TROUBLESHOOTING TACTICS TO IMPROVE PUMP MEAN TIME BETWEEN REPAIRS

by

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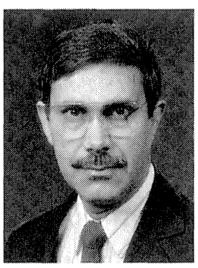

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ABSTRACT

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

Every process plant should assemble a task force to investigate pump reliability and repair quality. One of the primary goals of the task force is to establish quality criteria for process pumps that will drive mean time between repairs (MTBR) to as high a level as is practical. Seven years MTBR is a realistic target. Since time and space is not available, it is not the intent herein to compile a detail pump purchasing and repair procedure, but to outline the important subjects that should be addressed in such a document.

INTRODUCTION

The current process pump population in the Amoco Texas City Refinery is in the neighborhood of 3300 pumps according to MMS records, with an estimated 1900 or so running at any given time. For the past several years, the Texas City Refinery's Central Machine Shop and local vendor's repair facilities have had to repair between 800 and 1000 pumps each year. The annual repair bill for this group is approximately eight million dollars. The estimated mean time between failures (MTBF) worked out to a little more than two years for the average pump. The Texas City Maintenance Quality Action Team (QAT) investigated several competitor refineries and the top performers in this group are claiming MTBF of seven years.

In an environment of cost leadership, the Pump Repair Quality Task Force was formed to achieve seven years MTBF. The intent was to improve repair quality standards that should yield a nearly threefold increase in MTBF, up to the seven years target. The task force quickly recognized that much more was involved in obtaining a long, trouble free life span for any given pump than just repair quality.

DISCUSSION

The Pump Repair Quality Task Force is made up of participants from both hourly and salaried ranks. The team consists of both inside and outside maintenance folks, along with individuals from engineering, the reliability group, and operations.

The first order of business was to define the terms the task force will be using for communicating with the refinery. Again, the primary goal was to establish sets of criteria that would allow the refinery to achieve in the range of seven years MTBF. The question was asked by the group, "What is the definition of a FAILURE?" The task force decided that since cutting cost of repairs was the underlying theme, they needed to measure repairs in one form or another. After a time, they settled on defining a repair as any effort required for replacement of parts or materials. The only types of exclusion to these criteria are, say, a replacement gasket for a suction screen tee, replacement of filters in an oil system, oil changes, an occasional pressure gauge, etc. This type work will be measured as PM.

Repeated outages requiring bearings, seals, new pump components, and the like are measured as repairs, i.e., parts change define a repair event. The clock starts with each of these events. The short MTBR of these pumps provide the opportunities to look for root causes, to affect long term solutions, and to save on future maintenance costs. The repeat outages usually indicate a pumping system problem.

The analysis of pumping system problems should include an evaluation of each of the nine primary areas that can possibly contribute to symptoms. These nine areas include the following:

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

PROPER MECHANICAL AND METALLURGICAL DESIGN

The pump case is a pressure vessel and the OEMs design the case to withstand the operating temperatures, pressures, and the corrosive nature of the process. The standard carbon steel, API pump case is the basic design's starting point. By the nature of their business, the OEMs extend the basic design of a given pump case to higher pressures, temperatures, and/or more corrosive environments through metallurgical changes and, in some cases, wall thickness changes. In general, this is a safe and adequately thought-out method of providing a suitable product to the end user.

The goal of long, trouble free runs can be improved by providing the OEM accurate information up front on operating conditions facing a new installation. Stress relieving cases prior to final machining can eliminate severe case deflections in high temperature services. Adding stiffeners to internal passages can limit pump case deflections under pressure. Recognize that metallurgical choices are normally based on what are considered acceptable corrosion rates. Therefore, time in service and loss of wall thickness will reduce the ability of the pump case to withstand deflections due to pressure. In a severe situation, the case may not be able to contain the pressure. The condition of the case should periodically be inspected.

The bearing housings are preferred in cast steel to provide the ability to repair through welding. Housings should be manufactured to the type of lubrication system intended. If the user chooses oil mist for the field installation, return passages should not be cut or cast in the housings. The OEMs can, and should, test a pump intended for oil mist on the test stand with the OEMs oil mist system. During the test, the bearing housing temperatures should be monitored. The condition of the bearing housings, including aspects related to proper lubrication, should always be considered.

Selection of bearing type and size is fairly straightforward when the original purchase of a pump is made. The specs encourage 7300 series thrust bearing sets and generous sizing deep groove conrad type radial bearings. For overhung pumps with suction pressures above 200 psig, a preload can be specified into the 7300 series thrust bearing set or a specially designed directional thrust bearing can be considered. The use of 20 degree angular contact bearings on the inactive thrust bearing will help prevent failures due to skidding because of reduced bearing preload. Improper replacement bearings during an overhaul or changes in field conditions can require the reevaluation of the choice of bearings.

Proper pump shaft sizing during the original design will directly affect the ability of the pump to provide long, trouble free service. Most pumps should be operating at speeds well below their first lateral dry critical. The shaft stiffness should resist deflections in the seal areas during normal and off-spec operations. Most older pumps originally designed with thin shafts for packed service provide only limited seal run life when converted to mechanical seal service. The OEMs have provided little assistance in determining critical speeds and sensitivity to shaft deflection during these conversions. The seal manufacturer's solution is always a higher tech ($$$$$) seal.

The impeller provides the means to impart energy to the liquid. Hydraulic considerations will be discussed later. The mechanical design considerations should include the number of vanes, thrust balance provisions, corrosion and erosion resistance, ease of repair, and resistance to the mechanical damage induced by cavitation and recirculation.

Proper design of these components and potential problems associated with the pump case, bearing housings, bearings, pump shaft, and impeller should all be included in a thorough root cause analysis of pumping system problems.

PROPER DESIGN AND INSTALLATION OF THE PUMP BASEPLATE

Current specs adequately define the expected configuration for API pump baseplates including all provisions necessary to properly install the baseplate with epoxy/aggregate grout. The baseplate and epoxy on a concrete foundation are intended to provide support and stiffness for the pump. The epoxy is also very good at absorbing vibration energy. Pump problem areas associated with the baseplate and foundation are usually found in lack of support, lack of stiffness, and/or vibration associated with lack of bond or voids in the epoxy grout. Poor initial design or poor installation result in these problem areas.

Evaluation of the baseplate and its installation are key to solving some pump problems. Investigate the baseplate as a possible contributor to pumping system problems when conducting a root cause analysis.

PROPER HYDRAULIC FIT

Proper hydraulic fit means many things to many people. A significant amount of benefit can be gained throughout the company by educating operating, maintenance, field engineering, and project engineering personnel to understand the hydraulic limitations of the centrifugal pump. Experts at the highest levels in both the OEMs organizations and the user's organizations are pointing out that safe, sustained, trouble free operation of a centrifugal pumping system is very dependent on operations within a narrow set of hydraulic performance parameters. Excursions outside these parameters very quickly reduce the mean time between repairs.

The first operating parameter to limit is that of the flowrate, which is held very close to the best efficiency point (BEP) for the family of curves for the pump being evaluated. Note, this is not the highest efficiency for the trimmed impeller in use. How far away from the BEP flowrate a pump can safely be operated without incurring maintenance problems is dependent on the second and third hydraulic parameters. Curves are available to help calculate the "minimum flow for hydraulic stability" in any process pump.

The second parameter critical to a low maintenance pumping system is a significant margin between net positive suction head available ($NPSH_A$) and net positive suction head required ($NPSH_R$). The determination of $NPSH_R$ is performed during the performance test for the pump and is plotted as points where three percent head loss occurs for each of several flowrates as $NPSH_A$ (margin between suction pressure and vapor pressure for the liquid) is reduced. This is the Hydraulic Institute's standard for centrifugal pumps. What most people do not realize is these points are an indication of a low suction pressure condition that cause a three percent deterioration in the pump's ability to develop pressure. This loss in the pump's ability to develop pressure is caused by cavitation occurring in the eye of the impeller. The cavitation started at a much higher suction pressure condition ($NPSH_A$). The significant margin between $NPSH_R$ and $NPSH_A$ mentioned above can be 20, 30, or more feet above the $NPSH_R$ curve supplied by the OEMs. This can be seen by plotting the $NPSH_R$ test data for two percent head loss, one percent head loss, and zero percent head loss. Not only do the new curves rise significantly above the three percent head loss curve, the shape of the curves become more and more conservative.

The third parameter is called suction specific speed (N_{ss}). Suction specific speed is a dimensionless parameter that relates flowrate, rpm, and $NPSH_R$ for a given impeller design. Modifying an impeller's eye to reduce $NPSH_R$ can be achieved by increasing the diameter of the eye. This option was, and still is, very popular to project engineering teams in that it gives the impression that suction vessel elevations can be lowered to take advantage of the perceived benefit of a lower $NPSH_R$.

As $NPSH_R$ (three percent) is reduced for a given flowrate at a given speed, the N_{ss} rises. Users have determined that higher N_{ss} have a direct correlation to higher maintenance costs. Recognize that N_{ss} is an indication of a given pump's ability to tolerate flowrates less than BEP. The following flow limits should be placed on all high energy pumps. High energy can be defined as above 100 hp consumed by the suction impeller (Table 1).

Table 1. Minimum Flow Operations for Pumps.

Index (N_{ss})	Do Not Operate Less Than Percent BEP
<9,000	45%
> 9,000 - <10,000	50%
>10,000 - <11,000	60%
>11,000 - <12,000	70%
>12,000 - <13,000	75%
>13,000 - <14,000	80%
>14,000 - <16,000	85%
>16,000	90%

These limits are not absolute but the mean time between repairs is significantly better if these flows are considered strict limits.

So what is happening here? N_{ss} is a direct indication of the suction impellers tendency to exhibit flow recirculation in each of the impeller passages. The static pressure at the center of each recirculation cyclone drops below the vapor pressure of the liquid and a vapor pocket forms. The vapor pocket collapses at the wall of each vane, causing vibration and mechanical damage.

Many hydraulic conditions outside the narrow, satisfactory operating parameters can set the stage for mechanical damage like seal failures, bearing failures, and high vibration. How a pump fits the system requirements should always be considered in a root cause analysis.

It is appropriate at this point to mention points of interest that share in the two categories of mechanical design and hydraulic fit. First, a proper "Gap B" should be established (Figure 1). This is the gap between the outermost diameter of the impeller vanes and the innermost diameter of the volute lip or diffuser tip. The Gap B should be six percent of the impeller vane radius for diffuser pumps and 10 percent of the impeller vane radius for volute pumps. The benefit of a proper Gap B dimension is a significant reduction in vane passing frequency vibration, especially at off BEP flowrates. This reduction in vane passing frequency vibration can increase seal and radial bearing life. Another related characteristic that can help if a pump must be run at less than BEP flowrates is to overfile the vane tips to reduce the dead spot in the flow field exiting the impeller OD (Figure 2).

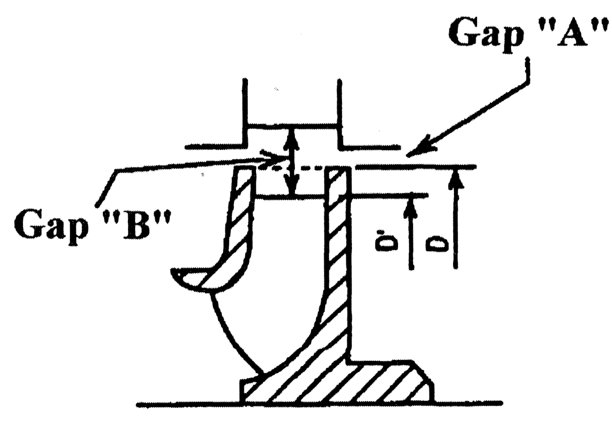
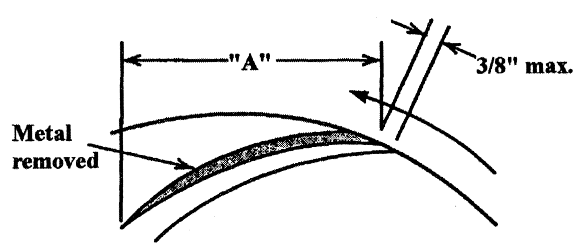


Figure 1. Impeller to Casing Gap "A," Gap "B" Relationship.



Length of blend for over filing	
Impeller diameter, in	"A" distance of blend, in
10 & below	1½
10 1/16 through 15	2½
15 1/16 through 20	3½
20 1/16 through 30	5
30 & larger	6

Figure 2. Impeller Vane Overfiling.

Pump designers are concerned for the loss of pump efficiency due to the increase in Gap B. Field checks have not shown a measurable loss of efficiency when this practice is used to reduce vane passing frequency vibration.

The second point that shares both the mechanical and the hydraulic categories is a proper "Gap A" dimension. Gap A is the radial distance from the outermost diameter of the shroud of the impeller to the corresponding circumference on the case (Figure 1). This gap should be established at 0.050 in for all centrifugal pump types that have one or two shrouds. The benefit provided by a proper Gap A is a significant reduction in the eddy currents associated with discharge recirculation at less than BEP flowrates. These eddy currents shuttle forces axially across the shrouds of the impeller and can cause axial vibration and seal and thrust bearing failures. Sometimes these forces are strong enough to cause fatigue failures in the shrouds between vanes of the impeller.

When the impeller must be trimmed to better fit the process requirements, trim only the vane, not the shroud. Before rushing out to check pumps for the proper Gap A, be aware that well over 90 percent of all single stage pump cases and over 60 percent of all multistage pump cases are cast without metal in the needed area to establish a Gap A.

PIPE STRAIN AND NOZZLE LOADS

This area is actually quite easy to describe and to design. First, the targets and the benefits will be discussed. Then, the means of achieving the targets will be discussed.

The target for pipe strain and nozzle loads should be as close to zero forces and moments as possible during the normal operating conditions for the process. But API allows nozzle loads much higher than zero. Yes, API does allow higher loads, but repeated field problems have convinced several of the industry's experts that a significant reduction in pump maintenance costs can be achieved by reducing nozzle forces and moments to as close to zero as possible while the pump is in operation. Excessive nozzle loads cause deflections within the pump case and severe misalignment between the pump and the driver. Both lead to higher than acceptable vibration levels and to seal and bearing failures.

When asked, pipe designers readily admit taking advantage of the published values for nozzle loads and start their pipe design from the pump. Essentially, they are using the pump as a pipe anchor. These same designers also state that if given the requirement of reducing nozzle loads to near zero, the task can be reasonably done through the use of unidirectional pipe shoes fixed to solid structures near the pump nozzles and proper sizing and placement of adjustable pipe hangers, spring-hangers, and spring cans. Expansion loops designed to move the piping away from the pump without exerting reaction forces on the pump nozzles can be achieved (Figure 3).

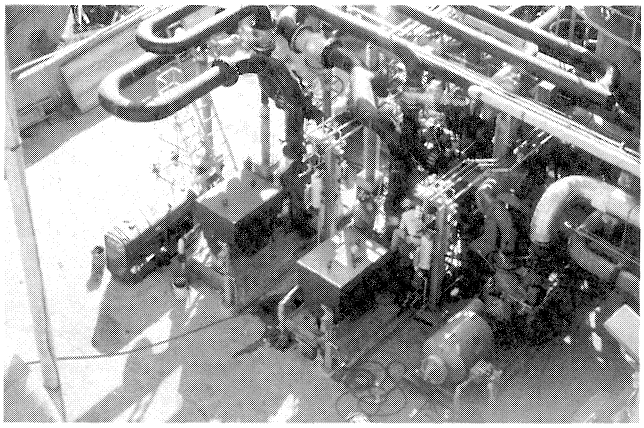


Figure 3. Expansion Loop in Suction and Discharge Piping of "Hot" Pump Similar to Steam Turbine Piping.

In the basic pumping situation of a process liquid at near ambient temperature, the pipe flange should hang above the pump nozzle with enough gap to slip in the gasket plus 1/16 in clearance. Light hand forces only should be needed to drop the studs through the flange bolt holes. The following rules-of-thumb should be adhered to when piping is attached to the pump to prevent shaft misalignment:

- Concentricity of the flanges should be such that the bolts can be inserted into the flange holes with finger pressure only (Figure 4). *No spud wrenches or "come-a-longs" are to be used to align the flange holes.*
- Parallelism of the flange gasket surfaces are to be limited to 0.002 in/in of nominal pipe size, with a maximum of 0.030 in (four inch pipe: 4×0.002 in $= 0.008$ in max). Pipe sizes less than three inches are flexible enough to allow a 0.008 in maximum out-of-parallelism on the gasket mounting surfaces. Vertical inline pump flanges less than three inches can have a maximum out-of-parallelism of 0.020 in without causing shaft alignment problems (Figure 5).
- The flange gap at the gasket surfaces is 0.063 in plus the gasket thickness (Figure 6).

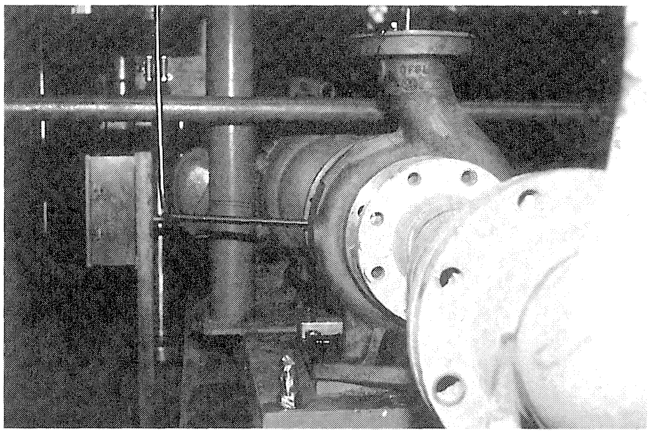


Figure 4. Concentricity of the Flanges Should Be Such That the Bolts Can Be Inserted with Finger Pressure Only.

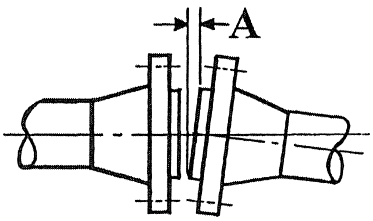


Figure 5. Rule-of-Thumb Tolerance for Parallelism of Pipe Mating Flanges.

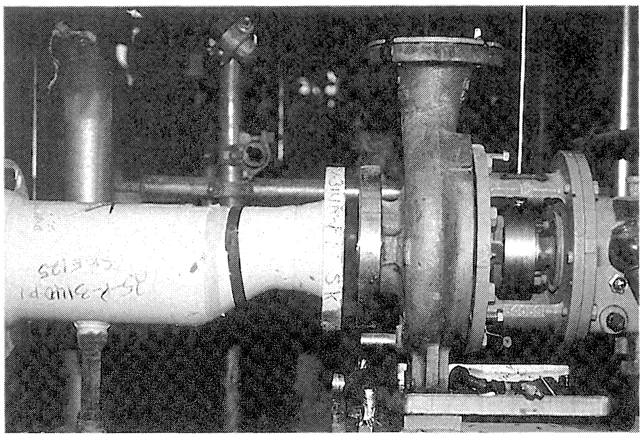


Figure 6. Maximum Flange Gap at the Gasket Surfaces Should Be 0.063 Inch Plus the Gasket Thickness.

There are steps that can be taken to minimize the affects of piping on pumps.

- The last 20 ft of piping to the pump suction and discharge flanges should be installed after the pump is grouted and aligned. Both suction and discharge piping flanges, with gaskets, should be four-bolted to the pump flanges and the piping field-fitted back to the pipe headers.
- Dial indicators should be installed from the driver to the pump to monitor movement when the piping is bolted up. The maximum acceptable movement is 0.002 in.
- Modify existing flange tightening procedures to the following. Tighten the flange bolts to one-third the design torque using a crisscross pattern. This reduces the possibility of cocking the flanges and causing shaft movement. Make a second pass on the bolts tightening them to two-thirds their design torque in a crisscross pattern. The bolts are then tightened to the designed torque level in a crisscross pattern. A final bolt torque is made in a circular pattern.

Where higher process temperatures are seen and/or the weight of the liquid becomes a factor in larger diameter piping, spring supports should be properly placed to allow the bolting to pull the flanges together. Even here, side forces should be very light. If vibration indicates misalignment or seals exhibit repeated failures after short runs, visit the pipefitters when they part flanges to blind the pump. See how far the piping springs when unbolted. Pipe strain and nozzle loads should be considered in any root cause analysis.

PROPER ALIGNMENT TARGETS AND ALIGNMENT PROCEDURES

Proper pump to driver alignment, in the running condition, is necessary to reduce vibration and to limit forces on bearings, seals, and coupling components. Setting the cold alignment targets beforehand and aligning to those targets within a tight tolerance is required to limit alignment induced forces to an acceptably low level. Most people suggest tolerances of so many mils misalignment per inch of coupling spacer length. From the charting method, one can easily determine the angular misalignment between the pump and the coupling centerlines and between the coupling and driver centerlines after allowing for thermal growth. The misalignment in the hot, running condition should not exceed 0.1 degree between the coupling and either the pump or the driver. Transients through the warm up to reach equilibrium temperature should not exceed 0.5 degrees misalignment between the coupling and either the pump or the driver (Figure 7).

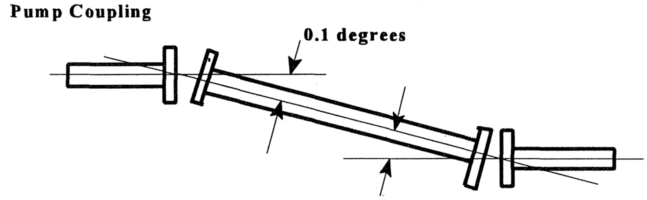


Figure 7. Alignment Tolerance Expressed in Degrees.

Remember that by the previously discussed piping criteria, pipe strain becomes a very low contributor to misaligning forces, especially at running temperatures. Therefore, most of the effort can be placed into determining appropriate cold targets to ideally achieve line-on-line alignment in the hot running condition. This must be done by surveying the pump and driver while in normal operation on a typical day to establish a temperature profile of the supports from the centerline of the shaft down to the top of the baseplate.

By knowing the materials of construction of the support structure and, thus, the linear coefficient of expansion, the cold alignment targets can be calculated. The alignment should be achieved by reverse indicator method as a minimum with laser alignment preferred if properly used. In the case of pumps operating above 700ºF and/or having hot piping movement, the pump hot alignment should be checked while running at load and temperature using optics or the Essinger Bar.

Variance from properly established alignment targets should be included in the root cause analysis.

PROPER REPAIR QUALITY

Proper repair quality criteria should be established to achieve the highest probability of a successful repair and a long run of the pump. The pump case and head(s) should be visually inspected for erosion and corrosion that can affect the integrity of the pressure vessel. Process pressures should be reviewed to determine if areas of thinning can lead to excessive deflections even if well above discard thickness. Squareness and perpendicularity should be checked to establish centerlines through the pump wear rings, seal chambers, and bearing housings.

Fits between the head(s) and the case should be checked to achieve minimal misalignment between case components. Squareness of the seal chamber gland face to the centerline will allow the seal faces to run true with minimal deflection.

The pump shaft should be checked for straightness. The impeller wear rings should be machined to provide proper clearances and to minimize indicated runout. The thrust bearings should correctly locate the impeller in the proper axial position in the case. The rolling element bearings should have the proper interference fit on the shaft and the proper clearance fit in the housings.

The rotating assembly should be dynamic balanced for all process pumps to limit vibration due to imbalance. This is a new criterion recently established to help achieve a post repair goal of less than 0.1 ips all pass vibration upon initial operation. Vibration readings in excess of this target for a recently overhauled pump can rule out imbalance and can then look for other contributors to vibration like alignment, off design operation, cavitation, recirculation, etc.

Establishment of proper overhaul and repair criteria for the pump and adherence to the criteria within acceptable tolerances should be considered in any root cause analysis.

PROPER OPERATING CRITERIA

Standard operating procedures should exist and be used for the startup, normal operation, and shutdown of each type of centrifugal pump in a process unit. These procedures should include periodic, frequent checks for proper lubrication, bearing housing temperatures, vibration levels, seal leakage, and operating flowrate relative to BEP flowrate. Excursions from acceptable flow limits should be strongly discouraged due to their effect on pump mechanical condition and MTBR.

Much can be said about this area but treatment of the pump by operators as a precise piece of machinery with limits to the operating envelope and the environment within which it exists can greatly improve run lengths. Operating criteria should be included in any root cause analysis.

PROPER DESIGN AND INSTALLATION OF AUXILIARY SYSTEMS

Auxiliary systems include such things as seal flush systems, cooling water systems, lube oil mist systems, gear boxes, drivers, suction screens, discharge check valve, suction and discharge block valves, etc. Proper repair of the pump and correct installation of the repaired pump should include checking each of the above items for cleanliness, proper arrangement, and proper operation. Nothing can mess up a good pump repair quicker than a seal flush system that is dirty, plugged, or vapor locked or a check valve that is leaking through enough to run the pump backwards as the operator pushes the start button.

A good root cause analysis will include evaluation auxiliary systems.

PROPER ARRANGEMENT OF WARM UP PIPING

Most hot pumps have a warm up line that bypasses the check valve and circulates hot process liquids from the pump discharge nozzle to the suction nozzle. In this arrangement, the hot liquids take the shortest path across the top of the pump case and do a very poor job of warming the bottom half of the case and rotor. Operations add to the problem by needing to start hot pumps very shortly after warm up has begun. Reasonable heat soak cannot be achieved with this arrangement. Even with extended warm up periods, most hot pumps will see significant temperature differentials from the bottom of the pump case to the top. Substantial clearances must be given to wear ring fits to allow a bowed rotor to run in a bowed pump case until temperatures equalize.

For all hot pumps, the warm up liquids need to be piped to the bottom of each chamber in the pump case. The hot liquids can then heat soak the case and rotor from the bottom up. More uniform warm up of the pump will occur. Proper warm up piping should be included in root cause analysis of hot pumps (Figure 8).

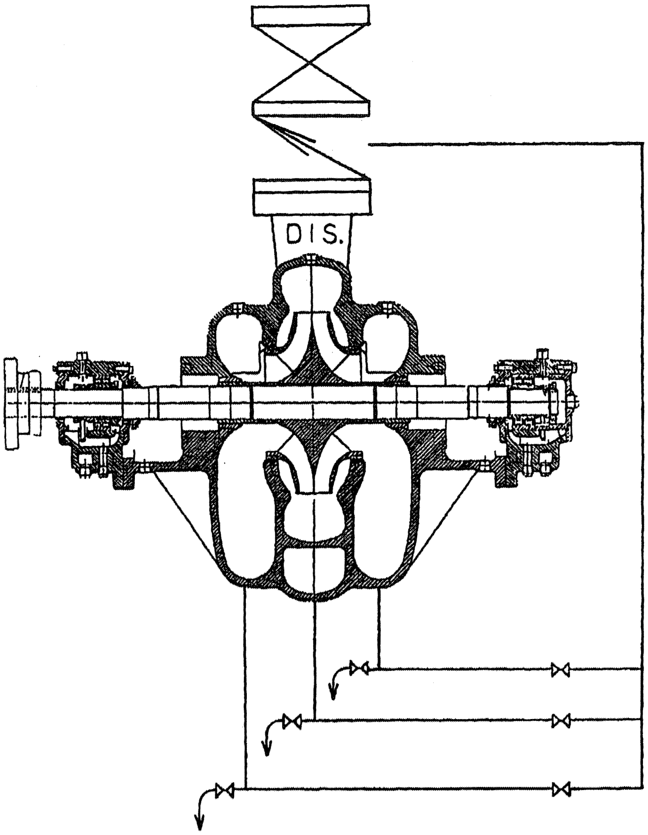


Figure 8. Warm Up Piping Arrangement for Hot Pumps.

CONCLUSIONS

Root cause failure analysis (RCFA) is the tool to use to achieve longer running pumps and to reduce maintenance costs. RCFA should be used to investigate repeat and high cost pump repairs for the foreseeable future. The basic procedure for RCFA is to close in on the real root cause of the failure. Keep asking the question "How can this happen?" until you reach the root cause of the problem that manifested itself as the symptoms you observed.

What the authors have tried to present herein is a set of topics that should be investigated for any pumping system RCFA.