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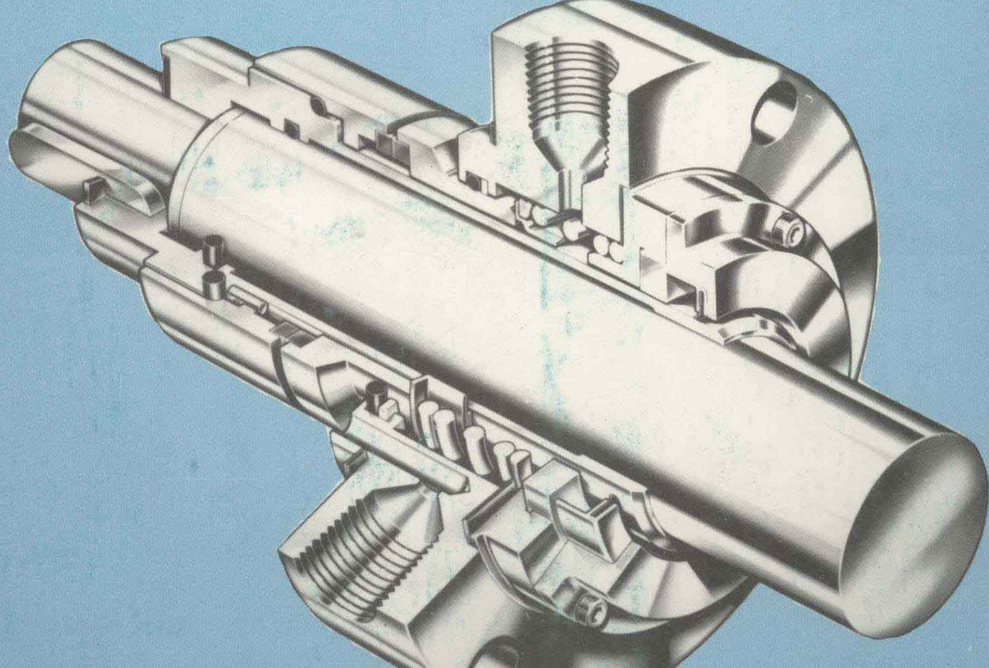
Shaft sealing in centrifugal pumps



Papers presented at a Seminar organized by the Fluid Machinery Committee of the Power Industries Division of the Institution of Mechanical Engineers, and held at the Institution of Mechanical Engineers on 12 February 1992.

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SHAFT SEALING IN CENTRIFUGAL PUMPS

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CONTENTS

Some aspects of seals for centrifugal pumps 1
B. S. Nau, N. D. Barnes, and R. K. Flitney

The development of more tolerant mechanical seals 9
J. M. Plumridge, B. J. Hyde, and R. L. Page

Oil production and power generation – some recent interesting seal developments 23
J. G. Evans

Chemical and petrochemical pumps with mechanical seals 33
P. R. Rogers

Mechanical seals for flammable liquids 51
P. Ansley-Watson

A sealing system guideline 59
A. J. Semple

Control of fugitive emissions of mechanical seals 69
E. Vanhie

Some aspects of seals for centrifugal pumps

B S NAU, BSc, PhD, FIMechE, CEng, FIMA, MASME,
N D BARNES, BSc, PhD, and R K FLITNEY, MASME
BHR Group Limited, Cranfield, UK

SYNOPSIS This is a selective discussion of some topics relating to the sealing of centrigugal pumps. After discussing the available options for type of seal attention is concentrated on mechanical seal issues. Topics covered include concepts for the rating of the severity of different seal duties, numerical examples of application duty parameters are given. Attention is then given to the seal environment in the pump and the control of emissions of volatile organic compounds with reference to recent actions in the USA.

NOTATION

Ai	sealing interface area
b	sealing face width
B	balance ratio
F_s	spring force
G	duty parameter(see eq 1)
G^*	transition value of G
N	shaft speed, krpm
p	sealed fluid pressure
$R_{thermal}$	thermal resistance of seal body
Re	seal chamber Reynolds Number
T_{BP}	boiling point (atmos.pres.)
$T_{chamber}$	fluid temperature in seal chamber
V	linear sliding speed between seal faces
W	total closiong force on sealing interface
ΔT	temperature differential
δT	temperature margin
η	absolute viscosity of fluid
η'	viscosity of fluid/ viscosity of water (20°C)
μ	sliding friction coefficient

1 SEALING OPTIONS

Three broad approaches offer themselves for the sealing of the impeller shaft of a centrifugal pump. These are 'Contacting' Seals, 'Non-contacting' Seals, and no seal at all.

'Contacting' seals in this context do not necess- arily suffer physical contact of the sealing faces during operation, a thin fluid film of the pumped medium generally maintains itself between the faces. This film may be no more than 0.5 to 2.0 micrometres in thickness and exists because either or both hydrodynamic action and hydrostatic action provide fluid pressure sufficient to support the closing force tending to bring the faces into contact.

Likewise, 'non-contacting' seals are not always non-contacting. The clearances are often towards the limit of what is achievable with normal machining tolerances and at least occasional contact must be accepted.

Canned pumps come in the category of 'no seal at all', but on reflection the gain is not quite all that it seems. The seal is traded for a bearing running in the pumped medium. Since a mechanical seal, for instance, is little more than a thrust bearing, the change is not as dramatic as at first seems. Even the types of bearing material can have much in common with seal face materials, e.g. the use of silicon carbide.

Returning to 'real' seals, these can be categorised as in Table 1.

Soft packed stuffing boxes tend to be dismissed today as 'low tech'. In reality they are too sophisticated for most users. The materials have very non-linear stress-strain characteristics and these are also time-dependent. The conseq- uence is that most users do not adequately appreciate the phase lag between adjusting the compressive load and the stabilisation of the gland's performance. Given adequate care and attention leakage rates can be as low as mechanical seals, as we have proved in rig tests in the past.

Table 1 Summary of rotary seal types for centrifugal pump shafts.

Passive sealing devices
- 'Contacting' seals:
  - Soft packed stuffing box
  - Polymeric lip-seal
  - Mechanical seal
- 'Non-contacting' seals:
  - Floating segmented bushing (e.g. segmented carbon ring)
  - Floating polymeric bushing (proprietary)
  - [Labyrinth]

Pumping sealing devices
- 'Non-contacting' :
  - Centrifugal ring
  - Viscoseal ('screw seal')
- 'Contacting':
  - Pumping lip seal
  - Pumping mechanical seal (proprietary)

Polymeric lip seals have a rather limited role in centrifugal pumps since they are essential low pressure seals (up to, say, 1 bar gauge). Also, they normally operate well only with liquids which are good lubricants, such as mineral or vegetable oils. Being polymeric their chemical and thermal compatibility is limited too. PTFE based materials are, however, exceptional in having excellent chemical compatibility and a reasonably high upper temperature limit.

Mechanical seals are undoubtedly the standard solution for sealing centrifugal pumps and will be given further attention later.

Passive non-contacting seals tend to have a rather specialist role. In particular, where reliability or low friction considerations outweigh leakage considerations. They are accepted on the basis of maintaining leakage within acceptable limits, which may actually be quite high. However, so long as the (high) leakrate is consistent and provision can be made to handle this efficiently this is not necessarily a problem. Indeed, even nuclear circulator pumps fitted with multistage mechanical seals may have a bypass flow around each seal stage, to control interstage pressures, and this controlled flow is accepted on the same basis as in a clearance seal.

Labyrinths are listed in Table 1 for completeness but rarely find application in centrifugal pumps as a primary seal due to their high leakrate. They sometimes have a role as a back-up seal to throttle a major leakage escape.

Pumping seals, as opposed to the 'passive' devices discussed so far, actively pump fluid back into the system from which it is leaking. The clearance seals in this category tend to find application at very high rotary speeds (e.g. 20,000+ rpm), since their head-flow characteristics are not impressive; however they can be effective for very viscous fluids at more modest speeds. Viscoseals have the added disadvantage of requiring tight machining tolerances over the screw lands. Both viscoseals and impeller seals can suffer problems due to air ingestion, the result of instability at the liquid/atmosphere interface.

'Contacting' pumping seals differ from the above in that they are only required to provide very low flow-rates, to pump back the limited fluid leakage flow passing through the thin interfacial film. Such seals do not suffer the disadvantages mentioned above.

In a vein not entirely facetious, one might consider the possibility of a pump having a smaller pump to pump back its leakage, the latter pump having a still smaller pump for the same purpose, and so on ad infinitum. Perhaps a small canned pump to restore the leakage from the mechanical seal of a conventional pump might be a practical solution in some circumstances!

2 MECHANICAL SEALS

It is not the intention in this discussion to be in any way comprehensive, but to address some issues relating to mechanical seals. The following topics will be considered in turn:

- Rating the severity of applications.
- The seal's environment in the pump.
- Emission control legislation.

2.1 Rating application severity

In a general way, most engineers with experience in the field of pumps and seals can identify some particularly 'difficult' duties and others which are 'easy'. But how does one quantify severity of a duty in an objective fashion?

There are several axes on which one would wish to measure, this is a multidimensional space! Some, if not all modes are listed in Table 2.

Table 2 Application severity modes.

Lubrication (wear, seizure, heat generation).
Leakrate requirement.
Fluid film thermal stability.
Continuity of operation.
Vibration.
Chemical reactivity.
Pump structure.

Lubrication :
Efficient lubrication is essential for successful operation of a mechanical seal. It limits wear, prevents seizure and controls frictional heating. In seals, the lubricant is commonly the pumped medium. If this is a low viscosity fluid, lubrication is likely to be less efficient, as also will be the case if the speed is low or the pressure-loading on the seal is high. A convenient combined measure of these effects is the dimensionless'Duty Parameter':

$$G = \eta \; V \; b \; / \; W \qquad (1a)$$

where η is the absolute viscosity (strictly speaking at the interface temperature), V is the linear sliding velocity, b is the sealing face width and W is the gross closing force on the interface (spring load plus hydraulic load). The larger the value of the Duty Parameter the thicker the lubricating film in the sealing interface is likely to be. For the present purpose it is convenient to simplify eq 1 by assuming a balanced seal with balance ratio of 0.7 and a spring load of 0.3 N/sq.mm of sealing face. This gives:

$$G = 24 \times 10^{-8} \; \eta' \; N \,/(p + 3) \qquad (1b)$$

where η' is the viscosity relative to water at room temperature, N is in krpm, and p in bar.

Experience shows that a value of G of about 0.5×10^{-8} marks the transition from a full fluid film separating the seal faces to partial asperity contact. This is an attractive operating condition to aim for in service since the leakage and friction are both a minimum. If we denote this transition value by G^* then for a particular application the ratio G/G^* indicates the nature of the lubrication mode, thus:

$G/G^* > 1$	full fluid film
$G/G^* < 1$	mixed film conditions
$G/G^* << 1$	solid contact.

It is interesting to consider the G-values for a selection of mechanical seal applications, Table 3. These values are based on information on specific applications reported in various published papers and seal manufacturers' literature. The viscosity was not normally given and has therefore been estimated. Notice how many of the applications cluster near the transition value of duty parameter.

Table 3 Examples of application Duty Parameters

Application	10^8 x G
Boiler-feed water	0.1 to 1.1
Oil pipeline	0.3 to 6.0
Carbamate	0.6
Mine-water	0.7
Flue gas desulphurisation	1.5 to 9.4
Water pipeline	1.6
Water pumps(Water Industry)	2.0 to 7.0
Paper pulp, cellulose	5.4 to 15
Gas reinjection (oil buffer)	14
NGL condensate (oil buffer)	36
Gas oil, crude oil	120 to 220

Leakrate requirements : In the field of gasketed joint design the trend is towards standardised categories for this parameter. Currently, the Pressure Vessel Research Council in the USA is proposing three target design levels for static gasketed joints which are given the names Economy, Standard and Tight. Each of these levels is 100 times less than the one before. With looming emission control regulations the definition of agreed standard levels having some technical significance could provide a useful basis for engineers specifying rotary seals for centrifugal pumps.

As a starting point for mechanical seal systems one might suggest the values in Table 4.

Table 4 Possible standard leakrate levels for mechanical seals.

Designation	Max. leakrate
Basic	10 ml/h
High grade	0.1 ml/h
Very high grade	1 ml/1000h

Fluid film stability :
Fluid film stability is important. If the film temperature reaches boiling point large volumes of vapour are suddenly and erratically created, blowing the seal faces apart. As the vapour is vented from the interface the faces equally suddenly collapse, which can cause serious damage and rapid seal failure.

Two physical characteristics are important in this context. One is the frictional heat generation in the sealing interface, in the range 100-1000 Watts for many mechanical seals. The other is the heat dispersal. This depends on the thermal resistance of the seal rings, the efficiency of heat transfer from the eal rings to the liquid in the chamber housing the seal, and the temperature of the bulk fluid in the chamber.

Table 5 illustrates the significance of seal body materials to thermal resistance. It shows, for various seal materials, the temperature differential required to drive a certain amount of heat through a seal ring. The figures tabulated clearly highlight the risk of film vaporisation.

However, it is not only the seal design and materials that are important in this context. Different fluids vary quite widely in the efficiency with which they can remove heat from the seal body, even for the same flow regime.

Table 6 shows this effect, taking water as a reference (100). Notice in particular how much better is the heat transfer for water than oil.

Table 5 Temperature differentials ΔT required to conduct 200W of heat through a typical seal ring (based on Nau 1990).

Seal ring material	ΔT, deg.K
Resin-impregnated carbon	140
Alumina (99.5%)	42
NiResist	32
Tungsten carbide	16
Silicon carbide (react.bonded)	3

Table 6 Relative heat transfer effectiveness of different fluids, water taken as 100. (based on Nau 1990)

Fluid	%
Water	100
Acetone	48
Ethanol	31
Mineral oil, light	10
Mineral oil, medium	4.1
Mineral oil, heavy	1.6
Air	0.21
Steam (saturated)	0.17

Clearly heat transfer is not just a matter of the physical properties of the fluid. The fluid flow regime in the seal chamber of the pump is also important.

Glossing over the details of fluid heat transfer, two useful thermal parameters relating to a mechanical seal application can be defined. One is the temperature differential available between the chamber fluid and the boiling point of the fluid:

$$\Delta Tavail = (T_{BP} - T_{chamber}) \qquad ...(2)$$

Atmospheric boiling point is a conservative value to use, rather than B.Pt. at chamber pressure for example). The other parameter is the temperature differential required by the seal. This is the product of the frictional heat generated in the sealing interface and the total thermal resistance (Rthermal) of the heat flow paths between the interface and the heat sink (normally the chamber fluid):

$$\Delta T_{req} = \mu W V R_{thermal} \qquad ...(3)$$

where μ is the friction coefficient, W the gross load on the sealing interface (i.e. $F_s+B\ p\ A_i$, see Notation) and V is the relative sliding speed of the faces. ΔT_{req} depends on a complex of operating conditions, seal face and body materials, shaft size and shaft speed.

The measure of application thermal severity, for a particular seal , is therefore:

$$\delta T = \{ \Delta T_{avail} - \Delta T\ req\} \qquad (4)$$

If $\Delta T \leq 0$ then film boiling is predicted and seal performance is likely to be poor. The information required to evaluate must come from both seal manufacturer and user. It would be very helpful if, as a matter of routine, seal manufact- urers provided the information needed by the user to evaluate eq 3 for ΔT_{req}.

Vibration : this has a ready made scale of severity in the British Standard vibration grades A,B,C,D defined in BS 4675 (see also ISO 2372).

Continuity of operation : whilst not easy to quantify, the continuity of operation is a significant factor in mechanical seal performance. Stop-start operation significantly reduces working life, as was shown by plant survey data collected by BHRA. In this context the persistence of the interfacial fluid film is a significant factor. In Nau(199)) film persistance times are tabulated for a variety of conditions and are shown to range from a few seconds to a few days, depending on fluid viscosity and the closing force - the latter would often be simply the spring force. Film persistance is important by virtue of its effect on wear, during starting and stopping. Wear at such times can modify the sealing face geometry (waviness and coning) thereby changing the seal's performance characteristics on successive restarts.

Chemical reactivity : this too has a ready made scale in the chemical pH acidity-scale. However some simplification can be envisaged e.g. by adopting a banding scheme for pH , such as:

<4.0 ,
4.0 to 5.9,
6.0 to 8.0,
> 8.0.

These may not be the best choice of ranges but they illustrate the principle.

Pump structure :
Structural aspects of the pump can impact on the behaviour of the seal, which must therefore be able to tolerate some degree of misalignment, runout etc. Table 7 summarises such factors. In considering such effects there are two sides to the coin. There is as yet no

general consensus concerning what limits the seal supplier can reasonably expect the pump manufacturer to achieve, nor the capabilities which the pump manufacturer can reasonably expect of the seal, as far as the seals ability to live with such effects. We ourselves are currently addressing the latter point in a program of test rig evaluations of mechanical seal capabilities.

Table 7 Pump factors affecting the seal.

Casing	
	rigidity
	pipe loads
	pump-mounting stresses
	squareness of seal mounting
	flatness of seal mounting
	concentricity of seal mounting
Shaft	
	stiffness and bending
	straightness
	alignment precision
Fluid	
	vibration when runnning off b.e.p.
	fluid pressure
Seal chamber	
	dimensions and shape
	injection/recirculation flows
	fluid pressure and condition in seal chamber

Clearly the various pump factors are associated with a cost factor. Providing a better environment for the seal costs money. What is not clear is how this can be quantified.

2.1 The seal's environment in the pump.

Under this heading attention will be concentrated on the fluid environment within the seal chamber. Details of the fluid flow pattern around the seal in the seal chamber affects seal performance for several reasons:

- chamber fluid is heat sink
- heat transfer from seal to fluid
- concentration of vapour or gas
- concentration of solids
- abrasive wear of seal (or chamber wall)

Several quite distinct flow regimes can exist around the seal, their occurrence is governed by the Reynolds Number Re (or the related Taylor number), among other things.

The Reynolds numbers (Re) for seal chamber flow varies over an enormous range (Nau 1990), depending on geometry, fluid viscosity and shaft speed. In heavy oil at low speed Re may be as low as 1 while in water at high speed it may reach one million or more. The consequence is that the flow regime may be any of the following:

laminar
Taylor vortex
fully turbulent
turbulent Taylor vortex.

The flow pattern is further complicated by radial and axial recirculations driven by radial surfaces, such as end walls of the chamber or radial surfaces on the sea (Nau 1990). Seal springs can further modify the flow, not to mention the effects of fluid injection or circulation. In this latter connection it may be pointed out that although it is common practice to talk in terms of volumetric flow-rates, it is equally important to consider the effects of velocity, which is affected by port size as well as flow rate.
The shape and optimisation of seal chambers is a major topic in its own right and has been the subject of considerable research in the last few years (Barnes,Flitney and Nau 1991, 1992). An interesting result of this work has been to highlight the effectiveness of tapered seal chambers for controlling build-up of vapour, gas or solids.

2.2 Emission control legislation.

Having considered the seal's environment we should now consider our own!

Regulations for the control of hazardous emissions from seals, amongst other equipment, are moving ahead rapidly in the USA, led by the state of California. In the latter, the South Coast Air Quality Management District has progressively tightened permissible limits. For concentrations of volatile organic compounds adjacent to the seal on a pump these have reduced from 10,000 ppm in the early 1980's to 1000 ppm in February 1991.

This stimulated US seal manufacturers to combine in the production of guidelines for meeting such regulations. Table 8 summarises some key recommendations from this document, these apply for seals up to 150mm diameter, pressures to 40 bar and speeds to 5600 rpm.

The use of double seals, with buffer fluid in between, raises the question of the handling of the contaminated buffer fluid, which may not be trivial. One might also comment on the use of concentration as a parameter. Whilst convenient for measurement, concentration is only a very indirect indicator of the quantity of emission escaping from the seal. It depends on the degree of ventilation of the space between bearings and seal, and the draught from the drive motor's fan.

Table 8 STLE advice on emission control from mechanical seals.

Specific gravity ≤0.40

use double seals.

Specific gravity >0.40

for permitted ppm<500 (10mm from seal):

use tandem or double seals.

for 500 > permitted ppm ≤1000 :

single seals sometimes acceptable,

else use double or tandem seals.

for permitted ppm>1000 :

use single, double or tandem seals.

REFERENCES

(1) Anon, Guidelines for meeting emission regulations for rotating machinery with mechanical seals. STLE Special Publication SP-30, September 1990.

(2) Nau, B.S. Research in mechanical seals. Proc.I.Mech.E. 204,349-376, 1990.

(3) Barnes,N.D., Flitney, R.K. and Nau, B.S. Considerations affecting the design of pump chambers for improved mechanical seal reliability. 12th Intnl. Pump Technical Conf., BPMA, April 1991.

(4) Barnes,N.D., Flitney, R.K. and Nau, B.S. Mechanical seal chamber design for improved performance.9th Intnl. Pump Users Symp.,Texas A&M Univ.,1992 (in press).

The development of more tolerant mechanical seals

J M PLUMRIDGE, CEng, MIMechE, MIEE, MRAeS, B J HYDE, and
R L PAGE, MBA
EG&G Sealol, High Wycombe, UK

The paper considers seal failure statistics and mechanisms. A comprehensive list of design features to improve seal performance and life is provided. Foremost amongst these will be the use of edge-welded metal bellows construction which is tolerant to extremes of temperature and rotational speeds. Examples are provided to cover the application range from simple seal solutions for lighter duties, to complex arrangements for both arduous duties and for those potentially hazardous installations which must be safely contained. It is argued that the least expensive seal is frequently selected, whereas quite often a modest cost premium will result in a significant improvement in seal reliability.

1. INTRODUCTION

In the mid 80's a large number of authors, particularly in the Refining Industry, presented papers equating maintenance spending with mechanical seal reliability. Whilst the data varied from one presentation to another, the message was generally the same.

- The Refining Industry spent around $6 billion per annum worldwide for maintenance (Ref 1).

- Of this, some 12% (or $700 million) of global maintenance expenditure was for rotating equipment.

- Approximately 70% of pump maintenance cost was seal related (Ref 2).

Within process plant, refineries are amongst the more demanding in terms of requirements for safety and component reliability. However, other Chemical Process Industries, Utilities and Municipalities are increasingly conscious of the need for safer, improved performance mechanical seals.

Seal manufacturers have recognised this, and have developed a steady parade of new products which have addressed the need for more tolerant seals.

2. WHY SEALS FAIL

Before considering why seals fail, we should first consider how failure is defined. Rarely is the performance expectation specified in leakage and life terms as part of a process sheet requirement. The seal manufacturer is often left with an impression that his product is expected to work in an infinite variety of operating conditions, should be completely leak free, last forever and cost very little.

In the absence of defined expectation, the author's company generally approaches a seal application with the following objective:-

- Recommend a seal arrangement which will operate with minimal leakage over a reasonable range of operating conditions for a minimum of 3 years, while making few/no demands on the operator to maintain a "seal support system", i.e. Design for the maximum tolerance.

So why do seals fail? Detailed failure mechanisms will be considered later on, but in the authors' view there are three principal reasons for failure.

2.1 Incorrect Selection

This can be the seal manufacturer selecting the wrong seal or materials. Quite often, though, the pump manufacturer or operator will select the wrong competitive tender. He will do so for one of two reasons; either he will not have the necessary detailed knowledge to make the correct selection, or will make the decision based mainly on the tender price.

An authority on mechanical seal reliability (Ref 3) suggests that pump application engineers will generally agree that seal and seal environmental system selection on many pumps is becoming more complex and time consuming than pump selection, and yet only 10 to 35% of pump engineering time is generally spent on the mechanical seal system. He further suggests that seal selections made entirely by the pump vendor have generally proven to be the least reliable. The pump vendor is concerned that his competitor will underbid him and that the engineer selecting the pump will only look at the bottom line cost without giving credit to superior seal components or seal system design. Consequently, the least expensive seal is often selected and plant operations or maintenance are then burdened with an inherently weak seal for the life of the equipment.

It is within this selection procedure that the seal manufacturer must bid. Whilst he will not knowingly propose a seal that will not meet the specification, market forces often demand compromise. We propose that seal selection should be based on qualified references and operator experience of existing successful installations, and that selection by the pump vendor alone should be discouraged. Quite often, a relatively small price premium for a superior seal will significantly improve the long term performance of the seal and hence the pump.

2.2 <u>Incorrect Installation</u>

This will commonly cause premature failure of the seal. In addition to education, training and fitting instruction discipline, seal design can be made tolerant to less skilled labour. Experience shows that cartridge (bench assembled) arrangements incorporating metal bellows seals can contribute substantially to error-free fitting.

2.3 <u>Incorrect Use</u>

Seal and system selection is based upon the process sheet, which tends to show single point (untoleranced) data. Most mechanical components perform well in unchanging conditions. Mechanical seals can be designed to work in arduous and difficult conditions and give long life. Others fail prematurely in much less arduous conditions. This is quite often due to transient changes in operating conditions outside the duty for which the seal was selected. Conversely, failure can be due to operating conditions not considered in the Process Sheet. In particular, continuous operation in a pump which provides a poor working platform - vibration, shaft eccentricity, housing squareness, surface finish, pressure pulsation can all affect seal types with low tolerance thresholds.

3. MECHANISM OF FAILURE

A recent survey at a major UK Refinery revealed causes of failure in line with Fig 1. The effects of incorrect selection, fitting and operational use usually reveal themselves in deterioration of the primary and secondary sealing areas.

3.1 <u>Deterioration of Wear Faces</u>

It has been estimated that 45% of mechanical seal failures or 31% of unplanned pump breakdowns are due to premature failure due to friction face deterioration. Leakage is the effect, and there are several causes:-

- Loss of lubricant
- Lubricant vaporisation
- Mechanical distortion
- Thermal distortion
- Abrasion
- Corrosion

Seal design, material selection and operator review should be undertaken based on the assumption that there will surely be off-specification upsets, and the seal should be able to cope with these transients.

To reduce problems at the friction face, the following guidelines are appropriate:-

- Use hydraulically balanced seal designs.
- Use seal types, arrangements, and materials which withstand the actual operating conditions, rather than modifying the conditions in the seal chamber to enable use of designs or materials which would otherwise fail.
- Ensure that the faces are subjected to fluid flow or turbulence, and they are not insulated from the heat flow paths (by packing etc).
- Use the best materials available and reduce the combination choice. Two combinations are sufficient for the vast majority of all duties:
  - Silicon Carbide vs Silicon Carbide for abrasive duties, and
  - Carbon Graphite vs Silicon Carbide for everything else.

Carbon is a forgiving solid lubricant which will tolerate fairly long periods of lubrication loss (dependant upon operating conditions). Silicon Carbide has all the ideal properties for the counterface. Extremely hard and wear resistant, capable of running against carbon at very high pressure/velocity combinations, has extreme temperature / corrosion / abrasion resistance, high thermal conductivity and modulus, low friction and thermal expansion coefficients, and is relatively inexpensive (Ref 4).

3.2 <u>Deterioration at the Sliding Elastomer</u>

It has been further estimated that 40% of mechanical seal failures or 28% of unplanned pump breakdowns are due to premature deterioration of the sliding elastomer/counterface. There are a number of causes:-

- Shaft or sleeve fretting
- Sliding elastomer wear
- A build up of leakage deposit on the atmospheric side of the sliding elastomer
- Deterioration of the "O" ring or dynamic packing itself (thermal or chemical effects causing swelling, hardening, compression set).

It is inevitable that mechanical seals will leak (if only small amounts), and this leakage will build up in way of the