Non-OEM Pump ReBuild Shops Part IV.: Case Studies

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How competent is competent? More importantly, how much might it actually cost your operations if you were to entrust your pumps to the wrong shop?

This article is the last installment in a four-part series based on a presentation delivered at the 2007 NPRA Reliability & Maintenance Conference in Houston, TX. As in the previous installments, (which ran in the July and September 2007 and February 2008), the authors discuss how to distinguish competent pump repair operations.

In this fourth and concluding part in our series on non-OEM pump repair facilities, we discuss two actual case studies. As you read on, please recall that we coined the acronym "CPRS" to convey the term Competent Pump Repair Shop.

Repair case study #1:

Two IR Type J4x 15 lean amine pumps The first of our two case studies concerns the repair of two IR Type J4 x 15 radially split, double suction, betweenbearing pumps purchased in 1982 for lean amine service. Figs. 1 through 3 provide specifics.

The pumps were to be repaired using new 316 stainless steel casings and heads furnished to a CPRS by the refinery client. The client had bought these parts from the "current" OEM—a successor company to the initial OEM. While one pump was being repaired, the other pump remained in service, operating without a spare. However, the new casings and heads required considerable rework before they could be used. This rework included:

1. Sleeving and re-machining an oversize stuffing box bore;

2. Re-machining the two spiral wound gasket faces;

3. Weld-repairing a sand inclusion on a stuffing box face;

4. Re-facing the stuffing box faces to remove steps caused by the milling operation;

5. Re-machining two stuffing box bores that had been damaged so that the seal gland pilot would not engage.

Based on the location of the bolt holes in the bearing housing mounting flanges, neither cast casing was symmetrical with respect to its shaft centerline. The new casings and covers also were drilled for the bearing housing alignment dowel pins. The casing and head for the second pump were returned to the "current" OEM after a pinhole leak was discovered while air testing the mechanical seals. The second pump was returned to the refinery with the original carbon steel casing and head.

Repair case study #2: Worthington Type 8 UZDL21 multistage ash sluice pump

A Worthington Type 8 UZDL21 two-stage pump from a power plant was received by this CPRS in 2006. It was disassembled and inspected and the names of the personnel involved in this work were recorded in the evaluation. By recording these names, the owner-operator ("client") of the pump was able to ascertain the experience levels of staff assigned to dismantling and inspection duties. Client representatives were in attendance for the inspection and met with CPRS personnel. A comprehensive Inspection Report & Repair Proposal was generated, submitted and quickly approved by the client. Repairs were started without delay.

The purpose of an Inspection Report and Repair Proposal is to evaluate repair options and/or design changes that may (or may not) be included in the repair plan, but which the examining engineer believes could increase the run time of the pump. The following design changes were implemented in this repair sequence:

1. Wear ring configurations were changed from "saw tooth" to "smooth."

2. All wear rings were coated with 88-12 tungsten carbide, 0.020" thick.

3. Thrust bearings were changed from MRC 8317 AB to SKF 7317 BEGAM.

4. Oil flinger ring diameters were increased from 7" to 7.50".

5. An appropriately sized oil drain hole was added in the bearing housing bore at the 6 o'clock position

6. Bolt-in bearing housing oil dams were eliminated.

7. The bearing housing oil level was changed from its previous level to an appropriate new level.

Based on the condition of the ash sluice pump and conversations with the client, the abrasive action of the ash sluice mixture limited the run time of the 316 stainless steel pumps to between 12 and 18 months. Based on the client's information package, this particular pump had operated for about 18 months and was taken out of service due to a catastrophic thrust bearing failure. In turn, this failure caused rubbing of the 2nd stage impeller back shroud to the casing and the shaft-to-thrust-end cover. The direction of thrust was away from the coupling and opposite to the preferred direction for the SKF/MRC "PumPac 8317 AB" bearing. The wear ring clearances were abnormally large and there was no evidence of oil in the thrust bearing or in its housing. Other damaged components included:

1. Coupling end sleeve nut;

2. Oil slinger rings;

3. Radial bearing; and

4. Cracked first stage impeller shroud.

The most probable and primary cause of the failure was (initially) determined to have been lack of lubrication to the thrust bearings. Nevertheless, the failed radial bearing showed a slight hint that there had, in fact, been oil in its housing. The CPRS examining engineer realized that the pump could not have operated for 18 months without oil in the bearing housings. Hence, lack of oil was judged a maintenance issue that would have to be addressed at the plant.

But, the examining engineer also believed that a thrust bearing failure could have occurred—even if the housings did have the required amount of oil. He knew that Type 8317 AB bearings, as had been installed, are designed to carry axial thrust toward the coupling. Calculations were made indicating that, with normal running clearances, the direction of thrust would be toward the coupling.

From common pump experience, the CPRS reasoned that wear ring clearances become larger with time; the direction of axial thrust then reverses. Further calculations demonstrated that if the pressure on the backside of the second stage impeller is 17-psi lower than on the front shroud, the axial thrust force will be approximately 3000 lbs away from the coupling. A 17-psi reduction in pressure can occur due to excessive leakage across the balance drum wear ring to the first stage impeller suction chamber. To overcome the thrust reversal problem, two recommendations were more closely evaluated by the CPRS:

1. Reducing the wear rate of the rings and bushings; and

2. Increasing the axial thrust capacity of the pump away from the coupling.

Comments on wear rate of rings and bushings...

The informative input from a CPRS explains why a certain course of action is recommended. The report generated during this case study discussed the following items:

1. Flushing One method of reducing the wear rate of the rings and bushings would be to reduce the concentration of abrasive matter forced through the running clearances. Flushing the rings and bushings with clean water would accomplish this. (Slurry pumps used in refineries are often designed with this feature.) This option was briefly discussed but did not seem to be feasible due to the lack of high-pressure clean water at or near these ash sluice pumps. Low-pressure water is available for bearing housing cooling and mechanical seal flushing. If this option were to be pursued, it would be necessary to determine the flow rate and pressure of the water needed so that a booster pump could be selected and the economics of the system could be evaluated.

Another source of flush water that might be investigated would utilize separators to remove fly ash from a small side stream of the pump's discharge. However, the maintenance cost associated with separator wear might make this option uneconomical.

2. Ring and bushing geometry The CPRS now considered various popular wear ring configurations. The examining CPRS engineer believed that the pump was last repaired with "saw tooth" type rotating rings and smooth stationary rings and bushings. For the same running clearance and differential pressure per length of seal, this style has the lowest leakage rate. The "saw tooth" geometry disrupts the flow causing high turbulence and thus increases the friction coefficient. During the inspection process it was noted that the rotating wear rings for this pump were oriented so that the direction of flow was opposite to normal. This would most likely cause the wear rings to become less efficient— to have higher leakage rates. In an abrasive-laden ash sluice service the saw-tooth wear ring profiles, stationary casing wear rings and balance drum bushings would show wear. The disassembled pump showed this to be the case. The outside diameter of the 300-series stainless steel impeller wear rings had a hard surface coating, but the inside diameters of the 300-series stainless steel case wear rings and balance drum bushing were soft. It was assumed that the stationary wear parts supplied by the client had a hard surface coating but that it had worn away.

This is where experience helps. The CPRS had recently repaired a multistage pump in coke cutting (abrasive) service. In this instance, the stationary wear rings and bushings incorporated "saw tooth" geometry and the rotating rings were smooth. The impeller wear surfaces had been overlayed with Stellite 6 (42 RC) and the stationary wear rings and bushings were made from 440 C material (48/52 RC). When, after three months of service, the pump was shut down due to a sleeve bearing failure caused by insuffi- cient oil, it was discovered that the smooth cylindrical wear surfaces on the impeller had become grooved due to the abrasive coke fines.

Based on these two experiences (and notice how the CPRS uses what it learns), the lead examining engineer believed that the "saw tooth" grooves trap and, therefore, locally increase abrasive particles. This then causes greatly accelerated wear. The trapped abrasive particles can originate from the pumpage and from the worn hard-coated surfaces. High turbulence created by the "saw tooth" geometry also increases the wear rate.

In an effort to reduce the rate of wear on the inside surface of the case wear rings and balance drum bushing, the lead engineer recommended changing to the normal smooth ring configuration traditionally used in pumps. This would reduce the impact angle of the abrasive mixture to zero, eliminate much of the turbulence and reduce the high local concentration of abrasives caused by groove trapping. The plain cylindrical surfaces also would be simpler to coat and, accordingly, have a higher bond strength.

Table I.

Component	Diametrical Design Clearance (in.)	Diametrical "As Built" Clearance (in.)
First Stage Impeller		
Outboard Wear Ring	0.028/0.030	0.030
Inboard Wear Ring	0.028/0.030	0.030
Second Stage Impeller		
Eye Side Wear Ring	0.028/0.030	0.029
Balance Drum Wear Ring	0.024/0.026	0.025

Hard surface coating...

The center stage bushing, impeller, case and balance drum wear rings were to be coated with 0.020" of 88-12 tungsten carbide (88% tungsten carbide, 12% cobalt) using the HVOF process. Having access to a good reference library, the CPRS knew this coating (70-72 RC) had been recommended in the Proceedings of the 9th International Pump Symposium ("Evaluation of Coatings for Abrasive Service") for slurry services. Running clearances were being increased over API-610 minimum standards to compensate for the reportedly low galling resistance associated with making both the rotating and stationary wear parts from 88-12 tungsten carbide. The design and "as built" running clearances are shown in Table I.

Bearing load capacity issues: thrust direction away from coupling...

When it was received at the CPRS facility, the ash sluice pump was found to be fitted with MRC (SKF) 8317 AB PumPac thrust bearings oriented for axial thrust toward the coupling. With the cracked thrust bearing end cover and worn back shroud of the second stage impeller, it was obvious that the thrust direction had reversed.

History is of interest here. PumPac bearings were developed in the mid 1980s to overcome ball skidding in heavyduty applications where the thrust load is in one direction only . Ball skidding becomes a more significant problem as bearing size and operating speed increase. At nDm (rpm "n" times mean diameter "Dm") values below 250,000, there is little risk of ball skidding. (This ash sluice pump with an operating speed of 1785 rpm and a mean bearing diameter of (85 + 180)/2 or 132.5 mm has an nDm value of 236,513.)

Table II.

MRC 8317 PumPac Thrust Bearing		
Load	"A" (40° contact angle in one direction)	"B" (15° contact angle in opposite direction)
3000 lbs/ 167 lbs	62,400 hrs	12,600 hrs
3000 lbs/ 250 lbs	59,400 hrs	12,200 hrs
SKF 7317 BEGAM (with 40° contact angle in each direction)		
3000 lbs/ 167 lbs	72,736 hrs in either direction	
3000 lbs/ 250 lbs	69,281 hrs in either direction	

Assuming over the life of the ash sluice pump that the axial thrust load changed direction, a pair of lightly preloaded 40° angular contact bearings would represent a better bearing selection. Nonetheless, to evaluate the improvement in L10 life, calculations were performed for the existing MRC 8317 AB PumPac bearing set and for a more conventionally applied SKF 7317 BEGAM bearing set. PumPac life calculations were performed by one of the bearing manufacturer's application engineers. His calculations were based on an axial thrust load of +/- 3000 lbs and radial loads of 167 and 250 lbs. They demonstrated that the values used by the CPRS for bearing life estimates were in the right league (see Table II, as follows). Again, in essence, this situation shows what happens when a CPRS facility involves competent suppliers of pump components in cooperative analyses: The pump user benefits.

The calculations assumed clean ISO VG 32 oil operating at approximately 160 F . It appeared, with the original design, that the oil lubricating both the radial and thrust bearings was trapped in its own sump with the oil level above the center of the lowermost ball. Consequently, the oil level in the original bearing sump was being controlled by the two ½" diameter radial drain holes shown in Fig. 4. By eliminating the bolted-in dams in each bearing housing, the bearing balls would no longer be submerged in the oil and churning would be reduced. The temperature rise would be less and the bearings would run cooler.

The CPRS opted for a larger diameter flinger (7.50" instead of 7"), which now makes it possible for the outside diameter of the flinger to be submerged in 0.25" of oil. A 3/8" diameter third drain hole was added to the bearing bore at the 6 o'clock position to drain oil from the cavity between the inboard bearing covers and the bearings. The inboard covers were notched at that position to provide an unobstructed opening to the drain holes. Again, these reflect small, but important, experience-based changes with major beneficial impact on uptime of the pump .

The CPRS calculated hydraulic thrust generation (in the axial direction) based on the following assumptions:

1. The first stage double suction impeller is axially balanced.

2. The back hub of the second stage impeller is essentially the same diameter as the balance drum wear ring, but has neither bushing nor case wear rings.

3. The head vs. capacity curve for each impeller is similar to the United L-10x23 TC proposal curves.

4. The specific gravity of an ash sluice mixture is 1.0.

5. Differential pressure at zero flow is 260 psi, at BEP it is 220 psi.

6. Each stuffing box is at or near suction pressure.

7. The pressure distribution on each impeller shroud is equal and has an average value of 0.75 times the pump differential head.

8. The impeller eye side ring diameter is 11.75", the shaft diameter is 3.937" and the spacer sleeve diameter is 4.937". 9. Because suction pressure acts throughout the pump, its effect does not influence the axial thrust and is taken as zero to simplify calculations.

At this point, the CPRS engineer listed his assumptions and submitted detailed calculations. At zero flow and P = 260 psi, the calculations corroborated his assumptions, yielding T = 3604 lbs toward the coupling. Similarly, at BEP and P = 220 psi, the calculations indicated thrust T = 3049 lbs toward the coupling.

While we elected to omit further details, the radial bearing loads were investigated in a similar manner. The message, once again, is that one should select a CPRS that will support its recommendations by readily providing the client with every relevant calculation or evaluation .

Furthermore, some pump issues deserve to be tackled just as one would approach a new design. The effects of operation with worn running clearances must be considered; with wide-open clearances, only the anti-friction bearings will carry the radial loads .

CPRS personnel also know that bearing life is reduced by incorrect bearing-to-shaft fits. Therefore, and in this example, relevant dimensions were recorded. Table III shows the bearing vs. shaft interference fits associated with the ash sluice pump. (Note that "T" stands for "tight.")

Considering the cracked first stage impeller shroud...

As identified in an Inspection Report and Repair Proposal issued by the CPRS, the outside diameter of the outboard shroud on the first stage double suction impeller was cracked at three locations. Each crack occurred adjacent to a discharge vane tip. These types of cracks are common, especially on impellers that have accrued relatively long run times . The CPRS knows this and will not try to sell new impellers where none are needed.

Cracking failures usually result from fatigue associated with pressure pulsations caused by the impeller vanes passing the casing volute tips. The magnitude of these pressure pulsations decreases with the clearance between the impeller and volute vane tips. The first stage impeller had a "B" gap that met the API-610 criteria—an important reassurance.

Another, less common, cause of shroud cracking is found on impellers having a natural frequency coincident with a multiple of the operating speed, most frequently the vane passing frequencies. With a 5-vane impeller and two volutes, the pressure pulsations occur at 5 and 10 times running speed of 8900 cpm (148 Hz) and 17,800 cpm (297 Hz). Thus, the natural frequencies of the damaged impeller were measured experimentally using an accelerometer and hammer and the results of this "ring test" cataloged. The lowest natural frequency above running speed was found to be about 658 Hz (22 times running speed).

No conclusive root cause for the cracked shroud was found. The impeller was weld-repaired, re-machined and rebalanced. The shrouds were not thickened as proposed in the original repair scope. The message here: CPRS facilities and their staffs apply scientific principles and analyses every step of the way. (Note: Recent publications, including several "Proceedings of the Texas A&M International Pump Users Symposium," have shown that new pumps suffering from different resonance phenomena have, on occasion, been delivered and put into service. So, in the case of this ash sluice pump, the testing done by the CPRS was justified, as was deviating from the original repair scope.)

At the conclusion of its work, the CPRS recapitulated and documented the repairs by restating problems and observations, and by again highlighting solutions that were both merely considered and actually implemented. It was a very thorough undertaking, much like this fourpart series, whose purpose, among other things, has been to alert you, the pump user, to the following very important fact:

As denoted by the acronym "CPRS," a truly Competent Pump Repair Shop, be it OEM or otherwise, will provide considerable value to you and your organization through its experience-driven and highly cooperative efforts. Choose wisely when it comes to entrusting your pumps to others.

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