reliability and improved the wear characteristics. Additional improvements that will be considered are other material changes to the impeller (electroless nickel or tungsten carbide coating), impeller and casing modifications, and improvements to the pump suction inlet piping. These improvements are expected to continue to assist in increasing the mean time between planned maintenance, as well as reducing the overall operation and maintenance costs.

Finally, the successful performance of end users' equipment should come with a working partnership with the equipment's manufacturer and local sales and service organization. It is in the best interest of both parties that their primary objective is properly operating and performing equipment. Though extensive quantitative data has not been produced, the qualitative results are important to meeting predetermined objectives.

REFERENCES

American Society for Testing and Materials, 1998, *Book of ASTM Standards*, American Society for Testing and Materials.

Budris, A. R. and Mayleben, P. A., 1998, "Effects of Entrained Air, NPSH Margin, and Suction Piping on Cavitation in Centrifugal Pumps," *Proceedings of the Fifteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 99-108.

Hydraulic Institute, 1994, *ANSI/HI 1.1-1.5-1994—American National Standard for Centrifugal Pumps*, Hydraulic Institute.

Hydraulic Institute, 1998, *ANSI/HI 9.6.1-1998—American National Standard for Centrifugal and Vertical Pumps for NPSH Margin*, Hydraulic Institute.

Ingersoll-Dresser Pump Company, 1988, *Cameron Hydraulic Data*, Seventeenth Edition, Second Printing, Liberty Corner, New Jersey, pp. 1-22 through 1-26.

International Standards Organization, *ISO Balance Standard DR 1940*, International Standards Organization, Geneva, Switzerland.

Karassik, I. J., et al., 1986, *Pump Handbook*, Second Edition, New York, New York: McGraw Hill.

BIBLIOGRAPHY

Benner, D. and Boyles, C. M., 1997, "Towering Above Breakdowns—Anticipating and Avoiding Cooling Tower Problems Through Predictive Maintenance," Benner Engineering and Plant Services Magazine, Plant Services Magazine.

Florjancic, S. S., Clother, A. D., and Chavez, F. J. L., 1993, "A Case History—Improved Hydraulic Design Lowers Cavitation Erosion and Vibrations of a Water Transport Pump," *Proceedings of the Tenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 81-90.

Hydraulic Institute, August 1997, "Cavitation Problems?" NPSH Margin Work Group, Plant Services Magazine.

ITT Industries, 1998, "The Mystery of Cooling Tower Pump Noise," Fluid Handling University/ITT Industries Website: http://fhs.ittind.com/press/mystery.html

Nelik, L., Silvaggio, J. A., Jr., Joseph, J. P. II, and Freeman, J., 1996, "The Use of Basic Hydraulics to Solve Field Problems," *Proceedings of the Thirteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 175-182.

Pembroke, J. P., Sabini, E. P., and Littlefield, D. E., 1998, "CFD Analysis of a Double Suction Cooling Water Pump," *Proceedings of the Fifteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 11-20.