

# FINDING AND ELIMINATING CHRONIC PUMP FAILURES

by

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## ABSTRACT

A Gulf Coast refinery decided to actively address rising pump maintenance costs and risk of production losses by using accumulated failure data and maintenance costs to identify bad actor pumps, then solving the problems causing repeat failures. The plant's computerized maintenance management system (CMMS) was used for the input of condition report data. Standard spreadsheet software was used to analyze data and expose the bad actors. The process of finding the root causes of the chronic problems began with classifying past failures by the agent of mechanical failure: time, temperature, force, and reactive environment. Two case histories are presented that were investigated using this process. The effect on the plant is also presented, including the reduction of bad actor failures by 67 percent.

## INTRODUCTION

Operations, maintenance, and reliability professionals have their own definitions as to what constitutes a bad actor pump. Risk to production goals, ease of maintenance, age of design, and nature of the product are factors that weigh on the three groups. In 1995, a Gulf Coast refinery brought all the concerned parties together and decided to achieve a 36 month mean time between failure (MTBF) on process pumps. When a plant bases its reliability strategy on MTBF, the underlying assumption is that every pump failure is of equal importance, so the most important pumps are the ones that have the most failures.

MTBF calculation varies throughout the hydrocarbon processing industry, but understanding the basic rules used by the plant should provide a basis for comparison. Motiva Enterprises LLC refineries agreed to two main rules for calculating MTBF. First, every pump and driver were combined to count as a single member of the pump population. Second, a pump failure was defined as the replacement of any part on either the pump or driver. MTBF was then calculated by the following formula:

$$\text{MTBF} = (\text{36 month window}) \times (\text{total pump train population}) \div (\text{pump failures in last 36 months}) \quad (1)$$

Raising MTBF meant finding the pumps with repetitive failures. Finding those bad actor pumps meant turning computerized maintenance management system (CMMS) data into information.

## FINDING CHRONIC PUMP FAILURES

To get a pump fixed, operators in this plant fill out a work order. Work orders designate the piece of equipment to be repaired, give a statement of the problem, assign the work to the appropriate maintenance zone (i.e., machine shop for rotating equipment, area maintenance for static equipment, and E&I for electrical and instrument problems), and assign priorities. Operation's supervision approves work orders. After approval, maintenance schedules manpower for the work, and charges the cost of labor to the work order. Maintenance charges materials to the work order. Out-of-plant repairs and expediting costs are charged to the work order.

When maintenance finishes work, they close the work order by filling out a condition report. The condition report includes the date the pump was actually worked, a fixed-field component code to show the part of the pump that failed, and a text section for description of any relevant facts.

### *Using Condition Reports to Enhance Troubleshooting*

Condition reports encourage troubleshooting in two ways. First, condition reports provide a history of what has been found, and what has been changed. The text repair notes are the keys to this history. Repair notes allow maintenance to indicate either of the following: that the pump failed during a unit startup, that the seal did not appear to have anything wrong with it, that a driver was bolt-bound, or that a non-OEM part was used. Rapid access to these notes is crucial to understand what maintenance believed was going on in the past. Problems that cropped up in the past can be checked again. For example, the text of the condition report in Figure 1 would be a clue that maintenance workers need to be trained in setting the tension of mechanical seals. Notes can help identify problems that have never shown up before, and the question, "What has changed?" can be researched. The repair notes of a condition report are the history of a pump, and make a great source of information for troubleshooting a single piece of equipment.

Component codes are the keys to "big picture" insights, because they are summaries of the repair notes. In a plant with thousands of pieces of equipment, where a thousand condition reports are generated in a year, re-reading all the text notes of all the condition reports in order to classify them is cost prohibitive. In this plant's system, the maintenance supervisor classifies the repair while writing the condition report, using a fixed set of component codes. This Gulf Coast refinery's CMMS puts all information into standard database tables. A standard commercial spreadsheet can attach to the database and download any of the fields from the tables. Table 1 is the result of a query of condition reports written in the first few days of 1994. Expanding the date criteria of the query to include all of 1994 is the first step in analyzing questions like, "Are we replacing more bearings or seals?" or, "Which pumps are failing the most seals?" or even, "Which operating units fail the most bearings?" Answers to these questions allow quick focus on the systemic problems of the plant. The data used to answer these questions also allow rapid generation of bad actor lists for action. Such a bad actor list identified all the pumps in the case histories presented here.

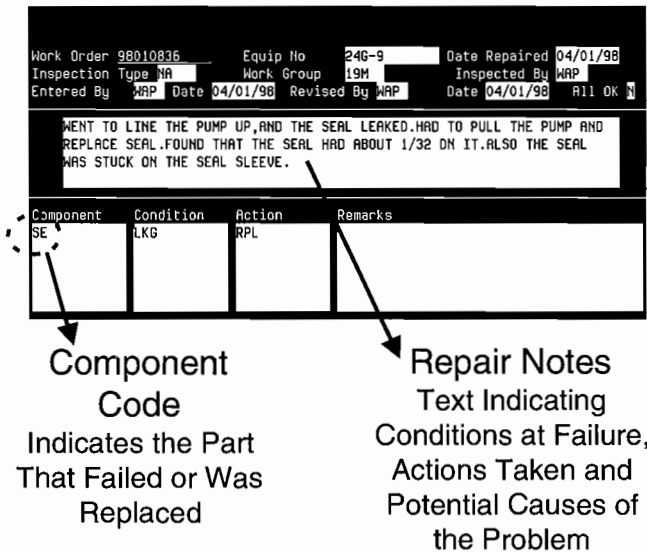


Figure 1. A Condition Report, Showing Component Codes and Repair Notes.

Table 1. An Example of Raw Data Downloaded From the CMMS.

Date	Equipment	Job Description	Unit	Work Order Number	Component Code
1/3/94	83P-136	Seal flush line plugged	VPS-2	9318070	OTH
1/3/94	70P-211	Cracked weld on east cylinder	H-OIL	9316248	SBX
1/3/94	70P-307	Increasing vibrations	H-OIL	9317573	SHT
1/4/94	14P-304S	Leaking seal	HTU-2	9400132	SE
1/4/94	17BP-107	Leaking seal	TGTU-1	9317880	SE
1/4/94	18G-36	Repair seal leak	ETU	9317863	SE
1/5/94	95P-101	Clean l/o res	HTU-3	9400262	RES
1/5/94	14P-302S	High vibrations	HTU-2	9400168	SEO
1/6/94	85P-123	Pump unusually noisy	HGU/W WTU	9400080	DIA

*Problems in the Condition Reporting System*

“Relevant facts” in a repair note description are of course subjective. Over time the repair notes tended to become more informative, as the writers began using them to jog memories or highlight problems. In the beginning of the CMMS implementation, repair notes tended to be terse, two word descriptions like “Replaced seal.” After years of usage, the writers of the condition reports included more and more of their observations on the conditions of the pump. This evolution will be apparent in the repair notes presented in the case histories of this paper. None of these problems come close to the value that repair notes provide in troubleshooting equipment.

The real obstacles to using the CMMS effectively have to do with complexity in the data. As an example, consider the naming convention of pumps in the plant. “79P-102” is a pump name, with “79” indicating the unit where the pump is located, “P” indicating the piece of equipment is a pump, and “102” is a serial number of pumps within that unit. The driver name is just the pump name with a “T” or “M” tacked on. So “79PT-102” identifies the turbine driver of pump “79P-102.” This naming convention becomes a problem when operations writes a work order against the driver, when the actual problem is a failure on the pump. CMMS links the condition report with the turbine driver, instead of the pump. Analysts then undercount the failures on a pump. Standard spreadsheet software can be programmed to turn “79PT-102” into “79P-102.” Analysts then count failures for an entire machinery train. Letting the computer do this transformation is very effective because this Gulf Coast plant generates 1000 condition reports in a year.

One of the problems of a fixed-field component code system is that there are plenty of code names from which to choose. Analysis requires reducing large amounts of data into the fewest classifications. This is the transformation of data into information referred to in Barringer and Weber (1995). In the system of this plant, there are nearly 50 different component codes. The component codes in Table 2 are the 11 different codes referring to mechanical seals. This complexity can be helpful in troubleshooting individual pumps: it is useful to know on a horizontally split between-the-bearings pump if all the seals that are failing are inboard seals, for example. In sorting data for an entire facility, complexity becomes an obstacle. Standard spreadsheet software was programmed to take component codes and put them into one of three classifications: seals, bearings, and other. The code to accomplish this task is presented in the APPENDIX. A pivot table was then used to quickly generate the data in Table 3.

Table 2. Component Codes Referring to Pump Seals.

Component Code	Component Full Name
ESL	Emissions Seal Leak
GSK	Gasket
OLS	Oil Seal
OS	Outer Seal
PKG	Packing
RES	Reservoir
SBX	Stuffing Box
SE	Seal
SEG	Gearbox Seal
SEI	Inboard Seal
SEO	Outboard Seal
SEP	Product Seal

Table 3. Percentage of Pump Failures by Failed Component.

Component	Percent of Total Pump Failures 1/1/94 Through 1/1/96
Seals	50%
Other	33%
Bearings	17%

*Analysis of Pump Reliability*

Data demonstrated that mechanical seals needed the most evaluation from the reliability group. Half the condition reports showed that the seal was the component causing the pump to be worked.

These data pointed to the need for the plant to focus attention and resources on eliminating seal failures. A corporate initiative to increase mean time between failure (MTBF) was crucial in winning the support of senior management for such a program. A reliability engineer was then assigned the 40 worst actor pumps

and took responsibility to reduce the number of failures on those pumps.

#### *Analysis of Process Service*

In addition to an analysis of component failures, a list of bad actors was generated. Pumps with the most seal failures were then classified by service, to help determine the services that were the most difficult to seal. The goal was to find services that were having the most seal failures.

Plant personnel interpreted the process service classification data of Table 4 to mean that most of the bad actor pumps were not in "marquee" or high profile services, i.e., the pumps did not immediately threaten production goals when they went down. Pumps in water service are extremely common in a refinery, and tend to be in auxiliary services that do not threaten production. Pumps in "sour" water (water and hydrogen sulfide) service are also very common, and are not directly linked to achieving process goals. Since most pumps in this plant were spared equipment, root cause analysis was not generally done on low profile pumps. Individual repairs of these low profile pumps were generally low cost, so queries looking for costly individual repairs did not identify them as bad actors. A more thorough analysis emphasized the total spent on a pump over a period of time. When this was done, the bad actors by number of failures were also the bad actors by total maintenance dollars spent.

*Table 4. Percentage of Bad Actor Seal Failures by Process Service.*

<b>Process Service</b>	<b>Percent of Bad Actor Seal Failures 1/1/94 Through 1/1/96</b>
Water & Sour Water	55%
Hydrocarbon	30%
Amines	10%
Other	5%

#### *Calculating Life Cycle Costs for the Purpose of Justifying Upgrades*

There is always a perceived conflict between improving MTBF and spending the least amount of money on a per-pump basis. Where expense budgets are tight, every reliability recommendation must be justified. Combining a financial return with maintenance's desire to eliminate pump rework and operation's desire to minimize production risk will, without fail, provide justification for upgrading pump reliability. Following up with regular cost and failure tracking of former bad actor pumps keeps the coalition of reliability, maintenance, and operations intact for future upgrade opportunities. Rising MTBF is then a manifestation of effectively spent maintenance dollars.

There are at least two techniques for justifying pump upgrades: calculating a return on investment like Hrivnak (1996) or calculating a life cycle cost like Bloch and Geitner (1995). Either method works effectively. We chose to use life cycle costs. Calculating life cycle cost drives home the point that if the root cause of failure is not eliminated, maintenance expenditures will continue as they have historically. In troubleshooting pumps, usually there are two general directions: eliminate the root cause of the failures, or continue to replace failed parts as has been done in the past. In other words, the options are to solve the problem, or do nothing. The CMMS of this plant had three years of reliable cost data, so historical costs were easily accessible. When the

troubleshooter is successful in eliminating the root cause, then the pump should run three years without failure. Because three years of failure free run time was the goal, the relevant time period in calculating the life cycle cost was three years. In other words, achieving a three year run meant the life cycle cost was cut from the expense budget.

Costs and dates of past expenditures were researched in the CMMS. After classifying costs by year, the average cost from the last three years was taken as the cost per year of the pump. The life cycle cost was calculated as the present value of the cost per year for the next three years at six percent interest rate, or life cycle cost = present value (six percent, three years, cost per year). The life cycle cost was then taken as the largest amount justified to upgrade the pump (Bloch and Geitner, 1995).

All pump costs in this paper are presented as dollars per pump horsepower (\$/hp), and are inflation corrected to constant 1982 dollars.

#### **ELIMINATING CHRONIC PUMP FAILURES**

At this point, the troubleshooter had in hand a bad actor list identifying the pumps requiring upgrades. For those bad actors, the life cycle costs had been calculated, so the maximum justified expenditure was known as well. The next steps were to go through the list of bad actors and, one by one, eliminate the root causes of chronic failures. Plant MTBF was expected to go up and maintenance expenditures were expected to go down.

The process of finding the root causes of the chronic problems began with classifying past condition reports by the agent of mechanical failure: time, temperature, force, and reactive environment. For example, condition reports referring to melted O-rings, discolored, or heat checked seal faces point to excessive temperatures as the most likely agent of failure. We classified all lubrication failures under temperature. Pusher seals with stuck rotating heads, fractured stationary faces, or severe, localized sleeve wear point to force as a failure agent. Hydraulic pump problems and poor maintenance practices usually unleash force as the agent of failure. Sticky or swollen O-rings, leaching of carbon faces, or any signs of corrosion point to a reactive environment. The agent of failure to strive for is time—the 10 years we should get out of our pump bearings. Where the text history is vague or terse, a process of elimination of the agents of failure gives the troubleshooter a good starting place checks and tests.

The checks used in this Gulf Coast refinery tended to look for problems in faulty design or maintenance deficiencies. Studying the pump while it is running allows the troubleshooter to quickly test whether or not operations is misusing the pump. Improper operation will always be one of the root causes of failure, but our experience is that most chronic failures are the result of faulty design or maintenance deficiencies. The failure cause distribution presented by Bloch (1990) singles out improper operation for only 12 percent of the total failures, leaving maintenance and engineering responsible for the remaining 88 percent of the centrifugal pump failures. Usually troubleshooting begins with an examination of a failed part. In fact, troubleshooters have a tendency to pay too much attention to a failed part. When the starting point was a bad actor list generated from condition reports, there was no guarantee the pump would be down during the time the reliability engineer would be looking for solutions. This was not really a limitation to the evaluation. The following checks were routinely done in this program:

- Review of condition reports
- Pump performance test
- Taking a temperature profile of the seal flush system
- Taking a temperature profile on the seal gland
- Taking measurements on the shaft stocked for the pump
- Verifying stocked O-rings were made from the correct material

- Auditing previous pump repair documentation (including alignment)
- Vibration analysis, including phase shifts
- Comparing the seal design with the API 682 (API, 1994) recommendation

#### Case History 1

Case History 1 demonstrates that even with “terse” historical notes, classifying previous repairs by the agent of failure is an excellent jumping off point for chronic failures. When the pumps failing the most seals were identified, a water wash circulation pump turned up with six seal failures in a two year period, for an MTBF of four months. The water wash circulation pump took the bottoms out of a settler tank and moved them into an amine header. Water made up most of the process fluid, with small amounts of amines and solids. The process temperature rarely got above 120°F. Seal flush moved through an API Plan 11 system, from the discharge of the pump to the seal gland. A pump test demonstrated the pump ran on its curve off its best efficiency point by perhaps 10 percent.

Cost data for the previous three years showed an average of 360\$/hp keeping this pump in seals. In other words, we could expect to spend \$360/hp each year for the next three years keeping this pump in seals. The present value of this series of cash flows was \$960/hp.

A study of repair notes requires the ability to interpret “terse” data. Terse data can result in nonsensical conclusions as to the causes of failure (Barringer and Weber, 1995). Using the four agents of mechanical failure (force, time, temperature, and reactive environment). In a process of elimination puts an analyst in a good position to understand and remedy chronic failures. None of the notes in Table 5 hint at the presence of high temperatures: there is no extreme wear of the faces, no melted O-rings, no discoloration. Considered with the low temperature of the process, temperature seemed unlikely as an agent of failure. None of the notes point to a reaction of O-rings to the process environment, making reactive environment an unlikely agent. A check of the O-rings assigned in stock to this pump verified the O-rings were compatible with the service. The seals lasted only four months, eliminating time as an agent. By process of elimination, force became the most likely agent of failure. Two clues in the notes point to force as the failure agent: maintenance went looking for missing antirotation pins on the 9/27 failure. This is generally evidence of the stationary face moving relative to the gland, a result of a problem in the assembly of the seal. Later, maintenance replaced a leaking seal on 8/21/95 after an 11 day run, and found no wear on the seal faces. When the rotating and stationary faces are not in contact, there will be no wear on the faces. Lack of contact can be caused by a problem in seal installation, or by a dynamic O-ring hanging up.

Alignment, seal installation, and seal design were the areas checked next as the likely sources of force-related seal failures. Maintenance alignment documentation was in order and all vibration amplitudes were low. The pump foundation had no voids. The baseplate was in good shape and met plant specifications.

In order to verify all the dimensions of the seal design, a drawing of the pump shaft was made. Shaft drawings are an indispensable pump troubleshooting tool. With a shaft drawing, the tolerances of the shaft at the bearing and at the seal sleeve can be checked. The stocked shaft bearing fits checked out, and the seal sleeve was dimensionally correct as well.

The seal design on the pump used a PTFE wedge ring as the secondary seal under the rotating seal face. This seal was considered less than satisfactory for the service, since PTFE wedge rings hang up, and will lose their elasticity over time. Addressing these problems with the seal design was believed to resolve the

Table 5. Repair Notes for the Water Wash Circulation Pump.

Date of Repair	Notes
8/17/94	Replaced seal.
9/27/94	Replaced the seal. Found that we changed this seal on 8/17/94. The seal did have some trash in it, but the seal seat has a hole for a pin. We looked at the seal drawing and it did not call for a pin.
7/19/95	Replaced the leaking seal.
8/10/95	Replaced seal on o.t. Found that the seal had some pits on the seal faces.
8/21/95	Replaced seal. Seal looked good. Looked like it did not run.
12/15/95	Seal was worked on a callout I never did see the seal that came out. The M/W said that the seal had a lot of trash on the faces.

issue in the repair notes as to why the seal would be leaking but not have any wear noted on the seal faces. A secondary goal of the MTBF program was to install cartridge seals everywhere possible. A rotating metal bellows cartridge seal similar to that of the Type A seal in Pellin and McCollough (1995), or the arrangement 1, type B seal in API 682 was purchased and installed in February 1996. This purchase used 240\$/hp of the total cost justified to upgrade the pump MTBF.

The seal removed in 2/96 did reveal a noncircular wear pattern on the seal faces. This was an indication that the troubleshooter had focused too narrowly on the part that failed, and had not found the root cause of the problem. The seal upgrade was installed anyway, but an evaluation of the factors that cause oval face wear on the seals was launched.

First on the list was excessive movement of the pump shaft. To get an idea of the amount of shaft movement, the  $L^3/D^4$  ratio was calculated for the pump (Figure 2).  $L$  in this shaft was 4.919 inches, the distance from the end of the shaft to the center of the radial bearing.  $D$ , the diameter of the shaft under the packing sleeve, was 0.875 inches. The  $L^3/D^4$  ratio was 203, indicating a very flimsy shaft. In a perfect world, the  $L^3/D^4$  ratio will be under 30. The bottom line is that  $L^3/D^4$  should be as small as possible. A flimsy shaft on a pump will cause a great deal (more than 0.002 inch) of shaft movement at the seal. If the pump is run off its best efficiency point, there will be even more movement for the seal to contend with.

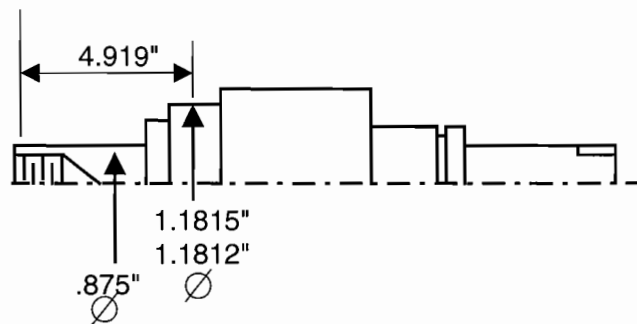


Figure 2. Diagram of the Water Wash Circulation Pump Shaft, Showing the Dimensions Used in the  $L^3/D^4$  Ratio.

Corrective action meant redesigning the shaft to get rid of the packing sleeve. Stated another way, the shaft diameter in the seal area had to be increased. The new diameter was 1.125 inches, bringing the  $L^3/D^4$  ratio down to 74, meaning the stiffness of the shaft increased by a factor of three. A new shaft was built, costing 100\$/hp. Total expenditure on the pump so far was 580\$/hp, short of the 960\$/hp justified by the life cycle cost to upgrade this pump. The redesigned shaft was installed when the upgraded seal failed after a six month run time.

The pump has now run for two years without failure. For the years 1997 and 1998 there were no maintenance dollars spent on pump repairs. The root cause of the problem on the seal was excessive movement of the shaft under the seal, a faulty design. Even with little historical information to start with, classifying the available notes by the agents of failure put the troubleshooter in the right place to solve the chronic seal problems.

#### Case History 2

In Case History 2, temperature was the agent of failure. Case History 2 also illustrates the usefulness of thorough condition reporting. A scrubber bottoms pump moved a solution of carbon fines and water into the quench water header of a gasification system. The solution of carbon and water was so abrasive that the unit was designed to have the carbon scrubber bottoms pumps on an API Plan 32 seal flush system. Plan 32 is the flush piped into the seal from an outside source. Generally Plan 32 flushes dilute the quality of the pumped product. On this unusual unit, Plan 32 dilution helped the operation. Boiler feedwater served as the flush medium. The boiler feedwater in the unit header runs at 1200 psig, and 290°F. The stuffing box pressure of the scrubber bottoms pump reached 975 psig, and the process temperature reached 450°F regularly. The pump design was single stage overhung. The seal design was a specially engineered pusher seal that was keyed to the seal sleeve because of the high pressure.

The equipment history of the pump is one of repetitive seal failures. Between 1/1/90 and 1/1/96, 30 seals failed. For brevity, only the repair notes after 1/1/94 are reproduced in Table 6. The pump was by far the worst actor in the entire plant, and perhaps in the entire company. Maintenance expenditures were frequent, but not costly on an individual basis. Average expenditure in each of the previous three years was 85\$/hp. Present value of the life cycle cost for the next three years was calculated to be 229\$/hp.

#### Analysis of the Repair Notes

Mechanical seals require three basic things in order to work properly. Seal faces must be flat, they must be pushed together, and there must be lubrication between the faces. Carbon faces phonographing, pitting at the inner diameter, and completely wearing the nose away are symptoms pointing to lack of lubrication between the seal faces. Such a lack of lubrication can be caused by excessive closing force, or by excessive temperature. Force and temperature were the agents of failure most likely to be causing the failures.

Excessive closing force squeezes the lubricant out from between the seal faces. Excessive closing force can be the result of maintenance deficiencies installing the seal. Any maintenance error that would overcompress the seal springs would cause the failures. If the keyway locking the rotating head to the seal sleeve were in the wrong place, the springs would overcompress. If the stuffing box face was a shorter distance from the end of the shaft because of prior maintenance changes, the springs would overcompress. All these potential problems were investigated. None of these errors were present.

Excessive force can be the result of faulty design. Balance ratio of the seal faces calculated at 77 percent, a reasonable amount for a gaseous service according to Robinson (1995). The pressure-velocity (P-V) of this seal was calculated to be 391,000 psi-ft/min, and the manufacturer claimed this point was on the correct side of

Table 6. Repair Notes for the Carbon Scrubber Bottoms Pump.

2/2/94	Found carbon face step worn away. Installed J/C seal. PB's cleaned seal flush cooler. Spoke with APC about seal flush rates. 14 month run. Disassembled and cleaned trip valve. Valve was sticking on startups keeping turbine from getting up to speed. This was done while pump was being worked.
3/9/94	Found carbon worn down and O-rings overheated. Had OEM in to look at seal and perform failure analysis. Installed a seal with 2-2.5 gpm seal flush rate. Seal flush temperature 225°. Original spec. sheet showed flush temperature to be 160°.
3/16/94	Found carbon pitted at I.D. and face. Seal worn and O-ring overheated. Installed seal. Seal flush temperature running at 225°. Seal manuf. to furnish failure report and make recommendation on seal.
5/15/94	Worked seal on weekend (call out). Seal faces were worn.
6/8/94	Sleeve gasket leaking, carbon face worn and scored, O-rings hard and brittle.
8/23/94	Seal faces worn and O-ring extruded/melted under carbon. This was probably caused by pump running dry.
9/9/94	Carbon face worn. Found pinhole leak in cover allowing process to get into cooling water system. Sent cover to OEM for repair.
9/12/94	Seal blown. Pump suction line and volute case plugged full of carbon. Installed seal. Operations believes carbon had settled in line.
11/21/94	Seal failed. Found carbon face worn and O-ring blown out under carbon. To analyze failed seal. Installed seal with new sleeve. This sleeve does not have keyway cut all the way through hook part of sleeve, this should eliminate problems with sleeve gasket failures.
1/3/95	Seal carbon and seat faces phonographed. Appears seal is losing flush or flush is flashing at faces allowing seal to run dry. Red to investigate (Clay Crook).
2/9/95	Pulled pump to replace seal and check throat bushing clearance per Red request. Throat bushing clearance .025, new throat bushing out of stock would have .027 clearance. Replaced impeller eye wear ring (washed out at top). Wear ring clearance .022. Installed seal. Clay Crook has seal that was removed.
3/15/95	Replaced seal. Faces looked OK, O-ring and backup ring in seal head was blown out.
5/23/95	Carbon face worn and scratched. Seal was dripping. Installed seal. Replaced wear rings. Clearance .018. Seal flush temperature approx. 240°.
7/6/95	Seal head spun on sleeve and seat was shattered. Replaced sleeve and seal.

the P-V curve. Based on these two calculations, and the dimensional check of the pump and seal drawings, excessive force was ruled out as the cause of the seal failures.

Excessive temperature can cause the liquid needed for seal lubrication to evaporate. Like any other wear parts, lack of lubrication of seal faces will cause wear and rapid failure. Repair notes mentioned melted and brittle O-rings, implying that excessive temperature was present. Field temperature measurements of the outside of the stuffing box and of the gland typically were over 300°F. Estimating that the pressure of the fluid declined linearly between the stuffing box side and the atmospheric side of the seal face (Figure 3), at least 10 percent of the seal face could be expected to have steam in between the faces. In other words, every place on the seal face below 100 psig and at

temperature of 300°F would have steam as the lubricant. In addition, as the fluid makes the phase change, the gas will expand and further reduce the contact area that is being lubricated. Since the pump had an API Plan 32 seal flush system, high temperature indicated a faulty design or the unintended operation of a seal flush system. Testing this idea required taking a temperature profile of the inlets and outlets of the seal flush cooler.

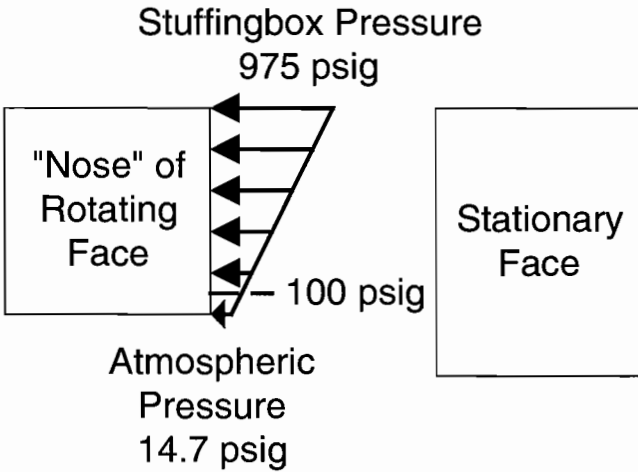


Figure 3. Sketch of the Estimated Pressure Drop Across the Seal Contact Area.

Using the standard Fourier equation for heat exchangers ( $Q = U \cdot A \cdot \text{LMTD}$ ), and the temperatures shown in Figure 4, the calculated overall heat transfer coefficient ( $U$ ) of the seal flush cooler was 99 Btu/(hr-ft<sup>2</sup>-°F). The exchanger supplier confirmed that the cooler was fouled, because  $U$  should be 250 Btu/(hr-ft<sup>2</sup>-°F) for an unplugged cooler in a water-to-water service. In perfect operation, the cooling water outlet temperature would still be well above the 140°F that the supplier recommended for the flush cooler used on the scrubber bottoms pump. Even with treated cooling water, water side temperatures above 140°F cause fouling of the coils of most off-the-shelf seal flush coolers. The higher the temperature above 140°F, the faster the fouling occurs.

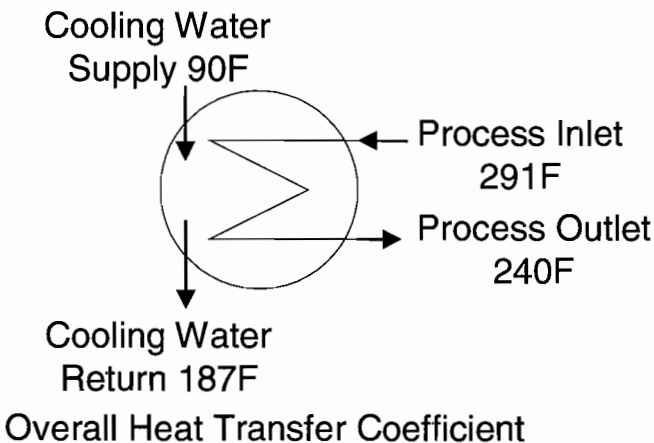


Figure 4. Process and Cooling Water Temperatures Used to Calculate the Overall Heat Transfer Coefficient.

To correct this design of the seal flush system, a finned tube exchanger with a much larger surface area for cooling the Plan 32 flush was selected and installed. Nearly the entire life cycle cost of 229\$/hp would be needed to install the larger exchanger. With the extensive failure history and life cycle costs, management approved capital money to correct the root cause of the failures.

After installation of the larger exchanger, flush temperature reaching the gland was less than 90°F. Gland temperature fell to 100°F. A lowered gland temperature allowed for improved operator surveillance. If the flush is working, the operator can touch the seal gland without gloves and not fear a burn. In the three years since the larger cooler was implemented, the yearly cost per horsepower has fallen to zero in 1998. The same pump that failed nine seals in 1994 has failed only once since 6/96.

The root cause of the pump failures was faulty design. In this case, faulty design of the seal flush system. Extensive condition reporting made it possible to quickly converge on the root cause of the chronic failures.

RESULTS

The effort to raise the plant's MTBF succeeded. All the pump upgrades executed in 1996 resulted in reduced pump failures in 1996, and in reduced failures and reduced maintenance expenditures in 1997 (Table 7). The goal was to reach 36 months MTBF by 1998; the goal was achieved in the first quarter of 1997, way ahead of schedule. Total plant failures fell by 124 between 1995 and 1997. Of that reduction, 80 fewer failures happened on the bad actors. The reduction of bad actor seal failures was 66 percent. As expected, the rest of the pump population continued to fail seals at the same rate.

Table 7. Results of MTBF Program by Year.

	1994	1995	1996	1997
Bad Actor Seal Failures	119	120	52	41
Other Pump Seal Failures	294	322	281	300
Total Plant Pump Failures	622	648	530	524
Plant MTBF on 12/31 (Months)	21.7	25	33	37

Reducing Maintenance Expenditures

Maintenance expenditures on the bad actor pumps fell as a result of the upgrade program. Maintenance expenditures on pumps in general, and bad actor seals in particular, had grown steadily up to 1994, but made a step change up in 1995. This step change was due to the large number of failures on some of the bad actor pumps. After reaching a high of 65\$/hp in 1995, the cost of bad actor pumps began falling in 1996 as upgrades eliminated failures. The cost of bad actor pumps in 1996 was 54\$/hp, including the costs of upgrading flush piping and seal designs. Expenditures on the original list of bad actor pumps have fallen to 21\$/hp in 1998. These original bad actor pumps are the only group of pumps in the plant that have seen the cost of repair go down in the last three years.

CONCLUSION

A computerized maintenance management system was used to record text repair notes, and fixed field component codes identified parts that had failed. Macros written for standard spreadsheet software took the raw component code data and turned it into information for identifying bad actor pumps. The text repair notes, evaluated with the four agents of failures, provided the basis for troubleshooting individual members of the bad actor list. Life cycle costs for the bad actor pumps were calculated, and used as the

maximum dollar amount justifiable for pump upgrading. After a series of checks and studies looking for design and maintenance flaws, root causes of the chronic pump failures were addressed. Two case histories demonstrating this process were presented. Results included plant MTBF rising by a full year, and maintenance expenditures on bad actor pumps falling to historical lows.

## APPENDIX

The following is the program for reducing the multiple component codes used in the CMMS down to three categories: seals, bearings, and other. The program was written in Microsoft Visual Basic for Applications, the macro language for Microsoft Excel.

### Option Explicit

'what this macro does is to take a list of components and convert them  
'into the classifications we track. That is, SE/OS/PKG/SBX/SEI/SEO  
'become seal failures. The same with bearings. Everything else is other.  
'This macro will add a column and put the converted data there.

### Sub Seals\_brgs\_other()

Dim StartRow As Integer, CompCol As Integer, CompCol2 As Integer  
Dim Counter As Integer

Dim Row\_End As Integer, Pot\_Belly As Integer

Dim CompName1 As String, SourceRng As String

Dim Mycell As Object, Mycell2 As Object

Dim CR As String \* 1

Const Stitle = "Feed Me!"

Const Etitle = "You're welcome."

Const Btitle = "You? You did that?"

'Preliminary stuff

CR = Chr(13) 'a carriage return

Counter = 1

On Error GoTo screwup

'First, let us get the component row

Set Mycell = Application.InputBox(prompt:= \_

"Select the cell containing the first component type. ", \_

Title:=Stitle, Type:=8)

SourceRng = Mycell.Address(ReferenceStyle:=xlA1)

'let's turn the screen off

Application.ScreenUpdating = False

'get the row and column numbers

Mycell.Select 'Select the cell they want

StartRow = Selection.Row 'Get the row of their cell

CompCol = Selection.Column 'Get the column of their cell

'get the last cell's row number

Selection.End(xlDown).Select 'Go to the end

Row\_End = Selection.Row 'Get the row

Mycell.Select 'back to the beginning

'Insert the column

ActiveSheet.Cells(StartRow, CompCol).EntireColumn.Select

Selection.EntireColumn.Insert

ActiveSheet.Cells(StartRow, CompCol).Select

CompCol = CompCol + 1

CompCol2 = CompCol - 1

ActiveSheet.Cells(StartRow - 1, CompCol2).Formula = "Component2"

'Add one to the compcol variable

'Initialize for the loop

CompName1 = Trim(ActiveSheet.Cells(StartRow, CompCol).Formula)

With ActiveSheet

\*\*\*\*\*The Loop\*\*\*\*\*

Do While CompName1 <> ""

'Update the status bar

Pot\_Belly = 100 \* ((Counter) / Row\_End)

Application.StatusBar = "Percent complete: " & Pot\_Belly & \_

"% "

Counter = Counter + 1

\*\*\*\*\*The Test\*\*\*\*\*

Select Case Trim(UCase(.Cells(StartRow, CompCol).Formula))

\*\*\*\*\*SEALS\*\*\*\*\*

Case "ESL", "OS", "PKG", "SBX", "SE", "SEI", "SEO", "SEP", "SEAL"

.Cells(StartRow, CompCol2).Formula = "Seal"

\*\*\*\*\*Bearings\*\*\*\*\*

Case "BRG", "IBB", "OBB", "TBR", "BEARING", "BSH"

'Bearings

.Cells(StartRow, CompCol2).Formula = "Bearing"

' This checks for reciprocating pump component codes.

' Case "BLR", "CON", "CSH", "DIA", "PD", "RDR", "VLV", "WR", "XHD"

' Recip pumps

' .Cells(StartRow, CompCol2).Formula = "Other-Recip"

Case Else

.Cells(StartRow, CompCol2).Formula = "Other"

End Select

'go to the next row

StartRow = StartRow + 1

'Set up the compname variable

CompName1 = Trim(ActiveSheet.Cells(StartRow, CompCol).Formula)

Loop

End With

Mycell.Select

Application.StatusBar = False

GoTo subend

'the error checking part.

'check to see if the user entered a range or hit cancel

screwup: If Len(Trim(SourceRng)) = 0 Then

MsgBox prompt:="Source range not entered, Procedure OVER!", \_

Title:=Btitle, Buttons:=vbCritical

Exit Sub

Else

MsgBox prompt:="Excuse me, I screwed up", \_

Title:=Btitle, Buttons:=vbCritical

Exit Sub

End If

subend:

End Sub

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