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CIRCULATING WATER PUMP MODIFICATIONS PERFORMED IN CONJUNCTION WITH COOLING TOWER REPLACEMENT DURING A 35-DAY OUTAGE

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ABSTRACT:

Westar Lawrence Energy Center Unit 5 in Lawrence, Kansas is a 375 mW unit that utilized a closed loop circulating water system with a wooden cooling tower and three horizontal circulating water pumps to provide cooling water to the condenser.

The wooden cooling tower experienced rapid degradation over the past five years of operation. Chlorine was added to the water chemistry to reduce attack of the wooden components, but the chlorine levels in the water caused accelerated corrosion of the pump components and base plates. Circulating water system capacity was reduced due to worn pump components and excessive clearances. Unit load was limited by the lack of adequate circulating water capacity.

This paper discusses the pump and motor repairs, design changes performed in parallel with demolition of the wooden cooling tower, and replacement with a new fiberglass tower designed to utilize the existing concrete basin. All of the changes were implemented during an outage in April-May 2009.

Contained in the paper are the details of the following:

- Assessment and analysis
    - Establish baseline performance for the (3) existing pumps
    - Determining hydraulic upgrades to the pump
    - Determining mechanical upgrades to the pump
    - Perform a hydraulic model study of the pump intake
- Implementation
    - Implementing upgrades to the pump concurrent during a planned one month outage
    - Refurbishing the motors
    - Repairing and replacing pump baseplates
    - Implementing intake modifications
- Pump start-up and performance verification

The project was completed on schedule, and the performance exceeded the design targets.

NOMENCLATURE:

TDH = Total Developed Head, Ft

H-Q = Head-Capacity, GPM - Ft

HISTORICAL BACKGROUND

Three 1960-vintage Ingersoll-Rand model 36AFV pumps are used to provide circulating water to the condenser. All are driven by Westinghouse induction motors rated for 1500 horsepower at 442 rpm.

The pumps are a single-stage double suction between bearing design (Figure 1) featuring a cast iron casing, bronze impeller and carbon steel shafting. Both the inboard and outboard bearings are grease lubricated spherical roller bearings.

Pump, driver and base condition was reported as deteriorating over time (Figure 2). All of the pumps had been refurbished, however, significant casing damage was observed within the hydraulic passages, critical fits and packing bores. In addition, the impellers revealed significant damage due to cavitation and subsequent vane breakage. One of the baseplates had previously been replaced due to corrosion; the other two also exhibited poor condition.

Prior pump modifications to increase capacity included a speed change (new motors and pump modifications) from 390 to 442 RPM. The pumps are designed for 50% capacity each, with normal full load operation on two pumps and the third as an idle spare.

UPGRADE METHODOLOGY

In preparation for the outage, prior to pump and motor removal, a methodology was determined to address the known problems.

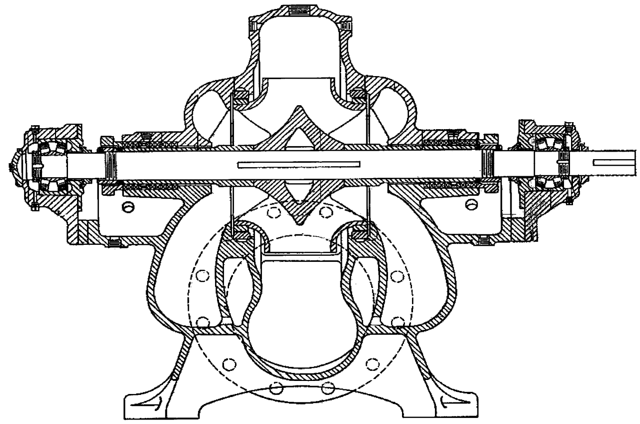


FIGURE 1: INGERSOLL-RAND MODEL 36A FV

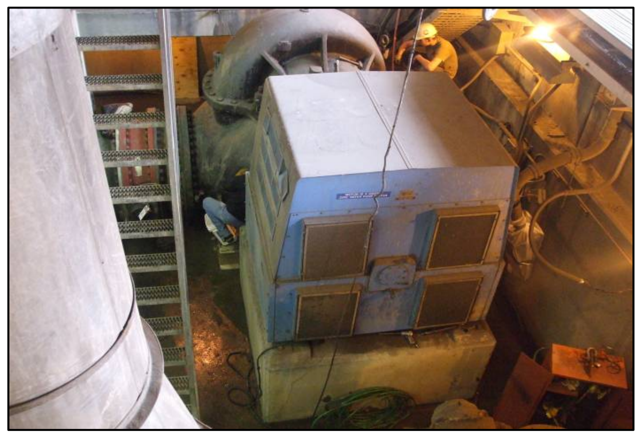


FIGURE 2: PUMP AND MOTOR MOUNTING

Planned Pump Repair:

- Disassemble and inspect
- Provide (2) new stainless steel impellers
- Inspect / Reuse (1) existing stainless steel impeller

- Provide new wear parts
  - Impeller rings
  - Shaft sleeves
  - Bearings
  - Gaskets / packing / miscellaneous
- Plane and rebore casings
- Coat casing internals with high grade epoxy
- Provide resistance temperature detectors
- Reassemble
- Inspect and repair / replace baseplates

Determine Existing Condition:

Perform Field Hydraulic and Mechanical Survey

Intake Model Study:

- Perform scale model test at a hydraulic laboratory
- Perform field hydraulic and mechanical survey

DETERMINE EXISTING HYDRAULICS AND PUMP INTEGRITY

Baseline testing was performed to establish current pump performance, define the system curve, and determine if any modifications were indicated.

Cooling water system capacity was measured on the twelve 30 - inch diameter fiberglass risers using a clamp-on (non-intrusive), ultrasonic flow meter (Figures 3 and 4).

The measured hydraulic data for the circulating pumps is displayed in Figure 5. Since the pump flow could not be individually isolated the flow was plotted as being even between the three pumps. It was believed to be lower for 501, basis visual observation of velocity through the trash racks, slightly lower amps (3%), and missing impeller vane segments found in the system.

The Total Developed Head (TDH) was 10.9 ft (11.1%), low compared to the expected head at the measured flow. This indicates that the pumps were worn with excessive ring clearances and/or impeller breakage.

FIGURE 3: FLOW METERING ON RISER #7

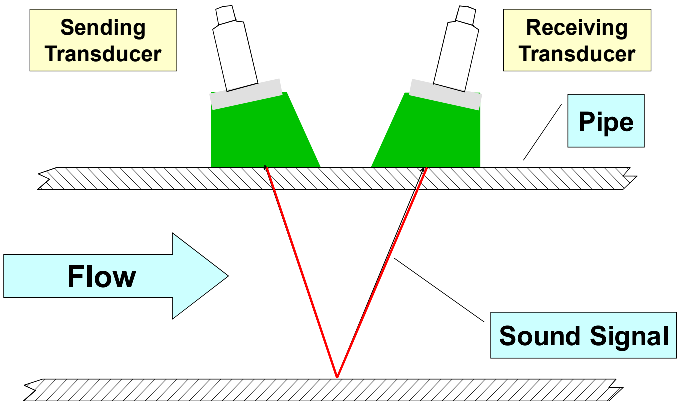


FIGURE 4: TWO PASS ULTRASONIC FLOW MEASUREMENT THE SOUND SIGNAL CROSSES THE FLOW STREAM TWICE FOR IMPROVED ACCURACY

The efficiency was also low by a similar percentage as the TDH, due to excessive ring leakage. The horsepower remained near expected for a “healthy” pump (i.e. the pump is doing work to pump flow through the ring clearances, but it is a loss since it doesn’t increase the H-Q that the pump is producing at the discharge nozzle).

Customer	Westar Lawrence 5	**Pump Size & Type**	36AFV	**Speed (RPM)**	442	**Capacity (gpm)**	56,666
Service	Circulating Water	**Number of Stages**	1	**Specific Gravity**	1	**Head (ft)**	75
Liquid	Water	**Npsha**		**Temperature (°F)**	Ambient	**Efficiency**	89.5%
Date	12/2/2008			**Motor**	1500 bhp 1.15 s.f.	**BHP**	1199

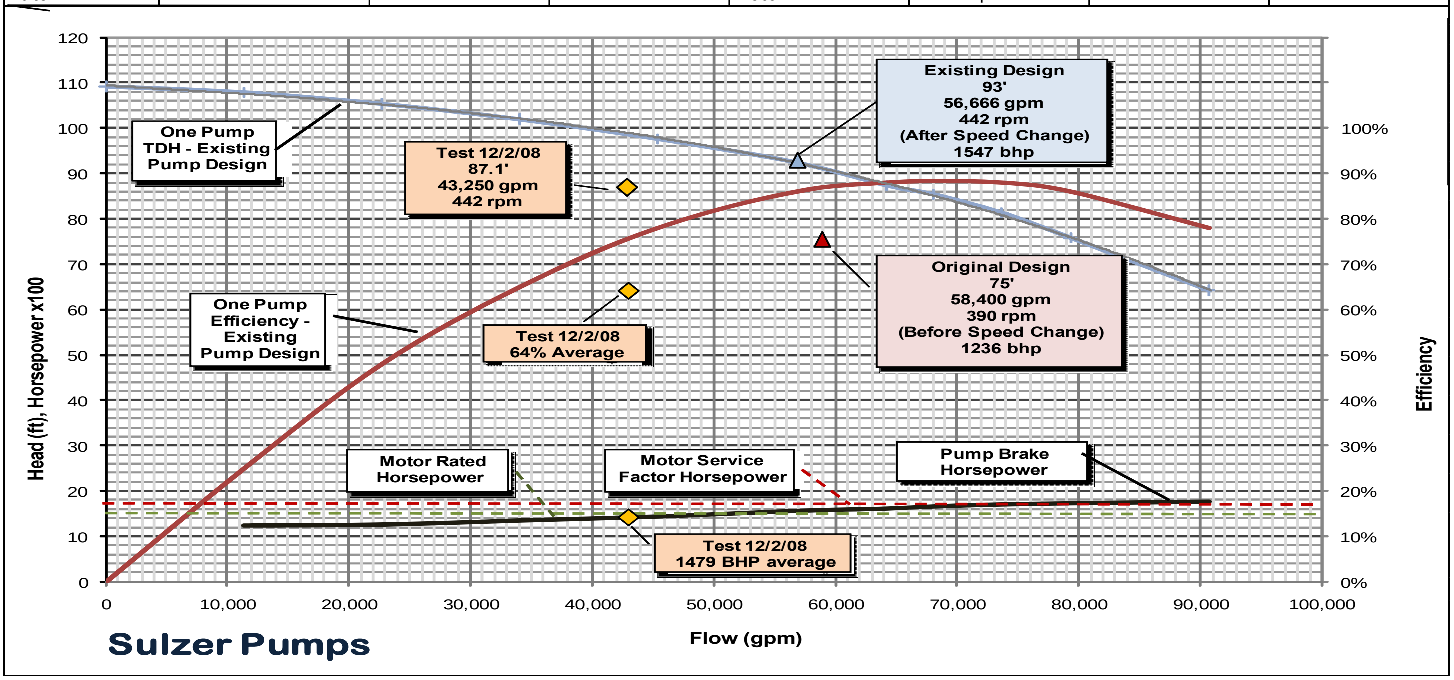


FIGURE 5: MEASURED HYDRAULIC PERFORMANCE

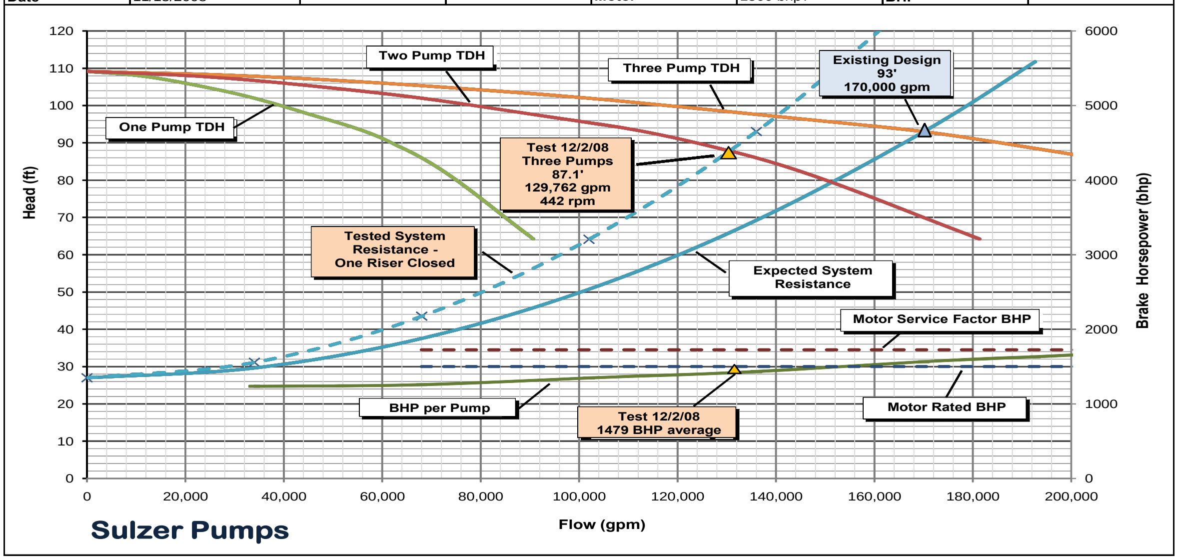
As shown in Figure 6 below, in addition to the pumps producing low TDH, the system resistance was higher than expected. This was likely due to one closed riser (increases friction loss in remaining risers), condenser pressure drop design value, and basin level below design.

CONCLUSIONS

1. The combined pumps were approximately 10.9 ft (11.1%) low in TDH at measured capacity of 129,762 gpm.
2. The pumps were tested with one riser valved out; the expected change with the riser open would be minor.
3. Condenser cleanliness was a factor during testing with:
    a. Design pressure drop 9.0 psi across condenser
    b. Pressure drop during testing 11.5 psi
    c. Pressure drop during last 6 months ranging from 10.6 to 13.0 psi at full load
4. The 501 pump appeared disproportionately low based on:
    a. Visibly lower velocity at trash racks
    b. Lower motor amps
    c. Impeller pieces found in system
5. The motors were operating near full load amps; however, the winding temperatures were relatively low with adequate thermal margin.
6. TDH design conditions of 93 ft @ 170,000 gpm total flow rate put the pumps slightly in the motor service factor (about 3% of 15% margin).

FIGURE 6: PUMP PERFORMANCE VS. SYSTEM CURVE

Customer	Westar Lawrence 5	**Pump Size & Type**	36AFV	**Speed (RPM)**	442	**Capacity (gpm)**	56,666
Service	Circulating Water	**Number of Stages**	1	**Specific Gravity**	1.00	**Head (ft)**	93
Liquid	Water	**Npsha**		**Temperature (°F)**	Ambient	**Efficiency**	
Date	11/13/2008			**Motor**	1500 bhp?	**BHP**	



SCALE MODEL INTAKE TEST AT A HYDRAULIC LABARATORY

On-going impeller damage and reduced pump performance were symptoms of the pump intake design issue. Clemson Engineering Hydraulics, Inc. (CEH) was contracted to conduct a physical hydraulic model (Figure 7) study of the pump intake.

The model was used to evaluate the hydraulic conditions within the intake and to determine if any circulation and or vortex activity was present in the intake or suction piping that would adversely impact the long term performance of the intake. In addition, the model was used to develop recommended modifications to remediate any adverse hydraulic phenomena which could impact pump performance.

Testing revealed overall conditions within the intake as turbulent and unstable. There was significant separation at the entrance to the three screen chambers which set up circulation within the common area downstream of the chambers (Figure 8). This increased the instability of flow entering the pump suction pipes. There was significant flow separation at the entrance to each of the pump suction pipes.

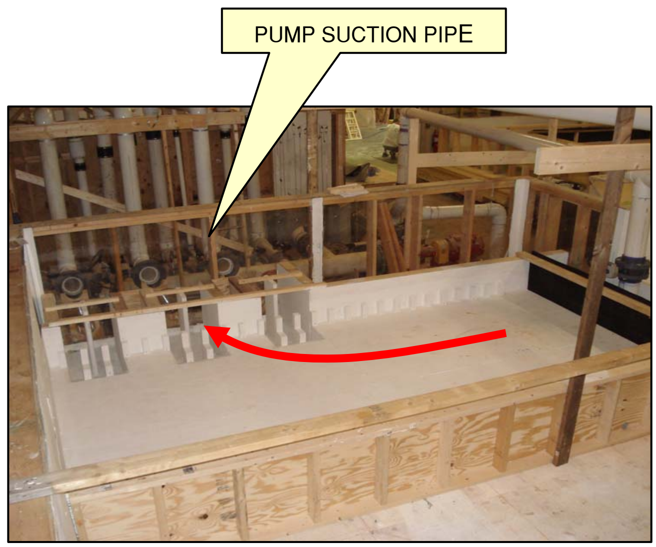


FIGURE 7: PUMP INTAKE MODEL

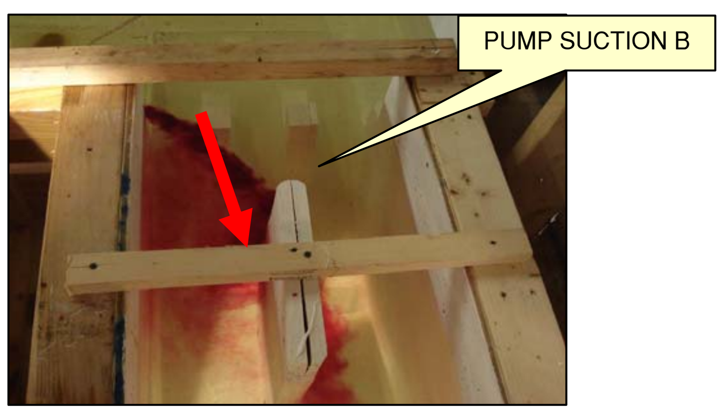


FIGURE 8: FLOW SEPARATION

The separation was most pronounced along the bottom of the pipe but was observed around the entire circumference. The separation actually caused circulation and upstream flow within the lower portion of the pipe. Frequent Type 3 to Type 5 (air entraining) surface vortices (Figure 9) were observed entering all three pumps during two different pump operating scenarios. Midflow vortices and intermittent floor vortices were also observed entering the pumps, but were less frequent and less severe than the surface vortex activity. Although pre-swirl was low, the recorded pre-swirl values for this intake were higher than expected for dry pit pump suctions of this type.

Focus was directed to modifications that could be pre-fabricated before the outage and installed relatively quickly once the outage started. A grating disk (Figure 10) was installed in the pumps at the entrance to each of the pump suction pipes (flush with the intake wall). The grating was effective at stabilizing the flow entering the pump and preventing separation within the piping. The grating was also effective at dissipating both surface and submerged vortex activity, as it entered the piping. However, the grating did not prevent air from being entrained into the suction pipe and subsequently reaching the pump. Installing a series of six (6) horizontal pipes (Figure 11) above each of the pump inlets prevented the surface vortex from fully organizing and entraining air into the piping. With the grating and horizontal pipes installed, velocity and turbulence fluctuations as well as pre-swirl were within criteria for all test conditions.

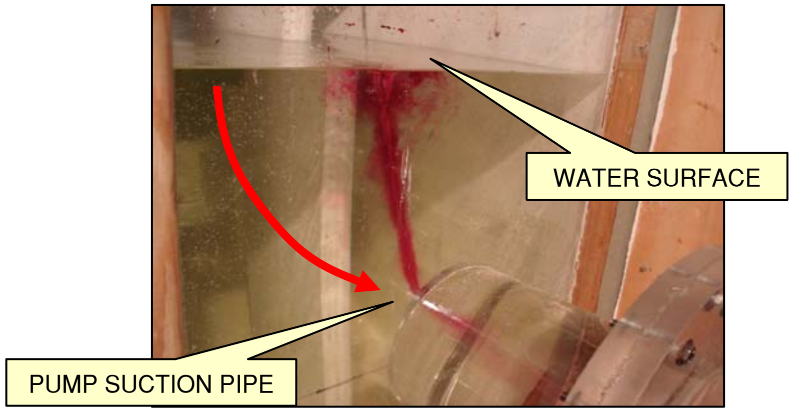


FIGURE 9: SURFACE VORTEXING

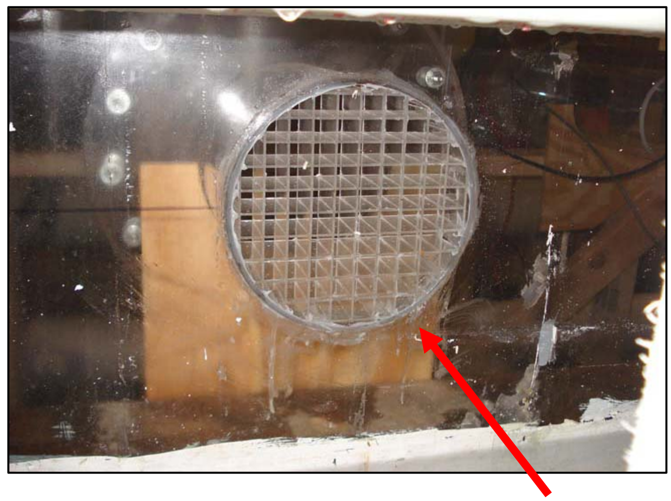


FIGURE 10: GRATING AT SUCTION

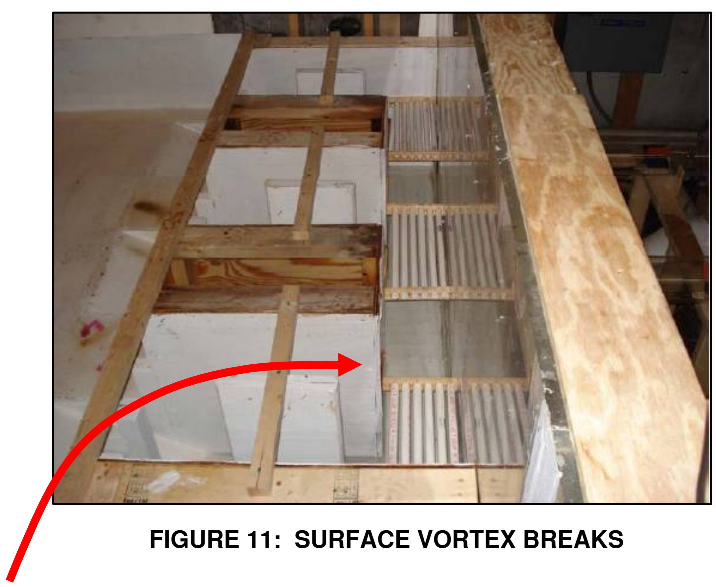


FIGURE 11: SURFACE VORTEX BREAKS

PUMP REFURBISHMENT AND UPGRADES

Prior to the outage spare parts pump inventory was gathered and forwarded to Sulzer repair shop. Major items included:

(1) Shaft
(1) Impeller
(2) Casing Rings
(1) Impeller Ring
(1) Inboard Bearing Housing
(1) Outboard Bearing Housing

The spare stainless steel impeller was reverse engineered (Figure 12), a vane layout and manufacturing drawing generated, and (2) additional impellers sourced – both in stainless steel. Minor hydraulic modifications were incorporated into both the spare and the new impellers. The balance of the spare parts were inspected and also reverse engineered in preparation for pump arrival

FIGURE 12: REPLICATED IMPELLER

Typical Findings – Pump 501 / 502 / 503 (Figure 13)

- Bronze impellers corroded with significant inlet tip damage, pump 501 was missing several large segments of vanes
- Shaft cracked at several locations
- Casing ring “spun” in casing (Pump 501)
- Excess wear
    - Impeller and casing rings
    - Shaft sleeves
    - Bearing housing bores
    - Stuffing box bores
- Casing corrosion throughout the hydraulic passages
- Cavitation damage on the casing at the impeller inlet

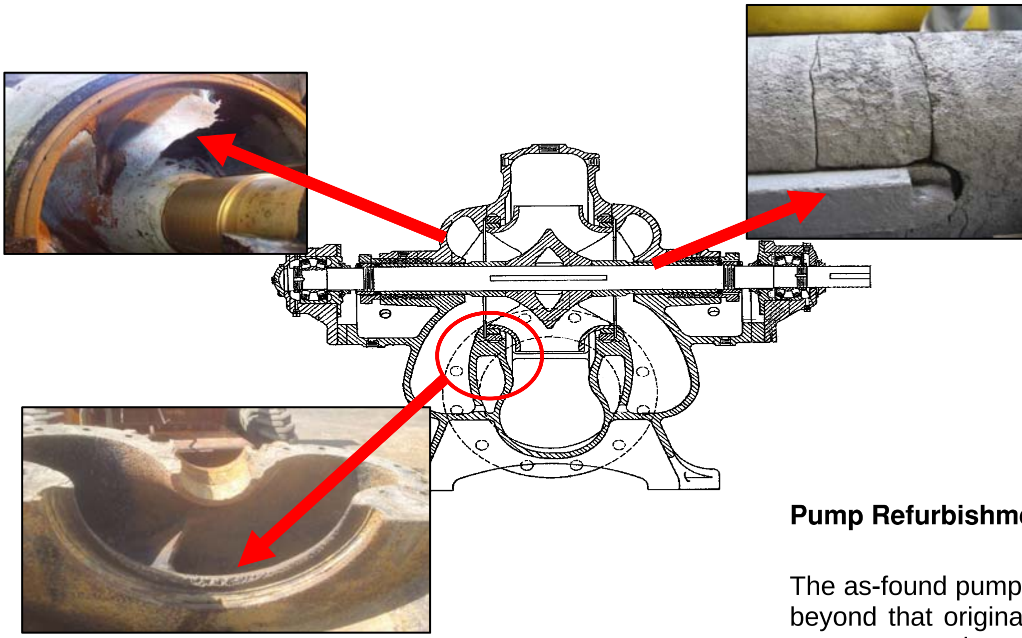


FIGURE 13: INGERSOLL-RAND MODEL 36AFV

Pump Refurbishment and Upgrades

The as-found pump condition presented a scope beyond that originally envisioned. Of particular concern was the condition of the casing at the casing ring bores (spun ring on one of the casings), extensive cavitation damage at the suction passage, and packing bores. Due to the unexpected damage, not all of the customer supplied spare parts could be used requiring redesign and expedited manufacture of casing rings, lantern rings, and gland rework.

Using the as - found pump geometry and the spare parts an engineering layout of the pump was generated (Figure 14). All damaged areas were thoroughly assessed and appropriate design modifications incorporated as follows (Figure 15):

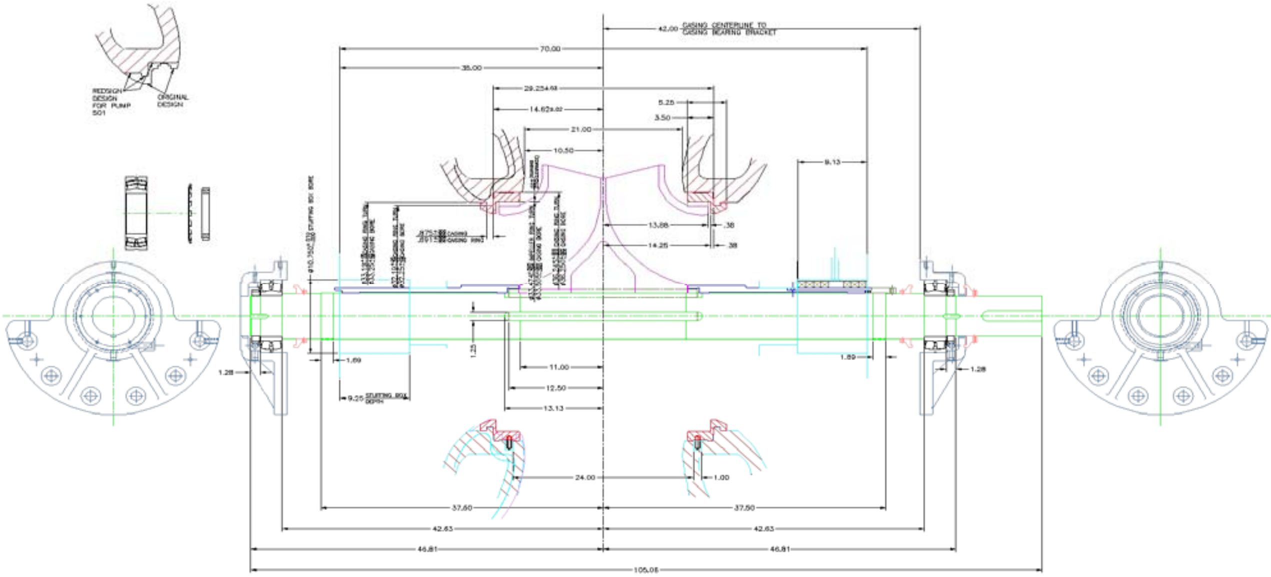


FIGURE 14: ENGINEERING LAYOUT

Casing and Casing Rings

The condition of the casing bore at the casing rings varied for each of the casings. The most significant damage was observed on pump 501 due to spun rings and cavitation damage. Customer supplied spare rings were not useable thus requiring oversizing the casing bores and manufacturing new 316SS rings with a revised design. The revised design incorporated both the ring and restoration of the cavitation damaged area of the suction passage.

Bearing Housings:

Inboard and outboard bearing housings were repaired by sleeving the bearing bore and machining for the Resistance Temperature Detector provisions.

Shaft Sleeves:

Bronze shaft sleeves exhibited extensive wear at the packing. Refurbishment was accomplished by shrinking and pinning stainless steel sleeves at the damaged area.

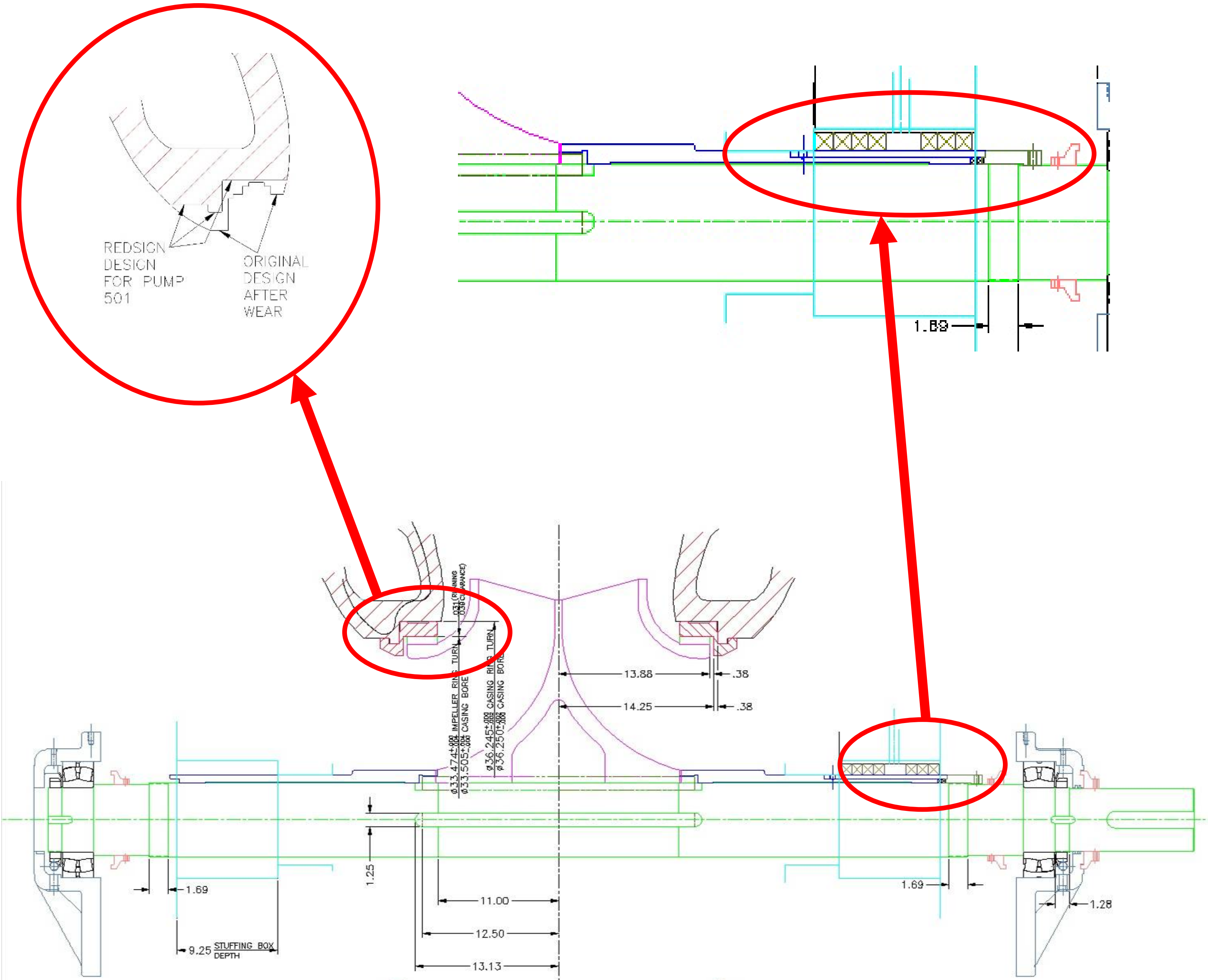
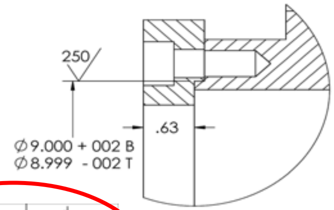


FIGURE 15: REPAIRED AREAS



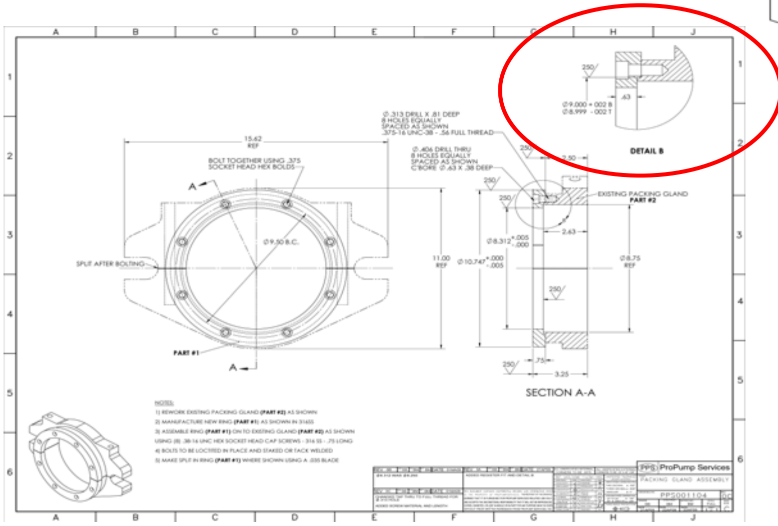
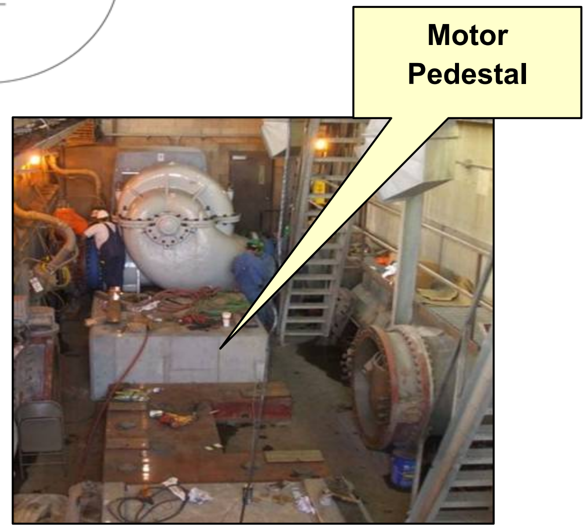


FIGURE 16: HALF GLAND MODIFICATIONS

Shafting:
New shafts were provided to replace existing shafts with significant cracking. Shafts were manufactured in 2 weeks to support the pump schedule.

Water Seal Cage and Stuffing Box Base Ring:
Damage to the casing stuffing bore required replacement of the noted components; a design was quickly established, manufacturing drawings completed, and orders placed to meet the required lead time.

Half Glands:
Damage to the casing stuffing bore also required rework of the half glands (Figure 16) to compensate for the enlarged stuffing box bores.



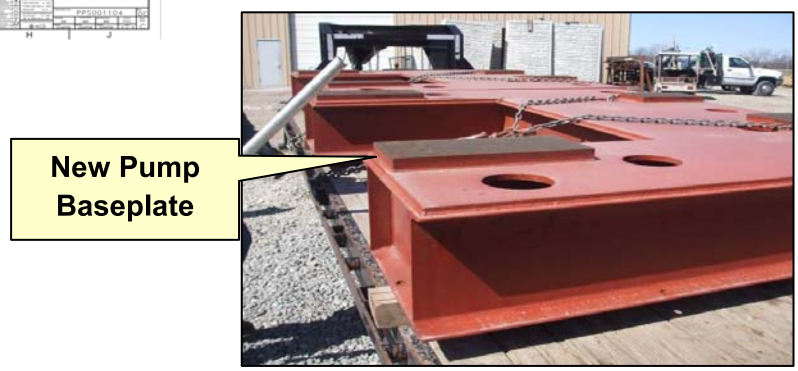


FIGURE 17: BASEPLATE AND SOLEPLATE

BASEPLATE REFURBISHMENT AND REPLACEMENT

The pumps are mounted to a fabricated steel baseplate grouted in place; motors are mounted on soleplates installed in a concrete pedestal. Concurrent with pump and driver repair, baseplate rework and replacement (Figure 17) was implemented. One baseplate 503 had been replaced during a prior outage, two required replacement. Baseplates were sourced prior to the outage. Immediately following pump and motor removal the baseplates were removed and the replacements grouted in place and field machined.

PUMP AND DRIVER INSTALLATION

Pump refurbishments were scheduled to accommodate ease of installation. The pumps exhibiting the least damage were completed first. The completed pump assembled weighs 25000#, is 9' long, 9' from flange face to flange face and approximately 9' high. Size and weight required partial pump disassembly at the vendor's facility for shipment (Figure 18).

Due to site crane limits, the pumps were shipped in three pieces:

- Lower half casing
- Rotor
- Upper half casing

Installation commenced with installation of the motor (Figure 19) on to the remachined soleplate and subsequent installation of the casing. With both installed the pump rotor and upper half casing assembly was completed. With the primary installation complete, peripheral work included pump to driver alignment, packing installation, and connecting the suction and discharge flanges.

A similar and consecutive methodology was used to install the second and third pump. Complete installation required 5 days for the pump and motor.

Unit scheduling required pump start-up of the two installed pumps in parallel with installation of the third pump.

The total time from the start of disassembly of the first pump to unit 5 restart was 41 days

POST REPAIR TESTING

Post repair testing was performed approximately one month after start up of the third pump.

The measured hydraulic data (Figure 20) for the Circulating Pumps is displayed in Table 1.

FIGURE 18: DISASSEMBLED PUMP

The replacement cooling tower did not include the external risers; therefore flow measurement similar to that performed 12/08 was not possible.

Table 1 chronicles data from both the control room and data acquired at both the pumps and condensers.

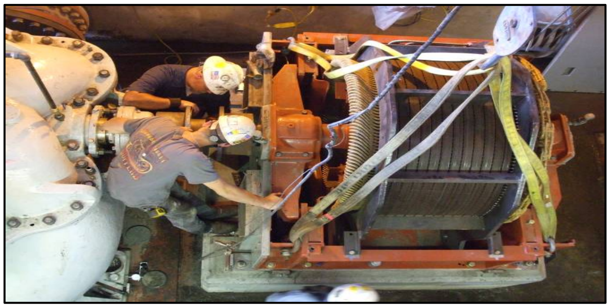


FIGURE 19: MOTOR INSTALLATION

FIGURE 20: SUCTION (TOP) AND DISCHARGE (BOTTOM) PRESSURE GAGES TYPICAL FOR ALL PUMPS

Table 1

	Pump / Motor 501	Pump / Motor 502	Pump / Motor 503	Condenser 501	Condenser 502
Motor Amps	208	209	202		
Pump Bearing Temp - IB(˚F)	140	94	128		
Pump Bearing Temp - OB (˚F)	108	93	110		
Condenser -P [psi]				9	8
Condenser -T [˚ F]				27	25
Turbine Backpressure [Hg]				3.4	3.4

The upgraded pumps provided a 10% flow increase while staying within the motor name-plate limitations. Vibration analysis, in conjunction with a noticeable decrease in audible noise, revealed a significant decrease in cavitation activity. Main steam turbine back pressure was returned to design levels for full unit loading during the summer season.

Summary

Pump and motor refurbishment were performed in parallel with cooling tower replacement including:

<u>Cooling Tower</u>
Wooden towers were demolished and replaced with fiberglass utilizing the existing basin.

<u>Pumps</u>
- Hydraulics and mechanical performance was restored.
- Select materials were upgraded to stainless steel for aggressive water
- Baseplates were replaced

<u>Intakes</u>
Modifications were implemented to reduce vortexing.

<u>Motors</u>
Motors had routine maintenance repairs performed.

The project was completed on schedule in approximately 7 weeks, and the performance exceeded the design targets.