TOWARD REDUCED PUMP OPERATING COSTS, PART 2—AVOIDING PREMATURE FAILURES

by

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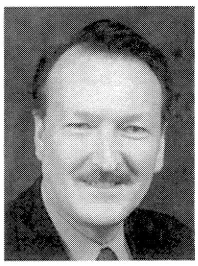

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ABSTRACT

Each year billions of dollars are spent annually on the premature or unnecessary repair and replacement of equipment. This is certainly a waste of not only our natural resources but the funds that can improve the operating profit of our companies. Improved reliability and operating efficiencies are being sought in every area. Rotating equipment and their components, seals, bearings, and couplings represent an opportunity for savings by increasing the mean time between failure.

The objective of this paper is to review equipment reliability in terms of infant mortality and to provide information on developing plans to avoid premature replacement or repair of equipment. Even though a plant may be constructed to the latest API specifications, additional plans must be implemented to ensure that the plant can be successfully started without any problems. Information is presented on a plant startup prior to API 682, and compared to the first API 682 plant startup. Understanding the effects of performance on reliability at startup are discussed as well as implementing a program to monitor progress. In addition to the startup experiences given for two different plant locations, specific

case histories are given for reducing life-cycle costs for pumps handling vaporizing liquids such as light hydrocarbon and ammonia.

The focus is on increasing mean time between failure, which will allow the plant operator to determine the actual time for equipment overhaul. Maximum mean time between repair is to be established just prior to the "wear-in phase" in the lifetime of equipment. This information can be used to determine entire plant shutdown for repairs. Cost savings for premature repairs are presented.

INTRODUCTION

This paper is directed to avoiding premature failures. Previously the subject of reliability engineering was reviewed in an earlier paper and achievements presented in increasing mean time between failure while reducing the cost of equipment ownership. Certain definitions were presented that will be reviewed and expanded.

In reliability engineering, the definition of *mechanical reliability* is the *probability* that a *component, device* or *system* will perform its prescribed *duty* without *failure* for a given *time* when *operated correctly* in a *specified environment.*

Terms that need further explanation:

Probability

In the past, the special significance of probability is that it cannot always be predicted with certainty that some event will always occur. Forecasting weather has become more accurate with satellites and advanced computer programs to model weather systems. So too in predicting when a mechanical failure will occur. Methods of collecting and analyzing test and field data are providing new insights on how and when equipment will fail. This information is a strong management tool to determine when plant turnarounds are required for equipment repair rather than waiting for equipment to fail.

Component, Device, or System

A *component* is the smallest part that would normally be replaced. Within a pump, seals, bearings, and couplings are components. A device comprises several (if not many) components. A device is a pump, compressor, turbine, gearbox, mixer, or agitator. A system comprises several (if not many) devices. For example: a process plant refinery, or nuclear electric generating station, and even an airplane is considered a system. However, also included in the system for a piece of rotating equipment such as a pump are the following items: foundation, baseplate, grout, piping, motor, and controls. Each item must carefully be considered along with equipment alignment. Ignoring any details of these items can result in short system life.

Duty

Actual service or *duty* expected of a component or device is of prime importance to its reliability specification as it describes the expected *stresses* during normal operation.

The design of a pump (device) or seal (component) for mechanical reliability, therefore, should require a detailed analysis of the likely operating stresses. The specifications must be examined for abnormal operating conditions as well. All too often, the purchaser does not know the entire range of stresses that the equipment has been designed for and the supplier does not know the range of loads for which the equipment is required.

Failure

Failure is when an item fails to perform the duty for which it is intended. A pump has failed if it can no longer move liquid safely in its intended environment. A mechanical seal has failed when it can no longer contain the liquid safely in its intended environment.

Infant Mortality Failure

Infant mortality failure is the failure of a device or component that occurs in the specified environment in less than one year. A failure that occurs every three months, four times a year, is considered an infant mortality failure. If the failure occurs at startup, it is also considered an infant mortality failure.

Premature Failure

A premature failure is any failure that occurs prior to the wear-out phase of the equipment.

Mean Time Between Failure

In the case of operating plants for pump, mean time between failure (MTBF) can be measured as:

$$MTBF = \frac{Total\,Number of\ Pumps}{Total\,Number of\ Failures} \times Review Period \tag{1}$$

Adjustments for spared pumps can be made (Wallace, et al., 2000).

Mean Time Between Failure for Seals

$$MTBF = \frac{Total\,Number of\ Seals}{Total\,Number of\ Failures} \times Review\,Period \tag{2}$$

Adjustments for between bearing pumps (two seals) can be made (Wallace, et al., 2000).

System Reliability

There are two types of systems:

- *Series systems*—Where the failure of one of the components means failure of the system as a whole
- *Parallel systems*—Systems that do not fail until all components have failed

For the purpose of this paper, the systems discussed and presented will be *series systems*. For information on *parallel systems*, the reader is directed to Wallace, et al. (2000).

Series System Example

A series system for an *ANSI* pump is illustrated in Figure 1. The total number of components is the seal, bearing, coupling, and shaft. In this case, if any component fails, then the entire pump is inoperable and must be repaired. The pump, in this case, has one effective MTBF based on the MTBFs of the individual components.

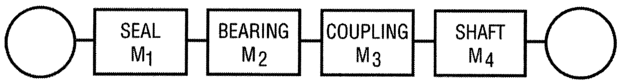

Figure 1. A Series System for an ANSI Pump.

The following equation will be used to determine the mean time between failure for the pump as a series system:

$$\frac{1}{m_s^2} = \frac{1}{m_1^2} + \frac{1}{m_2^2} + \frac{1}{m_3^2} + \frac{1}{m_4^2} \tag{3}$$

The calculated MTBF for a series system will be limited to the shortest component life. For extended MTBF, where the shortest component life is measured greater than five years, an additional 25 percent may be added to the MTBF to determine a more accurate value for estimating a plant turnaround for equipment repair or mean time between repair (MTBR). For example, if a mechanical seal fails every six months or 0.5 years, than the MTBF for the pump is 0.5 years. However, if the shortest component life is seven years, than the calculated MTBF is four years. An additional year or 25 percent of the calculated MTBF can be added to the calculated value to estimate when the equipment can be shut down for repairs. Time to preventive maintenance would be five years.

THE CONCEPT OF "BATHTUB" CURVE

A fundamental concept in reliability engineering is referred to as the hazard rate function. In simplified terms, this function takes the shape of the bathtub curve (Figure 2). Here, Phase 1 is defined as "Infant Mortality Failure," Phase 2 as "Chance Failure," and Phase 3 as "Wear Out Failure." In the operation of a plant, it is believed that it is impossible to separate each type of failure across the board. This statement is made based on the lack of knowledge on component failures. In fact, it is the job of maintenance engineers working together with the suppliers to continually monitor the equipment and identify the root cause for failure and identify corrective measures that need to be taken to increase MTBF.

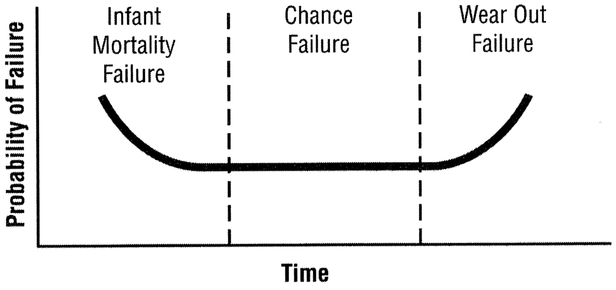

Figure 2. Bathtub Curve.

Pumps are purchased on the basis of a 20 year life. What then should the user expect in terms of maintenance? In a series system, if the seals are replaced every two years, then the MTBF for the pump is two. This means that seals would be changed out 10 times during the life of the equipment. The pump manufacturer recommends that the pump be overhauled every five years, adding to the maintenance costs.

Table 1 illustrates the cost to repair an ANSI pump between four different users. Even though the services are different, the average spend is important as well as the total for a 20 year repair cost. From the table, it can be seen that user A spends only $1600 per repair and the MTBF is only three to four months. Here components such as mechanical seals are failing. Seals are immediately replaced with no major effort to determine the cause for failure.

Table 1. Cost to Repair an ANSI Pump.

	COST OF REPAIR ($)	MTBM	AVERAGE REPAIR COST PER YEAR/ PUMP ($)	20 YEAR REPAIR COST/PUMP ($)
User A	1600	3 to 4 Mo.	4800	96,000
User B	2500	12 to 14 Mo.	2500	50,000
User C	3500	14 to 18 Mo.	3000	60,000
User D	4500	4 to 5 Yrs.	1125	22,500

This type of effort will only lead to the plant owners losing their competitiveness in the marketplace. It will become too expensive for them to manufacture their products. The maintenance cost of operating one unit for 20 years will be $96,000, plus the cost of equipment downtime. This type of failure occurring in three to four months is in the class of infant mortality.

Users B and C are doing better but there is still room for improvement. User D is doing well with components lasting a minimum of four to five years. By working together with suppliers and defining the failures, the first phase of infant mortality can be almost eliminated. Then the shape of the curve will be as shown in Figure 3. Here infant mortality failures have been eliminated. This is only possible by adopting an aggressive monitoring system and working with a supplier to correctly identify the reason for short equipment life. Also, in Figure 3, we begin to see the development of equipment life or MTBR. Repair would be scheduled just before the phase where equipment wear-out failures begin.

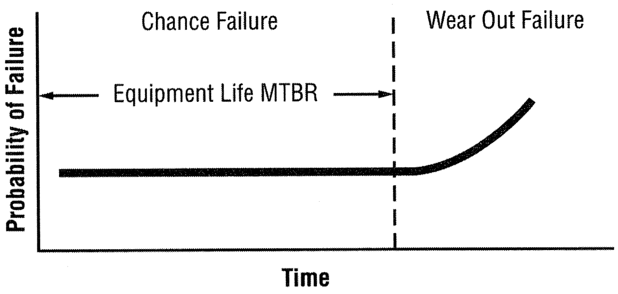

Figure 3. Modified Shape of the Bathtub Curve—Infant Mortality Phase Eliminated.

Figure 4 illustrates a modified bathtub curve with chance failures approaching zero. Here again, working with equipment suppliers in an aggressive monitoring program can have a major impact on reducing chance failures and reducing the cost of ownership of equipment. In the wear-out failure phase, improvements in design through a greater understanding of the wear mechanism for machinery are being made.

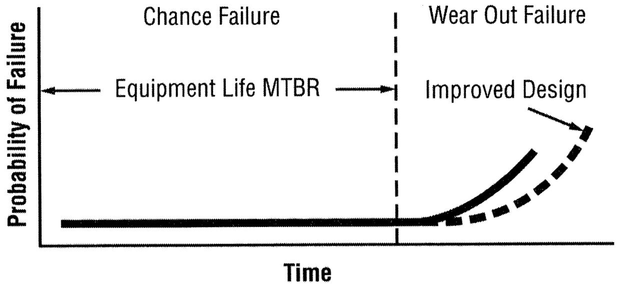

Figure 4. Modified Shape of the Bathtub Curve—Chance Failures Approach Zero.

The concept of the bathtub curve is being redefined through greater understanding of a failure and its root cause. More effort is being made to successfully identify infant mortality and premature failures. It is said that "wear-out" failures rarely apply to mechanical seals. This is due to the fact that many mechanical seals fail prematurely due to some other event happening.

ANALYSIS OF FAILURES

Operating Envelope

Each mechanical seal has an operating envelope defined by its design, materials of construction, and the fluid to be sealed. The operating envelope for a contacting seal is shown Figure 5. The upper limit is determined by pressure as well as the speed of the shaft. This limit is referred to as a pressure-velocity limit for the seal based on the materials of construction used for the seal.

Each seal must operate a given distance from the boiling point curve from the liquid being sealed or the liquid at the seal faces will flash or vaporize. If this occurs, the seal will fail in a short period of time.

Some fluids will carbonize rather than flash. In this case, the temperature must be kept below critical temperatures to prevent carbonization from occurring.

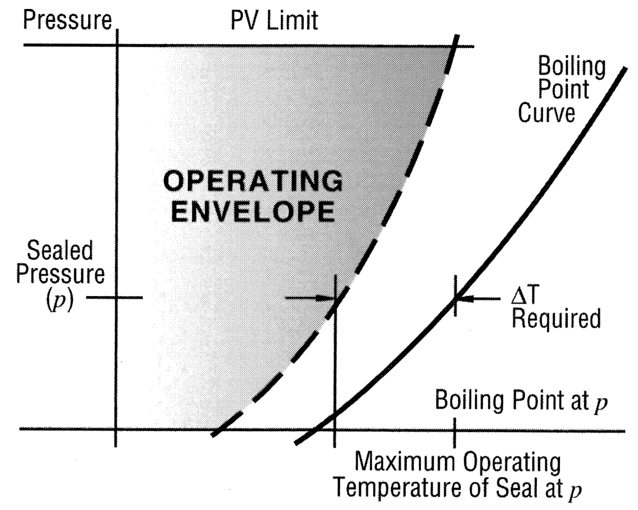

Figure 5. Operating Envelope for a Contacting Seal.

Flexibility

Another important concept in the design of a mechanical seal is flexibility. A seal within reason must be able to move to take into account equipment motions in the axial and radial direction as well as in the angular direction. When the limits of seal motion are exceeded, parts of the seal will be overstressed and fail.

The understanding of these concepts is necessary when purchasing a mechanical seal as well as maintaining the serviceability of the seal in the field.

A mechanical seal may fail for one of the following reasons:

- *Assembly*
  - Installation of the component to the device, i.e., seal to the pump
  - Installation of the device to the system, i.e., pump to the baseplate, foundation, piping, and motor
- *Operation*
  - Improper operation of equipment
  - Environment for the seal not fully defined
  - Process upsets
  - Cavitation
  - Low NPSH
  - Loss of seal flush
  - Improper venting
  - Operating too close to the vapor pressure of the liquid sealed
- *Selection*
  - Improper seal design for application
  - Improper selection of materials

Components such as bearings will have their life shortened by contamination of lube oil, high levels of vibration, process upsets, improper bearing fits, and incorrect installation. Coupling failures are normally due to misalignment problems.

When an entire system is considered, the estimated life of a component must be considered. For example, if a seal life or MTBF is always one year and the other components, bearings, coupling, and shaft are at two, five, and 10, the MTBF for the pump is still only one year. Originally, when the pump was specified and sold, all items were considered so that the pump would have an MTBF of a minimum of five years. When a component fails in less than its useful life, it is a premature failure.

Material within the component has become overstressed and failed. The component can no longer perform its function since it has operated outside its design range. Something then has been left out of the specification for the environment of the component or an error has been made in assuming what the conditions are in which the component must operate.

CATEGORIES OF PUMP FAILURE

Piping Stresses

In this example, a pump fails every three months. The user has changed seals at least three times without the help of either the pump or seal manufacturer. Each time the seal face, made of carbon, is severely chipped and shows signs of uneven wear. Face tracking on the hard mating seal surface shows light contact in one region to very heavy contact in the other. This ring is held stationary in the pump gland plate. This application involved sealing very hot water in a power plant.

To improve equipment life, a task force involving the user, pump, and seal manufacturers was formed. The team began to take measurements on the pump casing at full operating pressure and temperature. It was determined that the casing was deflecting as much as 0.016 inches. This in turn distorted the seal chamber and mating seal face. It was estimated that the angular distortion or out-of-squareness at the seal faces was greater than 0.012 inches. The shaft was turning at 1800 rpm. The seal had to flex 0.012 inches of travel 1800 times per minute.

The solution to this problem was to add an expansion joint in the piping in the suction line to the pump to eliminate the high load being transferred to the pump casing. This failure had nothing to do with the design of the components. Life has been extended to years of service. This type of misalignment would also have affected the service life of other components.

Piping Design

Short service life of a pump was being experienced. Service work was being done every three to four months. Seals were leaking badly and had to be changed.

Seals appeared to be running hot. Signs of surface distress were present. Also, there were some signs of vibration present. At certain times, the pump was noisy when in operation. It was determined that the pump was cavitating. The suction piping was changed to allow better inlet conditions to eliminate the problem. In this case, several different seals manufactured by different companies were tried without success. The problem was not in the design of the seal, but the environment in which the seal was required to operate. Until this was corrected, satisfactory life could not be achieved. Today the user is achieving 4.5 years between required maintenance.

Foundation Grouting and Piping

In this case, equipment life was increased from six months to 57 months by eliminating causes of excessive vibration that lead to short seal life. Early plant baseplate designs allowed the use of freestanding pump installations.

This cost savings feature would end up being extremely costly to the user in ongoing maintenance of equipment as well as the cost of making design changes to the existing plant structure to increase mean time between maintenance. The experiences of this user should be noted so problems in new construction can be avoided.

Operation Near Boiling Point

Operation near the boiling point of a liquid will lead to short seal life if the application is not well understood by the user and the equipment supplier. Many times when specifying a seal on certain fluids, the amount of heat developed by the seal can lead to flashing. This may be controlled by increasing the coolant or flush fluid to the seal.

However, the amount of flow still may not be enough to adequately remove the heat. If this is the case, a change in seal technology must be considered. If the heat cannot be removed, then the amount of heat generation must be substantially reduced by design. By working with the manufacturer, life can be increased from weeks to years on difficult seal pumps.

Having reviewed these items, it can be seen that performance of items not closely related to the pump must be considered. They can have a dramatic effect on performance. Knowing the influence of these items on the components of a pump can help to eliminate infant mortality and reduce premature failures.

DATA FROM A LARGE REFINERY

A continuous program of monitoring and identifying the cause for failure on mechanical seals has resulted in the following data shown in Table 2. The development of these data and the detail behind each listing must be done to establish current performance. These data will be used to set new performance goals to increase MTBF while reducing the cost of equipment ownership. The implementation of any improvement plan and the strict monitoring of progress must show continuous changes in increasing MTBF. Filtered out of the process must be those factors that influence component life such as piping stresses, foundation problems, and items that affect component life from plant to design. This should also include rotordynamic and natural frequency resonance checking of equipment, as well as continuous monitoring of alignment.

Table 2. Causes for Seal Failure in a Large Refinery.

AREA	%	SPECIAL ITEM	%
Operations	62.3	Chemical attack	5.7
		Support equipment failure	17.0
		Process failure	17.0
		Dry run	22.6
Maintenance	20.7	Mechanical damage	7.5
		Fitting error	1.9
		Bearing failure	11.3
Design	17.0	Worn out	1.9
		Hang up	1.9
		Face wear	

In Table 2, bearing failures are listed as 11.3 percent. The questions here are: did the bearings fail due to improper lubrication, or have the bearings failed due to additional stresses imposed on the system? The true cause must be addressed so that the percent of bearing failures will approach zero.

Similarly, in the case of dry run at 22.6 percent of the failures, what are the reasons for this occurrence? Is the major cause the pumping out of a tank and, when the tank is empty, is the pump allowed to run? If this is the case, then the proper controls need to be installed to prevent dry running. The only other option is to install seal technology that is capable of running when liquid is present in the pump or not.

In the cases of dry running, technology exists that can allow the seals to be run independent of the conditions in the pump.

In the case of dry running and preventing bearing failures, current performance would be improved by approximately 34 percent. The financial impact to the bottom line on operating profit would be substantial. Continuous monitoring progress will drive down the number of failures reducing the life-cycle cost of equipment ownership.

A survey of the refinery industry indicates that pump users will spend more money per repair and seal life will be far greater than other industries. Table 3 indicates the achievements of three very well run plants. User C has changed the seal design and environment for the seal.

In one area of the plant this has resulted in a total savings, including maintenance and process downtime, of just over $900,000 per year in plant operations. Continuous advances will be made in improving operations. However, there is now another measure that can be considered when making improvements in reliability. This is

Table 3. Cost to Repair an API Pump.

	COST OF REPAIR ($)	MTBM	AVERAGE REPAIR COST PER YEAR/ PUMP ($)	20 YEAR REPAIR COST/PUMP ($)
User A	5000	5 Yrs.	1000	20,000
User B	6000	7 Yrs.	857	17,400
User C	7000	6 Yrs.	1167	23,3 00

the requirement that the amount spent on the improvement have a payback in one year. When looking at the average cost to repair a pump in Table 3, the amount spent can vary from $857 to $1167. If a new improvement is determined to cost $5000, none of the refineries listed would consider the improvement. The reason being that the payback in one year could not be achieved. However, when the cost of process downtime is added to the repair cost of the pump, then the improvements can be considered.

An important observation from the refineries listed is that they have done a good job at minimizing or eliminating infant mortality failures and have done a good job reducing premature failures.

DETERMINING EQUIPMENT LIFE

Determining the life of equipment begins with determining the life of its component parts. As discussed, a series system for pumps is made up of the seals, bearings, coupling, and shaft.

The structure of the pump including the shaft normally has the longest component life. Generally 15 to 20 years is the expected life.

Couplings are the next longest life component if the equipment has been properly aligned. Soft couplings have a life of five years, while metal membrane couplings have a life of 10 or more years. Bearings for continuous operation are set at 60 months and spared operation at 120 months. Bearings need to be protected from the environment, particularly moisture, which can have a dramatic effect on life.

Mechanical seal life is not so easy to predict since this component runs directly in the product being sealed and is subject to mechanical motion from the equipment. Life can range from just weeks to over 20 years. How can one user have only marginal life while another achieves 20 years or more? The answer is in the environment in which the seal must operate. The example cited was life greater than 20 years for a finished oil products pipeline pump. This unit handled lube oil at moderate pressure and speed.

Temperatures are at ambient conditions. Much can be learned from studying seals that have been in service for extended periods of time.

In the seal industry, performance is judged on the ability of the seal to provide years of leak-free service in a given application. Mechanical seal manufacturers base the life of the seal on wear criteria that consider the pressure-velocity relationship (PV) at the faces for a category of process fluids, i.e., lubricating and nonlubricating liquids. For an actual application, its PV value would be compared to existing test data and from these data, a seal life would be estimated. PV is defined as:

$$PV = \{\Delta p(b - k) + P_{sp}\} V_m \quad (4)$$

where:

P = Pressure on the sealing surface
V, Vm = Mean velocity of the sealing face
b = Seal balance
k = Pressure gradient
Δp = Pressure differential across the seal face
P_{sp} = Pressure from spring load

The limits with respect to wear were established on a 3.625 inch diameter seal operating in tapwater at 115°F and 3600 rpm. Data were developed to define a two year life curve for seals. An application could then be determined greater than or less than two years of life. Testing of course took into account the materials of construction.

This method of establishing life has served industry well over the years and is certainly acceptable for light duty applications. However, as the focus on increasing equipment reliability and reducing life-cycle costs increases, more specific data on seal life are required.

Two important standards that have affected improvements for industry are:

- API Standard 682 (1994), Shaft Sealing Systems for Centrifugal and Rotary Pumps
- ASTM Standard F1511-94 (1994), Standard Specification for Mechanical Seals for Shipboard Pump Applications

Each standard requires the life testing of mechanical seals.

API 682

The mission of API 682 (1994) is to create a specification for seals that would have a good probability of meeting mission regulations and have a life of at least three years. To meet the requirements of API 682 (1994), testing would be done on a simulated refinery pump operation. This would include operating at continuous duty, pump shutoffs, fluid vaporization, or low flow and running the seals without a flush. Seals were expected to run meeting emission regulations and demonstrate a minimum of three years of life. The test conditions were as follows:

- Fluid sealed: Propane
- Pressure: 250 psig
- Temperature: 90°F
- Speed: 3600 rpm

Two and four inch seal sizes were tested. Test time was 100 hours. The results were excellent and have helped to define seal design, materials, and flush arrangements. In the field, seals to the API specification are exceeding the minimum life of three years.

ASTM F1511-94

ASTM F1511-94 (1994) covers the qualification requirements for mechanical seals used by commercial and the U.S. Navy for shipboard pump applications. The supplement to the standard addresses the design, materials, and performance expectations of a mechanical seal that must be in compliance in order to be on a NAVSEA contract or purchase order.

In addition to passing performance tests, seals must also have passed dynamic shock testing per MIL-S-901D. Performance testing to establish seal life was done to the following conditions:

- Test fluid: Seawater
- Pressure: 150 psig
- Temperature: 170°F
- Speed: 3600 rpm

Test time involves a total of 500 hours of dynamic testing of which 100 hours were done at offset conditions and 400 hours with the seal at normal conditions with at least 25 starts and stops. Seal designs tested were both the long and short versions of a full convolution bellows seal. Seal sizes tested were 1, 2, 3, and 4 inch seal sizes. This would allow sizes to 1 inch to be qualified, with test results for the 1 inch seal and 1.125 to 2 inch seals with results from the 2 inch test and so on. The estimated seal life is given in Table 4.

For the first time, we are beginning to see seal life established not on a maximum PV value, but essentially based on its specific operating conditions. This means that today's test information can be used to begin setting life limits on seals based on their operating environment. This test work does not include vibration from poor piping, pump installation, and other external causes. Therefore, when the MTBF for a seal or pump is substantially shorter than its estimated MTBF life, then the installation must be reviewed to completely eliminate other factors that would reduce seal life.

Table 4. Results of Performance and Wear Testing in Seawater Service.

SEAL SIZE INCHES	ASTM DESIGN	PV PSI X FPM	ACTUAL WEAR INCHES/500 HRS.	PROTECTED LIFE (YRS.)
1.00	Long	160,000	6.0×10^{-4}	9.7
2.00	Long	285,000	5.7×10^{-4}	13.7
3.00	Long	460,000	0.5×10^{-4}	15.6
4.00	Long	695,000	17.2×10^{-4}	4.5

Computer Modeling

Due to the different kinds of fluids on the vast number of operating conditions of pressure, temperature, and speed, testing at all conditions is impossible. Therefore, state-of-the-art computer tools are necessary in predicting the performance of a seal. These computer tools represent a suite of programs to analyze the performance of both contacting and noncontacting seal designs both in steady-state and transient conditions. This type of finite analysis considers all the operating conditions; fluid sealed, materials of construction, and seal geometry. The output from the program is seal distortion, temperature distribution, friction power, actual PV, leakage, percentage of face in liquid or vapor, and fluid film stability (Figure 6). These types of analysis require accurate fluid and material properties. The results from such programs will predict the success or failure of a given installation.

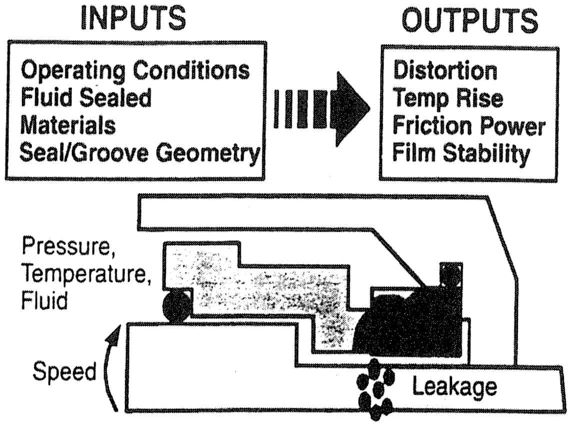

Figure 6. Fluid Film Model for a Mechanical Seal.

An example of a successful operating seal is shown in Figure 7. This seal is operating in a mixture of liquid hydrocarbon composed of ethane, propane, butane, and hexane. The seal is operating at 1300 psig, 70°F, and 3600 rpm. This study was made to determine the effect of the operating conditions on the seal prior to being installed. It has been reported that this seal has been operating successfully for more than four years. The success of this seal is due to maintaining face parallelity in service throughout its entire operating range and the ability of the installation to remove the heat that is generated at the seal faces.

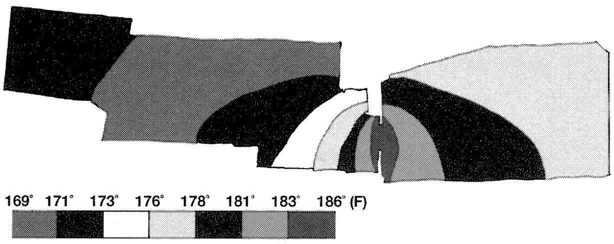

Figure 7. Computer Output of a Successfully Operating Seal.

In each case, the component of the series system for pumps should be designed for optimum life to meet the suggested targets for maximum MTBF for pumps.

PLANT STARTUP PRIOR TO API 682

Prior to API 682 (1994), plant operators were concerned with emission and pumps handling volatile organic compounds (VOCs). One user with 32 pumps wanted optimum safety and near zero emissions from all sources. This required a special program to ensure success. The program involved:

- Detailed installation requirements,
- Seal design requirements,
- Barrier system selection,
- Performance testing,
- Seal quality verification,
- Seal and pump installation,
- Field results.

Reasons for an aggressive program were:

- Increased personnel safety
  - Reduced exposure to the fluids with benzene
  - Reduced exposure to flammable liquids
- Reduced hydrocarbon emissions
- Increased equipment reliability

Data sheets for all pumps in the plant were reviewed. All pumps reviewed were identified as requiring dual mechanical seals. The criteria used for selection were:

- Pumps handling hydrocarbon fluid with a specific gravity of less than 0.8,
- Pumps handling product streams with benzene.

Pumps were built to user standards that included clearances and serviceability. Special thrust bearings were used to limit shaft vibration and displacement. This would improve the environment for the mechanical seal.

Pumps involved in the program were to API standards that existed from 1970 to 1989. This required a complete analysis of all pumps to determine if the seals could be properly fitted. The status of each unit was documented and areas identified where additional improvements could be made. The additional pump improvements made were:

- New thrust bearings to limit axial movement,
- Upgrading bolting material for low temperature services,
- Upgrading the lubrication system for pump bearings. A nitrogen purge system with an oil mist to bearings would provide for longer life, elimination of rust on the spare pump, operation with less heat, and elimination of moisture in the oil reservoir. A synthetic oil was selected for the applications. This synthetic oil is extremely clear and oil levels would be carefully observed.

Seal Design Requirements

The design requirements developed by the plant required that each seal selection be made with the highest degree of consideration to safety and equipment reliability. This included the following:

- Dual mechanical seal arrangement; tandem or double as required,
- All seals to be of cartridge design,
- Material of construction to be compatible with the process. This included low temperature requirements.
- A safety bushing required at the outboard seal on tandem arrangements,
- Nitrogen purge of all safety bushings required to:
  - Prevent icings of the outboard seal on cold pumps,
  - Isolate the outboard seal from dirt and dusty atmosphere.
- Incorporate a design feature into the seal face on some designs to minimize heat generation.
- Outboard seals fitted with nonmetallic pumping rings to provide force flow of coolant.

The three types of cartridge seals developed to meet the needs of the specifications are shown in Figures 8, 9, and 10. The most common design used is shown in Figure 8. Where possible, O-ring seals were used. All seals were fitted with low temperature materials. When aromatics were present, more corrosion resistant O-rings were used. When temperatures were extremely low, TFE wedges were used. These seals are shown in Figure 9. The surfaces under the TFE wedges were hard-coated to eliminate any wear or fretting corrosion.

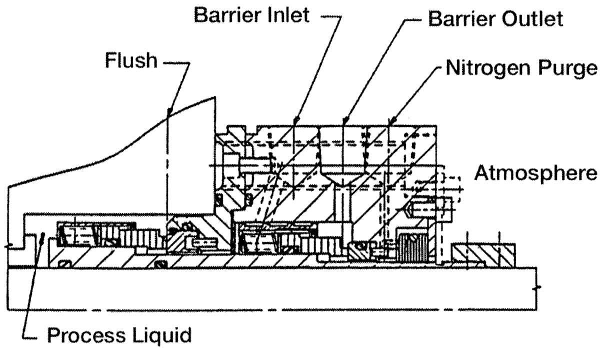

Figure 8. Cartridge Seal for Light Hydrocarbon Service.

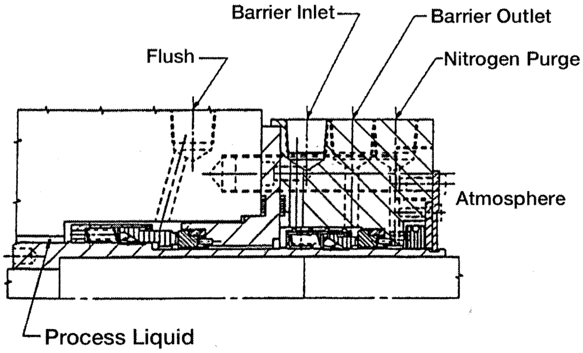

Figure 9. Low Temperature Cartridge Seal.

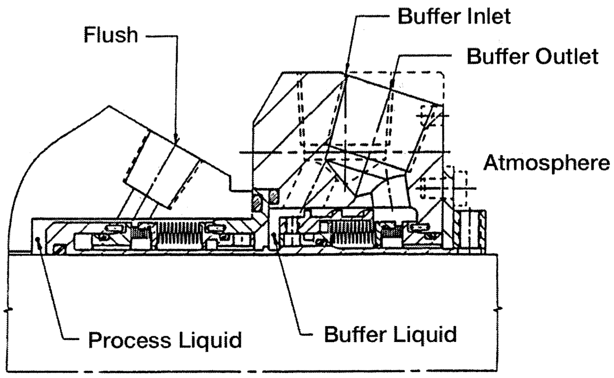

Figure 10. Cartridge Metal Bellows Seal.

A compact metal bellows with a built-in pumping ring was used on some higher temperature applications. This design is shown in Figure 10. This type of cartridge was selected to compensate for potential pressure reversal situations on these specific applications.

During the design phase of this project, it was clear that in some cases a higher strength material than carbon would be required for some faces. Due to pressure and space requirements, a high-strength silicon carbide with graphite was used. This feature allowed fitting of tandem seals on these units where axial space was limited and still be capable of handling full pressure.

Selection of Barrier System

After work was completed on each cartridge seal, attention was focused on the lubrication system, the barrier fluid to be used, and the method to fill the system. The requirement for the lubrication system included:

- Lubrication reservoir supplied with a vent system to flair,
- System fitted with trouble alarms,
  - Low level barrier switch
  - Pressure switch to warn of barrier problems
- System fitted with special rupture discs to release pressure within the reservoir under upset conditions. Any releases would not be made to atmosphere.

These features were included into the design of the lubrication system. Stainless steel was chosen as a material of construction for the reservoir and piping. Carbon steel was not considered due to the fact that if a leak were to develop, the carbon steel would become brittle and possibly fail during operation.

The use of threaded connections was eliminated in favor of welded connections. This reduced the number of potential leak points.

Reservoirs were sized based on developed heat load and volume of liquid for the seals. The design of the lubrication reservoir is shown in Figure 11.

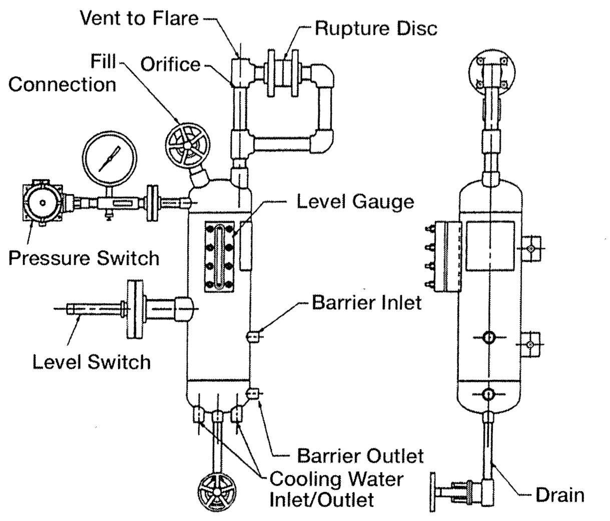

Figure 11. Lubrication Reservoir.

The selection of the barrier fluid was made on the basis that it be compatible with the process and not harmful to the environment. It was determined that n-propyl alcohol met the requirements and also had a good temperature range to handle all other plant applications. This fluid was also readily available.

To improve on the existing system, magnetic level gauges were installed.

Performance Testing

The most critical pump in the plant was identified by the user. Working closely with the supplier, the seals would be tested in the laboratory to confirm the design that was to be used. The service conditions required that the pump operate in liquid hydrocarbon at −143°F. The specific gravity was less than 0.4. Shaft speed and pressure were 3600 rpm and 480 psig, respectively.

The fluid and operating temperature could not be duplicated in the test lab. However, the pump and seals were performance-tested at room temperature for three hours with no leakage. Testing also included a hydrostatic test. No leakage was observed during this test as well. Test conditions were:

- *Hydrostatic test*—The pump filled with water was pressurized to 480 psig. Pressure was isolated so that a pressure decay could be used to indicate inboard seal leakage. In addition, the barrier fluid inlet at the bottom of the gland plate was left open so leakage could be collected and measured.
- *Dynamic test*—The schematic layout for the dynamic test is shown in Figure 12. Water was circulated through the pump and inboard seal. N-propyl alcohol was circulated in the outboard chamber by a pumping ring at a measured flowrate of 1.6 liters per minute. Pump suction pressure ranged from 150 psig to 200 psig. Discharge pressure was 300 psig. Shaft speed ranged from 2916 to 3497 rpm. The pump was run at slightly reduced speed to prevent the motor from overheating. Results were excellent with no leakage of water to the barrier fluid and no barrier fluid leaked into the water. This unit was specifically designed for low temperature.

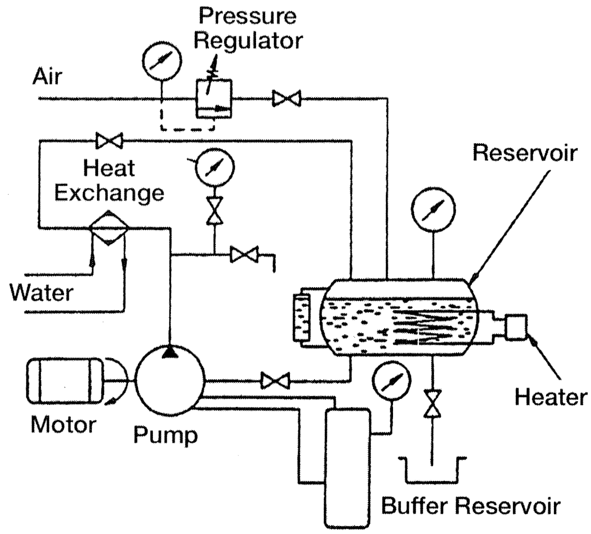

Figure 12. Schematic Layout for Dynamic Test.

Seal Quality Verification

All cartridge seals were 100 percent using a pressure decay method with a controlled volume of air at 50 psig. The acceptance criteria established specifies that 50 psig air pressure must be held for a minimum of 30 seconds with less than 2 psig loss of pressure. Results were excellent.

Seal and Pump Installation

The seals were assembled into the pumps at the plant. Each pump was then blinded and pressure-tested with n-propyl alcohol to operating pressure. The pump shaft was rotated. Visual inspection and gauges were used to verify the condition of the pump, piping, and primary seal. The outboard seal tandem seal was then pressurized. If all tests pass without leaks, then documentation is complete and the pump is ready for installation.

When the pumps were installed in the field, they were aligned using the reverse indicator method or the laser alignment method.

Field Results

Field results have been excellent. In the first 18 to 24 months, three seal cartridges have come in for repair. These were problems related to a power failure and not the seal design. It has been over 10 years since the plant was put into operation. Seventy percent of the seals are still in operation and have not been down for repair. This achievement in performance would not have been possible without a close working relationship between the user and supplier. This also points out that every step of the way attention must be paid to every detail.

First New API 682 World Class Refinery

A Thailand refinery was newly constructed, built to meet the requirements of API Standard 682 (1994). The experience from this refinery demonstrates how operator and vendor, working together, can quickly and effectively resolve problems and improve performance.

Refinery construction was completed in March 1996. Every advantage was taken to incorporate the latest technology to reduce operating costs and minimize environmental impact, where applicable mechanical seal selections were based on API 682 (1994).

During plant commissioning and early operation, mechanical seal "failures" were higher than expected and the company formed a task force to address the problem. The team included representatives from Operations, Maintenance, Integrated Machinery Inspection (IMI), and the seal vendor.

The first task was to establish a true picture of the situation, MTBF was found to be around 30 months with 15 "bad actors" identified. An initial target MTBF of five years was set with an ultimate objective of eight year's (12 month rolling sample) "pacesetter" performance. Each month the team would meet to review every seal replacement in the previous month. Operation's input at this meeting was significant as often they were able to provide details of the pump operation or product handled that influenced what changes if any were required. Only following agreement at these meetings were changes to materials, configurations, or pump, etc., carried out.

Through the team, failure modes have been identified and operator training undertaken to improve performance. Plant performance and improvements achieved have been reviewed around three measurement bases:

- Seal replacements by failure type
- MTBF/MTBR
- Repair costs

The greatest improvement has been seen in the reduction of operation's related failures. In the early days of operation, seal failure due to dry running occurred from:

- Incorrect valve operation
- Strainers being blocked by debris in the pipework
- Problems with the circulation flow in coolers (viscous plugging)

The flow to coolers was an operation problem, but also related to design. The seals were designed to run with a Plan 23 cooler. This worked well under normal operating conditions even though there is some sludge in the product, but when the pump was on standby, the product left in the cooler became highly viscous. When the pump was restarted, the pumping ring (API Plan 23) did not have sufficient head capacity to drive the viscous plug from the cooler. Consequently, cooling was minimal resulting in temperature increase in the seal chamber, vaporization at the seal faces, and seal failure. Using Plan 21 taken off first stage discharge and the same cooler, the inlet temperature was increased but is low enough for the duty. The Plan 21 has enough impetus to drive the plug from the cooler on startup. Note selection of Plan 23 was driven by API 682 (1994) without consideration of the viscosity of the product at cooling water temperatures.

In another case of operation failure related in this case to plant design, a dry running secondary seal was piped from the top of the gland plate via an orifice plate and check and isolator valves to the flare header located approximately 50 ft above the seal. Normal product leakage caused the seal to become permanently pressurized to approximately 1.3 bar (19 psi).

Under these conditions, the seal became hot and coke was formed, which led to hangup. In this case, the team removed the secondary seal and replaced it with a floating carbon bushing and steam quench piped to the drain system.

One final example of operation related failures concerns a double seal leaking barrier oil, smelling of H_2S from the outboard seal. Seal chamber pressures were found to be as designed. Even increasing barrier oil pressure did not stop the oil from being contaminated by the product. Pressure control for the system was being regulated, not by the pressure control valve, but a pressure relief valve, which meant that the barrier pressure was constantly dropping below that of the seal chamber for milliseconds before the pressure was restored. The compressor to which this seal was installed had a constant supply of seal water piped to suction that was a higher pressure than the seal chamber. By using this water, piped through the seal chamber and orifice to suction, not only was the unit made more reliable but large savings also resulted from reduced power consumption and elimination of barrier oil.

Not all problems were operational, for example, by training of the technicians the problem of silicon carbide faces being broken during fitting has been resolved and is no longer a problem.

The team also introduced some flexibility into the plant specification. By relaxing strict adherence to API 682 (1994), they were able to introduce PTFE O-rings for some applications where TFE/P copolymer or perfluoroelastomer O-rings had failed.

The original perfluoroelastomer O-rings exhibited problems of severe swelling in some seals; the replacement fluoroelastomer is performing satisfactorily. The original O-ring selection was driven by the presence of sulfur and H_2S on the data sheet diverting attention from the otherwise preferred material.

This last item is not really a seal problem but one of process design and miscommunication and perhaps more than any other indicates the benefit of bringing together expertise from operator and vendor.

The seal was a single bellows with carbon versus silicon carbide faces on API Plan 32; seal lives could be as short as five to six hours. The supply pressure of the Plan 32 was around 7 to 8 bar (102 to 117 psi) but the pump seal chamber pressure was found to be around 17 bar (250 psi); this resulted in the seal running on slurry rather than the Plan 32 clean injection.

The pump impeller was drilled with balance holes to get the seal chamber closer to the suction pressure. A floating carbon bushing was also fitted in the bottom of the seal chamber to slightly increase the pressure and reduce the usage of the Plan 32 flush. This was moderately successful, the seal now achieving lives of around four months.

It was then found that the bronze baffle sleeve under the bellows was becoming worn by radial movement of the bellows assembly, attributed to wet steam (in realty hot water); a steam trap gave little improvement due to the low usage rates. The Plan 62 was changed to a nitrogen quench, which stopped the wear on the baffle sleeve.

Wear on the carbon face was still a concern with lives still being relatively short. When the seal was inspected, traces of catalyst were still being found in the seal chamber. To make the seal more tolerant to catalyst, the carbon face was changed to a tungsten carbide, which further improved performance and increased service life to over 12 months.

It has been identified that there is a constant catalyst presence in the Plan 32 flush system and filtration is being installed to remove this. The last seals inspected had a brown glazed deposit on the tungsten face, which could be from the product or the catalyst. The option of using medium pressure steam for its cleaning and cooling properties is being considered.

Modification of the seal faces from carbon to silicon carbide is also being considered as the deposits are only found on the tungsten face.

While the first graph, Figure 13, indicates seal failures in real quantities, the curve, in Figure 14, gives a clearer insight into the drivers that are now influencing seal replacement. During the first year operational reasons accounted for over half of seal replacements, whereas this figure has reduced to around 10 percent at the last data issue.

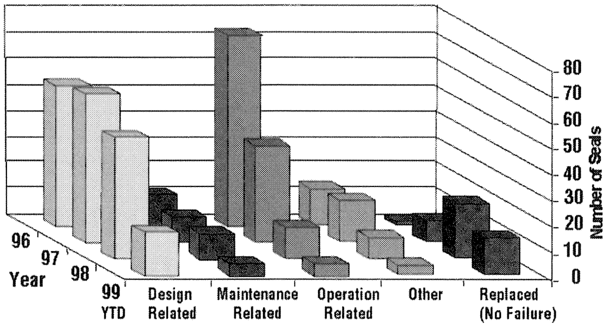

Figure 13. Seal Replacement by Failure Type.

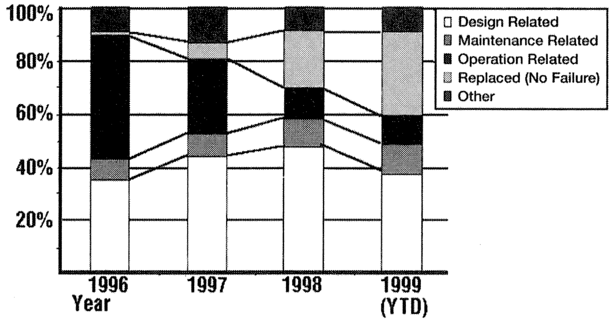

Figure 14. Seal Replacement by Failure Type as Percent of Annual Replacements.

Conversely, we now have a situation where around 30 percent of seals are replaced before their useful life is complete, which suggests that attention needs to be directed to other parts of the equipment. If we view this chart in combination with actual failure quantities, we see that operational experience and design improvements have led to a condition where operation's related failures are being substantiated by nonfailure replacements.

Perhaps the most frequently used measure for plant reliability is MTBF and this has been plotted for the life of the plant (Figure 15). The initial MTBF figure of 28 years is meaningless and is a function of the calculation method requiring time for stabilization but clearly within four months the value has settled to around three years. While this "meets API objectives," it is very low when compared with experience from modern plants. Eighteen months after startup, two developments can be seen in the graph. First, the plant MTBF starts to steadily increase, finally exceeding the target (pacesetter) value three years after startup. Second, while replacements "settle" (albeit with some natural fluctuations), failures have been separated out and appear to be settling at around 60 to 70 percent of all replacements.

This suggests that seal MTBF is no longer the overwhelming influence on pump MTBF and that other components are starting to effect overall reliability. This differential between replacements (which includes planned maintenance) and failures can be expected to increase as MTBF continues to rise.

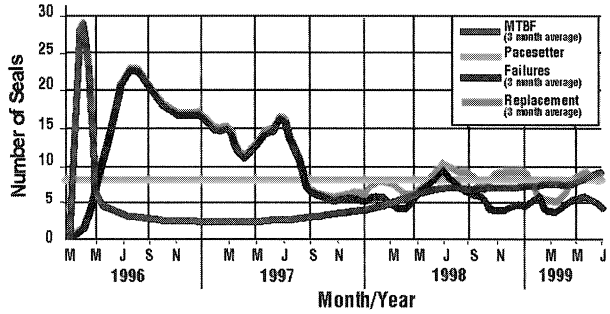

Figure 15. Mechanical Seal MTBF.

This can be seen more clearly (Figure 16) in the curves for MTBF and MTBR from April 1998. While the general form of the MTBR curve is similar to that for MTBF (naturally as it is highly influenced by it), there is an increasing divergence between the two. This status reflects that routine maintenance and failure of other components are starting to influence the curves more than actual seal failures and reflects the success of the program.

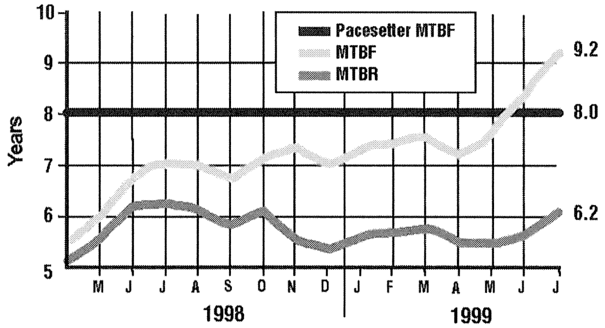

Figure 16. MTBF and MTBR Curves.

Previous papers on reliability have demonstrated the variation of MTBF seen on different units of a plant. This can also be seen in Figure 17.

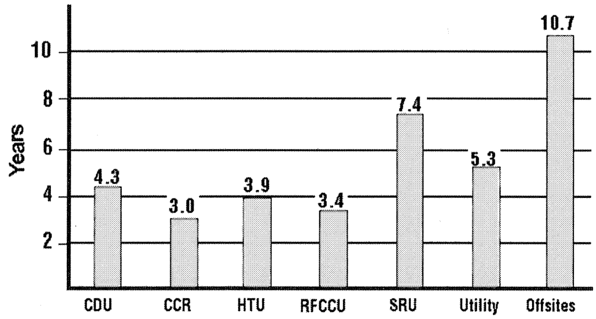

Figure 17. Mechanical Seal MTBF by Plant.

The lowest (12 month) MTBF is found in the continuous catalyst regeneration (CCR) unit, which includes some light hydrocarbon duties although, as there are a total of only 14 pumps, the results could be distorted by a relatively high number of seals that required change to O-ring materials. Three other units showed MTBFs close to or below four years. The crude distillation unit (CDU) includes high temperature applications as does the residual fluid cat cracker unit (RFCCU), the latter also including light hydrocarbon applications. By contrast, the hydrotreater unit (HTU) covers a wide range of applications and seal types.

Units that involve pumping of light hydrocarbons or high temperature fluids generally exhibit the lowest MTBF figures despite the fact that these are often given most attention at the specification stage. This gives rise to two questions.

- How bad could they be if they did not receive this attention? — A reminder to all that we should not be complacent because MTBFs are increasing.
- How good can general applications become if they are given the same level of attention?

While MTBF is a measure used extensively through the process industries, cost per seal installed (CPSI) is possibly of greater importance to the plant operator. Duty for duty, it is likely that a dual seal will give higher MTBF than a single seal (though this is not necessarily an automatic fact). Dual seals are, however, considerably more complex than a single seal and, therefore, more costly to operate/maintain/repair.

Figure 18 illustrates that seal repair costs have reduced considerably over the three years of operation, with current average monthly repair costs at less than 25 percent of the first year costs. The monthly cost, (Figure 19), reflects both seal repairs and nonfailure replacements and like the repair curve shows a reducing (but fluctuating) trend. From early 1998 these fluctuations tend to disguise the overall trend for costs so the graph also includes a polynomial that "smoothes out" the curve and indicates a steady average over a period of one year.

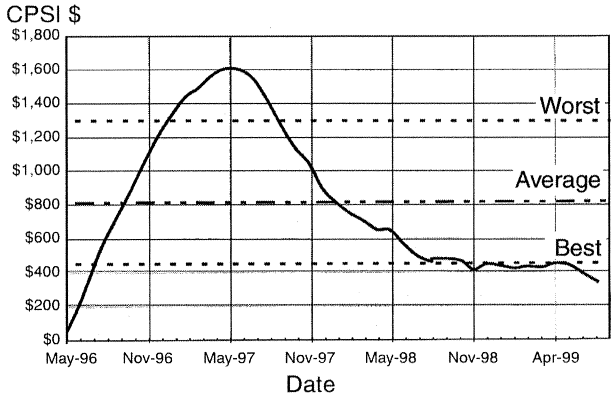

Figure 18. Cost Per Seal Installed.

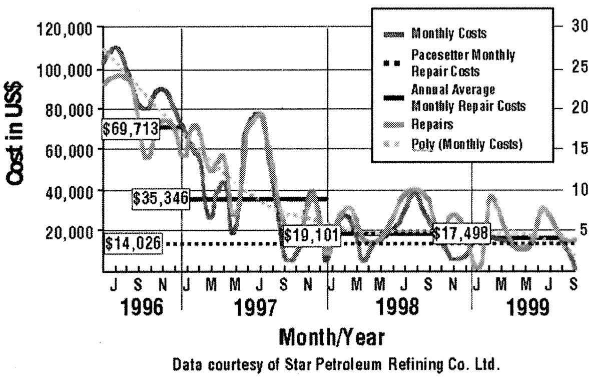

Figure 19. Monthly Mechanical Seal Repair Cost. (Data courtesy of Star Petroleum Refining Co. Ltd.)

Interestingly, while we have already seen that MTBF has exceeded plant pacesetter targets, the CPSI is still approximately 10 percent above target, which does suggest that the ongoing cost reduction indicated by the polynomial will be confirmed as more data become available.

A good measure of the success of an operator/vendor partnership comes from comparison with other plants. Wallace, et al. (1999 and 2000), reported MTBF achieved and (for 10 plants where data were available) cost per seal installed.

Comparing the current MTBF for the Thailand refinery with those plants shows that in two years it has gone from being in the lowest 15 percent to being in the top 6 percent of performers and with evidence of continuing improvements to come.

Equally significant and perhaps more important to the plant, CPSI has gone from being very poor to on a par with the best within the same period.

The benefits of operations and vendor working together are best summed up using the following bar chart (Figure 20).

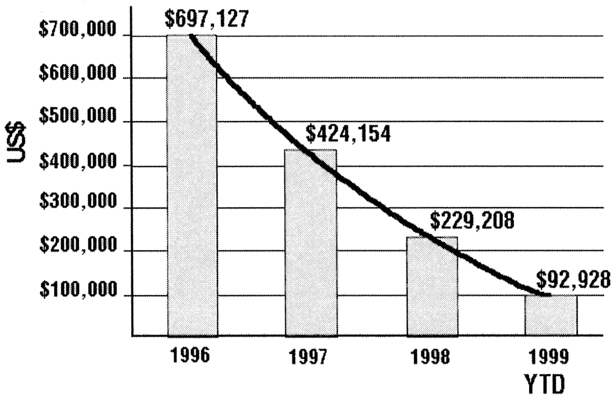

Figure 20. Annual Seal Repair Costs.

Whereas the annual spend on seal repairs was almost $700,000 during the first year of operation, this had dropped to below $230,000 within two years, an average savings for the operator of $234,000 per annum.

Summary

The Thailand plant was heralded as the first grassroots refinery built to the API 682 standard (1994). While this has undoubtedly helped the plant to quickly achieve world class levels of efficiency, it is clear that is not an automatic guarantee of success.

Commitment from both operator and vendor and a close working relationship between the two has been demonstrated to give major benefits, repaying the cost of implementing the scheme over and over again.

When an operator sees the damage done to a silicon carbide face due to running a pump with a blocked strainer, he can reduce the risk of it happening again. When there are copper particles in a seal on liquified petroleum gas (LPG), the operator can offer a possible source.

When a face is broken after two minutes, the maintenance personnel will identify if it was difficult to install or had to be done very quickly and perhaps this is how it got broken. They also start to understand operations and will tell the operators when they pass something that is not right, saving a potential failure.

The reliability people will say the seal is out for pump bearing problems, not seal failure, so the seal vendor does not spend hours looking at seal parts trying to find a cause for a nonexistent failure.

By working closely with all three, the seal vendor gets to understand more about plant operations and problems, can gather essential data there and then, and can offer more effective solutions for the future.

CASE HISTORIES— GENERAL RELIABILITY IMPROVEMENTS IN SEALS IN CRITICAL SERVICES

High Pressure Light Hydrocarbon Service

A major petroleum producer collects hydrocarbon gas from many different locations in the field. This gas is liquefied and pumped to shipping terminals located hundreds of miles away. The fluid is a mixture of ethane, propane, butane, and at times heavy ends, which included oils and tar. Existing seal installations only ran for two to three months before repair was required. The heat generated by the seals was enough to start the flashing process at the seal faces. When this occurred, it was only a matter of days before the existing seal installations would fail. To increase the reliability of the equipment, a change in seal technology would be required to prevent flashing. The seal selected and put into service is shown in Figure 21.

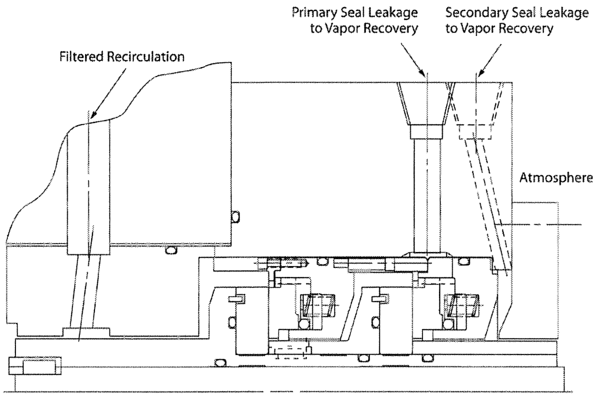

Figure 21. Noncontacting Seal for Light Hydrocarbon Seal Service.

The noncontact seal installation has been in service over 10 years. The savings has been substantial. The increase in life over the original installation is a factor 50.

Ammonia Service

Fertilizer plants have several pieces of critical equipment, one of which is an ammonia charge pump. Shaft speed is 7500 rpm and pressures to 412 psig. Temperature for the process is 50ºF.

Prior to the final selection of equipment, it was determined that the proposed liquid lubricated contacting seal would generate too much heat and the ammonia would flash to a gas. Noncontacting seals were analyzed and it was determined that they could be operated successfully. The design was successfully tested in the pump with ammonia prior to commissioning. Tandem seals were used with a dead-ended seal chamber.

CONCLUSION

Substantial progress continues to be made in reducing pump operating costs. Still much remains to be done to further reduce operating costs by continuous improvements in increasing MTBF. It is extremely important to develop a vigorous program that includes not only plant maintenance and reliability engineers but equipment and component manufacturers as well. Issues that influence equipment life must be identified and solutions that will substantially increase equipment life must be applied. When a new plant is constructed, the process must begin early in the specification and construction state to achieve the desired results. When the plant is in operation, a vigorous program to monitor performance must be in place. This can never be overstated. Focus on those areas where major savings can be achieved. The results of the plant program will be outstanding.

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ACKNOWLEDGEMENT

The authors would like to thank the management of the Star Petroleum Refining Company, Ltd., refinery in Thailand for permission to publish data from their plant on seal reliability. They would also like to thank Peter Bowden of John Crane EMA-AP for his help in developing the information on the refinery in Thailand.

PUMP RERATES – WHEN AND HOW

by

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During his career, Mr. McGuire has been involved in the application, design, manufacture, testing, and installation of single and multistage centrifugal pumps for the water, chemical, process, and utility industries. He has published several papers, articles, and two books, Pumps for Chemical Processing *and* Centrifugal Pumps *(coauthored with the late Igor Karassik). He earned his Bachelor's degree (Engineering) from the New South Wales Institute of Technology, and is a member of ASME.*

ABSTRACT

During the life of a plant, there often arises a need to change a pump's rating to match a new plant operating condition or raise the mean time between repair (MTBR) of an unreliable pump. In these cases, rerating the pump instead of buying and installing a new one, if that is feasible, offers the plant owner worthwhile capital savings. Often the capital savings are sufficient to lower the payback period and raise the return on investment (ROI) to the point where the project becomes financially viable. The tutorial first addresses the circumstances when a rerate is warranted, then how to determine the new rating, and finally, by way of three examples, the general techniques for carrying out rerates.

INTRODUCTION

Most refinery owners seek to maximize refinery throughput in order to realize a higher ROI. They accomplish this by progressive debottlenecking of the various units that make up the refinery. Debottlenecking frequently involves increasing the mass flowrate through a particular unit, raising the pressure at which it operates, or some combination of the two. This, in turn, means the pump or pumps that charge feed to the unit need to deliver a higher flow, higher head, or both. With ingenuity it is often possible to achieve the new pumping conditions by rerating the existing pumps. Doing so lowers the investment needed for the debottlenecking thereby improving its financial viability. At the same time, particularly in older units, the pump rerate can include basic machine design improvements to correct reliability problems and thereby raise MTBR. Such improvements also help raise unit ROI by lowering maintenance expenditure and raising unit availability.

In some instances, it is necessary to reduce the rate of a particular refinery unit to balance refinery operation. This means the pump or pumps charging feed to the unit have to operate at a lower flow. Depending on the degree of flow reduction and how it is achieved, operating units at lower rate can lead to unnecessarily high energy consumption or lower MTBR. When that is the case, rerating the pump so its hydraulics better match the actual operating flow is financially viable if the payback period on the investment to do it is short enough.

This tutorial provides guidance on pump rerates by first discussing the circumstances when a rerate may be appropriate. Next it deals with determining the new pump rating taking advantage of data from the operating unit. The last piece, how rerates are done, is a more complicated topic because each rerate is different in detail. Three examples, two for higher unit flow rate, the third for lower, are reviewed in broad detail to cover this aspect.

WHEN TO CONSIDER A RERATE

The following three circumstances warrant considering a rerate:

- *Increase in unit flow rate*—Pump ratings and the number of pumps installed generally have quite a degree of conservatism built into them (though this is falling with today's emphasis on project capital cost). It is therefore generally possible to achieve a worthwhile increase in unit flow rate just by using as much of that conservatism as the plant operators are comfortable with. If the existing equipment is operating at its practical limit before the desired unit flow rate is reached, rerating should be investigated to see whether that offers an economical means of achieving the desired flow rate.

- *Decrease in unit flow rate*—Occasionally it is necessary to reduce the flow rate in a particular unit to balance plant operation. In some of these instances, this can be achieved by simply throttling the pump, or if it is variable speed, lowering its speed. In other instances, the necessary reduction in flow, as a fraction of design, is so great that it risks lowering the pump's MTBR (or wasting energy if a permanent bypass is used). When that is the case, rerating the pump frequently offers the same MTBR while providing energy savings that help pay for the rerate.

- *Troublesome pump*—The root cause of short MTBR in many pumps is hydraulic in origin. Generally the pump's operating capacity is far from design and its energy level per stage is high enough for this to shorten seal, bearing, and running clearance life, and sometimes to cause premature erosion of the hydraulic parts, particularly the impeller. Plant management usually highlights such pumps as having high maintenance expense and reliability so low that it jeopardizes unit availability. Rerating is one means of restoring the pump MTBR and reliability to acceptable levels. That, in turn, lowers maintenance expense and can raise unit production.

DETERMINING THE PUMP'S NEW RATING

In general terms, determining the pump's new rating for a unit revamp requires the same basic discipline as for a new unit (McGuire, 1996), but with one important difference. The difference is that there is now a full-scale model available for examination, which if examined carefully can lead to an optimum new pump rating in terms of energy consumption and MTBR. How to carry out the careful examination is best addressed by reviewing some fundamentals of pump and system hydraulics.

Pump System Interaction

A centrifugal pump operates at the capacity given by the intersection of its head capacity curve and the system's head capacity curve (Figure 1). At this point the energy being added by the pump equals the energy required by the system. Note in Figure 1 that the energy required by the system is often increased, by throttling across a control valve, to allow variation of the pump's capacity. Note, too, that this means of flow control is feasible only with centrifugal and other kinetic pumps, those that add energy by raising the liquid's velocity.

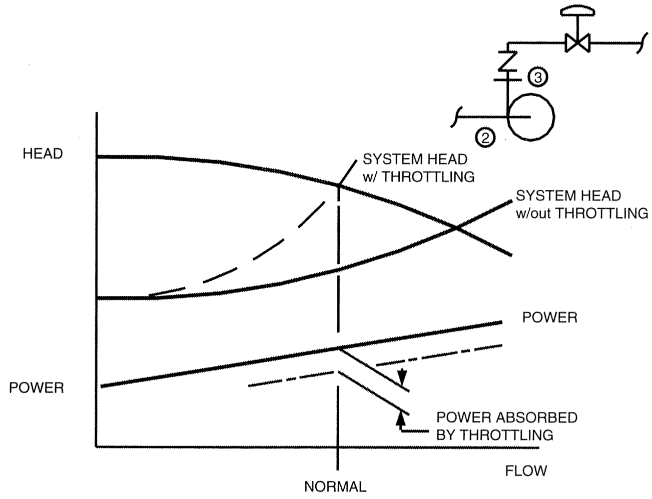

Figure 1. Centrifugal Pump Versus System with Control Valve.

Displacement pumps (Figure 2) deliver essentially a fixed capacity at a given speed, and consequently add as much energy as needed to move that capacity through the system. Care is therefore needed to ensure this energy can never be above the mechanical capability of the pump. In simple terms, this means displacement pumps must always be installed with a full capacity relief valve upstream of the first valve in the discharge system. And the relief valve must have an accumulation pressure (rise above cracking pressure to achieve full flow) that keeps the pump's discharge pressure and the corresponding power below the maximum allowable.

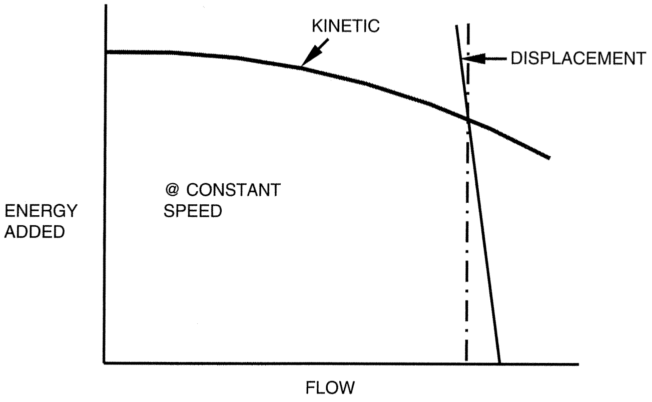

Figure 2. Flow Regulation, Kinetic Versus Displacement Pumps.

Learning from the Existing Installation

The system energy requirements for new units are estimated using various assumptions and margins. When centrifugal pumps are used, the system designer usually relies on a control valve to balance system and pump energy at the desired flow rate. In engineering a unit revamp, it is possible to determine the actual system characteristic with accuracy and thereby avoid the energy lost to conservative assumptions and margins. This loss can be on the order of 25 percent of pump power at rated capacity, far greater than that caused by the differences in efficiency between various pump selections for the same duty.

Determining the actual system head requires accurate measurement of:

- The pump's flow rate at one condition,
- The pressure at the pump's suction and discharge and at the suction and discharge vessels,
- The liquid levels, relative to some common reference, in the suction and discharge vessels, and
- The pressure drop across the control valve, if used, taking care to measure the downstream pressure some 10 diameters from the valve to avoid the influence of any flow distortion.

To make use of the pressure measurements, it is necessary to also determine the pumped liquid's specific gravity (SG) at each measuring point. This can be determined from liquid temperature provided the liquid being pumped is known with certainty. Using Figure 3 as a reference, the system head is:

$$H_{system} = \frac{(P_4 - P_1)2.31}{SG} + (Hz_4 - Hz_1) + HL_{1-4} \qquad (1)$$

where Hz_1 and Hz_4 are the static liquid levels, referred to the datum level, in the suction and discharge reservoirs respectively, and $HL_{1\text{-}4}$ is the total friction loss in the suction and discharge piping.

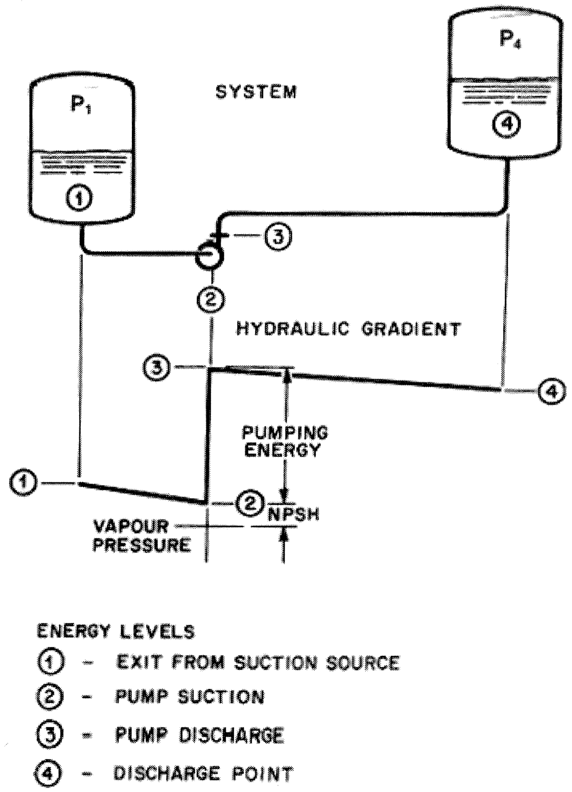

Figure 3. Hydraulic Gradient.

The pump's total head is:

$$H_{pump} = \frac{(P_3 - P_2)2.31}{SG} + H_z + HL_{2-3} + \frac{\Delta V^2}{2g} \qquad (2)$$

where Hz is the correction for gauge elevation, if any, $HL_{2\text{-}3}$ the friction loss between the suction and discharge pressure gauges, and $\Delta V^2/2g$ the difference in velocity head at the points of suction and discharge pressure measurement. The friction loss is significant when there are elbows, valves, or reducers between the gauge and the pump. The difference in velocity head usually only matters when the pump head is low and there is a difference of more than one pipe size at the points of pressure measurement.

Subtracting the static head components from the pump head (Figure 4), yields the system friction head, HL_f. When a control valve is used in the system, the head being lost to throttling across the control valve is calculated from the measured valve pressure drop, then subtracted from the total system friction to give the head lost to friction in the piping, including entrance and exit losses.

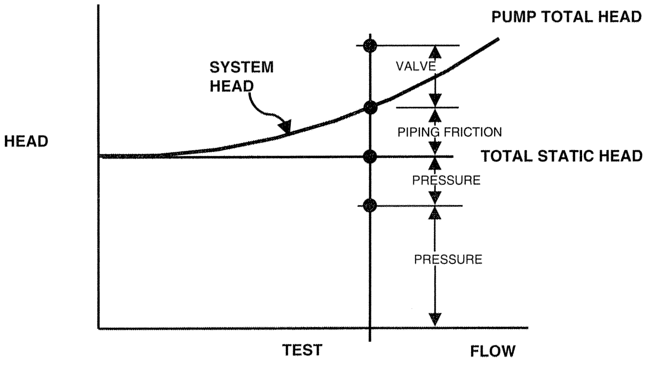

Figure 4. System Head from Measurements.

Recognizing that the head lost to friction varies as the square of the flow rate, the equivalent system friction at several other capacities can now be calculated and the system head characteristic plotted (Figure 4). If the static head varies with time, as it often does in a transfer process, then the range of system heads can be plotted after allowing for maximum and minimum liquid levels in the suction and discharge vessels.

The other critical aspect of the system to be verified using the measurements already made is the NPSH available at the pump. For measurements at the pump suction, again referring to Figure 3, the equation is:

$$NPSHA = \frac{(P_2 + P_a - P_{vap})2.31}{SG} + Hz + \frac{Vs^2}{2g} \tag{3}$$

where P_a is atmospheric pressure at site, P_{vap} is the vapor pressure of the pumped liquid at the pumping temperature, Hz is any correction for gauge elevation to the pump's reference level, and Vs is the velocity at the point of suction pressure measurement. The pump's reference level is the shaft centerline for horizontal machines, and the centerline of the suction nozzle for vertical machines.

Since NPSH available is also equal to:

$$NPSHA = \frac{(P_1 + P_a)2.31}{SG} + Hz_1 - HL_{1-2} \tag{4}$$

It is possible with the measurements already made to calculate the friction loss in the suction side of the system, and following the same procedure as used for the system head, develop the system NPSHA characteristic (Figure 5). If the liquid level in the suction vessel can vary with time, the range of NPSHA can be plotted in the same manner as the range of system heads.

With the net system head now known accurately, the pump head necessary to move the required flow and allow flow control can be kept to a minimum. At the same time, an accurate NPSHA characteristic eliminates hidden margins, which means an NPSH margin appropriate to the application can be set, thereby allowing the selection of an optimum hydraulic design. Keeping the pump head to the minimum necessary lowers energy consumption, and having an optimum hydraulic selection can contribute to both lower energy consumption and longer MTBR.

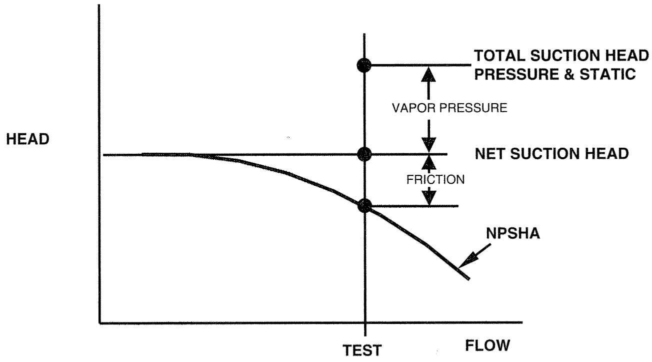

Figure 5. NPSH Characteristic from Measurements.

Before hastening off to prepare the pump specification, there are two more items to be addressed while at the site. The first is to study the suction piping. Many pump problems are caused by poor suction piping, so a unit revamp is an opportunity to correct that. And in the case of a revamp for higher unit rate, poorly laid out suction piping may need to be corrected to allow the pump to operate at the higher flow. The important features of the suction piping layout are the orientation of reducers, the proximity of elbows to each other when in different planes, the orientation of the elbow immediately upstream of double suction pumps, suction piping slope, and submergence over the vessel outlet. If it is thought there are problems in the suction piping, consult Karassik and Krutzsch (1986) or Karassik and McGuire (1998) for guidance on correction.

The second item is to ask the maintenance department about the pump's service history. What needs to be looked for in this review is evidence of problems with the pump's application, its materials of construction, or its mechanical design.

Poor application is usually evident from frequent shaft seal and bearing failures, rapid wear at the running clearances, frequent shaft failure, noise and vibration, or premature impeller erosion. All these are symptoms of prolonged operation at low flows. Whether this is the case can be determined by comparing known flow rates with the pump's performance curve to see where it has been operating relative to the pump's best efficiency point (BEP).

Better materials of construction are warranted if the pump has a history of components failing from general corrosion, corrosion-erosion, erosion, fatigue, or corrosion-fatigue. It is often hard to differentiate between causes of component failure, so it may be necessary to consult a metallurgist. In some cases, changing materials may not be enough; it may be necessary to either correct a problem in the process, for example lowering the concentration of abrasive solids or bringing the pH closer to neutral, or change to a more suitable type of pump.

Mechanical design becomes suspect only when the influences of application and the pumped liquid have been eliminated. (This is probably the reverse of common practice, but is the sequence to be followed in troubleshooting modern pumps.) A major source of mechanical problems is strain caused by piping loads. If the pump has had a high incidence of seal, bearing, coupling, or shaft failures, the cause might be piping loads. The question then is whether the piping loads are too high or the pump not stiff enough. A computer analysis of the as-built piping is the first step in resolving this question. If the piping loads are reasonable or high but cannot be changed, it is necessary to change to a pump with higher piping load capacity.

Short MTBR caused solely by the mechanical design of the pump is rare in modern designs, but not uncommon in many older designs (30 years or more). The usual difficulties are rotor stiffness, rotor construction, bearing capacity, bearing cooling, bearing housing stiffness, and casing and baseplate stiffness. These typically manifest themselves as frequent seal, bearing, and shaft failures, and rapid running clearance wear. Most of these are also symptoms of poor application, so care is needed in sorting out the true cause of the problem.

Pump Options for the Unit Revamp

Armed now with accurate data on the system head and NPSH available, and knowing whether the suction piping or pump need correction as part of the revamp, it is time to look at what has to be done and how best to do it.

First, data developed by the process designer need to be checked against the actual system head and NPSH available characteristics, and corrected where necessary. As already discussed, the decisions at this phase are how much pressure drop does the control valve need to control reliably or is it more economical to change to a variable speed pump, and what NPSH margin is necessary to ensure rated pump performance and expected life. The former is a question for the valve designer. A starting point for the latter can be taken from Table 1.

Table 1. Typical NPSH Margins.

Application	NPSH Margin $\%NPSHR_3$
Water, cold	10 – 35 [1] [2]
Hydrocarbon	10[2]
Boiler feed, small[3]	50
High energy[4]	100-200

Notes:
1. Depends on size; higher margin for larger pumps
2. Minimum 3 ft
3. Up to 2500 hp at 3600 rpm
4. U_1 greater than 100 ft/s

To meet the new conditions of service required for the unit revamp, there are three options: rerate the existing pump or pumps, buy an additional pump or pumps of the same design, or buy pumps of a new design. These choices may, in turn, be influenced by the operating history of the existing pumps. Table 2 summarizes the needs developed from investigation of the existing pumps and the usual options for satisfying them.

Table 2. Usual Options for Pump Changes in a Unit Revamp.

Need	Options				
	Rerate	Add	Replace	Mat'ls	Const'n
Lower flow	Δ				
Higher flow - small	Δ				
Higher flow - large		Δ	Δ		
Corrosion resistance			Δ	Δ	
Erosion resistance			Δ	Δ	Δ
Better mech'l design			Δ		Δ

Rerating the existing pump is the simplest course and the only one dealt with in this tutorial. To be successful, the rerate must be designed to meet the new conditions of service and at the same time overcome any deficiencies in the original application, such as being oversized for the normal flow or being of too high a suction specific speed.

Turning now to the mechanical aspects, a hydraulic rerate of older design pumps would typically be combined with a mechanical upgrade to raise MTBR. Most manufacturers now have standard upgrades available for pumps ranging from single-stage overhung to multistage.

If the existing pumps have suffered corrosion or erosion abnormal for the service and the class of pump, changing the materials should be considered. Conversely, if the corrosion or erosion appears more related to the type of pump than to the service, it may be better for the long term to change the pump.

In rare instances, it will be clear that the existing pump is the wrong configuration for the service, a circumstance that likely would be aggravated by rerating the pump to a yet higher energy level. A high-energy overhung pump on a severe service is a typical example. This will be obvious from the service history of the pump. Replacing it as part of the revamp is really mandatory because the success of the project will be jeopardized if an unreliable pump is retained in the unit.

Once the pump's new rating has been determined and its operating history established, these data need to be reviewed with the original equipment manufacturer (OEM) or a competent manufacturer of equivalent equipment to establish whether a rerate is feasible. This should be done even when the owner or his engineer think the pump should be replaced, because sometimes a rerate and radical upgrade might be feasible and more cost effective.

Whether the choice is a rerate or a new pump, the next step is to prepare the specification. The essential rule for a good specification is to keep it simple. Many a good solution has turned into a purchasing nightmare, to the detriment of the revamp project, because those preparing the specification for the pumps forgot this simple rule. The goal should be:

- A one or two page data sheet.
- Scope of supply summary, supplemented with a terminal point diagram if necessary.
- A schedule of events to complete the rerate.
- Agreed terms and conditions.

For a more detailed discussion of this critical phase and of the two means of purchasing the equipment, refer to McGuire (1996).

HOW A RERATE IS DONE

The objective is to achieve the required new pump rating while retaining as much of the original pump as possible, and correcting any reliability problems evident from operating history or deemed likely from engineering analysis of the proposed rerate. Three examples are discussed in broad detail to illustrate the overall approach for rerates involving more than just changing to maximum diameter impeller(s).

Gulf Coast Refinery—FCC Feed Hydrotreater

When first put into service in 1984, the fluid catalytic cracking (FCC) hydrotreater was rated at 55,000 bpd. Feed was charged to the unit by three half-capacity 12-stage double casing (API type BB5) feed pumps. Each pump was rated 1061 gpm, 5212 ft, 1370 hp at 3580 rpm.

A subsequent rerate raised unit capacity to 85,000 bpd. At the same time, reactor pressure was raised to increase the hydrogen content of the feed to the FCC, thereby improving its yield. Charging the unit at the higher flow and pressure was accomplished by running all three charge pumps in parallel, effectively turning them into three one-third capacity pumps.

In 1993, the refinery sought to raise the unit flow rate yet again, this time to 100,000 bpd. Preliminary process design put the required charge pump rating for the higher unit rate 1315 gpm, 5684 ft at 3580 rpm based on three pumps in parallel. This performance could not be achieved with the existing hydraulic design.

The operating history of the existing pumps had been acceptable (once some early difficulties with the shaft seals had been overcome), so the only engineering issue was finding or developing a set of hydraulics able to achieve the new rating and fit in the existing casing. A search of the barrel pump hydraulics available within the author's company yielded a "diffuser" type element of 11 stages that fit in the casing and needed only 10 stages to meet the rating, thus allowing a spare stage for a future head increase of up to 10 percent.

The one difficulty with the 11-stage element was that it was designed for counterclockwise rotation (viewed on the coupling) whereas the existing pumps were clockwise rotation. Since the pumps were motor driven, this fundamental issue was overcome by changing the motors' rotation.

To be able to test the new elements without disrupting unit operation and have a complete pump as a warehouse spare, the balance of the parts to make a fourth pump (casing, head, seal housings, and line and thrust bearings) were included in the rerate proposal.

With the projected investment needed to rerate the charge pumps on the order of $2.4 million, the refinery determined that the project was financially viable and so elected to proceed with it.

Following classical engineering practice, a detailed layout was made of the 11-stage element in the existing casing. Figure 6 shows the section of the final design. A key objective of the layout was to use as much of the existing pump as possible. This was accomplished with the following:

- Suction guide or spacer modified to match the existing casing.
- Casing head modified and a new discharge spacer provided to match the new element.
- New shaft designed to match the original bearings.

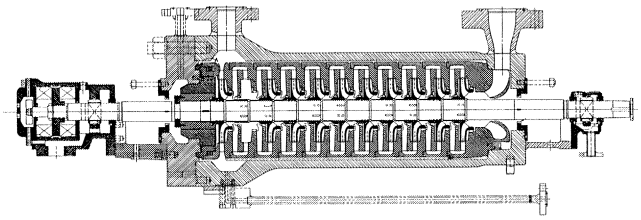

Figure 6. Section, 6 Inch Discharge 11-Stage FCC Hydrotreater Charge Pump.

Since the standard shaft for the 11-stage element was larger under the impellers and shaft seals than the original, the new shaft was about 36 percent stiffer than the original (L^4/D^2 2.4×10^6 in^2 versus 3.8). An analysis of the new rotor's dynamics showed that it was free from potential vibration problems with running clearances at "new" and two times "new" values.

Engineering analysis included a check of the pump's pressure boundary. The existing casing was checked using finite element analysis against ASME Section VIII, Division 2, allowable stresses and rerated for 2650 psig at 600°F. In the process of making that check, a potential resonance was identified in the casing wall, so the wall thickness of the new casing was increased to ensure this could not occur. There being no reports of such problems with the existing casings, they were left in their original condition. Analysis of the regions of the casing normally subject to suction pressure showed that their design pressure was limited to 1650 psig at 600°F, the limitation being seal housing bolting strength. The refinery's design practice required that such regions be good for maximum pump discharge pressure. Modifying the pump to achieve this would have involved a major redesign of the casing, hence three new casings and project delay, so the refinery elected to protect against accidental overpressure of the suction regions by other means.

Structural analysis established that the discharge end bearing bracket had to be stiffer to ensure adequate separation from possible exciting frequencies. A new design was included in the new pump. The bearing bracket of each existing pump was modified to meet the new design when its element was changed.

As engineering and manufacture were progressing, process design was finalized and the pump rating revised to 1195 gpm, 5503 ft, 1628 hp at 3570 rpm. The hydraulics were adjusted to meet this slightly lower rating.

Each element was shop tested in the new pump casing to verify its hydraulic performance and mechanical operation, the latter including axial thrust. The test curve for one of the pumps is shown in Figure 7. Rotor vibration on the test stand was limited to 1.25 mils peak-to-peak, including mechanical and electrical runout, at rated speed and within 10 percent of rated flow. This is slightly below the limit in API 610, Eighth Edition, which for this speed equates to 1.5 mils peak-to-peak from 70 to 120 percent of BEP.

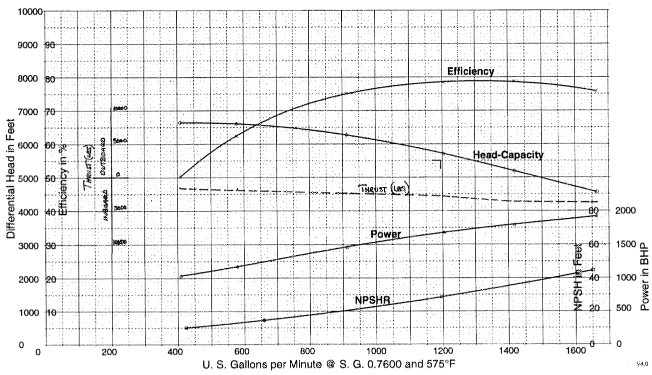

Figure 7. Test Curve of Pump in Figure 6.

The new pump was shipped directly to the refinery for installation, while the elements went to the local service center. Each of the existing pumps was then sent to the service center where it was dismantled, the casing inspected, existing parts modified where necessary, and finally reassembled with its new element.

In parallel with the charge pump rerate, the bottoms pumps for this unit were also upgraded. These radially split, single-stage, double-suction pumps, API 610 type BB2, had been a source of operating problems, often leading to low unit availability, since the unit first went online. Past experience with similar pumps had shown that the shafts were not stiff enough to ensure satisfactory shaft seal operation during adverse operating conditions. The upgrade therefore consisted of providing a stiffer shaft (larger diameter under the impeller and seals), modifying the impeller to accommodate the larger shaft, and providing new shaft seals.

The refinery's purchase orders for the rerate and upgrade were placed in July 1993. All the equipment was shipped from the manufacturing plant in January 1994, seven months from order placement. The hydrotreater went online at the higher rate three months later in March 1994.

The refinery's rotating machinery engineer for the project reports that since going back online the unit has operated as expected, in terms of both performance and availability. Speaking for future, similar projects, he emphasized the need for detailed coordination between the manufacturing plant and the local service center.

West Coast Refinery—FCC Feed Hydrotreater

After installing a "through the wall" hydrogen plant, the refinery found it had surplus hydrogen. Process design studied what could be done with this hydrogen and determined the best use was to simultaneously increase the rate of the FCC hydrotreater from 55,000 bpd to 65,000 bpd (with the possibility of running at 70,000 bpd in the future), and raise the pressure in the reactor to increase the hydrogen content of the feed to the FCC. Their estimate of the economics showed additional revenue of $13 million per year for an investment of $4.5 million.

The hydrotreater was served by two full-capacity charge pumps, each rated 1900 gpm, 5500 ft, 2955 hp at 3570 rpm. Preliminary process design required that the pumps be rerated to 2400 gpm, 6065 ft at 3570 rpm. Estimated power at the new rating was 3955 hp, which meant that the existing 3000 hp drivers also had to be replaced.

Operating experience with the existing charge pumps, 8 inch discharge, 11-stage barrel-type with an axially split, volute-type inner casing, had shown problems with leakage across the inner casing split joint to the lower pressure stages (typically second and third) and leakage past the radial seal between the outboard end of the element and the casing head (Figure 8). This leakage had often reduced hydrotreater flow rate and the erosion caused by it required welding and remachining to restore the inner casing. These difficulties were compounded by the inner casing being CG-8M stainless steel (equivalent of wrought type 317) to resist naphthenic acid corrosion, which meant a large "cold" clearance at the element to casing radial fits to allow for differential thermal expansion at the maximum operating temperature of 450°F. This posed a difficulty in centering the rotor since the stator centerline rose approximately 12 mils relative to the casing as the pump came up to temperature.

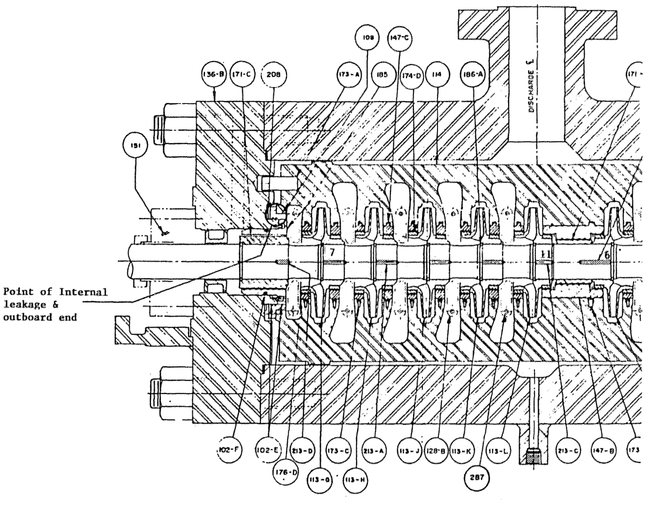

Figure 8. Point of Internal Leakage in Original Pump.

Taking account of the known internal leakage and rotor centering problems, the uncertainty of rerating the existing hydraulics (volute modification was limited), and the need for short delivery, it was decided to retrofit the existing pump with a radially split diffuser type element of existing design. Initial engineering showed that a "diffuser" type element of 10 stages would fit in the existing casing, and needed only nine stages to meet the rating, thereby leaving a "dummy" stage for a future head increase of up to 11 percent. To maintain complete interchangeability with the existing hydraulics, the new element was supplemented with new casing heads, seal housings, and complete line and thrust bearings. The final design is shown in Figure 9.

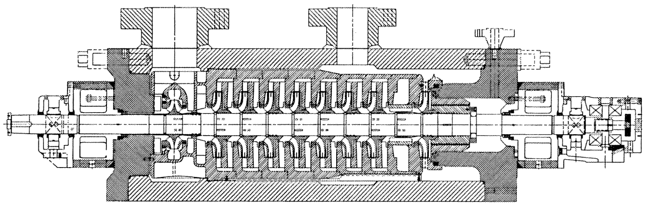

Figure 9. Section, 8 Inch Discharge 10-Stage FCC Hydrotreater Charge Pump.

Given the extent, hence cost, of the rerate and the owner's good past experience with the line of pumps from which the element was chosen, it was decided to rerate only one of the two installed pumps. This meant zero pump redundancy for operation above 55,000 bpd.

Because the existing pump had an opposed impeller rotor, its discharge was located midway along the casing. The 10-stage element has tandem (inline) impellers, so liquid discharged at the outboard end of the element has to pass back over half of the element to reach the discharge nozzle. To provide a large enough annulus for this flow, the OD of the latter stages of the element was reduced (Figure 9). Engineering analysis verified that the resultant compressive stress in the stage piece wall from radial pressure difference and axial force was still acceptable.

Being radially split with metal-to-metal face seals and shrink-fit assembled stage pieces, the new design element provided very good insurance against leakage from the discharge to the lower pressure stages. At the outboard end of the element, the pressure difference across the radial locating fit (Figure 9) is the pressure rise through the diffuser, which is negligible in terms of erosion potential.

To avoid the need for a large "cold" clearance at the element's radial locating fits, the replacement element was made from CD4MCu, a duplex stainless steel with a coefficient of thermal expansion much closer to that of the forged carbon steel casing.

Early in the engineering phase of the project, the pump selected for rerating was shutdown and its element removed so the casing could be inspected and measured. Inspection established that the critical fits and faces would need to be restored by overlaying with austenitic stainless steel and remachining. This was added to the work to be done when the casing finally came to the manufacturing plant.

Using the actual casing dimensions, the casing's pressure rating was checked by finite element analysis against ASME Section VIII, Division 2, allowable stresses. By this approach, the design pressure for the discharge regions of the casing was 2850 psig at 450°F. This was just enough to accommodate the new hydraulic design's 28 percent head rise (from rated to bypass) plus maximum suction pressure.

Once the basic hydraulic design was selected and a decision made to proceed, the critical steps in executing the rerate were:

- Material selection
- Material delivery
- Final pump rating
- Casing restoration and hydrotest
- Element assembly
- Pump test

To meet the owner's unit turnaround date, completing these steps in time became an exercise in concurrent engineering. That meant starting manufacture of the various parts before detail engineering was completed, then issuing engineering changes as manufacture progressed. With close cooperation between the owner, the manufacturer, and its suppliers this was achieved. The net result was that the tested pump shipped from the manufacturing plant 17 weeks after the owner authorized the first step in manufacture (production of waxes for the precision cast impellers).

At the point of being ready to start final machining of the hydraulic parts, process design settled on a pump rating of 2300 gpm, 5750 ft, 3450 hp at 3570 rpm, slightly below that used for the initial design. Figure 10 shows the rerated pump's "as-built" performance.

The unit went back online with the rerated pump in late May 1994. It has run since at rates up to the permitted 70,000 bpd and without any operating incidents attributable to the charge pump. The pump has demonstrated the ability to run out to higher rates, meaning that its rating is conservative, and when checked after 39 months operation showed no reduction in hydraulic performance.

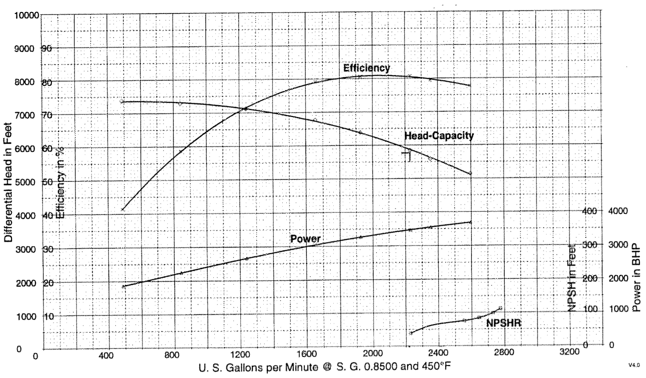

Figure 10. "As-Built" Shop Test Performance of Pump in Figure 9.

Latin American Refinery—Hydrocracker

The hydrocracker unit was to have two reactors operating in parallel for a total capacity of about 57,000 bpd. It was equipped with two full-capacity double-casing 13-stage charge pumps, each rated 1828 gpm, 8440 ft, 4530 hp at 3570 rpm. Rated capacity was at 79 percent of pump BEP.

For reasons not known, only one of the reactors was built, which meant that the charge flow required was about half the pumps' rating or about 40 percent of BEP. It was recognized that prolonged operation at such a low fraction of BEP would lead to shorter pump MTBR and so higher maintenance costs. To avoid this, the unit started up circa 1971 with a permanent bypass of 870 gpm to raise the operating charge pump's gross capacity to rated. From 1971 through 1993, the charge pump realized acceptable MTBR. To quite an extent, this was offset by very short MTBR, on the order of three months, for the bypass valve. This came to light in 1993 when the owner sought a multiple pressure reducing orifice to lower the bypass valve's pressure drop and so its erosion rate. Discussion of the problem suggested that it might be more economical over the long term to rerate the pumps rather than install a more durable permanent bypass.

The owner's process designer set the desired capacity for a rerated pump at 1140 gpm, based on a reasonable margin over the unit's actual operating capacity. Of the available hydraulic designs, the best fit for the new rating had rated capacity at 93 percent of BEP (Figure 11). Using $0.04 per kWh for the cost of electricity, a low value, and assuming continuous operation and 83 percent unit availability, the minimum savings in energy cost was estimated at $315,000 per year. This was sufficient to pay for a rerated inner element in just under nine months. The rerate required new impellers and diffusers, the former mounted on the existing shafts, and latter installed in the existing stage pieces.

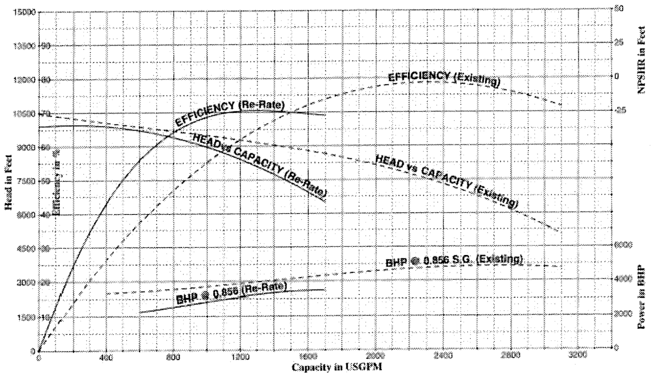

Figure 11. Original and Proposed Rerated Performance of Hydrocracker Charge Pump.

Because the shaft diameter at the impeller, d_1, was an unusually large fraction of the impeller outside diameter, D_2 (ratio $d_1/D_2 = 0.38$), care was needed with the impeller design. A search of the company's hydraulic database established that there were successful versions of such designs and therefore provided guidance on the design for the rerate. The section of the pumps with the new hydraulic components is shown in Figure 12.

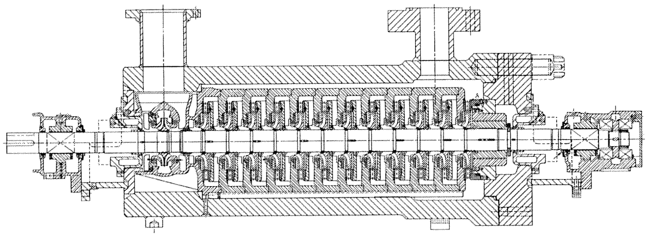

Figure 12. Section, 8 Inch Discharge, 13-Stage Pump Hydrocracker Charge Pump.

The first replacement element, produced by modifying the spare element, was completed in late 1994. It was not tested because both the casings were needed at the refinery. This element was installed and put into service in early 1995. Initial field testing showed capacity was well below expected. A check of the system piping found the 4 inch bypass line had been left in place to warm up the standby pump. Once the warmup flow was reduced to the correct value, the operating pump's capacity was as expected.

The second element was shipped in early 2000 and started up later the same year.

CONCLUSIONS

Rerating an existing pump can:

- Lower the capital cost of a unit revamp for higher rate thereby improving its financial viability,
- Reduce energy consumption when the actual operating capacity is far from the pump's design capacity, and
- Raise the MTBR of a pump whose reliability is poor.

The keys to a successful rerate are:

- Determining the actual system head for the pump as now operating,
- Checking the pump's operating history,
- Establishing the new rating for the pump,
- Careful review of the required rating and operating history by the OEM or a competent manufacturer of similar pumps,
- A simple purchase specification, and
- Competent project management.

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