CORROSION IN PUMPS AND OTHER DAMAGE MECHANISMS

by

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ABSTRACT

The successful application of pumps in handling a variety of fluids depends upon knowledgeable design considerations, and the selection of the correct materials for the liquid environment. A previous tutorial "Corrosion in Pumps" addressed many of the considerations that must be taken into account to prevent accelerated degradation resulting from corrosion. Other mechanisms of damage that can limit the useful life of these machines are examined. Mechanical damage mechanisms can render a pump useless if proper design and material selections are not done upfront. As is the case with most pumps, a corrosive environment will influence the useful life of the material in the pump. The engineering challenge is to design and build a reliable pump where both corrosion and mechanical damage mechanisms are attempting to render the pump useless.

INTRODUCTION

Today, the dependability of pumps is taken for granted. Until one takes a closer look at the many complexities encountered in moving a fluid at high pressures and temperatures, it is easy to view these dependable machines superficially, while failing to appreciate the design and material developments that went into engineering and building the pumps. Pump producers have had many unfortunate experiences where an overlooked detail or forgotten variable omitted from specifications has turned a routine application into a disaster. The more information concerning an application that is available during the inquiry portion of the order, the higher the potential for a successful application. In reality, pumps may experience premature damage due to corrosion, erosion, and/or wear mechanisms that could have been avoided through proper identification of all the fluid environment variables. If sufficient detail of the environment is provided upfront, a proper materials selection process will choose a material that will give sufficient corrosion and/or erosion resistance for the expected life of the application. Mechanisms of material degradation are well enough understood by both the pump producers and users so that few inappropriate applications should be encountered.

For pumps to perform their task, precisely dimensioned parts must move at high relative speeds in close proximity to one another. Damage may occur with dramatic results if the parts make contact under load. Accelerated material loss or catastrophic seizure of these components will result in costly repair or replacement. In addition to corrosion damage frequently encountered in pumps, mechanical wear can render a pump useless in a short period of time. Knowledgeable selection of materials to increase the likelihood of surviving component contact is important to the successful operation of these machines. High temperature applications pose an additional challenge with respect to ring clearance design. If the clearance is too small at room temperature, component contact can occur as a result of thermal transients encountered during startup. Thermally induced growth of seal components can close the already tight clearances, forcing these parts to rub one another with excessive force. Therefore, one aspect of survivability depends on the adhesive wear characteristic of the two contacting materials. The successful use of materials in adhesive wear environments is based upon both past field experiences and extensive testing. These experiences have lead to adhesive wear "rules of thumb" that are generally accepted industry wide.

Wear is a complex phenomenon. Wear or loss of material of pump components can result from more than one mechanism simultaneously. Wear mechanisms have been categorized into more than 20 individual processes [1]. The introduction of solids into a fluid stream increases the challenge for a materials engineer in making appropriate selections of materials. Solids entrained in the fluid stream can prematurely degrade a material by abrasion or erosion. The various wear mechanisms that are reviewed include: adhesive wear, abrasive wear, erosion, and fretting.

Since pump components are subject to alternating loads, the subject of fatigue cracking is also addressed. Fatigue cracking causes and remedies are well defined. If pump component damage is the result of this mechanism, appropriate steps can be taken to avoid catastrophic failures.

ADHESIVE WEAR

Centrifugal pumps often utilize a set of wear rings for the purpose of sealing interstage pressure. A rotating impeller ring runs at high speeds with a small clearance between it and a stationary casing ring. This includes a stationary casing ring running against a rotating impeller ring. Because of the importance of these rings to efficient operation, a search for improved material performance is a goal of all pump producers. By studying and understanding the mechanisms which produce wear ring material loss and at the extreme catastrophic seizure, increased reliability of centrifugal pumps can be achieved. One of the primary causes of material loss on wear components in a centrifugal pump handling clear liquids is adhesive wear. This is material loss due to metal to metal contact. Components that are in relative motion with the possibility of contact in other types of pumps are also prone to this form of material degradation. Therefore, knowledgeable selection of material combinations based on experience and testing is vital for serviceability of these components. Two important characteristics of a pair of materials that may come into contact are their adhesive wear traits and galling threshold. Data from wear tests can provide the necessary information, so that a high degree of confidence can be attained by the pump manufacturer in utilizing a particular wear combination.

Adhesive wear can be defined as the progressive loss of material resulting from local mechanical failure of highly stressed interfacial asperities. When two surfaces are brought together, the actual area of contact consists of the plastically deformed surface asperities and is much less than the apparent total area of contact. This is illustrated in Figure 1. This results in very high stresses at these junctions. In addition, at these contact points, the molecules on opposite sides are so close together that they exert strong intermolecular forces on each other. This phenomenon is known as surface adhesion. In many cases, when the stresses are large enough, the contact points become cold welded. This is what occasionally occurs when an austenitic stainless steel impeller is shrunk on an austenitic steel shaft. The making and breaking of intermolecular bonds and cold welds between two surfaces in relative motion, produces a net resistance to motion. This resistance force contributes to the total force known as friction.

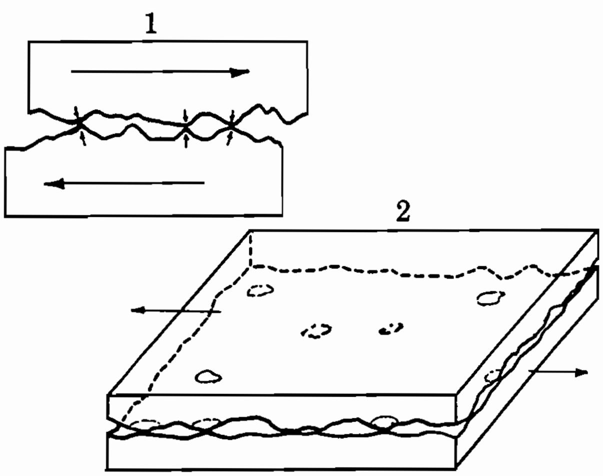

Figure 1. Contact between Two Surfaces is Along Asperities (1) Making the Actual Area of Contact Very Small (2).

Adhesive wear occurs when a strong junction and weak subsurface region of the sliding material combine to generate a wear particle (Figure 2). This adhesive wear model implies that the magnitude of adhesive wear is related to the probability of producing a wear particle, instead of breaking the intermolecular bonds at the interface of the two surfaces.

The earliest adhesive wear theory, proposed by Holm [2], suggested that wear took the form of single atom transfer between surfaces in contact. About the same time, a similar adhesive wear theory was presented by Archard [3], which assumed transfer of material to be comparable in size to the junctions formed by the asperities in contact. Radioactive tracer experiments later confirmed the wear particle size to be in the assumed range proposed by Archard's model. However, both theories developed an almost identical general wear equation. Regardless of assumed particle size, the variables contributing to wear are load, distance and material hardness. The significance of Archard assuming a hemispherical wear particle is a constant shape factor of three found in

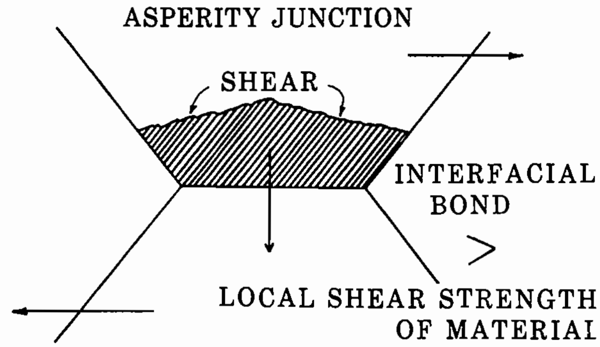

Figure 2. Illustration Showing the Formation of an Adhesive Wear Particle Due to a Strong Interfacial Bond Between Asperities.

the denominator of the wear equation. To eliminate the confusion surrounding an assumed particle size and shape, Rabinowicz [4] used Holm's equation for analysis of adhesive wear.

The general adhesive wear equation is therefore:

$$W = \frac{KFD}{H} \quad \text{(Holm's Equation) [4]} \qquad (1)$$

Where: W = Volume of Material Worn Away
K = Wear Factor
F = Applied Load
D = Distance Moved
H = DPH Hardness of Material Worn (Diamond Pyramid Hardness)

Because velocity and time are two variables controlled in the test, Equation (1) can be written:

$$W = \frac{KFVT}{H} \quad [4] \qquad (2)$$

Where: V = Velocity
T = Time

Rearrangement of the wear equation yields:

$$W = \frac{WH}{FVT} \quad [4] \qquad (3)$$

All of the variables on the right side of Equation (3) are either a controlled test parameter or property of the material being tested. "K" is, therefore, an empirical value based on tests conducted in the laboratory. Rabinowicz, who is a proponent of Archard's adhesive wear model, suggests that the wear factor, K, is a number relating to the probability that an asperity junction will form a wear particle. Therefore, dimensionless Ks can be used to compare various material's adhesive wear characteristics and ultimately used for relative ranking based on this number. The material combination that produces the smallest numerical value of K is expected to exhibit the best resistance to adhesive wear in the regime that the wear tests are conducted.

After the wear factor has been established, two determinations result. One is the elimination of many tests by varying the test parameters F, V, T, along with as the test material's hardness to determine the adhesive wear characteristics of a given material couple. As long as the wear regime remains the same, K does not change. Secondly, and of engineering importance, is the ability to predict wear rate (W/T) from the wear equation when operational parameters are known. Because the test parameters were selected to approximate the wear environment of a pump, the use of the determined wear factors applies.

Wear of two surfaces in relative motion is very complex. Therefore, in addition to the adhesive wear model described above, some alternate theories of sliding wear have been proposed. They are: surface delamination theory, a fatigue model, oxidation theory, and combinations of several theories mentioned. However, only the adhesive wear theory offers a general wear equation to quantitatively predict wear, and it provides a means to rank materials with respect to their wear characteristics.

Application of the adhesive wear theory requires data usually generated by properly controlled wear tests. These tests are usually performed in a controlled environment designed to generate the desired damage at an accelerated rate. The ability to produce a meaningful accelerated adhesive wear test is a controversial subject of depending upon point of view and prior experiences. The trick is to accelerate the damage process without changing the wear regime that is normally expected in a application. An adhesive wear testing procedure was developed at Ingersoll-Rand through a series of test runs using two resulfurized stainless steels, AISI-416, commonly used as wear rings in centrifugal pumps. This material combination was tested by varying the speed, applied load, temperature of test fluid, and test duration until a wear surface that duplicated returned field wear parts was achieved. This surface appearance qualification indicated that the test parameters used duplicate the wear regime occurring in pumps under normal wear of components. The ability of this wear test procedure to achieve an identical wear scar on wear test specimens and field returned parts is illustrated in Figure 3.

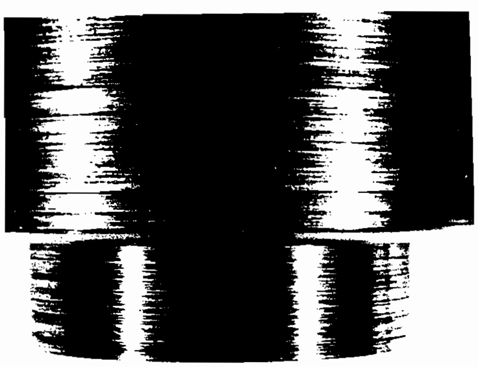

Figure 3. Laboratory Produced Adhesive Wear Test Surface as Compared to a Field Returned Balancing Drum of the Same Material.

Using these parameters for testing all material combinations available currently and new materials as they are developed, a comprehensive data base can be generated to compare their adhesive wear characteristics. Test results for these tests along with information gathered from the literature has assisted in the selection of candidate materials for pump applications. Some adhesive wear test results are shown in Table 1. This ranking of material combinations is intended to provide a means to compare all other material combinations tested under the same controlled laboratory test parameters. After selecting a candidate material pair, which, through testing, appears to be a good running combination, it is run in an actual pump for final assessment. Therefore, the laboratory adhesive wear testing is used as a screening process for potential applications.

Table 1. Calculated Wear Factor in Distilled Water Specific Gravity 1.0

Material (Ring/Block)	*Hardness DPH* (Ring/Block)	*Wear Factor* K
Leaded Bronze/ASTM A48 Class 30 CI	80/205	$0.17X10^{-4}$
Ni-Resist/Ni-Resist	120/120	$0.41X10^{-4}$
Nitronic 50/Nitronic 60	195/190	$0.76X10^{-4}$
90-10 CuNi/ASTM A48 Class 30	87/210	$1.14X10^{-4}$
Stellite 12/Stellite 6	440/395	$1.71X10^{-4}$
Ampco 18/Ampco 18	155/155	$2.40X10^{-4}$
AISI 410/ASTM A743-CZ6NM	300/270	$2.45X10^{-4}$
AISI 410/AISI 416	290/430	$2.97X10^{-4}$
AISI 416/AISI 416	430/360	$3.57X10^{-4}$
In Alcohol Specific Gravity 0.87		
Nitronic 50/Nitronic 60	195/190	$0.62X10^{-4}$
Leaded Bronze/ASTM A48 Class 30	80/205	$1.54X10^{-4}$
AISI 410/AISI 416	290/430	$7.38X10^{-4}$
In Iso-octane Specific Gravity 0.69		
Leaded Bronze/ASTM A48 Class 30 CI	80/205	$0.41X10^{-4}$
Nitronic 50/Nitronic 60	195/190	$0.69X10^{-4}$

The galling resistance of a material combination is another important consideration for a successful wear ring. Galling thresholds for a variety of materials have been published by Schumacher [5], based upon standard laboratory testing. Table 2 is a listing of a few galling thresholds published by Armco. In any pump application, it is desirable to select a combination with a high galling threshold for components with close running clearances. If a combination with a low galling threshold cannot be avoided, the running clearance should be as large as possible. Most pump

Table 2. Galling Resistance of Alloys [5]

Metals in Contact			Threshold Galling Stress (ksi)
Silicon Bronze	(200) vs Silicon Bronze	(200)	4
Silicon Bronze	(200) vs Type 304	(140)	44
Waukesha 88	(141) vs Type 303	(180)	50+
Waukesha 88	(141) vs Type 316	(200)	50+
Waukesha 88	(141) vs S17400	(405)	50+
Type 410	(322) vs Type 420	(472)	3
Type 416	(342) vs Type 416	(372)	13
Type 416	(372) vs Type 410	(322)	4
Type 440C	(560) vs Type 440C	(604)	11
S17400	(311) vs Type 304	(140)	2
S17400	(435) vs Type 304	(140)	2
Nitronic 50	(205) vs Nitronic 50	(205)	2
Nitronic 60	(213) vs S17400	(313)	50+
Nitronic 60	(205) vs Nitronic 50	(205)	50+
Nitronic 60	(205) vs Stellite 6B	(415)	50+

manufacturers have developed standard running clearances with this material limitation in mind.

An example of galling in a reciprocating pump is shown in Figures 4 and 5. Galling of an AISI Type-440C stainless steel valve stem is shown in Figure 4. This is depicted as heavy material transfer along the contact zone due to loading above the material combination's galling threshold. The mating AISI Type-440C bushing surface is shown in Figure 5 with evidence of material galling. This is in contrast to the normal adhesive wear expected for this component shown in Figure 6. The normal adhesive wear shown consist of uniform wear with light transfer of material shown as black streaks in this figure.

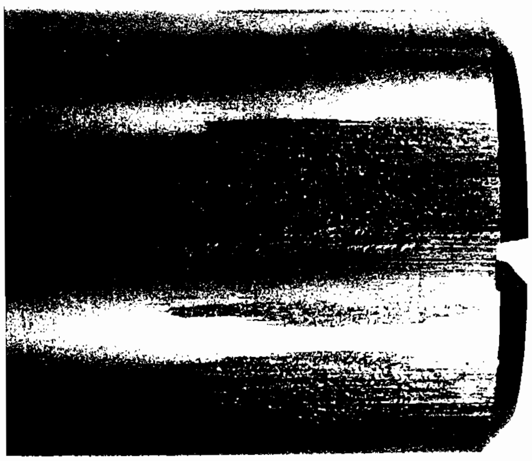

Figure 4. Galling Along the Stem of a Reciprocating Pump Valve.

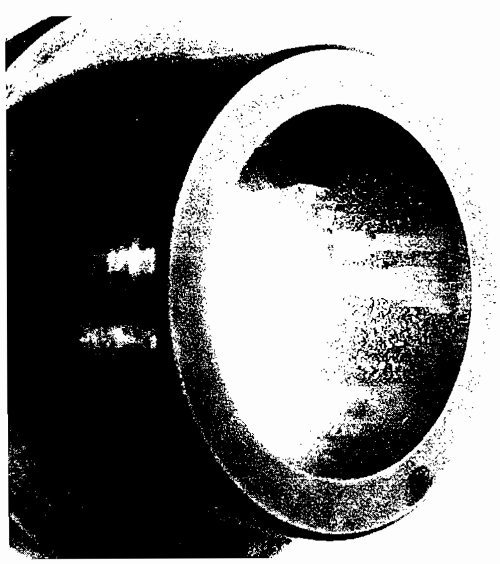

Figure 5. Galling of the Mating Sleeve in the Reciprocating Pump Valve. Note the heavy material transfer.

Based on empirical testing and field experiences several sound rules of thumb have been developed through the years to avoid catastrophe when selecting wear ring materials. Three factors can be used to select materials for wear surfaces in clear liquid environments. They are:

- Corrosiveness of the fluid,

Figure 6. Normal Adhesive Wear Along the Stem of Another Reciprocating Pump Valve. Light material transfer and uniform wear is evident.

- Amount of wear allowed and
- Galling stress.

Corrosion determines the class of material to be used. These classes generally fall into three groupings. These include: noncorrosive, mildly corrosive, and corrosive. Of course, there are additional constraints in selecting an appropriate material within the corrosive material grouping that will need to be addressed by application experience.

For noncorrosive environments, a variety of inexpensive materials can be considered. Ambient noncorrosive waters, oils, and solvents fall into this category. The most commonly used materials for these services are cast irons and leaded bronzes. Both of these materials contain second phases which decrease the coefficient of friction, thereby enhancing that combination's wear resistance. Free graphite (carbon) in cast iron, and lead (Pb) in the bronze alloys act as solid film lubricants or friction modifiers when contact occurs. It was common practice to use cast iron and leaded bronze in combination in noncorrosive applications; recent concern with the hazardous biological effects of lead has resulted in legislation to minimize, if not eliminate, their availability in the future. For this reason, nonmetallic substitutes for wear ring applications are getting a lot of attention in the pump industry.

The mildly corrosive environments are handled with martensitic stainless steels. These materials have a relatively low galling threshold if used at the same hardness below 450 Brinell. The rule of thumb is to select a combination that has at least a 50 Brinell hardness difference. If the material hardness is greater then 450 Brinell hardness, the combination can have the same hardness. Standard wear components for mildly corrosive services are often manufactured from AISI type 410 or AISI type 416 stainless steels. The rotating rings are usually heat treated to have a hardness value slightly below 300 Brinell and the stationary components are heat treated to an approximate hardness of 400 Brinell. The softer rotating component is needed to resist stress corrosion cracking or hydrogen embrittlement of impeller rings in environments containing chlorides or hydrogen sulfide. This is also true for services such as ultra-pure high temperature waters where embrittlement can cause delayed cracking resulting from absorbed hydrogen. These rings are placed in tension, because of the usual practice of shrink fitting them onto impeller hubs, and are susceptible to embrittling mechanisms if through hardened above 302BHN. High surface hardnesses have been achieved by selectively hardening these components with a laser. Laser hardening allows for the selective heat treatment of controlled carbon AISI type 420 stainless steel to a value of greater than 500 Brinell. Since this is a surface hardening process, the ring is not susceptible to embrittlement by the environment even after it is pressed onto an impeller hub.

Corrosive fluids such as seawater or those with high concentrations of chlorides require the use of austenitic stainless steels or nickel base corrosion resistant alloys. These materials have lower galling thresholds. In such environments it has been the practice of most pump manufacturers to utilize an antigalling austenitic stainless steel alloy commercially known as Nitronic-60. This material has a published galling threshold in excess of 50 ksi in combination with the other austenitic stainless steels [5]. If additional corrosion resistance is necessary, the Nitronic-60 component can be made from Waukesha-88. Waukesha-88 contains discrete pools of tin and bismuth for lubrication and also has a high galling threshold [5]. Another common practice for corrosive environments is to use a standard austenitic stainless steel with a stellite overlay. These materials are preferred when sufficient solids are present to cause three-body abrasive wear of these components.

Some special applications have produced unique material applications for given environments. These include low specific gravity applications where the use of mechanical carbon materials are desirable because of the nonlubricating nature of these fluids. A common practice is to make the stationary component a metal-filled graphite if the specific gravity is 0.5 or less. Stationary mechanical carbon components are also used in liquid CO_2 services and other potential dry start applications such as the upper bearing in vertical pumps. Currently, nonmetallic wear components of advance polymers are being looked at to solve nagging problems encountered in a variety of applications. Usually these nonmetallics are glass-filled with some having various additives to enhance their wear resistance.

ABRASIVE WEAR

Often abrasive wear is categorized into two main classifications. They are two-body abrasive wear and three-body abrasive wear. The susceptibility of a material to abrasive wear is best addressed by looking at the mechanism that produced material loss in an environment containing solid particles. The degree of material damage, due to this mechanism, depends upon the bulk hardness of the material and the characteristics of solids present. Important particle characteristics include: size, shape, hardness and mass.

Two-Body Abrasive Wear

Most texts designate abrasive wear by either two-body or three-body abrasive wear mechanisms. Two-body abrasive wear occurs when a hard body slides over a soft surface [6], producing grooves due to ploughing and micro cutting of the softer material. This is usually associated with a hard material with relatively sharp peaks which focus the forces transmitted through parts in relative motion. This type of abrasive wear damage is usually not a serious consideration for pumps. Most pump design engineers avoid intentionally making one component detrimentally affect the life of another by designing a file like wear surface. However, two-body abrasive wear can occur when pump components become rough along their contacting surfaces. This can produce an irregular rough file or rasp appearing geometry. This surface texture modification is usually the result of prior adhesive wear events, when for example: transferred wear particles adhere to the face of another component. These new wear particles present themselves as hard sharp surfaces that can abrade the softer counterface material. The transferred material usually has a higher hardness, because of cold working or metallurgical transformation, which can occur during the transfer process. These hard surface anomalies will cut furrows in the softer material during contact. Although one would expect two-body abrasive wear damage to be common in a pump, in reality this form of abrasive wear is not considered to be a real problem. It is generally concluded, in the literature, that the resistance of a material to two-body abrasive wear is increased with increased material hardness. Equations predicting the amount of two-body abrasive wear can also be found in the literature [6, 7].

Three-body Abrasive Wear

A second and more predominant form of abrasive wear found in pumps is referred to as three-body abrasion. This occurs when a third body, an abrasive particle, enters the space between two parts in relative motion. These hard particles will plough and machine one or both of the materials while passing between them. This is most pronounced when solids in the fluid stream are able to flow through the clearance between rotating and stationary components such as wear rings or impeller hubs and diffuser rings.

Two variables must be taken into consideration to minimize three-body abrasive wear. One is a design consideration of the pump itself. The wear ring clearance can encourage or impede this form of material destruction. The relationship between the size of particles in the fluid stream and the gap into which they can enter is important in determining the magnitude of and inclination for the abrasive wear damage. This is graphically illustrated in Figure 7, where three possible cases of particle-to-gap relationships are shown. Condition "A" is logically the most damaging three-body abrasive case. When particles are just large enough to fit between the two bodies in relative motion interaction is bound to occur. At this location, they become ground between the two rings or seal faces producing a high rate of damage. Two geometric circumstances that can mitigate the three-body abrasive mechanism in a pump are shown as condition "B," very large particles relative to the ring clearance and "C," very fine or relative small particles with respect to the gap between the components in relative motion. As shown, the very large abrasive particles will not be allowed to enter the zone between the components in relative motion and therefore, cannot cause material damage. This condition allows the particles to flow with the fluid stream through the eye of the impeller and exit the pump without producing three-body abrasive wear. The fine particles in contrast, condition "C," will not be entrapped and ground between the rings and will not result in collateral damage of the components. These illustrations are, of course, ideal cases.

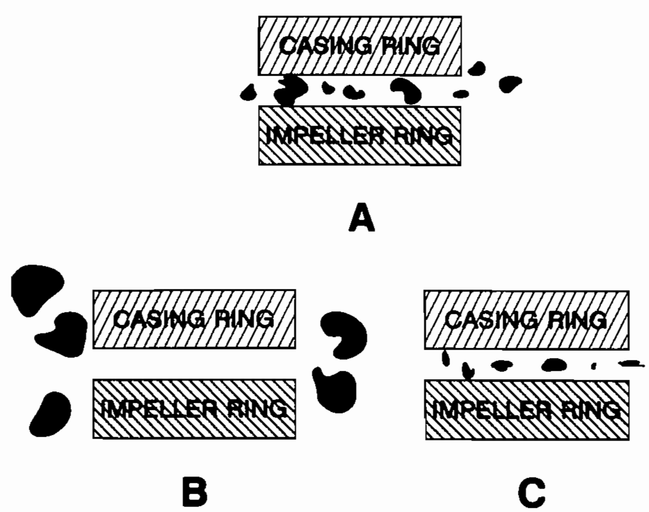

Figure 7. Three Possible Conditions Between Wear Surface Clearances and Solid Particle Size. Condition "A" is conducive to maximum three-body abrasive wear.

In reality, particles will exist in a range of sizes so that for all services where solids are encountered in the fluid stream, all the conditions shown in Figure 7 can occur simultaneously. To ascertain the tendency for condition "A" to exist, a characterization of the particle size and distribution should be known. This is relatively simple to accomplish by having the solids extracted from a fluid sample and performing a sieve analysis. The percentage of solids present in the fluid stream is extremely important and will be reviewed later with guidelines given for appropriate material selection. The sieve analysis will delineate the fraction of total particles in the fluid stream falling into the "A" category. In this manner, the design engineer can determine if a problem exists and can take suitable action concerning ring clearance and material hardness.

Particle hardness is also extremely important. Common sense tells us that if particles are soft and friable, such as talc, little damage would be expected to occur on the metal pump components as a result of three-body abrasive wear. However, if the particles are very hard, such as welding scale or silicon dioxide (SiO_2), which is sand, the amount of damage is expected to be greater. Geometry of the particles also determines the amount of damage that can result in three-body abrasive wear. Often, particles of SiO_2 are found in a rounded condition. Water from rivers and pumps used to handle seawater on ships frequently encounter these configurations. These hard and round configured particles are less damaging than particles of equal hardness with angular sharp configurations such as fly ash.

In order to determine a material's resistance to abrasive wear, several standard tests are available. Each test is unique in that it attempts to provide the mechanism that most appropriately addresses the class of abrasive wear. Materials that are resistant to two-body abrasive wear are generally resistant to three-body abrasive wear also. Two-body abrasive testing is for the most part conducted using a pin type abrasive wear test [8]. This test uses a rotating pin that is forced against a fresh commercial abrasive paper in an oscillating fashion. Three-body abrasive testing is usually performed using a rubber wheel tester. In this test, a specimen of material is forced against a rotating rubber wheel while an abrasive is introduced along the rubbing face. These tests can be performed to the guidelines of ASTM G-65 which describes the test apparatus, specimen configuration, rubber wheel type, sand quality, and recording information.

Test results from both types of tests show that for metals, the primary property for increasing resistance to abrasive wear is the bulk hardness of the alloy. Zum Gahr has provided test results to graphically illustrate this fact [7]. They have shown that subtle microstructural differences, alloying and surface condition differences, within alloy groups, also influence a materials abrasion resistance. These differences are:

- Abrasion resistance is increased with increasing bulk material hardness.
- At the same bulk hardness steels with higher carbon content have higher abrasion resistance.
- Cold working, which increases a material's surface hardness, does not significantly increase abrasion resistance of the alloy.
- Precipitation hardening increases bulk material hardness and abrasion resistance of an alloy.
- Gray cast irons show decreasing abrasion resistance at higher hardnesses. (Only a slight decrease is shown.)
- Softer austenitic white cast irons exhibit improved abrasion resistance over martensitic white cast irons.
- Carbides are important for wear resistance of steels and chromium alloyed white cast irons.
- A carbide volume fraction of 30 percent maximizes abrasive wear resistance for materials with a soft matrix.

What does this mean with respect to pump components that may be susceptible to three-body abrasion? As mentioned before, the primary location within a centrifugal pump that may experience this form of damage is at the wear rings. An example of three-body abrasive wear is shown in Figure 8. The wear of a laser hardened impeller ring is shown after approximately one year of service in a mine dewatering operation. The abrasive wear was caused by fine tailings which were abundant in this mine application. To increase the survivability of a material at this location, a material with maximum hardness is a good start. That is why pump producers use coated rings in applications where abrasives are present. However, depending upon the severity of the service, a choice of a ring material containing carbides, or at least having high hardness, is a good starting point for this application. Some antiabrasive wear materials used for these services are:

- Ni-Resist, generally good adhesive wear and due to of the chromium carbides in the matrix can survive mildly abrasive services.
- Hardened AISI-440C, high hardness (50-55 R_c), has good abrasive wear resistance.
- Selectively surface hardened AISI-420 (Laser hardened 50-55 R_c), has good adhesive and abrasive wear resistance in mildly abrasive services and is not susceptible to hydrogen embrittlement or stress corrosion cracking.
- Carburized and hardened rings, good for mildly abrasive services if 12 percent chromium stainless steel is the base material.
- Stellite or Colmonoy coated (hardfaced) austenitic stainless steel rings, good for abrasive services.
- Solid stellite rings, same as above.
- Tungsten carbide inserted rings, good for abrasive services, however, caution must be exercised in the retaining mechanism to avoid applying tensile stress during operation or due to thermal transients that may be encountered in the pump during operation.
- Silicon carbide inserted rings, same as above.
- Partially stabilized Zirconia (PSZ).

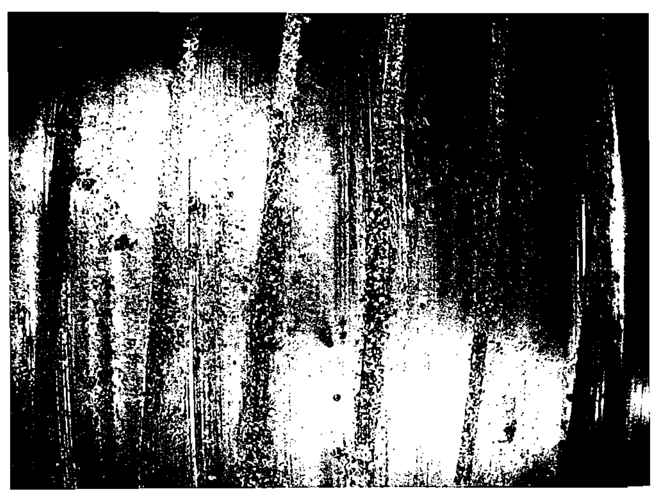

Figure 8. Three-Body Abrasive Wear of a Laser Hardened Shaft Sleeve in an Abrasive Service. Note fine concentric scoring of the hardened surface. Helix pattern is the laser beam overlapped zone produce by the laser process.

This list is ever expanding with developments by pump manufacturers and coating specialists. The use of nontraditional pump materials and surface alloy modifications is providing increased resistance to abrasive wear damage, thereby increasing the life of pump components in hostile environments. Recent advances have been through the use of: ceramics, metal-matrix composite materials along with laser surface alloying and laser surface modifications to a substrate that could not survive in an abrasive service.

EROSION

Theoretically, most fluids handled by pumps are void of any significant amount of solid particulates. These fluids are often referred to as clear liquids. The materials needed to handle these fluids are based solely on the corrosive nature of the fluid. Guidelines for many of these services are embodied in "Corrosion in Pumps" [9]. In reality, many fluid handling applications requiring pumps involve far from clear liquids. Solid particulates may be unavoidable in pump application without costly filtration systems that must work flawlessly. Solid particles may be introduced to the pumping system during the manufacturing of the pump. Casting sand, heat treat scale, corrosion product, and weld slag are a few examples of debris within the pump which can become entrained in the fluid. Another source of suspended solids is from the piping systems. Again, many of the solid debris items listed for pumps can be originating in the fabricated piping system. An example of naturally occurring suspended solids are those found in water sources such as river water or seawater. These solids which are usually rounded silica (sand) particles, are not as abrasive as others encountered in pump applications. Suspended solids can cause premature erosion if their nature (geometry), size distribution, and hardness are not known during the material and pump selection phase of the procurement process.

Pumps of many types have encountered premature damage due to solid particle impingement. Both centrifugal and reciprocating pumps can lose excessive amounts of material, rendering them useless in relatively short periods of time. The characteristic features of solid particle erosion damage are usually recognizable. However, when an aggressive fluid with respect to corrosion is present, solid particle impingement effects may not be easily identified. These effects can appear very much like corrosion-erosion which is a fluid velocity controlled damage mechanism with no solids. If this damage is misdiagnosed, an improper material substitution can be made which may not solve the real problem. This can be very frustrating to a maintenance department. Conversely, a more likely situation is where the observed damage resulting from erosion-corrosion is misinterpreted as solid particle erosion. This can be equally frustrating to a maintenance department. A full understanding of pumpage including fluid velocity, fluid corrosiveness, content, and nature of the solid particles present is necessary for the appropriate action to be taken in improving the life of a damaged pump.

A good example of solid particle erosion in a pump is shown in Figure 9, and Figure 10. As illustrated, severe erosion damage is the gouging of the casing along surfaces that were either directly impinged or scoured by glancing blows of the solid particles in the fluid stream. This pump was in a bauxite service where the percent (and velocity) of Al_2O_3 and sand were too high for the carbon steel casing and CD4MCu impeller. A CF3M impeller in a fly ash service is shown in Figure 11. The fact that erosion damage is a function of the particle velocity is shown clearly in this figure. The greatest damage to the impeller is at the outer periphery which corresponds to the highest velocity of the fluid. The least amount of damage is near the impeller inlet eye. Closer examination of the impeller eye shown in Figure 11 illustrates the relationship of fluid velocity to the magnitude of material damage. This figure shows

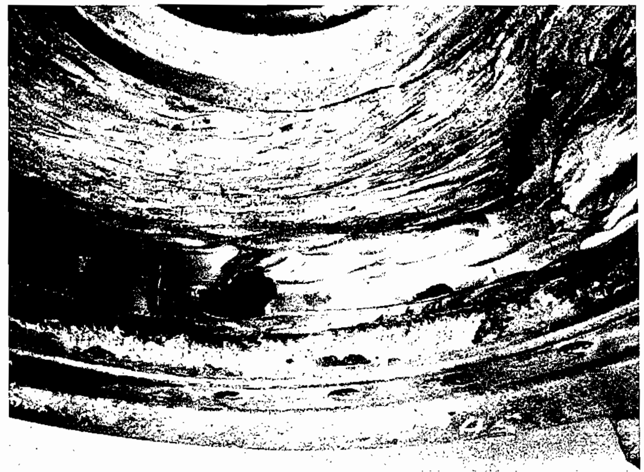

Figure 9. Severe Erosion of a Carbon Steel Casing in 17 Percent Bauxite and Sand Service. Note gouging due to local turbulence of the slurry.

Figure 10. Erosion at Exit Vane Tips of a Duplex Stainless Steel, CD4MCu, Impeller in the Bauxite Service.

Figure 11. Austenitic Stainless Steel Impeller in a Abrasive Fly Ash Service Shows Severe Erosion. Increased erosion occurs with increase fluid velocity along the impeller.

that even within the lower velocity region of the impeller inlet eye, the relative velocity of the fluid at the impingement surface is important. Note that the damage increases near the outside of the inlet eye and is almost nonexistent at the impeller hub where the fluid velocities are lower.

Erosion damage can also be encountered in reciprocating pumps. A good example of this is the erosion damage of a ball valve in a coal slurry application. Extensive erosion of an AISI type-440C stainless steel ball is shown in Figure 12, after the ball became "stuck" and unable to rotate. Rotation is important to produce uniform material loss of the entire ball in an abrasive service.

Figure 12. Erosion Damage of an AISI type-440C Stainless Steel Ball Valve in a Coal Slurry Service.

Macro material loss due to the erosion mechanism is principally due to the interaction between solid particles, which have obtained a certain velocity by the moving fluid with the surface of a pump component. In this case, potential energy is converted into kinetic energy, which produces material loss by the transfer of energy from the particle to the component. The amount of material damage on an individual particle scale depends specifically upon particle velocity and mass (kinetic energy), and the particle's geometry and hardness. Of course, the pump part which absorbs the kinetic energy resulting from particle impact has a role to play also. The material hardness and/or resilience of the pump component in absorbing the particle's impact energy will also determine the amount of material loss.

Chen and Hu [10] have performed laboratory tests on materials while changing the particle variables described above. Their test results show the following:

- Increased particle hardness increases material loss to 1500 kg/mm^2 microhardness (>75 R_c) where a decrease in wear occurs. This is most likely the result of the hard brittle particles fracturing, which absorbs some of the kinetic energy.

- Sharp angular particles increase the wear rate over round particles.

- Increased concentrations of abrasive particles increases the amount of erosion.

- Increased fluid (and particle) velocity increases the wear rate due to erosion.

- The impingement angle of the abrasive with the material is important with respect to wear rate. The minimum wear occurs at an impingement angle of 0.0 degree (tangent to the target surface) and increasing to a maximum amount of wear at a 65 degree impact angle.

The solid particle impingement angle versus amount of erosion has been plotted by several authors [11, 12]. A common characteristic of these plots is that erosion increases with increased impingement angle to a maximum material loss at an angle of approximately 25 degrees. Then the erosion damage decreases to the 65 degree impingement angle mentioned above. This behavior is for ductile materials. The plot for brittle materials, such as glass, is quite different. From an impingement angle of 0.0 degree to 90 degree, the volume of material loss is continuously increasing for erosion of brittle materials [11, 12].

These laboratory test results are very predictable based upon the type of mechanical damage which takes place. In fact, two forms of particle interaction with the impacted material have been formulated. They are [13]:

- Rubbing, shock or deformation wear.
- Cutting or shearing wear.

Other researchers have developed formulas for both forms of erosion. Bitter [14] has shown that if equations for each form of particle interaction are taken into account, good correlation exists between them and actual test results. Bitter also matches the bulk material property of yield strength because it is assumed that wear only takes place when the yield strength is exceeded.

Erosion damage, once identified, has a limited number of solutions to prolong the life of pump materials. There are six considerations when encountering fluids containing suspended solids. They are:

- Hardness of particles
- Quantity of particles
- Size of the particles
- Nature of the particles (geometry)
- Velocity of the pumpage
- Angle of fluid impingement

The first four items listed deal with the suspended solids, which can vary greatly from application to application. The hardness of the particles is important to understand in determining the best materials necessary to give acceptable life of the pump. Hardness can range from relatively soft substances, such as cellulose fibre, to very hard abrasive particles such as diamonds. Abrasivity of hard particles can be described using the Miller number [15].

The Miller number was developed to determine the relative abrasivity and attrition of a slurry. This slurry characteristic was deemed important in a test loop designed for testing the life of fluid ends in a reciprocating pump intended for slurry applications [15]. In a closed test loop, the abrasivity of the particles become less damaging with time due to particles fracturing and rounding (friability). The Miller number is, therefore, reported with two numbers. The first number characterizes the abrasivity of the particles and the second number is the loss of abrasivity (attrition) of the particles during the slurry test. The abrasivity portion of the Miller number is useful in practical applications, while the attrition number has found little use other then characterizing a test loop's influence on a slurry [15]. A slurry having a Miller number less then 50 is not considered abrasive in a reciprocating pump [15]. Some examples of slurries with a Miller number below 50 are: limestone, sulfur and detergent. The Miller number tests have provided insight into the influence that the particles' physical characteristics have upon abrasivity of a slurry. It has been determined that a slurry consisting of finer particles is less abrasive than one containing larger particles. Test data shows that Corundum at 220 mesh is about four times as abrasive as the same material at 400 mesh [15].

Particle velocity and impingement angle are design factors which can be used to mitigate erosion. The challenge in the coal liquefaction program investigated by the Department of Energy in the 1970s was to develop a high speed pump for handling coal oil slurries [11]. This was attempted because traditional slurry pumps are usually slow moving large machines which increase capital and operating costs of pilot plants built during this era. The majority of the slurry pump industry utilizes large slow moving single-stage pumps to address the solid particle erosion problem.

Materials

Materials of construction for pumps that handle high concentrations of suspended solids is based upon high bulk hardness. In many applications, coatings, hard liners and weld overlay are used to specifically increase the surface hardness of the internal wetted portions within the pump. However, surprising as it may sound, many of the slurry applications use nonmetallics because of their unique qualities.

Nonmetallics

Contrary to "harder is better," a good number of slurry pumps use nonmetallic materials such as rubber, which absorbs the kinetic energy of the solid particle through large elastic deformation of the surface. One case in the Unites States where a nonmetallic replaced metal is in the phosphate industry which flourishes in southern Florida. Urethane reverse-engineered replicas of single-stage metal pumps were produced by a small job shop. These nonmetallic pumps out performed harder cast materials used by the OEMs for this service.

Natural rubber is the most commonly used material since it provides good wear resistance with abrasive particles less than approximately 1/4 in to 3/8 in in size. Rubber linings pose a problem in the bonding of this outer shell to a metallic substrate. This is particularly true for cutwater areas of a casing and of course attachment to metal skeletons of an impeller. Care must be taken also in considerations of the liquid phase of the slurry, which can degrade the rubber, and the temperature of application. In general, rubber lined pumps should be limited to 250F.

Metals

In mildly abrasive services, carburizing of materials is often used to increase the wear life of components. Carbon is diffused into the surface of carbon steel, which after a hardening heat treatment can achieve surface hardness of 60 R_c. This gas diffusion heat treatment (carburize and harden) can produce high hardness layers which penetrate the outside surface of the pump component to a depth of approximately 0.080 in to 0.090 in. However, after carburization the materials are impossible to weld repair without cracking. Using a special process, usually a vacuum furnace, carburizing has been employed in the surface hardening of the 12 percent chromium stainless steels such as CA15 for abrasive service where mild corrosion is expected.

The most commonly used materials for severe slurry services are the abrasion-resistant cast irons found in ASTM A532. These specifications are broken into three main classes of hard cast materials. There are essentially three classes and several types of alloys covered in this specification. The most widely employed in slurry applications is the Class III hard irons. A brief description of this class of erosion resistant iron is as follows:

ASTM Class I-Type A:

This grade is a lower chrome cast iron containing one to eleven percent chrome and 3.0-7.0 percent nickel. This material is referred to as Ni-Hard. Class I alloys are heat treated to produce a martensitic structure containing secondary hard phases of chrome and iron carbides. They have a typical Brinell hardness of 500-600.

ASTM Class II-Type A,B,C,D, and E:

This grade is a higher chrome cast iron containing 11 to 23 percent chrome with the addition of 0.5 - 3.5 percent molybdenum. Class II alloys are also heat treated to produce a martensitic microstructure with chrome and iron carbides present. Class II alloys are frequently annealed to reduce the hardness to approximately 450 Brinell for machining. This class of material can also be hardened to approximately 600 Brinell The molybdenum in this class increases the material's hardenability for use of thicker sections.

ASTM Class III-Type A:

Straight high chrome cast irons with between 23-28 percent chrome, are referred to as 26 percent chrome irons. They are sometimes referred to by their initial trade name HC-250. Class III alloys are also martensitic when heat treated, and contain chrome and iron carbides. The material can be heat treated from 400 to 600 Brinell depending upon the desired properties.

In general, it should be stressed that machining and welding these three classes of materials is difficult. Another important consideration is the role of carbon content on corrosion, erosion and fracture resistance. For a given chromium content, high carbon reduces corrosion resistance because any chromium tied up as chrome carbide is no longer available to form a protective chrome oxide layer. While beneficial with regard to erosion and abrasion resistance, high carbon content increases susceptibility to breakage by thermal and mechanical shock. To counteract this problem, a number of precautionary measures must be adopted. First, slow warm-up cycles must be instituted, typically around 100°F to 150°F per hour [16]. Another strategy to enhance serviceability of this class of materials is to lower the hardness from about 600 to 400 Brinell by a partial anneal. This measure reduces brittleness, but at the expense of decreased erosion resistance.

Linings, Inserts and Coatings

The low ductility and toughness of A532 cast irons does not permit their use for primary pressure boundaries per ASME code and API regulations. Therefore, it is necessary to use steel pressure casings with liners of hard material, which restricts the use of hard irons to internal wetted parts. A common slurry pump consisted of HC-250 impellers and replaceable HC-250 wear liners for the volute, and for both the inlet and outlet ends of the pump casing.

Since pump erosion is very often quite localized, in some instances it is more practical to install replaceable mechanically attached inserts at high wear areas, such as the cutwater. These are typically made of sintered tungsten carbide or some other very hard material. Newer materials, such as ceramic composites and toughened ceramics, should perform better than the "cermets" used in the past. One problem with inserts is protecting the fastening device against erosive wear. Another problem is their tendency to act as turbulence raisers due to imperfect fit-up or erosion-induced crevices and offsets. Limited success has been achieved with weld-applied overlays of stellite and other hardfacing materials. Drawbacks of weld overlays include a propensity for cracking, debonding resulting from preferential corrosion of the bond line, dilution of the hardfacing material with the substrate, and potential uneven thickness after machining.

Thermal spray coatings, along with diffusion surface treatments, have been used in pump applications for fluids containing high concentrations of suspended solids. Spray coatings are restricted to accessible areas of applications within a pump. This is because spray type coatings are limited to line of site application. Diffusion produced coatings are not limited by this constraint. One drawback to diffusion processes is that they are performed in a high temperature furnace which can negatively influence base material properties of the component. Diffusion coatings can range from traditional gas carburizing to the diffusion of high chromium alloys. These coatings increase the surface hardness of the component, and depending upon the process can increase the material's surface hardness to values in excess of 60 R_c. Diffusion layers can be produced to a depth of approximately 0.100 in. One item of caution: these coatings usually render the material unweldable after application. Future weld repairs are not possible. For this reason, steps must be taken to protect areas of anticipated welding, such as attachment piping.

Through the years, developments in thermal spray equipment have enhanced the acceptability of this surface modification process. Thermal spray processes employ the transfer of a material onto another by raising the temperature of the hardfacing material, usually in powder form, and projecting it against the component that requires the additional erosion resistance. The bond strength between the hardface material and the substrate material is directly influenced by the maximum velocity that the particles of molten material achieve in a given thermal spray process. The greatest bond strength is achieved by the highest velocity process. Typical thermally sprayed materials used in pumps to resist solid particle erosion damage are:

- Nickel chromium boride coatings.
- Cobalt base hardfacing coatings.
- Tungsten carbide coatings.
- Solid particle tungsten carbide loaded (1) or (2).

Process	*Typical Particle Velocity*
(1) Flame Spray Process	100 Ft/Sec.
(2) Plasma Spray Process	800 Ft/Sec.
(3) D-Gun Process (Union Carbide tradename for Detonation Gun Process	800 Ft/Sec.
(4) HVOF (High Velocity Oxy-Fuel) Process	3000 Ft/Sec.

The severity of the service usually dictates the process used. In the coal liquefaction experiences of the 1970s, thermal spray coatings and diffusion coatings were tried in centrifugal slurry pumps. The thermal spray coatings, for the most part, were tungsten carbide and the diffusion coatings tried at the time were high in borides. Today, it is common to use carburization of carbon steel or 12 percent chromium stainless steel centrifugal pump components for mildly corrosive environments. It was found that for spray coatings, increased performance could be achieved by applying them over erosion resistant substrates. This is a challenge because the high chromium carbon abrasion resistant materials are thermal crack sensitive. The main shortcoming of coatings is their thinness, which can translate into an inadequate life. Another coating shortcoming is the lack of bond strength and the difficulty in keeping coatings in place. Overlay coatings, if applied several times thicker, are more prone to cracking, chipping and spalling.

Coatings are frequently used in reciprocating pumps for slurry services. In these services, they are used principally for increasing the life of plungers.

Another process for application of hardfaced materials is the laser consolidation process. This process can be accomplished in two different ways. The first case is where a laser beam is used to melt an applied coating placed upon a substrate via one of the thermal processes. Another process used is to simultaneously melt the substrate while applying a hardfaced material. In either case the principle is to use the hardfaced material as a consumable in a laser welding operation. Since the laser is a rapid process, very little dilution of the hardfacing material is produced. This allows for use of much thinner coatings that are less prone to thermally induced cracking during operation. In addition, since there is little dilution, the hardness and chemistry of the coating are very consistent. This provides for uniform erosion resistance throughout the entire coating thickness.

FATIGUE

In general, pumps are machines that either have fluid or mechanically induced cyclic loading on their components. Centrifugal pumps are, for the most part, steady state rotational equipment, but pulsations or fluctuating applied stresses are encountered. The source of these cyclic stresses can be from the fluid interaction between the impeller exit vanes and diffuser vanes. In a volute pump, impeller vanes and the casing cut water interaction with the fluid produces cyclic pressure loading. Mechanically induced cyclic loading also can be experienced. These forces are due to bending moments acting on the pump shaft or possibly the result of component imbalance in the rotor assembly. Reciprocating pumps experience cyclic loading of internal and external components from the very action of the machinery. In fact, these pumps can be thought of as large fatigue testing machines due to the pulsating action of the pumping process.

The one essential parameter in component fatigue is the presence of an alternating or cyclic load. When cyclic forces are applied to materials in a pump over a period of time, a crack may initiate from the component's surface. After initiation, the crack will grow with continued cyclic loading until the part finally fractures. The concept of material fatigue, which is a human trait, is associated with this mode of fracture because the magnitude of the cyclic loading necessary to cause this damage is well below the material's ultimate strength.

Corrosion, often the primary cause of pump material damage, can increase the likelihood of fatigue cracking. Corrosion assisted fatigue is a name give to this special type of cracking. Corrosion damage can change the surface texture of pump parts and significantly increase the local stresses acting on the pump component. If the corrosion damage is severe enough to produce a sharp notch in a region of high cyclic loading, then fatigue cracking of the component is inevitable. Corrosion is not the only mechanism of surface degradation that can promote this form of cracking. Surface disruptions through fretting or wear contact may also provide the sites for fatigue crack initiation.

When materials are subjected to cyclic loading, they can fracture even though the loading is far less than the tensile strength of the material. Engineers have been aware of this potential mode of component fracture for many years and have developed design criteria which take into account this anomaly. The study of cyclic loading and material behavior based upon cyclic stress history and flaw size is beyond the scope of this text. It should be noted, however, that the field of fracture mechanics offers an engineering design tool which can predict the life of an engineered component. This science uses the principles of cyclic loading and flaw sizes often found in commercial materials, to estimate the survivability of a particular design.

Fatigue was first systematically studied by August Wohler in 1852 [18]. Integrated in Wohler's work was the concept of alternating applied stress *vs* the number of cycles applied to a sample until fracture occurs (fatigue). This is the basis for today's S/N curves used by engineers in designing components. A typical laboratory generated S/N curve is shown in Figure 13. This curve shows the fatigue limit (sometimes called the endurance limit) of nickel aluminum bronze to be 42,000 psi. This curve was generated using smooth bar rotating beam test specimens. These specimens are carefully machined to avoid any type of metallurgical notch on their surface that will lower the applied stresses required to produce a failure. The lower curve shown in this graph is the result of introducing a corrosive media to the test environment. Corrosion will produce a degradation of the specimen's surface which will lower the fatigue curve. There is, in fact, no true fatigue limit for materials in a corrosive environment. Therefore, corrosion fatigue life of a material is usually published with cautionary statements. This is because corrosion is a time-dependent event. If given enough time, corrosion can penetrate completely through a fatigue test specimen and result in data points at zero load and zero cycles. For this reason, corrosion influenced fatigue test results usually specify the corrosive media, test temperature, details of sample preexposure to the corrosive media, and the test frequency with respect to the applied cyclic loading. These tests should never really show a true run-out condition. Published data varies because laboratories that use a low frequency of applied stresses increase the influence that corrosion has upon the test specimens and determined endurance limit. This is in comparison to laboratories that conduct these tests at a high frequency which minimizes the influence of the corrosive media. Published values for the corrosion fatigue life of various alloys are shown in Table 3 [19].

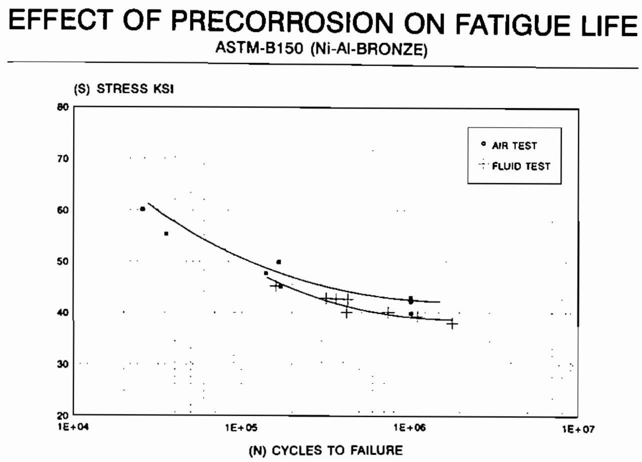

Figure 13. Example of a Laboratory Generated S/N Curve for Nickel Aluminum Bronze.

Wohler's investigation shows that fatigue life can be reduced with the presence of a mechanical notch [20]. This is schematically shown in Figure 14 as a shift in the S/N curve below that produced by a smooth bar specimen. The severity of the notch determines the divergency from the smooth bar curve. Fatigue is a three stage process consisting of crack initiation, crack propagation, and final fracture as suggested by Wohler [20]. Other researchers have delineated the precrack initiating stage into a separate category. Klesnil and Lukas [18] indicate that the fatigue process can be broken into three consecutive partially overlapping stages. They are fatigue hardening and/or softening, microcrack nucleation, and crack propagation ending in final failure. Stage one of this process accounts for the degradation of the material at the atomic level. Here, the applied load causes crystal lattice defects to grow in the material until they reach macro size in the form of small surface flaws or notches. These flaws become the initiation sites for fatigue cracks. In the case where a notch is already present, the first stage of either model will be shortened which shortens the entire fatigue cracking process. This is the reason for the shift in the fatigue curve for notched specimen.

The influence that mechanical notches and corrosion have on a S/N curve is shown in Figure 14. Each form of surface degradation mechanism lower the stress needed to produce specimen failure

*Table 3. Corrosion Fatigue of Alloys in Sea Water**

Alloy	UTS	CFS
Ti-6Al-4V	154	88
Inconel 718	189	60
Inconel 625	149	50
Hastelloy C	108	32
Monel alloy K-500	176	26
Ni Al bronze (cast)	115	15
304 Stainless	79	15
316 Stainless	85	14
304L Stainless	75	14
316L Stainless	79	13
17-4PH - cast		10
70-30 Cu-Ni (cast)	83	9
Ni Mn Bronze	82	9
MN Bronze	73	8
D-2 Ni-Resist		7.5
Mild steel		2

*Ambient temperature = 1750 rpm, about 2 to 3 ft/sec. All values in ksi at 100 Mc, about 48 days.

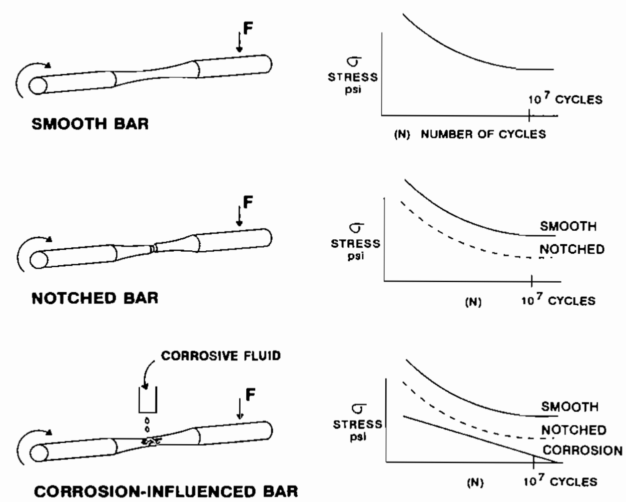

Figure 14. Sketches of Rotating Beam Test Specimens Including: a smooth bar, a notched bar and a corrosion influenced bar. The effect each specimen condition has upon the endurance limit is shown in S/N curve alongside each bar.

after a certain number of cycles. This in turn reflects a lowering of the material's endurance limit. In the case of introduced corrosion, no true endurance limit is reached, as illustrated in the bottom figure.

Fatigue fractures are easy to identify, especially if no other secondary damage masks their tell-tale appearance. Once identified, fatigue cracks can be addressed through either a material change, surface treatment, or design modification to decrease the magnitude of the applied stresses. Usually the three stages of fatigue cracking can be observed on a fatigue fracture face. A high magnification view of a smooth bar fatigue specimen after fracture is shown in Figure 15. Arrow "A" shows a single origin on this specimen. This fatigue crack propagated across the entire specimen diameter until a final fracture occurred. The area of the

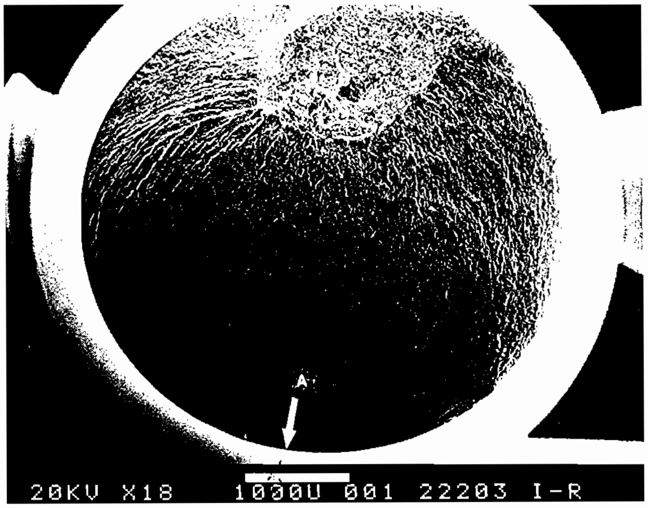

Figure 15. SEM Photo of a Fracture Smooth Fatigue Test Specimen with a Single Origin. The fatigue crack origin is at arrow "A" and the final fracture zone is at location "B."

fracture face is sometimes referred to as the ductile overload zone or fast fracture zone. The final fracture zone of the test specimen is shown as a small circle at arrow "B" in Figure 15.

An example of an actual fatigue fracture of a pump shaft is shown in Figure 16. The arrows shown in this figure indicate the location of many crack origins. Multiple origin fatigue fractures are often associated with rotating components. The relatively flat and smooth surface appearance of this fracture face is a characteristic of fatigue fractures. To the casual observer, this flat fracture appearance is sometimes mistaken for a brittle fracture because no evidence of plastic deformation is observed on or near the break. On this macro viewing scale, the fracture face indicates the magnitude of the cyclic loading that propagated the fatigue crack until it broke apart. Shown almost directly in the center of the fractured shaft section is a very small area of ductile overload marked "A" and "B." Because this area is very small, almost nonexistent, the loading causing the crack to propagate was very small. Looking at it another way, the only material holding the two halves of the shaft together was the last area to fracture, which is a mere fraction of the total cross sectional area of this shaft. This type of fatigue fracture is referred to as a multiple origin high cycle fatigue fracture.

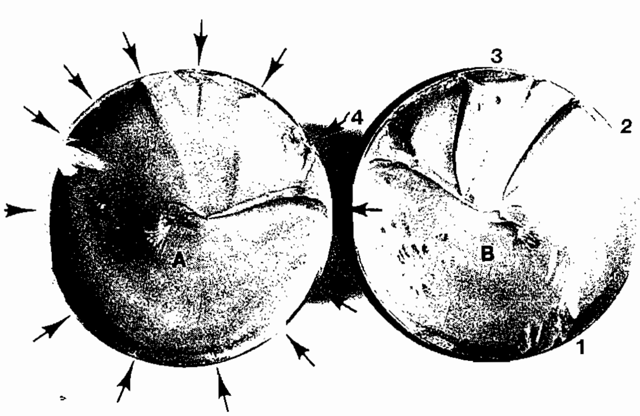

Figure 16. Multiple Origin Fatigue Fracture of a Pump Shaft. Arrows show locations of the many fatigue crack origins. "A" and "B" are corresponding final fracture zone of each fracture face.

Fatigue fractures also can occur in components that are of nonuniform geometry. For example, a fatigue fracture of an impeller shroud is shown in Figure 17. The arrows show the directions of fatigue crack propagation. Even though this geometry is complex, the three stages of fatigue cracking are still evident. As mentioned before, an investigator uses the relative size of each fatigue crack stage to determine the magnitude of the loads acting on the component. The identification of the crack origin is also of prime concern in conducting a failure analysis. The crack origin is important to determine if the fatigue crack initiated from a flaw in the material, a notch produced in service, or during manufacturing.

Figure 17. Fatigue Fracture of an Impeller. Arrows show the direction of fatigue crack propagation.

Corrosion plays a major role in the cracking of components in a pump. If the environment is sufficiently aggressive, the component can fail from corrosion assisted fatigue. In these cases, the corrosion mechanism is responsible for the fatigue crack initiation. In some cases, the propagation phase is also influenced by oxidation, which can mask the telltale features of the fatigue mechanism. Corrosion oxides, which form along the crack face, can produce a wedging effect which mechanically increases the local tensile forces acting on the crack tip. This will increase the crack propagation rate. An example of corrosion associated fatigue is shown in Figures 18 and 19. These figures show an impeller that experienced fatigue cracking of the front shroud wall at two locations. Evidence of corrosion pitting on the surface of this impeller indicates a strong possibility that corrosion influenced the mode of fracture. Further investigation of this fracture showed that both of the shroud wall fatigue fractures initiated at corrosion pits located in highly stressed areas of the impeller. The alternating loading was the result of fluid pulsations acting on the exit vane tip of this impeller.

Once the mechanism of fatigue cracking has been identified, suitable corrective actions can be implemented. These include:

- *Higher Strength Material*-as shown on any S/N curve, each material has a fatigue endurance limit at which point the component will have essentially infinite life. This limit has been correlated to the mechanical property of a vast number of materials. This data suggests that a good approximation for endurance limit of a metal is 50 percent of the material's tensile strength. This is for high cycle fatigue where no macro plastic loading is experienced. This rule of thumb is shown in a graph by Hertzberg [20]. If a

Figure 18. Overall View of a CF-3M Impeller that has Two Corrosion Assisted Fatigue Fractures in the Front Shroud Wall.

Figure 19. Closer View of One of the Fatigue Fractures that Originated in a Corrosion Pit at the Exit Vane Tip and Shroud Intersection. Additional corrosion pitting can be seen on the impeller in this figure.

higher strength material is used, an increase in endurance limit can be expected.

• *Design Modification*-the stress acting upon a component can be reduced with increased section size. Reducing the stress on a component will, of course, increase the parts life. Design criteria for mean stress in a alternating loading environment can be determined using several analytical models. Since components are subjected to a range of loadings (not a constant amplitude), a fluctuating mean stress is encountered in real life. Anticipated load history can aid in the design process to avoid fatigue fractures. Prediction of potential component life can be base upon a fluctuating mean stress design criteria referred to as the Pamgren-Miner cumulative damage law [20].

• *Surface Treatments*-that introduce compressive stresses to the surface of a part increase the fatigue life of a component. This is usually performed at a crack sensitive region such as sharp corners or notches. Fatigue cracking occurs as a result of applied tensile stresses. If compressive stresses are introduced into the surface of a material, cyclic tensile stresses in excess of the compressive stress value are needed to cancel their effect before fatigue damage can occur. Therefore, any form of compressive stress introduction will benefit a component with respect to fatigue cracking. Compressive stresses can be introduced by (1) cold working, (2) shot peening, or (3) a local heat treatment that introduces beneficial compressive residual stresses (such as laser hardening or induction hardening).

• *Increased Corrosion Resistant Material*-is beneficial for cases where corrosion has influenced the life of a component by degrading the parts surface condition. As mentioned before, notches formed by corrosion will increase the components susceptibility to fatigue crack initiation.

FRETTING

Fretting can be considered a special case of adhesive wear. It occurs when two parts in contact experience repeated small amplitude relative motion between close fitting surfaces. The potential of small amplitude motion between pump components can be at loose fitting impellers, beneath loose bearings, between impeller wear rings and the impeller hub. The design engineer does not intentionally create a circumstance that will generate this type of motion but, when it occurs, fretting damage can lead to other problems.

Fretting can usually be easily identified by the red powdery oxide that forms along the fretted surface. This is obviously not the case in a fluid environment that will wash this unique evidence away from the mating area. However, a tell-tale damaged surface appearance will develop on the fretted surfaces. This damage is often described as having a mottled appearance. It is best depicted as a flat, eroded looking surface with no directional appearance to the damage. Fretting damage of a pump shaft is shown in Figure 20. The fretting damage is limited to where the impeller was oscillating due to a loose fit. During operation, the impeller apparently was wobbling slightly producing the damage along the impeller fitted area of the shaft. Since the motion necessary to cause fretting can be of small amplitude, large vibrations of the pump may not be present. This makes detection of fretting during operation difficult. Although not available at the time of this investigation, the impeller for the example above would have similar degradation along its bore.

Although the oxide may be washed from the surface, some staining of the adjacent component can be observed after disassembly of the pump. This has led to the misinterpretation that fretting is a corrosion mechanism. However, fretting is the result of a special wear phenomenon. The fluid environment does not have to be present for fretting to occur.

Researchers have described fretting damage as being a four stage event [21]. The four stages include:

- adhesive wear of the asperities of the mating materials,
- abrasive wear caused by the wear debris produced by step one,
- abraded particles fill the asperity valleys, and
- elastic contact occurs producing cold working of the surface and micro pitting.

Avoidance of fretting damage is relatively simple. The first step is to eliminate or prevent the possibility of motion between the two components. This can be achieved through tighter clearances or for best results, providing a shrink fit of the assembly to increase the clamping force thereby preventing the oscillatory motion. If fretting is unavoidable due to design constraints, other methods of mitigation can be used. These include the use of various coatings or providing the contact zone with an appropriate lubricant. Coatings that have been successful include flame spraying high nickel alloys and silver plating one or both of the faces in contact.

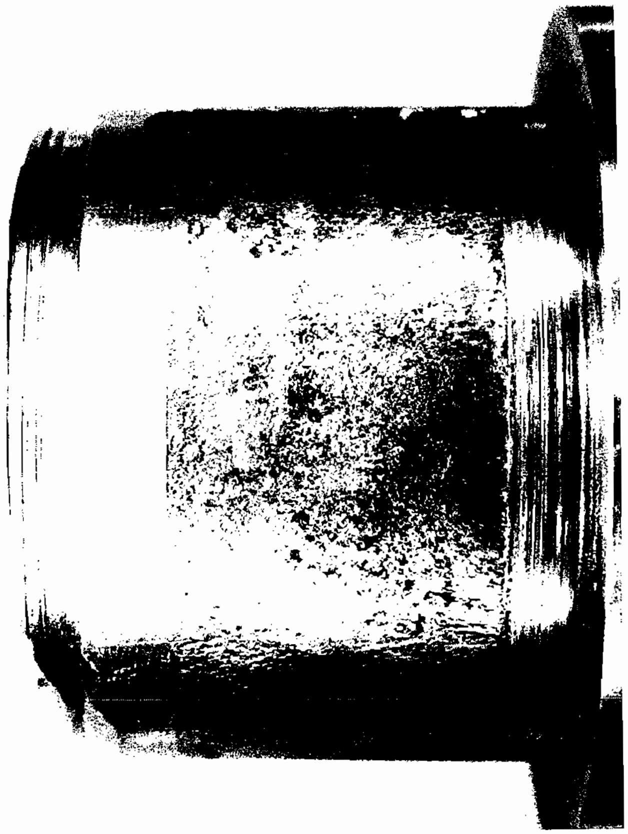

Figure 20. Fretting Damage of a Shaft Beneath an Impeller that Experienced Small Amplitude Motion.

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