TOWARD REDUCED PUMP OPERATING COSTS

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ABSTRACT

Pump components such as seals and couplings affect the reliability and operating costs of pumping equipment directly. By reference to basic principles, those effects will be quantified and compared with the overall running costs of pumps. Available options will be examined and guidance given on the best choices.

Examples of how the performance of mechanical seals in critical service has been improved will be given and experience in improving the reliability of large populations of pumps in plants around the world will be outlined.

RELIABILITY AND OPERATING COSTS FROM FIRST PRINCIPLES— BASIC RELIABILITY CONCEPTS

Reliability Engineering

Reliability engineering is a mathematically based, esoteric applied science. It is full of probability, uncertainty, and the theory is difficult to understand. Surely then it is the prerogative of mathematicians! Not so, our advice is "get involved, get your feet wet, and let the theory catch up with the practice later!" All plant, equipment, and components have a finite life and, eventually, even the best of them will fail.

Failure

Failure is defined as the inability of a component or system to carry out its specified function.

Reliability

Reliability is the probability that a component (e.g., seal), device (e.g., pump), or system (e.g., plant) will perform its function without failure for a given time when operating properly in a specified environment.

• Example—A seal on a methanol pump. The answer can be expressed in one of three ways:

- 100% (or 1) — The seal will definitely *not* fail in the period.
- <100% (e.g., 50%) — There is a definite chance that the seal will fail in the period.
- 0% (0) — The seal will definitely fail in the period.

A Definition of Mechanical Reliability

We need an unambiguous and precise definition of reliability. It is not good enough to have just good (reliable) and bad (unreliable) products. The following definition has been developed specifically for mechanical equipment:

• *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.*

Some of the *terms* need further explanation.

Probability

The special significance of probability is that it cannot always be predicted with certainty that some event will occur. (Weather forecasting, tossing a coin, betting on a horse, or predicting when a mechanical failure will occur.) Such events are probabilistic. Probability can be considered as the chance of some event happening (Table 1).

Table 1. Hypothetical Failure Pattern of 100 Machines on Test.

Time Period	Event	Reliability R
10 weeks	No machines fail in first 10 weeks	1.0
30 weeks	5 failures in weeks 10/20 10 failures in weeks 20/30 **Total = 15 failures**	0.85
60 weeks	15 failures up to week 30 20 failures in weeks 30/40 30 failures in weeks 40/50 25 failures in weeks 50/60 **Total = 90**	0.10
70+ weeks	90 failures up to week 60 10 failures in weeks 60/70 **Total = 100**	0

The relative "frequency of failure" is the number of machines, $n(t)$, failing in a time interval divided by the total number on test. The histogram (Figure 1) leads to the definition of the probability of a single machine failing between an upper time limit of t_2 and t_1. The probability of a single machine failing in t_2/t_1 is:

$$= F(t_2,t_1) = \{n(t_2,t_1)\}/N \tag{1}$$

where $n(t_2, t_1)$ is the number of failures in time zone t_1 to t_2, and N is the number in the sample. For example, the probability of failure in 40/50 weeks = 30/100 = 30 percent, shown as 0.3 in Figure 1.

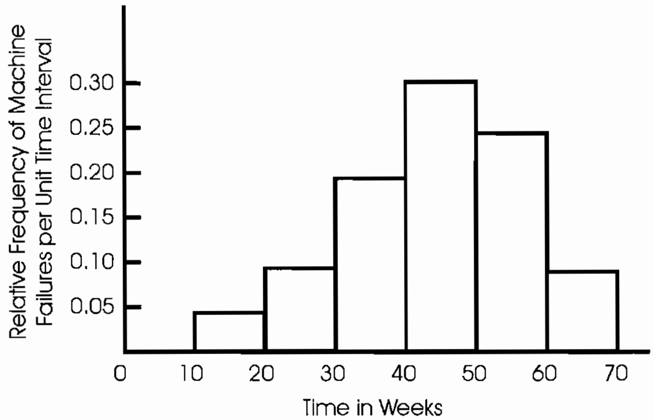

Figure 1. Histogram Charting Table 1 Information.

With a very large number of parts, components, or devices and small time intervals, a smooth shaped failure distribution curve can be drawn (Figure 2). Then the "relative frequency" term is replaced by the "probability density function" (PDF), which is a function of time and represented by $f(t)$. In this case, the probability of a single machine failing during the time increment, dt (that is, $(t, t+dt)$), is given by:

$$F(t,t + dt) = f(t)dt \tag{2}$$

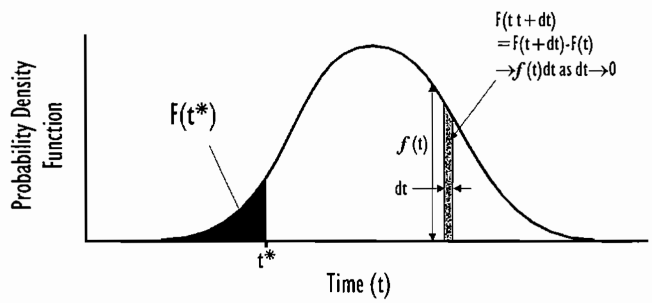

Figure 2. Probability Density Function (PDF).

For Figure 2:

• Probability during time increment dt (t, t+dt) is F(t, t+dt) = f(t) dt

• Probability in increment (0, t*) = f(t)dt = Area 1 (shaded area to the right on Figure 2)

• Probability of failure F(t)

• Reliability $\frac{R(t)}{1}$

• Probability of failure in time t + Probability of survival in time t = 1

$$F(t) + R(t) = 1 \tag{3}$$

Component, Device, or System

A *component* is the smallest part that would normally be replaced. A *device* comprises several (if not many) components, e.g., pump, compressor, gearbox. A *system* comprises several (if not many) devices, e.g.. a process plant, an airplane. This is important as the mathematics are different for components, devices, and systems.

Duty

The actual service, or *duty*, expected of an artifact is of prime importance to its reliability specification as it describes the expected stresses during *normal operation.* Mechanical systems are very different from electronic systems, which are basically benign.

Consider a pump. During startup, it can consume more than three times its normal power to overcome inertia and break out friction. It can be operated at nominal duty point or:

- Toward full head
- Toward full flow
- At overspeed
- Under conditions of cavitation, etc.

The design of a pump for mechanical reliability will therefore require a detailed analysis of the likely operating stresses. Too often, with standard equipment, the purchaser does not know clearly the range of stresses for which the pump was designed, and the supplier does not know the range of loads for which it is required.

Failure

What constitutes the *failure* of an item to perform its duty? It is often possible for a piece of mechanical equipment to perform all or part of its function in an impaired state (e.g., a leaking valve). To improve the classification of failures, it is normal practice to relate to their modes, causes, and effects. A *failure mode* in a mechanical seal might be external leakage or overheating or emission of noise. The *cause* might be abrasives in product or loss of circulation or dry running. The *effect*, however, must be enough to stop the device from sustaining its prescribed duty.

Time

It can be possible to measure the *time* to failure in time units or in cycles (on/off operation of a switch). Normally, however, time is the most convenient measure.

Correct Operation

Misuse or abuse of a piece of equipment can seriously affect its reliability. It is important, therefore, that the equipment supplier specifies the limits within which the equipment should be used, which, in turn, depends upon an accurate knowledge of customer requirements. It also necessitates high standards in the design and use of operating and maintenance manuals.

Specified Environment

The more severe the stress imposed on a piece of equipment, the more likely it is to fail. Specifying the operating *environment* correctly is clearly a most important first step to attaining high reliability.

In developing a better understanding of mechanical reliability, we will now look at "interference diagrams" and the "bathtub" curve.

Interference Diagrams

The strength of a component can vary between limits due to tolerances in materials and machining. The load imposed on a component can also vary within a range. Strength and load may be represented as probability density functions (PDF) on an interference diagram (Figure 3).

Strength and load are measured in identical units. If the PDFs of strength and load are separated, (Figure 3), then theoretically the component will be absolutely reliable.

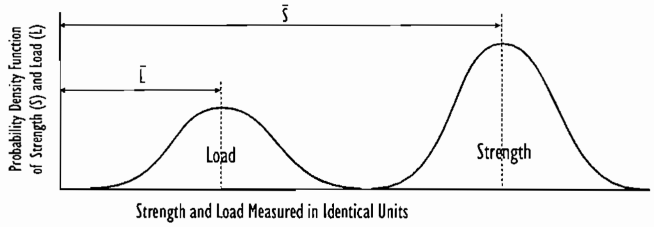

Figure 3. Strength/Load Interference Diagram with Complete Separation. ($\bar{S}$= Mean value of component strength, $\bar{L}$ = Mean value of imposed load, σ_S = Standard deviation in components strength, σ_L = Standard deviation in load imposed)

If, however, the PDFs of strength and load interfere, as in Figure 4, then there is a probability that the load imposed will exceed the strength of the component and it could fail. The numerical value of this probability is the area of overlap. The larger this area, the higher the probability of failure. An established technique is the use of the safety margin (SM).

$$SM = \frac{\bar{S}-\bar{L}}{\sqrt{\sigma_S^2+\sigma_L^2}} \tag{4}$$

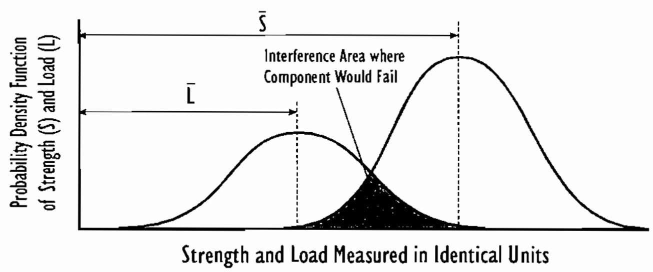

Figure 4. Strength/Load Diagram with Interference.

The Bathtub Curve

A fundamental concept in reliability engineering is the "hazard rate function," *z(t)*. This is *not* the "failure rate." The hazard rate function is used to describe the mathematical function describing the behavior of nonrepairable components that form part of a system. The instantaneous value of the hazard rate function is the hazard rate.

The failure rate implies that the time to failure distribution is exponential and is used for repairable systems. The "hazard rate function" is, then, a measure of the probability that a component will fail in the next time interval, given that it has survived up to the beginning of that time interval. It can be proved that the probability of failure in the next time interval:

$$= z(t)dt = \frac{F(t + dt) - F(t)}{R(t)} \tag{5}$$

which can be simplified to:

$$z(t)dt = \frac{f(t)dt}{R(t)} \tag{6}$$

so that the hazard rate function:

$$z(t) = \frac{f(t)}{R(t)} = \frac{\text{Probability density function}}{\text{Reliability}} \tag{7}$$

The hazard rate function is commonly assumed to take the shape of the "bathtub curve" (Figure 5).

For Figure 5:

- Phase 1— "Infant mortality" phase

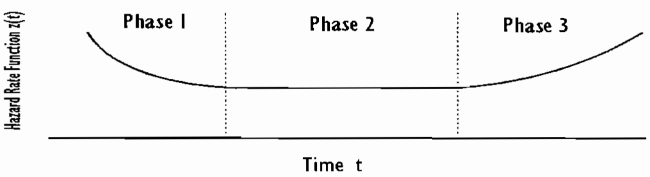

Figure 5. The Conceptual "Bathtub" Curve.

- Phase 2—Approximately constant *z(t)* due to chance failure through unexpected overstressing (dropping a plate), "normal operating life." The constant value of *z(t)* is usually called the failure rate and denoted by the symbol, λ.
- Phase 3—An increasing *z(t)*, meaning that the probability failure is increasing (like people aging), "wear out phase."

This curve should not however, be taken too literally and can vary dramatically between different types of components. Reliability life testing in mechanical products tends to be confined to mechanical components due to the cost and difficulty. Reliability attributes tend to be measured "in service."

FAILURES, RELIABILITY, AND LIFETIMES—A CLOSER LOOK

Failure Rate

Failure rate, λ, has the dimension of number of failures/unit of time. For example, if 10 out of 1000 components fail in 5000 hours, then the proportional failure rate = $\frac{10}{1000} \times \frac{1}{5000}$

$$\lambda = 2 \times 10^{-6} \text{ of total/hr} \tag{8}$$

$$\% \text{ failure rate} = \frac{10}{1000} \times \frac{1}{5000} \times 100 = 0.0002\%/\text{hour} \tag{9}$$

Failure rate is often expressed as percent per 1000 hours, i.e., 0.2 percent/1000 hours.

Reliability equations are:

Constant failure rate
Probability of no failures can be expressed as $R = e^{-\lambda t}$
λ is constant failure rate
R is reliability (the probability of no failures in time, t)
Q is unreliability (the probability of failure in time, t)

$$R + Q = 1, \text{ so} \quad Q = 1 - e^{-\lambda t} \tag{10}$$

Number of survivors/failures
Ns = Survivors at time, t,
No = Original number

$$Ns = No\, e^{-\lambda t} \quad Ns/No = R = e^{-\lambda t} \tag{11}$$

Mean time between failure (MTBF(m)) usually refers to a situation where the failure rate, λ, is constant (Figure 6).

$$\text{MTBF} = \frac{1}{N_O}\int_0^\infty N_S dt = \frac{1}{N_O}\int_0^\infty N_0 e^{-\lambda t} dt = \int_0^\infty e^{-\lambda t} dt \tag{12}$$

$$= \left[-\frac{e^{-\lambda t}}{\lambda}\right]_0^\infty = \frac{1}{\lambda} \tag{13}$$

$$\text{MTBF (m)} = \frac{1}{\lambda} \tag{14}$$

$$\text{MTBF} = \frac{\text{total number} \times \text{time period}}{\text{no. of failures in time period}} \tag{15}$$

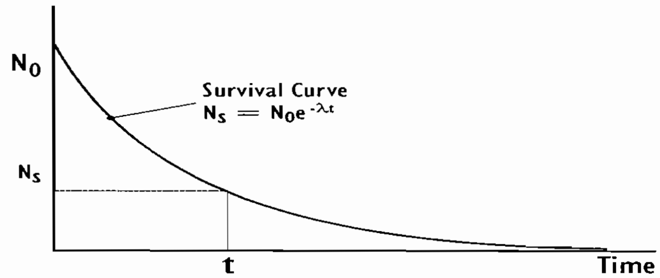

Figure 6. Survival Curve.

For example, 100 units in service with 50 failures in one year:

$$\lambda = \frac{50}{100} \text{ per year} = 0.5 \text{ per year} \tag{16}$$

$$\text{MTBF} = \frac{100}{50} \times 1 \text{ year} = 2 \text{ years} \tag{17}$$

RELIABILITY DEPENDENCIES— PUMPS, SEALS, AND COMPONENTS

In the case of pumps in process plants, we measure:

$$\text{MTBF as } \frac{\text{Total number of pumps}}{\text{Total number of failures}} \times \text{Review period} \tag{18}$$

To complicate this, there is a high proportion of spared pumps (A and B and even C). If we assume that the proportion of pumps spared is K_s, then a truer $\text{MTBF}'_{(\text{pump})}$ would be:

$$\text{MTBF}'_{\text{pump}} = \text{MTBF}_{\text{pump}} \times 1/(1 + Ks) \tag{19}$$

and K_s is typically 0.6, so:

$$\text{MTBF}'_{\text{pump}} = \frac{\text{MTBF}_{\text{pump}}}{1.6} = 0.625 \text{ MTBF} \tag{20}$$

In the case of seals, MTBF is often calculated conveniently as follows:

$$\text{MTBF(seals)} = \frac{\text{Total number of pumps}}{\text{Total number of seal failures}} \times \text{Review period} \tag{21}$$

In this case, there are two complications; a number of pumps (K_B × total) are between bearings (two seals) and, as before, a number of pumps are spared. In this case:

$$\text{MTBF}'\text{(seals)} = \text{MTBF(seals)} \times \frac{(1 + K_B)}{(1 + K_S)} \tag{22}$$

K_B is typically 0.15 for a refinery, so:

$$\text{MTBF}'\text{(seals)} = 0.719 \text{ MTBF(seals)} \tag{23}$$

Similar allowances can be made for bearings and couplings, but for convenience, the general definition for pumps and pump components is often accepted as follows:

$$\text{MTBF} = \frac{\text{Number of pumps}}{\text{Number of component failures}} \times \text{Review period} \tag{24}$$

As long as the method of calculation is left constant, then the principal objective of obtaining comparative data is achieved. Care must clearly be taken when comparing data from different sites that may have been done on a different basis, although adjustments can invariably be made.

• *Key point*—The basis of MTBF calculations must, however, always be clearly stated.

THE PRACTICALITIES OF MTBF

The theorists ask that MTBF be applied to chance failures (Figure 7). In a real process plant, it is impossible to do this as it is impossible to separate out infant mortalities, chance failures, and "wear out" failures across the board. Sometimes it is almost impossible to collect any data and so it is accepted that the MTBF values produced for pumps, seals, bearings, and couplings are normally a mixture of infant mortalities, chance failures, and wear out failures.

• *Key point*—Having said that, proper segregation of failures into those categories will greatly aid in the understanding of failures and the consequent improvements in MTBF through corrective actions.

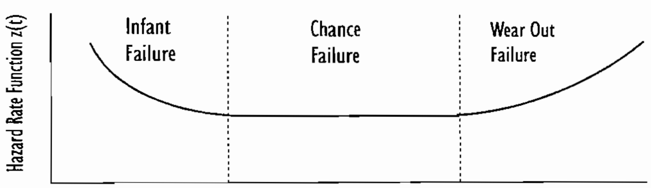

Figure 7. Bathtub Curve.

It should, therefore be the objective of plant operatives who are really serious about improving reliability and saving money to produce more comprehensive data and analyze it more carefully. Once again, close cooperation between user and supplier is a key element in achieving success.

Confidence limits are such that, as in all things, calculated MTBF values are average values and subject to uncertainty and variation. Variation is normally in accordance with normal distributions for wear out failures (Figure 8). "Wear out" may apply to car tires but rarely applies to mechanical seals, for example. Other time dependent effects like O-ring deterioration or accumulation of coke and hangup may be regarded as wear out effects, as they are time dependent. A clear objective is to systematically:

- Eliminate infant mortalities
- Minimize chance failures
- Achieve and then extend wear out failures

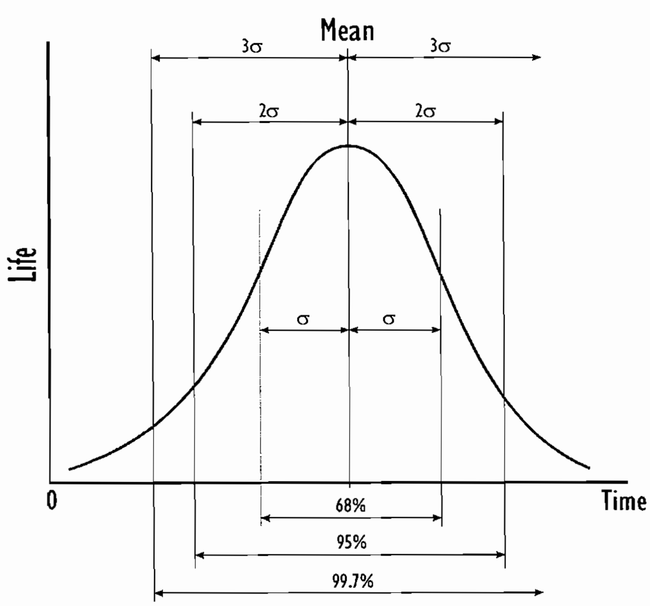

Figure 8. Wear Out Curve—Normal Distribution.

SYSTEM RELIABILITY

There are two types of systems:

• *Series systems*—Where the failure of one of the subunits or components means failure of the system as a whole

• *Parallel systems*—Systems that do not fail until all subunits or components have failed.

Individual pumps are normally series systems but can have parallel components (discussed later).

Series Systems

If X and Y are two independent events, and P_X is the probability that X will occur, and P_Y is the probability that Y will occur, then the probability that both events will occur is given by:

$$P_{(XY)} = P_X \times P_Y \qquad (25)$$

The reliability of a series system or probability of survival of the system is the probability of all the components surviving, since a failure of only one component means overall system failure. For constant values of λ (chance failures), it can be shown that:

$$\text{MTBF } m_S = \frac{1}{\lambda_S} = \frac{1}{(\lambda_1 + \lambda_2 + \lambda_3 + \text{etc})} = \frac{1}{\left(\frac{1}{m_1} + \frac{1}{m_2} + \frac{1}{m_3} + \text{etc}\right)} \quad \text{Rule 1} \qquad (26)$$

Example—ANSI Pump Component Lives

In the "pump world," a series system can be schematically represented as shown in Figure 9. If any component fails, then the pump becomes unserviceable and must be repaired. The pump, thus, has one effective MTBF, based on the MTBFs of the individual components. This can be illustrated using public domain data on ANSI pump reliability (Table 2).

$$m_S = \frac{1}{\left(\frac{1}{1.2} + \frac{1}{3} + \frac{1}{4} + \frac{1}{15.4}\right)} = 0.675 \text{ years} \qquad (27)$$

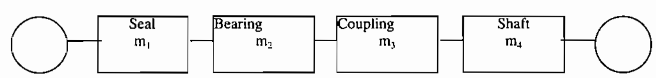

Figure 9. Schematic Showing a Pump as a Series System.

Table 2. ANSI Pump Reliability Data.

Component	MTBF (Years)
Mechanical seal	1.2 (m_1)
Ball bearing	3.0 (m_2)
Coupling	4.0 (m_3)
Shaft	15.4 (m_4)

However, the data normally produced for a pump is *not* at constant failure rate and the following equation is commonly, alternatively used.

$$\frac{1}{m_S^2} = \frac{1}{m_1^2} + \frac{1}{m_2^2} + \frac{1}{m_3^2} + \frac{1}{m_4^2} \quad \text{Rule 2} \qquad (28)$$

$$\frac{1}{m_S^2} = \frac{1}{1.2^2} + \frac{1}{3^2} + \frac{1}{4^2} + \frac{1}{15.4^2} \quad m_S = 1.07 \text{ years} \qquad (29)$$

The effect of varying seal life can be seen in Table 3. Taking the maximum reasonable values for bearing and coupling life of 10 and 20 years, respectively, and proposing seal lives of three and five years, the effect on pump life, using Rule 2, can be seen in Table 4.

Table 3. Effect of Varying Seal Life.

Mechanical Seal Life	Pump Life Rule 1	Pump Life Rule 2
0.8	0.527	0.758
1.2	0.675	1.070
1.6	0.785	1.326
2.0	0.871	1.523
2.4	0.939	1.687
2.8	0.995	1.809
3.2	1.040	1.905

Table 4. Pump Life as a Function of Seal Life.

	Life years	
Seal	3	5
Bearing	10	10
Coupling	20	20
Shaft life	15.4	15.4
Pump	**2.797**	**4.2**

Parallel Systems

If the two events can occur simultaneously, the probability that either X or Y, or $X + Y$ occurring is $P_{(x+y)}$:

$$P_{(X+Y)} = P_X + P_Y - P_X \times P_Y \tag{30}$$

The reliability of a parallel system is the probability of at least one component surviving. If R_P is system reliability and R_1 and R_2 are component reliabilities:

$$R_p = R_1 + R_2 - R_1 \times R_2 \tag{31}$$

This formula can be extended to more than two components. It can be shown that the MTBF mp for a two unit system can be given by:

$$m_p = m_1 + m_2 - \frac{m_1 \times m_2}{(m_1 + m_2)} \tag{32}$$

A clear example of a parallel system is a number of pumps operating in parallel, but this is beyond the scope of these notes.

Another example is a seal "subsystem" where a tandem (unpressurized dual) seal system is used. The seal system is viable until both seals have failed, hence it could be considered as a parallel system (Figure 10).

In the context of the other pump components (a series system), the schematic now becomes what is shown in Figure 11, and,

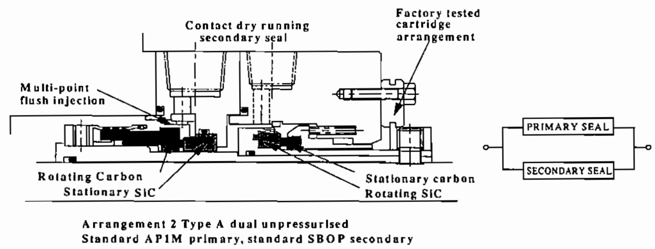

Figure 10. Unpressurized Dual Seal as a Parallel System.

following the ANSI pump example again, and assuming constant failure rate, we can compare the resultant pump MTBF for the (series) and parallel systems. First, look at the seal subsystem with the primary seal having the same life as before, 1.2 years, and the secondary seal having a slightly reduced life of say 1.0 years (Figure 12).

$$m_p = m_1 + m_2 - \frac{m_1 \times m_2}{(m_1 + m_2)} \tag{33}$$

$$m_p = 1.2 + 1.0 - \frac{1.2 \times 1.0}{(1.2 + 1.0)} = 1.65 \text{ years} \tag{34}$$

Then for the system, refer to Table 5. With the overall being calculated typically as:

$$\frac{1}{m_S^2} = \frac{1}{m_p^2} + \frac{1}{m_2^2} + \frac{1}{m_3^2} + \frac{1}{m_4^2} \tag{35}$$

$$\frac{1}{m_S^2} = \frac{1}{1.65^2} + \frac{1}{3^2} + \frac{1}{4^2} + \frac{1}{15.4^2} \tag{36}$$

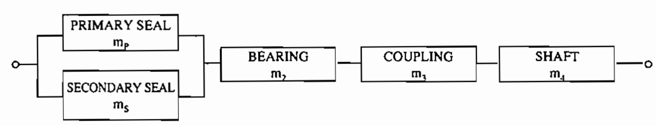

Figure 11. Schematic of Figure 10.

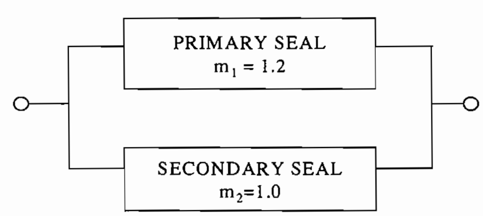

Figure 12. Primary and Secondary Seal Schematic.

Table 5. MTBF Comparison of Tandem Versus Single Seals.

Component	MTBF (Years) Tandem seal	MTBF (Years) Single seal
Mech seal	1.65	(1.2)
Ball bearing	3.0	(3.0)
Coupling	4.0	(4.0)
Shaft	15.4	(15.4)
Overall	1.35	(1.07)

While it is not a serious proposal, the addition of a secondary seal, purely for the reason of extending seal MTBF, can extend pump MTBF by a factor of 1.35/1.07 = 1.26, or 26 percent.

The purpose of this exercise is to illustrate the power of a parallel subsystem and illustrate the gains available, even if the secondary element has a shorter lifetime than the primary.

Operational and Cost Implications

The effects of maintenance policy on in-service reliability are:

- *Preventive maintenance*—For those circumstances where it is necessary to take a component out of service to return it to "as good as new" (AGAN) condition, it can be proved that such a policy is only justified if two conditions prevail:
  - z(t) should be increasing (it is wearing out).
  - It can be shown that the effect of failure is such that preventive maintenance can be justified on economic, safety, environmental, or moral grounds.
- *Corrective maintenance*—The repair or replacement of a component only takes place after a component has failed in service and should be used in all cases where the above two conditions do not apply.

Reliability centered maintenance (RCM) is a formal technique outside the scope of these notes.

The "AGAN" Philosophy

Research has shown that, after repairs, mechanical equipment is not "as good as new" (AGAN) in a reliability sense.

Devices or systems that have been given a thorough overhaul or reconditioning as the framework of the device or system often exhibits "wear out effects." If the equipment is repaired locally, maintenance and reassembly standards are rarely as good as in the original factory. As a result, the device is rarely restored to AGAN in a reliability sense. Most literature on reliability engineering ignores this phenomenon.

Conversely, it is actually possible to improve the reliability of equipment with successive repairs. This often occurs during the early operating experience with new designs or applications where there is successive development to design out earlier failures by:

- Design
- Operating procedure
- Maintenance deficiencies

Whether *z(t)* increases, decreases, or oscillates with time depends upon the relative impact of improvements through learning and the effect of successive maintenance actions. The mean time between failures (MTBF) for a sample of mechanical devices can decline, rise, or oscillate, depending on which of these effects is stronger. This can depend as much on the quality of maintenance management and service as it does on any intrinsic design or reliability attributes of the new device or its spare parts.

LIFE-CYCLE COSTS

The costs associated with a piece of mechanical equipment through the whole of its "life-cycle" are known as its life-cycle costs (LCC). These costs include:

- Design costs
- Technical and market research and development costs
- Prototype fabrication and testing costs
- Production costs
- Warranty costs
- Equipment and capital costs
- Operating costs
- Repair and maintenance costs
- Downtime or lost opportunity costs
- Replacement costs
- Retirement and disposal costs

All these costs are influenced to some degree by equipment reliability.

One of the best opportunities for minimizing LCC is at the design stage, where all options can be considered. With established products and technology, it is easier (though not easy) to minimize LCC. With leading edge technologies, it is much more difficult and LCC is normally higher.

The technology of mechanical reliability needs to be developed much more before LCC for a brand new product can be minimized from an early stage. Two prerequisites for achieving this exist:

- Adequate recording and analysis of in-service reliability data
- The feedback of all performance and reliability data to the designers in order to influence:
  - Design modifications for existing products
  - Design improvements for new generation products
  - Development of relevant and cost effective data banks that form an essential link in the search for engineering excellence

There are tradeoffs in the search for minimized life-cycle costs:

- Normally, increased reliability includes increased capital cost, but there is a law of diminishing returns where additional cost does not bring a useful improvement in reliability.
- Improved reliability will reduce other LCC costs.

This can be usefully represented in a life-cycle cost model, which is shown in Figure 13. An example of pump life-cycle costs, based on information provided by a North American oil company, is shown in Table 6.

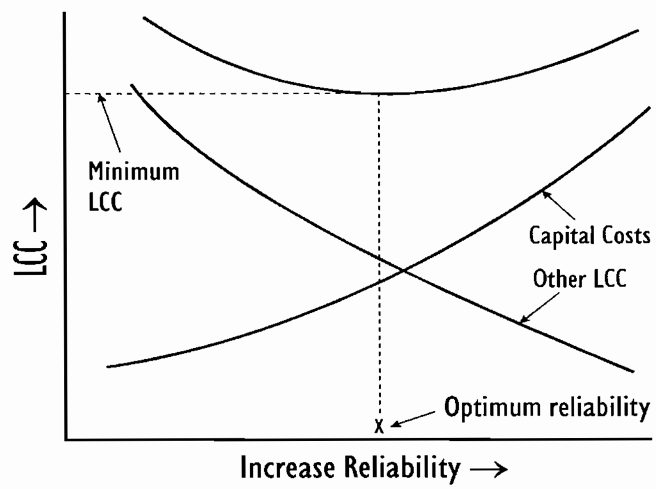

Figure 13. Life-Cycle Cost Model.

Pump Capital Cost

As can be seen, the initial cost of the pump is relatively small in comparison to the total cost. Most of the cost is incurred in operation. Pumps are often purchased on a least cost basis, but this is not always the best choice. Often a more expensive pump will have better efficiency and reliability. This is especially important in situations where downtime can incur high financial penalties resulting from lost production.

Table 6. Example of Pump Life-Cycle Costs.

API Pump, 50 HP, Single Seal, Base, and Coupling, 20 Yr Life		
Pump	$40,000	7%
Startup spares	9,200	2%
Procurement	15,486	2%
Installation	60,000	10%
Operation	384,315	67%
Maintenance	56,500	10%
Disposal	12,000	2%
Total cost of ownership	**$577,501**	**100%**
Less pump and startup spares	**$528,301**	**91%**

Startup Spares

Typical startup spares include gaskets and joints, bearings and mechanical seals, a shaft and spare impellers. Such parts are essential because accidental damage can easily occur during commissioning when proper operating procedures are not in place.

Procurement

Before a pump is purchased, a number of areas must be evaluated. The engineering content whereby power rating, head, flow, NPSH, motor speed, and system design are to be considered. Once these are defined, tender and specification documents are prepared, bids are evaluated, and an order is placed. Then the pump is dispatched and freight, insurance, import and duty fees, storage and inspection costs are incurred. The total cost of procurement can be extremely high in cases where special requirements are made or when the engineering content is particularly involved.

Installation

As this is usually priced into a civil or mechanical contract, these costs are not often attributed to the pump. Installation costs include civil works for foundations and plinths, isolation valves, gauges, and pipework interfaces. Electrical costs can amount to the same, again in cases where soft start and variable speed drives are used. Further costs are incurred through commissioning and inspection. Installation costs often exceed the cost of the pump, yet poor installation is frequently the cause of pump operating problems.

Operation

Energy costs are by far the largest proportion of the total cost of ownership for most pumps. Therefore the efficiency of a new pump has far more impact on the lifetime cost than does the capital cost of that pump. As energy costs continue to rise, so will the pressure to take these factors into account. Other operating costs include services such as lubrication systems, flush water for seals, barrier system gases, and monitoring.

Maintenance

During its lifetime, the pump will require regular maintenance in order to counteract the effects of wear and to reduce the risk of unexpected failures. Assuming good operating conditions, a pump can be expected to run for five years between overhauls and two years between seal replacements. An overhaul can cost up to 60 percent of the pump cost and seal replacement can cost from $300 to several thousands. Routine servicing such as oil changes, leak checking, and barrier system flushing should be performed on a regular basis. Periodic vibration monitoring and real-time analysis are needed in order to plan maintenance schedules and assess pump condition. Skilled staff and suitable facilities along with parts and materials are required for all these functions.

Many companies maintain a stock of spare parts in an effort to reduce downtime, but this introduces further costs. A large capital investment is needed to stock a warehouse, especially in plants with a wide variety of pump models and sizes. Maintaining and tracking stock levels incurs management costs and other factors such as security, insurance, and cost of warehousing must also be accounted for.

Mechanical Seal Arrangements, Failure Probability, and Power Consumption

Typical seal arrangements used in industry are single, double, and tandem seals. Extensions of these designs include dry running outboard tandem seals and double gas lubricated seals. Each of these designs has been used on various applications to extend MTBR. Basically, the more complex the design and the higher power losses at the seal faces, the shorter the system life. When considering the system life or reliability, consideration must also be given to the cost of operation through the life of the equipment. A comparison to API plan and seal size is given in Table 7. Here, the cost of power losses is given for two and four inch diameter seals operating in light hydrocarbon at 20 bar pressure and 3600 rpm.

Table 7. Estimated Cost of Operation Based on Seal Horsepower Losses Through a 20 Year Equipment Life.

API PLAN	SEAL ARRANGEMENT	SEAL SIZE INCHES	HORSEPOWER CONSUMPTION	FLUSH RATE GPM	KILOWATTS	POWER COST IN DOLLARS 1 YEAR	20 YEARS
11	SINGLE SEAL	2.000 4.000	1 3.50	1.14 4.57	0.7457 2.610	523 1831	10,464 36,624
11/52	LIQUID TANDEM SEAL	2.000 4.000	1.21 4.29	1.58 5.6	0.9023 3.2	633 2,244	12,660 44,890
54	LIQUID DOUBLE SEAL	2.000 4.000	1.44 4.87	1.88 6.36	1.073 3.63	753 2547	15,060 50,940
	DRY TANDEM SEAL	2.000 4.000	1.02 3.53	1.29 4.51	.7606 2.632	534 1847	10,680 36,940
	DOUBLE GAS SEAL	2.000 4.000	0.04 0.29	4SCFH 6 SCFH	0.03 0.216	21 152	420 3043

EST. COST

From Table 7, it can be seen that power losses at the seals can be significant for a given installation. A dramatic increase occurs with size, as evidenced in the comparison between seal sizes in the table. The cost of power can range from $0.05 to $0.10 per kilowatt hour. This table is based on power costs of $0.0877. The most efficient sealing arrangement is the double gas lubricated seal. Here there is a significant reduction in power losses, since the seal is operating on a gas film, eliminating contact.

Additionally, there are hydraulic power losses when the fluid pumped is used to cool the seal and is flushed from discharge through the seal chamber to suction. This value should be added to the power losses at the seal faces.

The point of this portion of the tutorial is to show that real cost is not simply the cost of the spare seal and labor to change the seal, but also includes the operating costs, which are significant and must be considered in the reliability analysis.

THE EMISSION IMPLICATIONS OF PRIME ENERGY CONSUMPTION

Emissions in relation to pumps and mechanical seals have been very topical over recent years and legislation has had a profound effect on seal design and operation. This extensive subject is beyond the scope of this document, but it is important to register the fact that, generally speaking, seals that give good lifetimes are seals with low emission characteristics. Additionally, process plant operators want simple seal arrangements and the fears of several years ago have mostly been allayed. The prospect of fitting double seals and/or sealless pumps everywhere has mostly been allayed as existing seals and simple arrangements with good materials, etc., have been shown to meet legislation satisfactorily.

The effect of prime energy consumption on our environment must be remembered if we are to be effective in our collective endeavors. It is important not to solve an emissions problem in the plant with a less efficient technology when a much worse emissions problem is created at the power station. Energy consumption costs money and creates and aggravates environmental emissions problems.

A METHODOLOGY FOR ACHIEVING IMPROVEMENTS IN PUMP COMPONENTS

A systematic and sustained approach is an essential prerequisite of achieving performance improvements in an existing plant. There are many alternative methods, but they all depend on the basic elements of establishing current performance, setting goals, establishing and implementing an improvement plan, and monitoring progress.

Outline of What Can Be Achieved, Setting Targets

Improved reliability of seals, couplings, and bearings leads to improved reliability of pumps, which, in turn, leads to reductions in pump operating costs; the objective of the exercise. However, although extremely important, it must be remembered that it is not the only way to achieve the goal. Other measures should be taken to reduce running cost, reduce inventory, etc., and these possibilities will be dealt with. On the question of improving reliability, however, the authors recommend following a logical sequence of events:

- Understand what is being and what can be achieved in relevant or equivalent situations
- Set appropriate outline targets and time-scales for your own plant
  - Determine what is being achieved in your plant
    - Recruit the help of your suppliers as appropriate
    - Conduct a survey
    - Analyze the data
  - Set up a performance improvement plan
    - Identify the bad actors
    - Agree the plan with revised targets and time-scales as necessary
  - Set up the monitoring and reporting system
  - Collect and analyze data regularly
  - Compare to target
  - Revise the plan
    - Identify today's bad actors
    - Agree the revised plan with revised targets and time-scales as necessary
  - Collect and analyze data
  - Compare to target
  - Etc.

Hard data relating to what is being achieved is not always easy to find for pumps and pump elements, although it is not too difficult to come up with a reasonable assessment that allows targets to be set.

Before getting into the detail of improving MTBF, it is worth broadening the perspective by looking at some of the cost saving statements and outline improvement plans that some users have come up with, so that a balanced approach to cost reduction can be formulated.

- International chemical company—In 1998, reduce costs by $9 million by increasing production and/or reducing operating costs.
- Major British steel company—"The 25 drive": Reduce the cost of steel by $40 (£25) per ton
- Canadian based international aluminum company—Increase overall efficiency by 10 percent
- An old, existing UK refinery—Increase pump mean time between repairs (MTBR) to 45 months. Current levels of achievement are shown in Table 8.

Table 8. Current Pump MTBF Achieved on an Old UK Refinery.

Pump Vendor	No of Pumps	MTBR (Months)
A	52	9
B	40	15
C	130	11
D	137	7
E	184	8
Total	**543**	**9**

A Middle East Refinery

The following bullet points summarize the brief given to the lead author's company for a site survey and improvement plan for seals. Objective: to provide recommendations on total management of seals from purchasing to maintenance to allow:

- Today's standards of performance to be achieved for reduced cost of ownership
  - Safety
  - Reliability
  - Parts rationalization
  - Inventory reduction
- To allow comparison with peers
- Increase the existing MTBF of all pumps by 25 percent over a two year period
- Identify the pumps with low seal performance and recommend upgrades to improve
- Minimize the number of installed seal sizes over two years
- Make seal rationalization proposals to reduce parts inventory
- Reduce maintenance costs of seal failures by 20 percent in two years
- On existing seals, review auxiliary services and make recommendations to improve reliability by:
  - Clean flush from external source
  - Recirculation/dead-ending
  - Cyclone separator
  - Steam/water quench
  - Control of steam flow and pressure

- Recommend standby or tandem seals with alarm devices for:
  - Light hydrocarbons below 0.55 SG
  - Toxic duties
  - Hazardous duties
- Recommend new seals for conversion of water service pumps

For the record, this refinery was already heavily involved in extensive upgrading from soft packing to mechanical seals and from gear coupling to metal membrane couplings. To that point (1993) improvements in seal life had been achieved as shown in Table 9.

Table 9. Improvements in Seal Life.

Year	1990	1991	1992	1993
MTBF (months)	48	53	55	60

PUMPS

At the end of the day, it is the MTBF of the pump that determines the number and frequency of costly maintenance interventions. The pump MTBF is, itself, very dependent on the lifetimes of its components as illustrated by the MTBF formulas for series systems described previously in Equation (28). The effect of this can be seen in the simulation below.

The figures are taken for a population of ANSI pumps from data published in 1980 following a survey, but where proper references were not available at the time of writing. The simulation (Figure 14) includes data shown in Table 10. Since that time, there has been a lot of progress on improving the lives of pump components and that will be reflected later in calculating suggested MTBF targets for pumps.

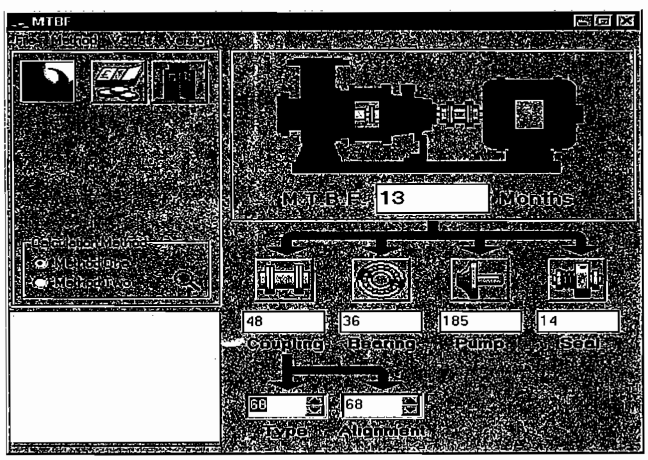

Figure 14. Pump MTBF Simulation.

At this point, it is suggested that a target of 48 months for pumps is typical of what can be achieved in a well run plant. In contrast, one refinery is achieving six months and is targeting two years.

General Targets

On a subjective level and based on market experience, the target lifetimes shown in Figure 14 and Table 11 are seen as reasonable.

SEALS

In 1976, a survey on seal life was conducted by BHRG over some 5000 pumps in a range of process industries. Average seal lives were only around 12 months, and the users were very dissatisfied and demanded improvements. The sealing industry responded.

Table 10. ANSI Pump Component Lives.

Component	MTBF (Years)
Mechanical seal	1.2
Ball bearing	3.0
Coupling	4.0
Shaft	15.4
Pump	**1.07**

Table 11. Suggested Target Pump MTBFs.

Pump Type	Target MTBF (Years)
Boiler feed pump	8
Cooling water pump	8
HPI (bottoms pump)	2
Reactor charge pump	3
Process pump	4
Chemical pump	2
Other	4

Many companies, particularly in the oil and gas industries, started working very closely with the seal vendors in setting up seal MTBF improvement plans. The lead author's company took a very proactive part in that and cooperated in over 30 major refinery surveys, the results from 21 of which are tabulated in Figure 15. These data have been combined with equivalent data from the user, which has allowed the overall seal MTBF chart to be produced. (Adjustments had to be made to combine the data since the vendor MTBF was based on the number of pumps and the user data, on the number of seals. The final data are based on the number of seals, which involved multiplying the vendor MTBFs by 1.15.)

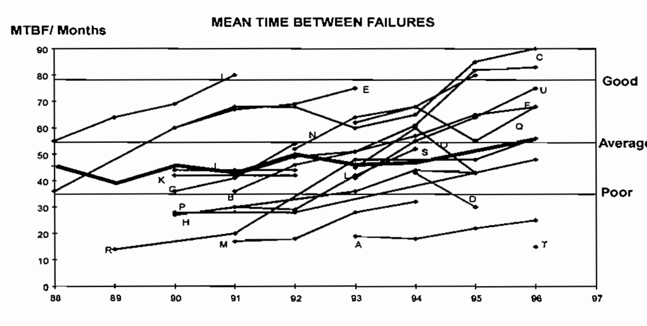

Figure 15. Results of 21 Major Refinery Surveys.

Combined MTBF Values Based on Over 12,000 Seals in 36 Refineries

The chart in Figure 16 includes upper and lower quartile values that are useful for suggesting target performance values (Table 12).

The average MTBF improvement rate in Figure 16 is 30 months in six years, or five months per year. That should be regarded as the base rate that is the average of a large number of plants, not all of whom were on improvement programs. Furthermore, one should expect large percentage improvements initially as the worst "bad actors" are dealt with.

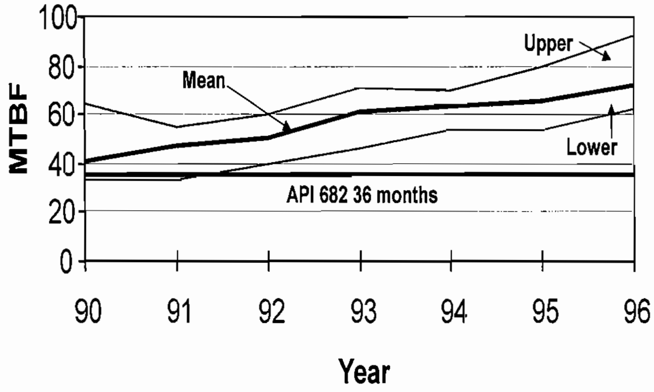

Figure 16. Summary MTBF Data for 36 Refineries.

Table 12. Suggested Seal Target MTBFs.

Target seal MTBF	
Excellent	> 90 months
Very good	70/90 months
Average	70 months
Fair	62/70 months
Poor	< 62 months

Not all plants are refineries, however, and different results can be expected elsewhere. In chemical plants, pumps have traditionally been "throw away" items as chemical attack limited life. Things have improved in recent years, but the limited space available in DIN and ASME stuffing boxes does limit the type of seal that can be fitted to more compact and simple versions. Lifetimes in chemical installations are generally believed to be around 50 to 60 percent of the refinery values.

OTHER PERFORMANCE INDICATORS

For over a decade, MTBF has been the primary means of assessing mechanical seal reliability, mainly because other data are hard to find. Measuring MTBF on one site in the same way offers an excellent guide to performance and allows trends and changes to be monitored. To compare different sites, however, requires great care. With that in mind, an additional approach was proposed by a UK refinery engineer to provide a simple "yardstick" with which comparisons could be made and which looked at money; cost reduction being the primary objective. That "yardstick" was "cost per seal installed" (CPSI).

"Cost per seal installed" can provide an alternative view that permits an overall appreciation of what is happening on one site and allows comparison with another site. How accurate are the base data?

- The number of mechanical seals installed? Almost all sites have this logged to an accuracy of 95 percent or greater.

- Annual spend on mechanical seals? Virtually all purchasing departments will have this to an accuracy greater than 99 percent.

If a very simple equation is used:

$$\frac{\text{All mechanical seal spend } \$250{,}000}{\text{All seals on site} \qquad 500} = \$500 \text{ CPSI} \tag{37}$$

This very simple equation uses readily available data and yields the actual cost of using any manufacturer's mechanical seal. (Based on the purchased cost of parts and services from the vendor.)

Example 1

A survey of 10 refineries CPSI was carried out using only one vendor's mechanical seal data. It showed extremely interesting results and certainly put a cost perspective on the traditional MTBF form of measurement. The fact that all seals in this example are from one vendor makes the comparison between refineries particularly accurate (Table 13).

Table 13. CPSI for 10 Refineries with the Same Seal Vendor.

Refinery	Number of Seals	Annual Spend	Cost/Seal Installed
Ref 1	550	$696,390	$1266
Ref 2	380	183,700	483
Ref 3	660	584,500	886
Ref 4	250	208,750	835
Ref 5	180	116,900	649
Ref 6	250	183,700	735
Ref 7	90	43,420	482
Ref 8	250	197,060	788
Ref 9	280	116,900	418
Ref 10	500	283,900	568
Totals	**3390**	**$2,615,220**	**$771**

At the highest end of the scale, Refinery 1 has a CPSI of $1266 per annum, which, in this particular case, equated to an MTBF of less than 30 months. Refinery 9, with the best performance, has a CPSI of $418 per annum and an MTBF of 60 months. Even allowing for the 100 percent improvement in MTBF *(which would reduce CPSI to $633)*, there is a further 26 percent improvement in cost management. In this case, those savings had been brought about by using the most suitable, and therefore most cost effective, sealing solution for each application, but closely linked to maximum rationalization.

Example 2

Finally, it is interesting to look at the combined results of two sites (same operator) with three different seal vendors (Table 14).

Table 14. CPSI for Three Different Vendors.

Vendor	CPSI
A	$ 264
B	1013
C	897

Clearly, it is interesting to compare the results from different vendors, and CPSI can give a meaningful insight when the different vendor's seals are used in similar services. In the case shown in Table 14, the figures are a little misleading, as the vendor with the worst CPSI has all the hot hydrocarbon services (high temperature bellows seals). CPSI is a useful and meaningful indicator but, as with MTBF, care must be taken.

COUPLINGS

It would not be fair to say that gear couplings have failed every time they need lubricating, even if the pump has to be stopped outside a normal shut down opportunity. Very often the pump has to be stopped specifically for that purpose. On a refinery in the Middle East, which underwent a major conversion from soft packing to mechanical seals and gear couplings to membrane couplings, it had been the practice to clean, inspect, and regrease gear couplings every four to six months. The couplings were replaced every five years or so, and could be said to have an MTBF of around five years.

The cost of coupling ownership must however be calculated properly by including both maintenance and replacement costs in any calculation to show real-life cost benefit. As far as this aspect of reliability is concerned, flexible element couplings have two very significant benefits:

- They require no lubrication.
- They have a very long theoretical life.

Additional benefits can include low imposed loads on the shafts that extend bearing life and no wearing parts, which means retention of dynamic balance and low vibrations for extended mechanical seal and bearing life.

Note: The BHRG survey in 1976 identified vibration as the number one cause of reduced seal life and increased leakage levels.

The main enemy of membrane couplings is excessive misalignment, which can result in fatigue failures. However, some flexing element couplings (metal membrane) are able to take typically 0.5 degrees per membrane bank (one millimeter of lateral misalignment per 100 mm of shaft separation), which is many times greater than the level couplings can be easily aligned to. The modified Goodman diagram, shown in Figure 17, illustrates the margins normally applied to metal membrane couplings, which means they can expect to run maintenance free for many years.

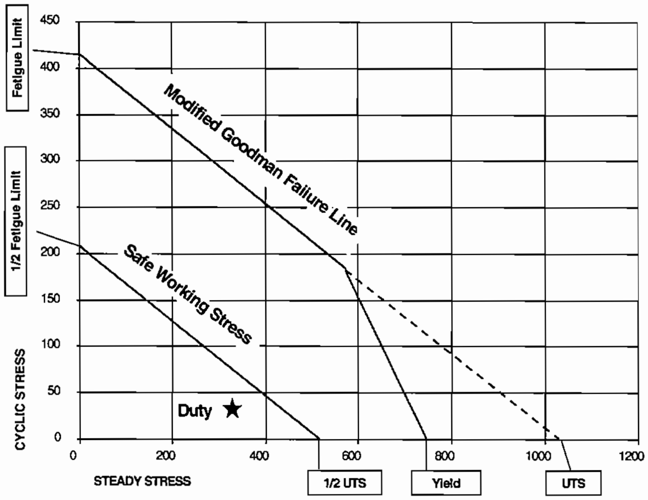

Figure 17. Modified Goodman Diagram for AISI 301 Half Hard Membranes.

In short, metal membrane couplings can reasonably expect to last the life of the plant, unless they are overloaded or badly aligned on installation or due to some change such as foundation movement. Clearly, however, a bearing failure can result in severe coupling damage.

BEARINGS

A rolling element bearing has a finite life and eventually will fail due to fatigue, even if operated under ideal conditions. The operational life of a rolling element bearing, limited by fatigue, is a statistically calculable parameter. The required service life of a machine can therefore be matched to the bearings. Provided that a large enough bearing can be fitted and that satisfactory operational conditions are maintained, service lives of many years are the norm.

The fatigue life is governed by the operating speed and radial and axial loads applied to the bearings. The generally accepted method of defining it is the L_{10} life. The "L_{10} life" is the expected number of cycles, or hours ("L_{10h}"), without evidence of fatigue that 90 percent of a group of apparently identical bearings will achieve when operating under the same conditions of load and speed. The basic calculation for this is described in ISO 281 (1990). Basic rating life in millions of revs:

$$L_{10} = \left(\frac{C}{P}\right)^p \text{ or } L_{10h} = \left(\frac{1{,}000{,}000}{60n}\right) L_{10} \text{ hours} \tag{38}$$

for ideal conditions, where:

C is the basic dynamic load rating depending on the bearing design
P is the equivalent combined dynamic bearing load
p the exponent = 3 for ball bearings and = 10/3 for roller bearings
n is the speed in rpm

Unfortunately, most industrial installations are not under ideal conditions, and bearings often fail well before reaching their theoretical design life. To ensure reasonable, practical operating periods rotating machinery standards based on operating experience require L_{10h} bearing life to be typically as follows:

- 25,000 hours for spared machines in general operation
- 40,000 hours for unspared machines with two years or more continuous operation
- 100,000 hours for high reliability, remote, and unattended continuous operation

API 610 (1995) specifies three years design life for bearings:

- 25,000 hours continuous operation
- 16,000 hours at maximum axial and radial load

Published statistics indicate that two-thirds of all rolling element bearing failures that are replaced in service fail prematurely. One-third of bearings fail due to fatigue spalling. These tend to be the longest running bearings where operational conditions have been consistently satisfactory. One-sixth of bearings fail due to incorrect fitting, incorrect selection, damage due to external causes such as vibration or electric currents, or overloading. One-third of bearings fail early due to lubrication problems, usually because of an insufficient quantity of grease or loss of oil from the bearing. One-sixth fail due to contamination entering the bearing. Solid contaminants cause surface damage by indenting, wear, and early fatigue by bridging the oil film. Moisture can cause corrosion and can reduce the effective viscosity of the lubricant, again causing wear and reducing the fatigue life.

The effect of water contamination on basic bearing life can be seen in Figure 18. As can be seen from Figure 18, if the water content of the oil increases through contamination from the nominal (ideal?) 100 ppm to 400 ppm, the bearing life will be more than halved. Similarly, solid contaminants will reduce the life by varying amounts depending on the amount, distribution in the oil film, and particle size relative to the oil film thickness.

Some bearing manufacturers have refined the basic bearing life calculation to include factors to account in some measure for the degree of contamination. However, these factors show what improvement in rating life can be achieved with known, controlled levels of contamination. They still depend upon the *bearing being protected from the environment by effective sealing.*

Overloading of the bearings is prevented by selecting a suitable type of large enough size, but bearing size is limited by the operating speed and lubrication as well as by space considerations! So it becomes necessary to control and limit the loads and

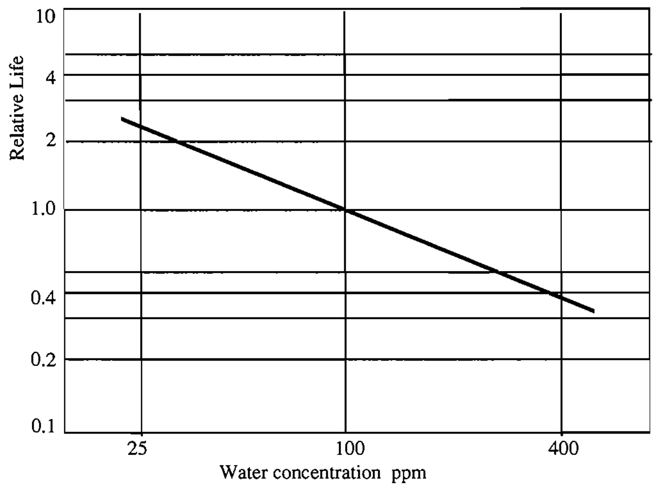

Figure 18. Effect of Water Contamination on Basic Bearing Life.

moments that may be imposed upon the bearing. In addition to the basic rotating weight distribution of the machines, significant forces can be applied to bearings by misalignments between coupled machines. *Careful consideration of the type, weight, and especially flexibility of the coupling selected can minimize these imposed loads and moments.*

SUMMARY SUGGESTED LIFETIME TARGETS (MTBF)

It is important to have distinct targets for MTBF improvement. The following suggested values are based on the figures presented previously in this section.

- Seals (Table 15)

Table 15. Goals for Seal MTBF Improvement.

MTBF	Refineries	Chemical & Other
Excellent	90 months	55 months
Average	70 months	45 months

- Cost per seal installed—Excellent: < \$500 per annum; Average: = \$770 per annum
- Couplings—10 years minimum with membrane couplings; > five years with gear couplings
- Bearings—Continuous operation: 60 months; Spared operation: 120 months
- Pumps—48 months

Note: Assuming a 90 month seal life, 120 month coupling, bearing, and pump part life, pump life by Rule 2 is 50 months. So to achieve 48 months, all components need to be working at their best.

MONITORING PROGRESS

Monitoring progress is of absolute importance to measure what is going on so that improvements can be proven. The importance of having a good system for data logging, information storing, performance measuring, and reporting cannot be overestimated.

IMPROVEMENTS IN PUMPING EQUIPMENT ACHIEVED

Historical Data

In previous papers, certain applications in refineries have been identified as the most troublesome following detailed surveys of refineries worldwide. Those applications include refinery "hot duties," light hydrocarbons and crude over 150°C.

Experiences Worldwide

A major oil company has implemented mechanical seal performance improvement plans in their installations worldwide over the last few years. The substantial data collected serves to illustrate the levels of reliability and the rate of improvement that are currently being achieved.

The Challenge of Refinery Hot Duties

In the early 90s, the oil company started a review of mechanical seal reliability across its refineries worldwide. This was part of an overall reliability improvement initiative for mechanical equipment, and for centrifugal pumps, mechanical seal reliability was seen as the single most important factor. Also at that time, two other issues were focusing attention on mechanical seals in the industry. These were the introduction of the Environmental Protection Agency (EPA) emissions legislation in the USA (1990 Clean Air Act, 1990) and the development of API Standard 682 (1994) for mechanical seals. In particular, the requirement of API Standard 682 (1994) for "three years uninterrupted service, while complying with emissions legislation" was seen as a new benchmark for the industry, and very relevant to the reliability review. Further, because the majority of pumps in critical service have always been spared, it was felt that seal reliability had been given insufficient attention in the past.

While seal reliability improvement programs had been in place for a number of years, it was clear that there were significant differences in perceived reliability across the refineries. However, there was no standardized method for measurement nor was there general agreement on seal life expectancy, both of which needed to be addressed. Also in view of the large numbers of pumps in similar services, albeit in different locations and with seals from different manufacturers, it was felt that overall experience was not being adequately captured and used to its best effect.

Evaluation of Seal Performance

For evaluation of seal performance, a collective site index was considered to be the most appropriate, i.e.. the number of seal failures across the whole site. A very clear requirement from all concerned was that whatever system was chosen, it should be relatively straightforward to introduce and easy to maintain, primarily in terms of data collection. The index that was chosen was:

$$\text{Mean time between failure} = \frac{\text{Total number of seals}}{\text{Total number of failures per month}} = \text{MTBF in months} \qquad (39)$$

based on a moving average of 12 months.

By using this simplistic approach, it is clear that the MTBF generated will not be the "real" seal running life, as there will be variations as to the percentage of the total number of pumps in service at any one time. However, to introduce service factors for each refinery to compensate for this was considered an unnecessary complication, and in view of the relatively large sample size, the approach adopted was considered to be valid for the review program.

Total Number of Seals

The number of seals used in the calculation is the total number installed in equipment that is available for plant operations, regardless of service or service factor. Seals in tank farms and oil movement areas count equally to seals in process units. During refinery or unit turnarounds, the seals on the unit that is shut down should continue to be counted.

Pumps with two shaft penetrations and two seals are counted as two. Double and tandem seals count as one seal, as do single seals with dry-running backup seals.

Definition of a Seal Failure

Bearing in mind the requirement for the review program to be simple and straightforward, a seal is deemed to have failed if it needs to be replaced for whatever reason. Any attempt to categorize seal failures against established criteria was considered to be unworkable.

Seals that are replaced as part of a major overhaul of a pump but had not failed, are not counted. Similarly, seals that fail as a result of failure of another component, e.g., bearings, are not counted. Failures that occur during recommissioning after seal replacement are counted.

Data Collection

The total number of seals on each site was established at the start of the program. Each refinery provides on a monthly basis:

- The number of seal failures, and
- The number of additional seals installed in that month.

The MTBF is then established on a 12 month rolling average basis to remove monthly fluctuations resulting from small numbers of seal failures per month. In addition, it is sufficiently short to allow the effect of improvements to show up without unnecessary delay.

Program Results

Figure 19 shows MTBF for the individual refineries plotted on a monthly basis, plus the refinery-wide average. In terms of numbers of seals per site, this ranges from 216 to 1206 seals. Since the program began, there has been a marked improvement in MTBF across all sites, and it can be seen that the site-wide average has increased significantly, from approximately three years in 1991 to more than five years at the end of 1997. At the end of 1998, this had increased to six years. During 1998, additional monitoring programs were introduced for services below 210°C (410°F) and above 210°C (410°F) and Figures 20 and 21 show the data for 1998. While the data is for one year only, it is clear that there is scope for improvement in the reliability of hot service seals.

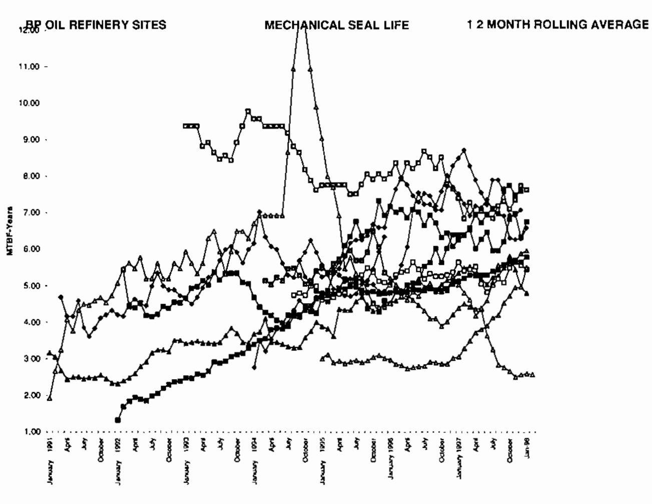

Figure 19. Oil Refinery Sites Mechanical Seal Life 12 Month Rolling Average.

The improvement in seal reliability has been achieved entirely by the various seal reliability improvement programs at each site. The contribution of the MTBF monitoring program has been to provide an immediate reference of relative positioning and to indicate where action on seal reliability is required.

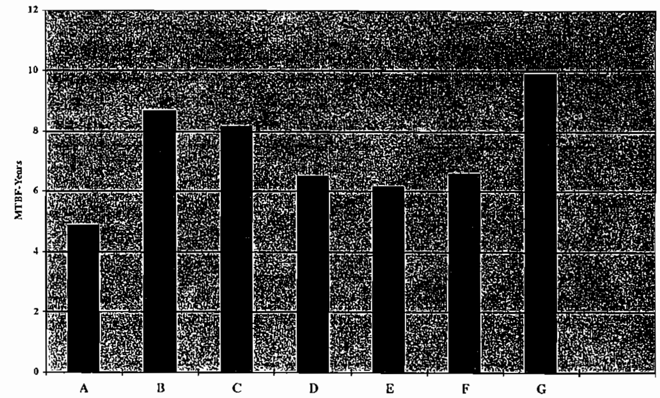

Figure 20. 1998 MTBF for Seals on Services <210°C.

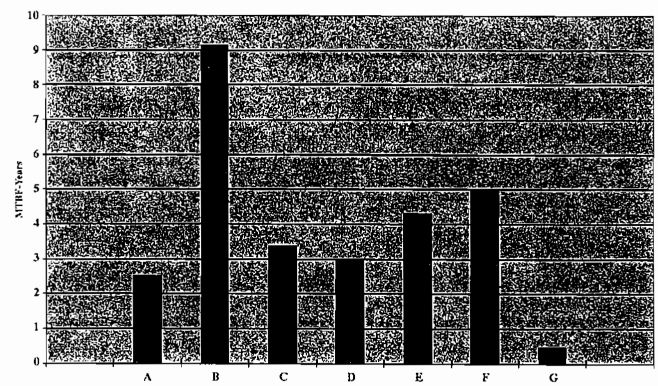

Figure 21. 1998 MTBF for Seals on Services >210°C.

The improvement in reliability has been achieved for several reasons. Probably the single most important factor has been the attention to the environment in the sealing chamber. For those seals with poor reliability, the application is reviewed in detail, in particular to establish the viability of maintaining a stable fluid film across the seal faces. In many cases, primary and secondary seal component materials have been changed, which is in line with the general trend to upgrade to "premium" materials. From a mechanical point of view, attention to accuracy of pump build/assembly has contributed significantly to the improvement in seal reliability, as has the move to cartridge design for all seals.

Close cooperation with the individual seal manufacturers throughout has been an essential part of the reliability improvement program. The involvement of and cooperation with the manufacturers varies from site to site, and seal alliances are in place at many of the refineries.

THE FUTURE

The program is continuing into 1999 and beyond, with the new, enlarged oil company discussed in the previous section, which has in excess of 13,000 mechanical seals. In terms of expected MTBF in the future, this appears to be approaching six years, which by international comparison, ranks very high. Taking service factors into account, the API 682 (1994) expected reliability of three years, is therefore, in general, being achieved.

New technology is expected to contribute to the anticipated improvement in seal reliability in the future as is cooperation between seal suppliers, operators, and pump manufacturers. Noncontacting gas seals are an example of new technology, which is yielding very promising results. Similarly, many of the alliances between operators and the seal manufacturers are contributing significantly to the improved seal reliability that has been achieved.

CASE HISTORIES—GENERAL RELIABILITY IMPROVEMENTS IN SEALS IN CRITICAL SERVICE

HF Alkylation Service

A survey of the refinery industry indicates that users will spend money per repair and achieve greater MTBR when compared with other industries. The reason, plant maintenance engineers and equipment suppliers are taking the time to identify and correct areas for improvement. The cost to repair a pump in an alkylation unit can range from $6000 to $7000 per event. In some cases, seals have lasted less than one year. Over a 20 year life of the pump, maintenance costs can reach $140,000 and higher. This does not include the cost of any process downtime.

The purpose of the HF alkylation process is to increase the octane rating for a fuel. This requires hydrofluoric acid, which is a strong toxic and corrosive fluid, to be used to obtain the desired chemical reaction. To successfully seal this fluid, special consideration is given to the seal design, arrangement, and materials of construction.

The most common type of seal used in an HF alkylation unit has been a single seal with an external flush to API Piping Plan 32. The external flush, normally isobutane, enters the seal chamber at a pressure greater than the acid being pumped. This keeps acid from entering the seal chamber. A bushing at the bottom of the seal chamber is used to limit flow into the seal chamber when a loss of seal flush occurs. In the event of primary seal leakage, a close clearance floating bushing in the seal gland limits emissions to atmosphere. A single seal has provided good results over the years.

In addition to the safety aspect of this dangerous application, there are two important industrial standards that have challenged engineers to provide solutions to increase equipment reliability while protecting the environment. These are:

- 1990 Clean Air Act (1990)
- American Petroleum Institute Standard API 682, "Shaft Sealing Systems for Centrifugal and Rotary Pumps" (1994)

The Clean Air Act (1990) is a regulatory standard that defines the emission limits of rotating equipment to the environment and is enforced by the Environmental Protection Agency in the United States. API 682 (1994) is an industrial standard developed by users with input from equipment manufacturers. The goal of the standard is to create a specification for seals that would have a good probability of meeting the emission regulations defined by the Clean Air Act and have a life of at least three years. In practice, these standards have presented industry with solutions to difficult problems in the field of sealing technology. This has resulted in a major increase in equipment reliability and significant savings to the user.

The use of dual seals at three very well run refineries is presented. These installations have resulted in an increase in mean time between maintenance (MTBM) from just under one year to over five years. These specific installations are:

- API 682 dual pressurized seals
- API 682 single seal with a noncontacting gas lubricated tandem seal
- Double seals with a pressurized barrier liquid

API 682 Dual Pressurized Seals

Prior to the installation of API 682 (1994) dual pressurized seals, a North American refinery was experiencing an MTBM for the alkylation unit of less than one year. Vibration inherent with the equipment was also excessive. Aggressive goals were set that required a complete upgrade to current API 682 specifications. This included changes to the seal chamber, bearings, and the installation of a dual API 682 seal, shown in Figure 22. The inboard seal is designed to handle pressure at both the outside or inside diameter of the seal faces. A very reliable isobutane stream is flushed through the outboard seal cavity at a pressure slightly greater than the pressure on the inboard seal. The flowrate is kept at 3 to 4 gpm. Isobutane from the outboard cavity is injected into the inboard seal cavity. The seal faces are kept exceptionally cool and there has been a significant improvement in controlling vibration. The result is seal life that is approaching six years of service.

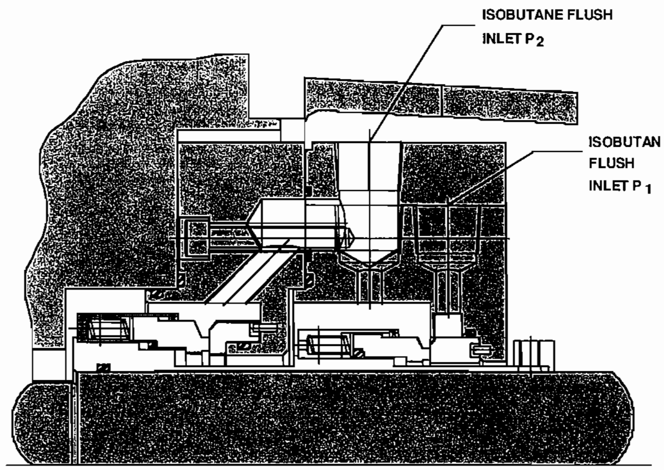

Figure 22. Installation of a Dual API 682 Seal.

API Single Seal with a Noncontacting Gas Lubricated Tandem Seal

This installation, also in a North American refinery, requires an API 682 single seal installed with Piping Plan 32. Isobutane is flushed over the inboard seal. The difference in this design is that the area at the inboard seal is blanketed with nitrogen gas. The outboard seal is a noncontacting gas seal. This seal is designed to handle pressure at the outside and inside diameters of the seal faces. The seal basically pumps a small quantity of dry nitrogen gas to the outside diameter of the seal face. To control the loss of dry nitrogen to atmosphere, a floating bushing is used. The space between the seals is vented to an acid relief header. This design is illustrated in Figure 23. In the event of a primary seal failure, the outboard seal will perform as a contacting seal, preventing leakage to atmosphere. Results with this installation have been a significant increase in mean time between maintenance.

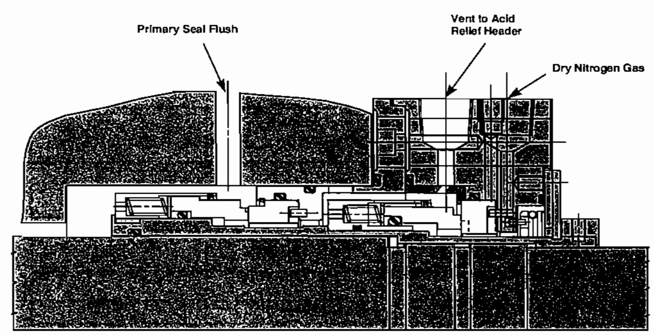

Figure 23. API Single Seal with a Noncontacting Gas Lubricated Tandem Seal.

Double Seals with a Pressurized Barrier Liquid

Double seals have been chosen as a standard by several refineries. This design is illustrated in Figure 24. This seal arrangement is capable of handling reverse pressure at the inboard seal in the event of the loss of barrier pressure. An alkylate barrier fluid is circulated through the seal chamber at a flowrate to remove the developed heat. The pressure is slightly higher than the pressure against the inboard seal. An isobutane flush is used at the inboard side to prevent acid from coming into contact with the inboard seal. A bushing in the gland plate is used to restrict emissions to atmosphere. The space between the outboard seal and bushing is vented to a vapor disposal area. This type of seal has been in service in one European refinery since 1991. With the exception of two pumps that were assembled wrong, there has never been a failure of a double seal. A total of 15 pumps are in service. However, this refinery goes through a shutdown every three years, at which time double seals are overhauled.

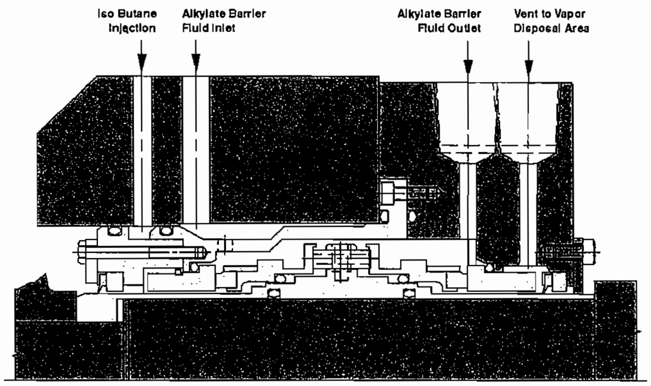

Figure 24. Double Seals with a Pressurized Barrier Liquid.

The keys to success in sealing HF alkylation units are:

- Control and remove the developed frictional heat.
- Eliminate equipment vibration.
- Use of proven materials of construction.
- Use of proven seal designs.

THE RELIABILITY AND POWER SAVING BENEFITS OF GAS SEALS AND STANDBY SEALS

The usefulness of gas lubricated sealing technology in increasing MTBR is becoming more apparent with the passage of time. More and more diverse applications are being successfully sealed with noncontacting seals. These applications range from cryogenic services to higher temperature light hydrocarbons. This type of seal design can improve MTBR from months to years. Table 16 is a sample of possible improvements in MTBR.

Table 16. Sample of Possible Improvements in MTBR.

Fluid	Original Life	Current Life
Light hydrocarbon	2 months	11 years
Cryogenic fluids	4-6 weeks	5 years
Hydrocarbon	2 months	6 years

All these installations are still in operation. Current life has resulted in substantial reduction in maintenance costs. As indicated previously, power savings over contacting seals are considerable. This reduction in power losses at the seal faces accounts for the long seal life.

CONCLUSIONS

Substantial progress has been made in increasing equipment reliability to improve plant operations. Every issue that can influence equipment life must be identified and the proper solution applied. As discussed in this tutorial, by adopting a vigorous program in monitoring seal performance, this serves to focus on those areas where major savings can be achieved.

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