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MECHANICAL SEALS IN PROCESS PLANT: A CRITICAL REVIEW OF SEAL LIFE AND SEAL FAILURE

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ABSTRACT

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MECHANICAL SEALS IN PROCESS PLANT:

A CRITICAL REVIEW OF SEAL LIFE AND SEAL FAILURE

by Phillip Allan Conner

Mechanical seals are complex devices which are widely used in process industries to seal shafts in centrifugal pumps. Around ten years ago, the industry first recognised and attempted to improve mechanical seal life and reliability.

Over the past ten years excellent new seal face materials have been developed (eg silicon carbide); bringing about a general improvement in mechanical seal life.

An oil refinery indicates a maintenance cost of £1200 pa (per pump, in 1990 pounds) relates to a mechanical seal MTBF of 1 year. The maintenance cost is 25% of the total cost of operating a seal, if indirect costs are included.

Two oil refineries provide evidence of improving average seal life, when compared to the BHRA survey in the early 1980's. However although nearly 35% of mechanical seals at Plant A survived longer than 30000 hours (only 3% in the BHRA survey), almost 50% still failed within 5000 hours (almost identical to the BHRA survey).

The sealed fluid has a significant effect on mechanical seal life; a 6:1 MTBF ratio for the best and worst sealed fluids.

Weibull analysis shows that the failure distributions are almost identical if a dimensionless life parameter is used, for a wide range of sealed fluids. This indicates that the most significant seal failure mechanisms are common to a wide range of sealed fluids, and only a small subset of the fluid properties significantly affect mechanical seal life.

Long seal lives is extended through better seal face materials, better quality seals, and more complex seals (ie double and tandem seals). Short seal lives are the result of poor pump/seal overhaul procedures. Correct installation, and reducing misalignment and vibration at the seal to a minimum, will substantially reduce the number of short seal lives.

Seal failures can be analysed in many different ways. The delta-T and Duty Parameter provide the best dimensionless groups for predicting performance; checking if duties fall within a good seal operating regime. Statistical methods have an important part to play in establishing the relative importance of a large number of seal operating parameters, classifying seal failures, comparing seal failure distributions, and seal failure modes. There is no doubt that existing statistical methods, given suitable data, can lead to a much better understanding of why mechanical seals fail.

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SYMBOLS and NOTATION

MTBF	Mean lifetime between failures
IMechE	Institute of Mechanical Engineers
ICFS	International Conference on Fluid Sealing
BHRA	British Hydromechanics Research Association
NCSR	National Centre of Systems Reliability
pa	per annum
Pf	Seal face pressure
Pp	Sealed fluid pressure
Psp	Seal spring (compression) pressure
Psf	Sealed fluid pressure
Pmin	Minimum seal face pressure
Tsf	Temperature of the sealed fluid
Tmax	Maximum sealed fluid temperature to avoid vaporisation
Tvap	Sealed fluid vaporisation temperature at Pmin
b	Seal balance ratio
k	Pressure gradient factor (seal face contact)
T[a]	Seal face temperature rise
T[b]	Difference between the sealed fluid temperature and the sealed fluid vaporisation temperature, at Pmin
G	Duty parameter
f	Seal face friction coefficient
V	Mean seal face sliding speed
B	(section 5.5 only) Seal face width
W	Total closing force at the seal faces
Lx	The age when x% of the population has failed
F(t)	Cumulative failure distribution
To	Origin of the failure mode
h	Characteristic life (age when 63% of the population has failed)
t	Age at failure
B	Weibull index (shape factor)
N	Total number of seal failures in a distribution

GLOSSARY OF TERMS

AUXILIARY SEAL SYSTEMS

A collective term to describe a **quench**, **flush**, **recirculation**, **barrier fluid** (double seals), or **cooling** system for a mechanical seal installation.

BALANCE RATIO

The proportion of the **seal chamber** pressure that is applied to the **seal faces**.

BALANCED SEAL

A mechanical seal design in which the seal face and **shaft sleeve** geometry produces a **balance ratio** less than 1 (typically 0.6 - 0.7).

BARRIER FLUID

A non-hazardous fluid injected between the two mechanical seals forming a **double seal**.

COOLING

A cooling system is used to remove heat from the seal faces, especially to avoid **vaporisation**. A liquid from an external source is circulated through a cavity in the **stationary seal**, or another cooling element in the seal chamber.

DOUBLE SEAL

A seal installation where two mechanical seals are installed in series in the same **seal chamber**. The two seals seal in opposite directions. If the **floating faces** are adjacent, the double seal is termed "back-to-back". If the **seats** are adjacent, the double seal is termed "face-to-face". Double seals have been used predominantly on toxic or hazardous fluids, where zero leakage of the **sealed fluid** must be ensured. A **barrier fluid** is supplied to the region between the two seals at a pressure higher than the sealed fluid (ie seal chamber pressure). The second seal prevents excessive leakage of the barrier fluid, and provides a back-up if the first seal fails. The barrier fluid pressure is monitored to provide a warning if the first seal fails.

DYNAMIC (ELASTOMERIC) SECONDARY SEAL

This is the secondary seal used to prevent leakage between the shaft (or **shaft sleeve**) or seal housing, and the **floating seal face**.

FLOATING SEAL FACE

The sprung face of a mechanical seal, which enables limited axial movement: to accommodate wear of the faces, minor shaft misalignment, and shaft end float.

FLUSH

A "clean" liquid injected into the seal chamber, to prevent damage of the seal faces by **sealed fluids** which are corrosive or contain solids. A flush connection can also be used for **cooling** purposes.

HANG-UP

A term to describe the seizure or sticking of a **dynamic (elastomeric) secondary seal**, under the applied spring and hydraulic forces.

IMPELLER BETWEEN BEARINGS

A centrifugal pump design in which the impeller is mounted at the centre of the shaft, supported between bearings. This design provides a stiffer shaft, reduces misalignment, reduces vibration, and is always used on multi-stage pumps. Two mechanical seals are required (or two sets of **double seals**), to seal the shaft on both sides of the impeller.

METAL BELLOWS

A metal bellows is used in place of a spring(s) and a **dynamic (elastomeric) secondary seal**. This type of mechanical seal design is used for high temperature, or high pressure duties. Bellows designs are also used on fluids which are liable to cause **hang-up** or rapid deterioration of a dynamic (elastomeric) secondary seal.

MECHANICAL SEAL

A mechanical device for sealing rotating shafts. It consists of two plane faces perpendicular to the shaft, which form a seal across their radial width. One face rotates with the shaft (the **rotating face**), and the other is fixed in the seal housing (the **stationary face**). One of the faces is flexibly mounted to allow movement in an axial direction (the **floating seal face**).

OVERHUNG IMPELLER

A centrifugal pump design in which the impeller is mounted at one end of the shaft. Only one mechanical seal is required. This design is used on many single stage pumps (ie one impeller). An overhung impeller design has a lower shaft stiffness than an **impeller between bearings** design; causing higher vibration and greater misalignment.

QUENCH

A fluid (often steam) introduced to the atmospheric side of a mechanical seal. A quench is used to prevent crystallisation, icing, or coking.

RECIRCULATION

Mechanical seals may have a recirculation system. Some of the **sealed fluid** is circulated through the seal chamber to provide cooling, and reduce access of solids to the **seal faces**.

ROTATING FACE

The seal face which rotates with the shaft. It may or may not be the **floating seal face**.

ROTATING SEAL

A mechanical seal design in which the rotating face is the **floating seal face**.

SEAL CHAMBER

The space in which the mechanical seal is mounted. The seal chamber is often called the stuffing-box.

SEAL FACES

These form the primary sealing path in a **mechanical seal**. The seal faces refers to both the **rotating face**, and **stationary face**.

SEAL RING

This term has not been used in this study due to some confusion about its definition. A general term for the **seal faces**, or a term to specifically describe the **floating seal face**.

SEALED FLUID

This is specifically the process fluid which the centrifugal pump is designed to pump.

SEAT

The face which is not the **floating seal face**. The seat may be the **stationary face** or **rotating face**.

SECONDARY SEALS

The secondary seals in a mechanical seal prevent leakage along the **rotating face**/shaft path, **seal sleeve**/shaft path (if applicable), and the **stationary face**/seal housing path. The **seal faces** are the primary seal in a mechanical seal.

SHAFT SLEEVE

A sleeve fitted over the shaft to provide a step in the shaft for **balanced seal** geometries, and a replaceable wear-resistant contact area for the dynamic secondary seal.

SINGLE SEAL

A seal design with only one pair of **seal faces**. A single seal may have other seal types (eg bush, lip seal, etc). Single seals are used on non-hazardous fluids, where the effect of leakage is not a threat to health and safety. Environmental legislation will probably preclude the use of single seals on many process sealing duties; due to limits on fugitive emissions and leakage. At present single seals are by far the most common type of mechanical seal design (and the cheapest).

STATIONARY FACE

The face fixed to the seal housing, which does not rotate with the shaft. The stationary face may or may not be the **floating seal face**.

STATIONARY SEAL

A mechanical seal in which the **floating seal face** is the **stationary face**.

TANDEM SEAL

A design where two mechanical seals are mounted in series, and seal in the same direction. An **auxiliary seal system** is required to circulate fluid between the two mechanical seals. However the auxiliary seal system is cheaper than in a **double seal**, because the required fluid pressure (supplied between the mechanical seals) is much lower. A tandem seal provides a "belt-and-braces" sealing arrangement; but the auxiliary system only dilutes leakage of the

sealed fluid through the first mechanical seal. A tandem seal will still produce a very small leakage of the **sealed fluid** (ie diluted) through the second mechanical seal.

UNBALANCED SEAL

A mechanical seal where the seal face and shaft/**shaft sleeve** geometry produce a **balance ratio** greater than or equal to 1.

VAPORISATION

Heat generation due to viscous shear in the fluid film and asperity contact of the seal faces, raises the temperature of the fluid film (ie **sealed fluid**) between the **seal faces**. There is a pressure drop across the seal faces, which lowers the fluid film vaporisation temperature. The temperature rise across the seal faces, due to heat generation, may be great enough to cause the fluid film to achieve its vaporisation temperature. This phenomenon is what is termed "vaporisation". Vaporisation can cause severe seal face damage, due to thermal cracking, thermal shock, and damage through the seal faces slamming together.

1.0 INTRODUCTION

1.1 Objectives

The study concentrates on mechanical seals fitted to centrifugal process pumps in the petrochemical and refining industries, with shaft diameters in the range 40-100 mm. The study has been carried out to fulfil the following objectives.

(a) Use statistical and analytical methods to predict seal life with greater confidence and accuracy.

(b) Quantify the cost of operating mechanical seals, in terms of maintenance running costs and costs incurred through poor mechanical seal reliability.

(c) Relate seal failure modes to seal life data.

(d) Present, discuss, and interpret new (unpublished) data from two large U.K. process plants (a refinery/petrochemical plant, and a refinery). These plants are referred to as Plant A and Plant B in the text.

1.2 Background

Rotary shaft sealing may be accomplished using packing, labyrinths, lip seals, and many other more exotic seals (fig 1.1). However for high integrity sealing on arduous duties the mechanical seal has no peer. A mechanical seal is capable of sealing gases and liquids under static or dynamic conditions at extremely low leakage rates. The mechanical seal is unique in being capable of sealing simultaneously, high pressure, extremes of temperature, and high shaft velocity. In addition a mechanical seal requires no maintenance under normal operation, since wear is accommodated automatically.

A mechanical seal consists of two plane faces perpendicular to the shaft, which form a seal across their radial width. One seal face rotates with the shaft. The other face is stationary and fixed to the machine's casing. One face is sprung axially to accommodate wear and minor misalignment of the seal faces. Elastomeric seals (eg 'o'rings) prevent leakage along the secondary leakage paths.

The majority of process pumps in the refining and petrochemical industries have shaft diameters greater than 40mm. Plant failure data suggests that 60-75% (23) of all maintenance costs are attributable to mechanical seal failure. In recent years plant operators have taken a strong interest in the performance of mechanical seals. Most pumping duties have a spare pump to avoid loss of throughput when a failure occurs. As a result most mechanical seals are probably run for 50-60% of their installed life. Even in industries where quality assurance is vigorously applied (eg Nuclear Industry), unexplained premature and random mechanical seal failures occur. In terms of sealing efficiency and versatility of application, mechanical seals represent a significant improvement over any of the other sealing techniques. Mechanical seals are capable of operating efficiently for several years without any maintenance. However despite their almost universal use for sealing shafts in the process industry, they appear to suffer a high incidence of premature failures. The mean time between mechanical seal failures is 8-13 months (29)(installed life). This is well below the life expected of rolling element bearings, which are the other main element of a pump to cause an outage. So although mechanical seals are the accepted way of sealing rotating shafts on process pumps, they are unreliable and have a short life compared to other pump elements.

1.3 Format of the Study

The author was employed by a large oil company (8/87 to 8/88) at a U.K. oil refinery, and became involved in a Total Quality Management project in January 1988. The aim of this project was to reduce refinery operating costs by targeting efforts to improve the least reliable operations. Centrifugal process pumps were found to be the highest priority, confirming the general trend in the process industry. Study of maintenance records soon revealed that mechanical seal failure was the most common cause of pump failure (followed by bearing failure). The author compiled a computer database covering the period 1969-1988, of pump duties and corresponding seal lives. This database contains about 300 pumps and 1200 mechanical seal failures. The database was used to group mechanical seals by duty features (ie generically), and target the refinery's efforts at those groups of seals with the shortest life.

The same oil company provided support for the author to carry out further studies of mechanical seal performance at Southampton University. In addition to the mechanical seal database, details of actual plant costs were made available so that an analysis of mechanical seal operating costs could be performed. This information and refinery experience formed the basis of the author's BEng thesis (7). The thesis provided some preliminary work and data for this study.

In the late 1970's and early 1980's the significance of mechanical seal failure on maintenance costs was being recognised, and causing serious concern. The IMechE held seminars during 1983/84 which culminated in a handbook (29) for engineers being published, based upon data gathered from seal users, seal manufacturers, pump manufacturers, and research bodies. The BHRA carried out two major surveys (23,24,25) of mechanical seal performance in process plant. These three references constitute the main sources of published data on mechanical seal performance in process plant. In recent years many plants have begun to keep detailed records on mechanical seal failures, but at present very little of this data is published and remains largely inaccessible. The database at the National Centre of Systems Reliability was able to provide very little operational data on mechanical seal failures (21).

In November 1989 the author began this study, to carry out a more comprehensive appraisal of mechanical seals in the process industry. A large new mechanical seal database was made available to the author. This database contains mechanical seal failure data from over 1200 centrifugal process pumps, with over 3000 seal failures, for the period October 1986 to December 1989. This large database has provided the opportunity to carry out a good statistical analysis of mechanical seal performance, to determine relationships between life, seal design, and operating conditions.

1.4 Format of this Report

The report uses real data on mechanical seal life and reported seal failures, at two large process plants in the U.K. These plants are referred to as Plant A and Plant B, throughout the report. Plant B is the refinery at which the author worked during 1987/1988. Plant A is a large refinery and petrochemical plant. The mechanical seal data from Plant A and Plant B has not been published before.

Chapter 2 puts mechanical seals into a historical context, provides a broad overview of mechanical seal development, and describes some of the literature which has been published on the subject of mechanical seal life and reliability.

Chapter 3 investigates the costs associated with mechanical seals fitted to centrifugal process pumps. Direct maintenance costs and indirect costs associated with operating mechanical seals, are discussed in detail. The direct and indirect costs are quantified in a case study, using real data from Plant B. The case study quantifies relationships between seal life, seal type (ie single seals, double seals), direct costs, and indirect costs.

Chapter 4 discusses failure mechanisms associated with mechanical seals. Mechanisms are divided into those internal to the seal, and those external to the seal. A case study using field data from Plant A reveals the most common failure mechanisms in reality, and refers to the characteristics (also see chapter 9) of the mechanical seal failure distribution from Plant A. The majority of mechanical seal failures are due to mechanisms associated with the seal faces. Seal face materials are described in detail in the context of their material properties, historical development, and mechanical sealing properties (for various material combinations). The mechanical, thermal, and chemical failure mechanisms associated with the seal face materials are listed in detail.

Chapter 5 assesses the way that dimensionless groups have been used in the analysis of mechanical seal data; to improve the understanding of mechanical seal behaviour, and extend mechanical seal life through better selection methods. Five dimensionless groups are discussed in detail. The conclusions from studies using these groups, are used to evaluate whether these dimensionless groups are relevant or useful to the analysis of mechanical seal data; with the purpose of increasing life and reliability.

Chapter 6 assesses the value of statistical methods, along similar lines as chapter 5. Statistical measures of seal life (eg. mean time between failure) are frequently used, but they are examined in detail to assess their scope, value, and limitations. Regression analysis, weibull analysis, and discriminant functions are explored in detail, including references to other studies of mechanical seal data using these statistical methods.

Chapter 7 discusses the considerations for collecting field data on mechanical seals. There is a discussion of the relevant types of information that should be recorded in a mechanical seal database, the impact of inaccurate or missing data, and the types of database that can be established.

Chapter 8 is the "key" chapter in the report. Data from Plant A and Plant B is used extensively. Chapter 8 identifies ways of improving mechanical seal life, and mechanical seal reliability. The cost benefits of bringing about this change are discussed and quantified. Weibull analysis is used to establish relationships between seal life, cause of failure, seal face materials, and the sealed fluid. The most suitable (ie for improving mechanical seal life) dimensionless groups, statistical methods, and data for a mechanical seal database, are defined. The way in which these methods can be applied to improve mechanical seal life is explained.

Chapter 9 provides case studies to assess the life of mechanical seals in refineries and petrochemical process plants. This chapter is aimed at providing a "benchmark" to establish the existing life and reliability of mechanical seals in process plants. There is a description of the two new data sources presented in this report (ie Plant A and Plant B). This mechanical seal data is compared with the data presented in the BHRA survey (23), published in 1987.

Chapter 10 draws together concluding comments based upon key points in this report.

Chapter 11 indicates the priorities for further work.

Appendix A1 contains weibull plots of mechanical seals on seven different sealed fluids at Plant B. Appendix A2 contains weibull plots of mechanical seals with six different seal face materal combinations at Plant A. These appendices are referred to and discussed in Chapter 8.

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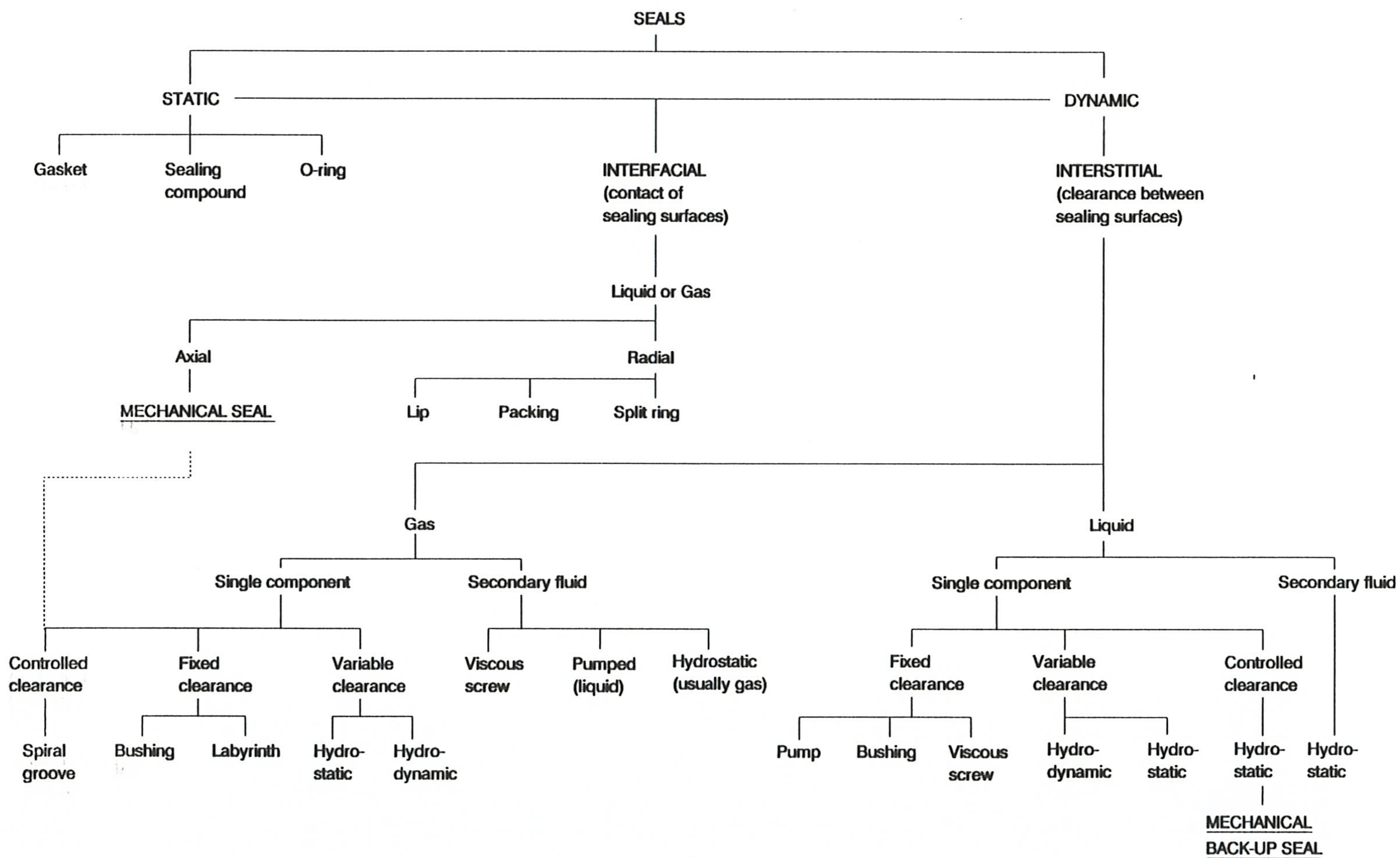


Fig 1.1 : Seal Spectrum

2.0 HISTORY

The commercial development of mechanical seals began after the Second World War. Mechanical seals have virtually replaced gland packing as the usual type of seal on rotating shafts in centrifugal pumps and compressors. The Second World War generated rapid developments in materials technology. After the War materials with suitable properties (low friction, low wear, corrosion and heat resistance) became cheap enough to make mechanical seals a viable alternative to gland packing. Material Technology has continued to advance rapidly and mechanical seal manufacturers have always looked for new materials to improve the life, range of operating duties, and reliability of mechanical seals.

The principle elements which form a mechanical seal have not changed significantly from the first mechanical seals. However the detailed design has advanced considerably with the aid of powerful computer programs to analyse the fluid dynamics and thermodynamics inside the seal. Research on test rigs and field studies also input into the development of new mechanical seals. Computer controlled manufacture (CNC) has improved the quality and tolerances on mechanical seal and pump components. This has generally reduced the level of vibration and misalignment at the mechanical seal.

Despite their widespread use there has been concern about the life and reliability of mechanical seals in service. Mechanical seals are capable (and some do!) of operating without any maintenance for several years, even on severe duties. However there is a very high incidence of premature failures, which results in seal behaviour which is unpredictable. It is very common for a pump to have mechanical seals of identical specification, which have lasted anything between 10 days and 4 years. The IMechE(7) consensus was that the mean time between mechanical seal failures is 8 to 13 months. This is much shorter than the life expected of rolling element bearings, and significantly shorter than a normal period between shutdowns. Mechanical seal life and reliability represents a significant penalty in terms of downtime, spare equipment, maintenance cost, and potential hazards from leakage. As operating margins have fallen, the cost of new capital equipment (eg spare pumps) and maintenance has become increasingly important.

Many process plants are now collecting mechanical seal failure data to enable a better evaluation of seal performance. Very little of this work has been published so actual operating data on mechanical seals is still scarce (21,23,24,30,36). The IMechE seminars (29) concluded that seal manufacturers, pump manufacturers, and seal operators, regarded the exploitation of existing operating knowledge the best way to understand seal failures; rather than further research into seal behaviour. Clearly this has not happened in the most effective way, since plants have exploited their own operating knowledge rather than "pooling" their operating experience. This situation has led to little improvement in the overall understanding of factors affecting mechanical seal performance, in the past 10 years. The industry has tackled the whole problem in a "piecemeal" way, by simply trying to improve seals with the shortest lives. If all the existing plant operating data was combined into a single accessible database, statistical and analytical methods could be applied much more effectively, to identify and quantify the factors affecting mechanical seal life and reliability.

3.0 COSTS ASSOCIATED WITH MECHANICAL SEALS FITTED TO CENTRIFUGAL PROCESS PUMPS

3.1 Cost Definitions

Mechanical seal costs can be divided into two main categories. Direct costs such as maintenance and initial capital cost form the first category. Indirect costs resulting from the limited life of mechanical seals form the second category.

Direct cost = [component cost] + [labour cost]

Indirect cost = [lost throughput] + [standby equipment cost] + [inventory value]

This chapter looks in detail at the various elements which contribute to the direct and indirect costs of operating mechanical seals in centrifugal process pumps. Costs are quantified from data relating to shaft diameters in the range 40-100mm at Plant B (an oil refinery). All the cost data is based upon 1990 values.

3.2 Direct Operating Cost

3.2.1 Mechanical Seal Maintenance - Component Cost

The initial capital cost of the seal is relevant since a correctly specified and correctly installed expensive high quality seal will usually have a longer life than a cheap low quality seal (similarly specified and installed). Seal faces and elastomeric secondary seals should always be replaced when carrying out mechanical seal maintenance. The other mechanical seal components need only be replaced if worn, broken, or corroded.

3.2.2 Refurbishment of Seal Components

Many mechanical seal failures are due to damaged seal faces, or the build-up of hard deposits which restrict the movement of the floating seal face (ie seal hang-up). Experience shows that at least 20% of failed mechanical seals are suitable for refurbishment. The mechanical seal is dismantled and all the components cleaned and polished. The seal faces are ground to remove face damage (eg chips, scratches, and grooves) and regain their flatness. A series of lapping operations are needed to satisfy the original specifications for flatness and surface finish. The mechanical seal is always reassembled with new elastomeric secondary seals. Refurbishment of the seal faces is a specialist job requiring high precision lapping plates. Seal manufacturers and a few specialist firms are best suited to carrying out seal refurbishment. The cost of refurbishing a mechanical seal to its original specification is about 25% of the cost of a new seal. Properly refurbished mechanical seals are as good as new seals.

3.2.3 Mechanical Seal Maintenance - Labour Cost

When a centrifugal pump fails due to mechanical seal failure alone, the man-hours required to gain access and overhaul the mechanical seal form the labour cost. The man-hours will cover pump isolation, pump removal, pump stripdown, and then a refit with a new mechanical seal. Experience (Table 3.1) suggests that less than 20% of the labour cost is incurred during the pump stripdown and refit operations. Most of the labour cost is associated with taking the pump off-line so that it can be worked on. Around 30% of the labour cost is related to safely taking the pump off-line, and around 50% during the reinstatement of the pump. If a pump is already stripped down for another reason, the labour cost to replace the mechanical seals is negligible.

Pump size has little effect on the labour cost. The type of labour can significantly affect the labour cost. Contract labour will generally take considerably longer to do the same job than dedicated maintenance teams employed by the plant, because they will have to follow more lengthy procedures to obtain equipment (eg cranes) and permits to work.

3.3 Indirect Operating Cost

3.3.1 Lost Production

Spare or standby pumps are often installed in the process industry, because of the very high costs associated with reduced throughput and lost production. In many plants (eg refineries) the individual process units are linked so that a loss of throughput in one unit will have a "knock-on" effect which affects the whole plant. This is particularly true in modern integrated refineries.

Mechanical seals rarely fail without some warning. Visual observation is the basic (and often only) form of seal leakage detection. On hazardous or toxic duties where a double seal is used, seal failure can be detected by monitoring the barrier fluid pressure. In the majority of cases seal failure is detected quickly enough to switch to a standby pump with little or no loss of throughput. The greatest chance of lost production occurs when a standby pump is run-up after several months standing idle. It is not uncommon for the mechanical seal to set solid, especially on heavy hydrocarbon duties. Hard deposits can easily form unless careful priming and shutdown procedures are employed.

3.3.2 Standby Equipment

Mechanical seals are the major cause of centrifugal pump outage in the process industry (23). The bearings and other pump elements also cause pump outage, so standby pumps will always be installed on the most critical duties. Bearings which are the second most common cause of pump outage, have L10 lives (ie 10% failed) in excess of 10,000 hours (1.14 years). If the life and reliability of mechanical seals could approach the performance of rolling element bearings, then it would probably be economical to not install standby pumps on most process duties (ie the cost of lost production due to the small number of pump failures would be less than the cost of installing standby equipment).

Installation of standby equipment is very expensive. This equipment consists of the pump, motor, plinth, valves, pipework, electrical distribution, MCC (motor control centre), circuit breaker, etc. There is a considerable cost associated with installing this equipment. Cost savings would be achieved through smaller overall plant size, if less standby equipment was required.

At present the L10 life of mechanical seals is very low (10-50 days), and the mean time between failures lies in the range 8-13 months (29). These installed lives are typically twice the running life, since standby and duty pumps are usually run alternately. A considerable improvement in mechanical seal life is necessary to make it economical to not install standby pumps on most process duties.

The real cost of standby equipment is the loss of revenue that could have been obtained if the same capital had been invested elsewhere. This type of equipment is often written-off financially, linearly over a ten year period.

3.3.3 Inventory

A large number of seal types and material variations are required to seal the wide range of duties found in the process industry. Unpredictable life and the risk of sudden failures has led many plant to stock spare mechanical seal components. The inventory becomes large due to the wide range of seal sizes and duties found in most process plants.

An inventory represents a capital investment, in financial terms. Like standby equipment (3.3.2) the real inventory cost is the loss of revenue that could have been obtained if the same capital had been invested elsewhere. A large inventory is financially undesirable, but has been necessary to avoid the risk of lost production through mechanical seal failure. This "catch-22" situation can be overcome by using consignment stock. Consignment stock is the name given to items held in the plant inventory, but not paid for until they are actually used. This arrangement is becoming increasingly common, with the concept of "preferred suppliers" being used by many large companies to ensure the quality of supplies.

3.3.4 Health, Safety, and the Environment

The process plant environment may contain many potential health and safety hazards. In most types of process plant (refineries and chemical plants) very stringent safety procedures are laid down by law and codes of practice. Maintenance tasks take longer because strict work procedures and "permits to work" are required.

Many process fluids are hazardous or toxic. Double mechanical seals, with a barrier fluid circulating within the seal, can ensure that there is no leakage of the sealed product to the outside environment. Double seals are considerably more expensive (initial cost) than single seals. However the seal cost is secondary to the health and safety of people (in case of seal failure), if the sealed product is toxic or hazardous. At present there is no specific legislation to control the emissions and leakage of process fluids from mechanical seals, into the environment. Growing public concern over the leakage and emissions from industrial plants, is causing political pressure which will probably result in new legislation to limit the leakage and emission from all sources. The performance of a mechanical seal is almost always measured against leakage in the liquid phase. Many single mechanical seals with very low liquid leakage rates have significant leakage in the vapour phase. Tougher environmental legislation will require reductions in both liquid and vapour leakage. Using current technology the most obvious way to meet tougher environmental legislation will be to install double seals. A cheaper alternative would be to install "backup" seals, which provide a limited sealing life when the primary seal fails. A backup seal will not reduce the level of product vapour emissions like a double seal. The relative value of double seals or backup seals will depend on the actual emission and leakage constraints set out in the new environmental legislation. A considerable cost will be incurred by the process industry to bring centrifugal pump seal specifications up to the standard required to meet tougher environmental legislation.

3.4 Case Study - Mechanical Seal Costs At Plant B

3.4.1 Introduction

The case study quantifies both the direct and indirect costs associated with operating mechanical seals in 700 centrifugal process pumps, at a U.K. oil refinery (Plant B). Fifteen pump installations were studied in detail, covering a wide range of duties including water, heavy gas oil, naphtha, and LPG. The fifteen pumps are all of an overhung impeller design (one mechanical seal per pump). Over 90% of the pumps at Plant B are of this design. Four different makes of seal had been installed on these pumps, and seal failures covered the period 1969 to 1987. Workshop records gave details of which mechanical seal components were renewed after each failure.

Mechanical seals at Plant B fall into three main categories:

Group 1 - Single seals with a sprung face and elastomeric secondary seals. These seals have the lowest component cost and tend to be used on the least severe hydrocarbon and water duties.

Group 2 - Single seals with a metal bellows secondary seal. These seals are mandatory on duties with sealing temperatures in excess of 200 deg.C. They are also used on the more severe hydrocarbon duties or where elastomeric seals are chemically incompatible with the sealed product. Group 2 are more expensive than group 1 seals.

Group 3 - Double seals with a barrier fluid system. These seals are used on toxic and hazardous duties such as LPG and hydrofluoric acid. Group 3 seals are the most expensive, due to the complexity of the seal itself and the cost of the barrier fluid system.

3.4.2 Analysis Of Mechanical Seal Operating Costs At Plant B

Seal Group

Pumps operating with group 1 seals experienced a wide range of MTBFs. The direct seal operating costs vary similarly (fig 3.4). As expected most group 1 seals are cheaper (30-50%) to operate than group 2/3 seals, since they have the lowest component cost; the primary reason for their selection! The study also demonstrates some pump installations in which a group 1 seal is clearly not suitable (very short seal life). The direct seal operating costs on these duties is upto 50% higher than the typical cost for a group2/3 seal.

The sample of pumps operating with group 2 and group 3 seals was small. These pumps showed a small range of MTBFs and associated direct operating costs (fig 3.4). This suggests that group 2/3 seals operate more predictably in terms of life and direct operating costs, than group 1 seals.

There is a cost benefit in optimising seal group selections. Although the more expensive seals (group 2/3) behave more consistently, it is much cheaper to select a group 1 seal if possible. However, if a group 1 seal is poorly chosen or not suitable, then the seal operating costs may be 50% higher than if a more reliable and expensive group 2/3 seal had been chosen.

Maintenance - Labour Cost (see 3.2.2)

The maintenance labour cost is inversely proportional to mechanical seal life. Seal type has very little influence since over 90% of the labour cost (table 3.1) is generated in gaining access to the mechanical seal. The mechanical seal labour cost is generally independent of pump size.

JOB STEP	DESCRIPTION	MAN-HOURS
1	Obtain a permit to work	2
2	Isolate electricity and processes	6
3	Disconnect the pump	4
4	Remove the pump to the workshop	2
5	Strip-down the pump	6
6	Fit new seals and rebuild the pump	4
7	Transport pump to site	2
8	Reconnect pump	5
9	Align the pump	4
10	De-isolate electricity and processes	6

Table 3.1

Table 3.1 indicates the typical job steps and man-hours involved in a mechanical seal overhaul on a centrifugal process pump at Plant B. The labour rate would be about £11.50 per hour, including basic, overtime, and National Insurance. So a typical mechanical seal overhaul costs about £470 (1990). The inverse proportional relationship between labour cost and mechanical seal life (MTBF) is shown in fig.3.1. The empirical relationship is approximately:

Mechanical seal labour cost (per pump) = (470*365) /MTBF
[£ pa] [1990] [days]

Mechanical seals at Plant B have an overall MTBF of 474 days, so the average seal labour cost (per pump) is about £360 pa using the relationship above.

Maintenance - Component Cost

Mechanical seal component costs are inversely proportional to seal life (MTBF) (fig 3.2). Generally component costs are greater than labour costs in mechanical seal maintenance (fig 3.3). The empirical relationships are approximately:

Mechanical seal component cost (per pump) = (700*365) /MTBF
[£ pa] [1990] [days]

and,

Mechanical seal component cost = 1.5 * Mechanical seal labour cost

Mechanical seals at Plant B have an overall MTBF of 474 days, so the average cost of renewed mechanical seal components (per pump) is £540 pa using the relationships above. The component cost assumes that 20% of the seal components were refurbished, at 25% of the cost of a new component.

Lost Production or Reduced Throughput

Standby equipment is installed on all the critical process pumping duties at Plant B. Lost production due to mechanical seal failure is negligible. The value of product lost through emission and leakage is not quantified.

Standby Equipment

There are about 100 standby centrifugal process pumps at Plant B. The typical cost of installing a pump at an oil refinery is between £50,000 and £100,000 for a motor driven type. The costs associated with the electrical distribution work are very high, typically £40,000 for the hardware and labour costs. The cost of the electric motor, pump, and pipework tie-ins is additional. A medium sized pump and electric motor would cost around £10,000 each. These costs are written off over a 10 year period.

If each standby pump represents an initial capital cost of £75,000, Plant B has invested £7.5M. This investment is spread over 10 years. So the annual cost is £750,000. Since around 70% of centrifugal pump outage is due to mechanical seal failure, this proportion of the standby equipment costs (£525,000) shall be attributed to the indirect mechanical seal operating costs. This represents an average indirect seal operating cost of £750 pa(1990) when divided between the 700 process pumps at Plant B. The true cost is higher, since the £7.5M could have been invested at an average growth rate (above inflation) of around 7% pa. Over 10 years this investment would have resulted in a capital growth to around £14.7M. Due to the need for standby equipment this investment potential is lost. So the true cost of the standby equipment (per pump) is probably nearer £1500 pa.

Mechanical Seal Inventory

Plant B held a substantial inventory of mechanical seal components on site, to maintain the 700 centrifugal pumps operating with mechanical seals. The capital value of the mechanical seal inventory was £250,000 (1990). The plant did not operate a consignment stock arrangement at the time of collecting these costs. The inventory represents an indirect mechanical seal operating cost (per pump) of £360 pa.

3.4.3 Summary of the Direct and Indirect Mechanical Seal Operating Costs At Plant B

The costs set out below are based upon an oil refinery with 700 centrifugal process pumps (100 standby), at 1990 prices.

Direct cost -		
	maintenance - labour	250,000
	- components	380,000
Indirect cost -		
	lost production	0
	standby equipment	1,000,000
	inventory	250,000
		£ 1,880,000 pa

Average total mechanical seal operating cost (per pump) £ **2,690** pa

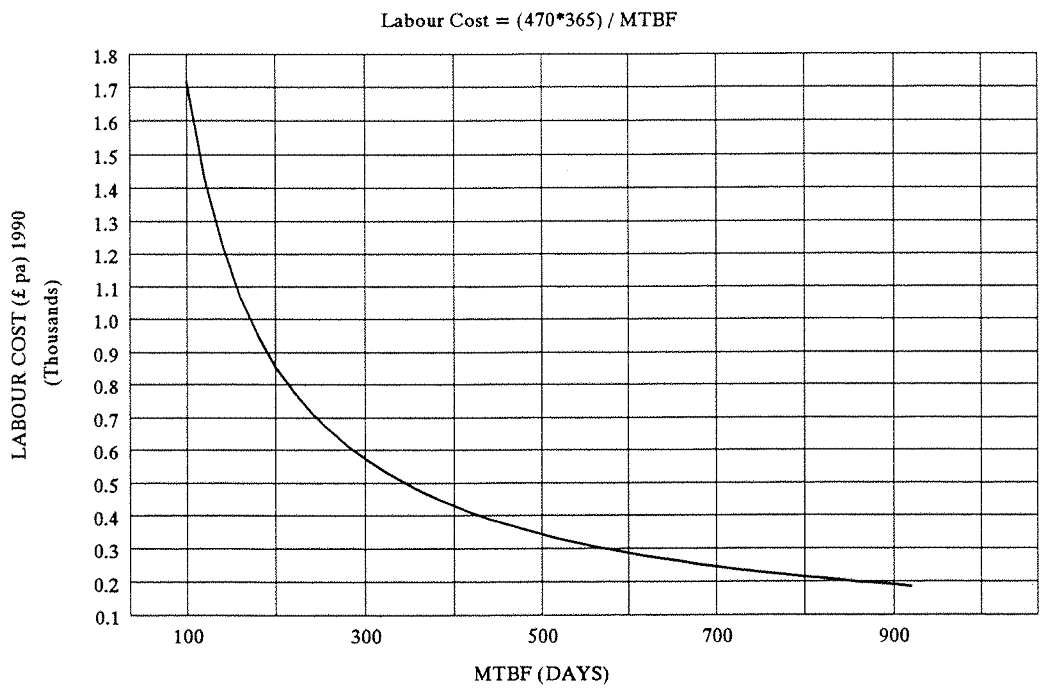


Figure 3.1 : Seal Labour Cost (per pump) v Seal Life, at Plant B

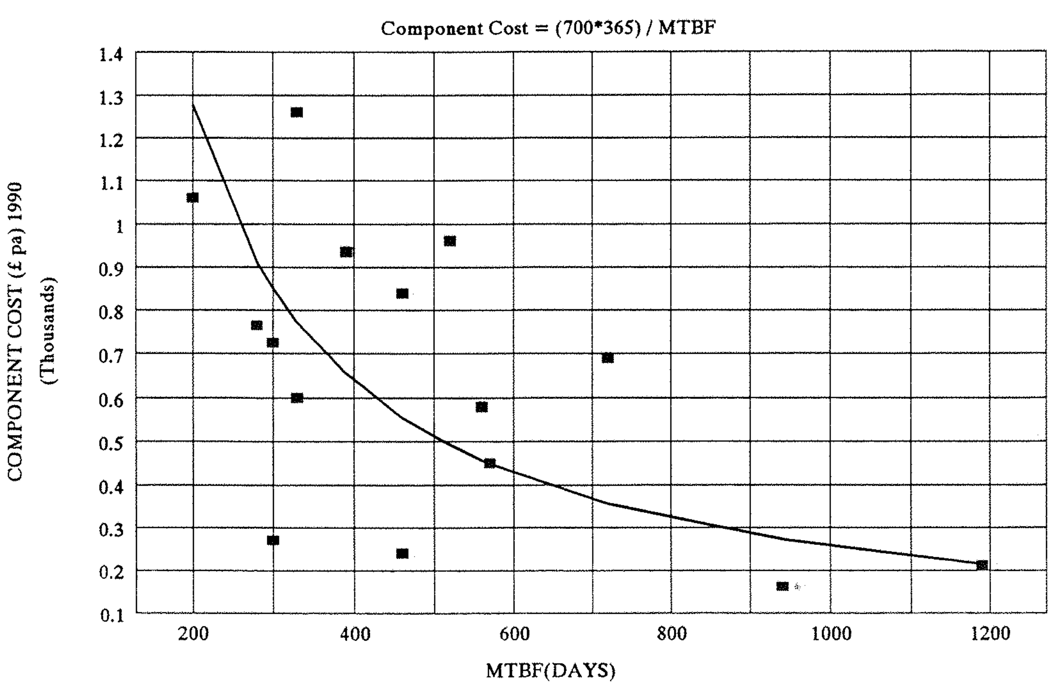


Figure 3.2 : Seal Component Cost (per pump) v Seal Life, at Plant B

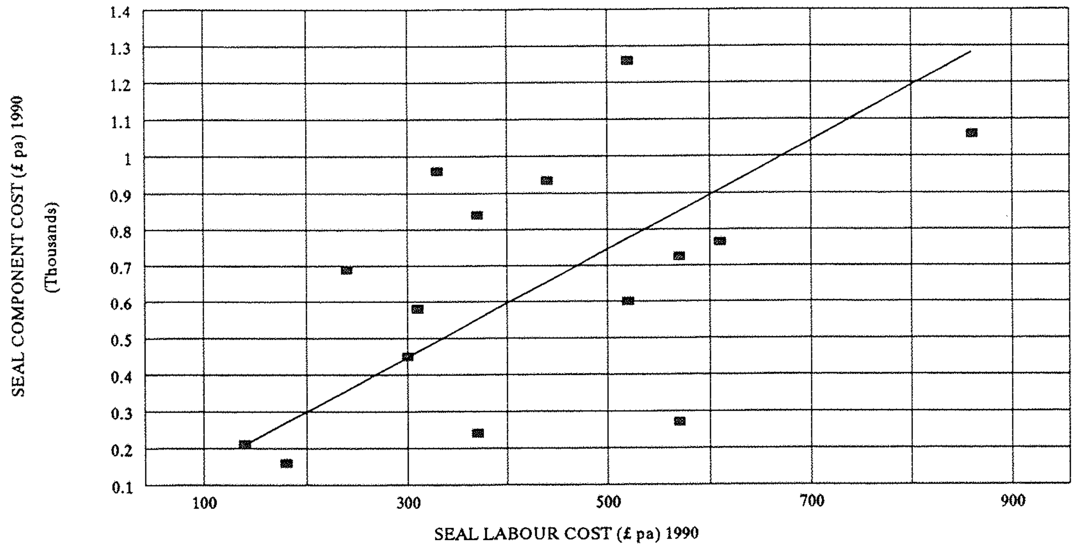


Figure 3.3 : Seal Component Cost (per pump) v Seal Labour Cost (per pump), at Plant B

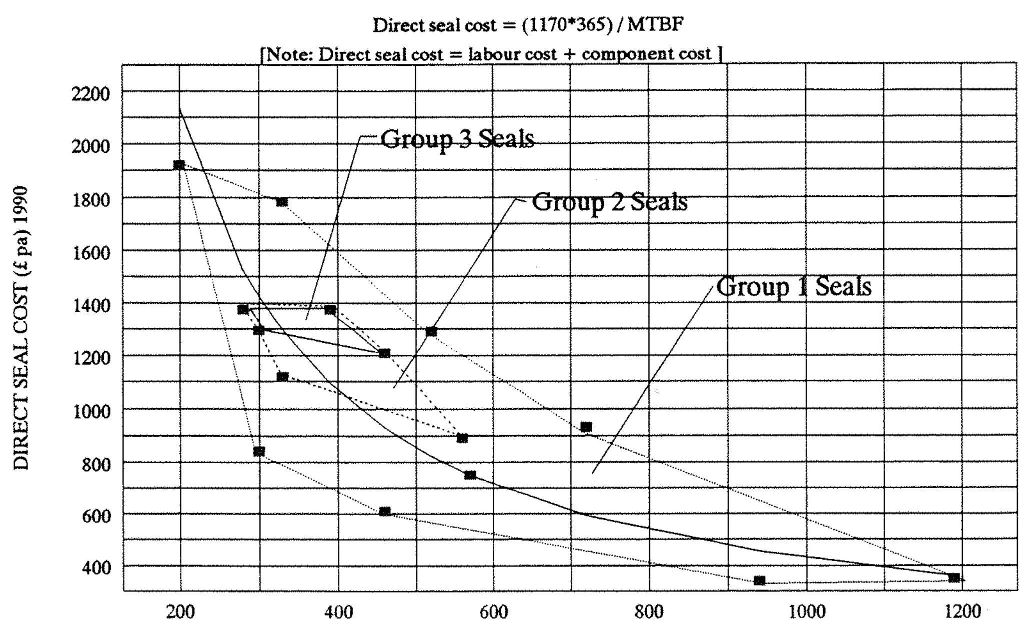


Figure 3.4 : Direct Seal Cost (per pump) v Seal Life, at Plant B

4.0 FAILURE MECHANISMS IN MECHANICAL SEALS

4.1 Introduction

Mechanical seal failure is a complex process. Failure is not an absolute term since failure is usually defined as "unacceptable" leakage. There are a wide range of failure mechanisms, due to the sensitivity of mechanical seals to adverse operating conditions. The mechanisms can be grouped into failure mechanisms internal to the seal, and failure mechanisms external to the seal. External failure mechanisms generally result from operator errors or malfunction of another element in the pump or plant. The most significant internal failure mechanisms relate to the seal face materials.

4.2 Definition Of Mechanical Seal Failure

Seal failure is most commonly defined in terms of leakage rate. All shaft seals leak a small amount of liquid or vapour (providing there is a pressure differential across the faces). This indicates that a thin fluid film exists between the seal faces which is necessary for good mechanical seal life. A seal fails when the leakage rate becomes excessive. The maximum acceptable leakage rate may depend on the cost of leaked product, hazard to people and the environment, danger of ignition, necessity to keep pump on-line, odour, or plant appearance.

4.3 Failure Distributions

A failure distribution expresses failure rate against element life. The three fundamental failure modes (fig 4.1) are premature failure, random failure, and wearout failure. A failure distribution will indicate the general modes of failure of a mechanism. Most mechanical systems have the characteristic "bathtub" distribution (fig 4.2). The bathtub curve is made up of the three fundamental failure modes:

(1) **Premature failure.** A falling failure rate and short lives. There are always some premature failures on mechanical systems due to installation errors, start-up errors, and damage prior to installation.

(2) **Random failure.** A steady failure rate upto the design life. A low random failure rate is inevitable as a consequence of external system failures, human error, or inadequate quality control during manufacture.

(3) **Wearout failure.** A rising failure rate after the design life has been exceeded. The failure rate increases as the system reaches its design life, and the parts of the system wear excessively, fail due to fatigue, or creep excessively.

In a reliable system strict handling, installation, and start-up procedures keep premature failures to a minimum. Some random failures are preventable through better training of system operators, and better control systems on pumps which ensure the mechanical seal is operated correctly (ie cooling, flush, and quench systems) when starting and stopping the pump. In a reliable system the majority of failures would occur through wearout, some time after the design life had been achieved. If maintenance is planned so that systems are renewed when their design life is exceeded (ie through a planned maintenance programme), then very few wearout failures will occur.

Failure distributions (see fig 9.1, 9.2, and 9.3) suggest that the majority of mechanical seals fail prematurely. Very few mechanical seals fail in a wearout mode so it is difficult to plan mechanical seal maintenance. Improved mechanical seal life and reliability will be achieved by tackling the causes of premature and random failure modes.

4.4 Types of Mechanical Seal Failure

4.4.1 Internal Failure Mechanisms

This type of failure occurs as a direct result of the seal selection or specification. Failures of this kind are listed below.

(1) Material problems (see 4.5)
- not compatible with the sealed fluid.
- not suited to the operating conditions (pressure, temperature, or shaft velocity).
- inadequate thermal shock resistance.

(2) Seal instability
- in balanced seals a different balance ratio or spring rating may remedy this problem (see 5.2).
- excessive heat generation due to the wrong seal face materials, which leads to vaporisation of the fluid film.

(3) Seal quality and design
- insufficient cooling, or easily blocked cooling lines.
- poor manufacturing tolerances.
- poor quality or defective materials.

(4) Wearout
- the most desirable type of failure if it occurs with long seal life.

(5) Seal hang-up (seizure of the floating seal face)
- incorrect secondary seal specification.
- wrong seal type.
- inadequate auxiliary seal lines (ie quench, cooling, etc).

There is strong operational evidence (8,23) to suggest that poor selection accounts for a high proportion of mechanical seal failures in the process industry, and wearout with long life is uncommon. All the internal failure modes listed above are premature failure modes (except wearout, if it is accompanied with long life).

4.4.2 External Failure Mechanisms

This type of failure occurs as a result of an operating malfunction which puts the seal under conditions for which it was not designed. Typical failures of this kind are listed below.

(1) Excessive vibration
- poor pump alignment.
- poor condition of the pump bearings.
- poor balancing of the shaft and impeller.

(2) Installation errors
- seal not fitted correctly.
- damage to the seal.

(3) Handling damage
- seal not packaged and stored correctly.

(4) Operator error
- wrong pump startup and shutdown procedures (ie failure to check the correct auxiliary seal line conditions), causing periods of dry-running.

(5) External component or system failure
- bearing failure .
- pressure surge.
- excessive temperature of sealed product.
- pump cavitation.

Failures due to external component or system failures are random failure mechanisms. Failures resulting from installation, handling, or operator errors are premature failure mechanisms. Improved training, skill, and care of the maintenance workforce should prevent many of these premature failures. Handling and damage to seals in storage can account for a surprisingly large number of premature mechanical seal failures (26).

4.4.3 <u>Case Study - Seal Failure Mechanisms at Plant A</u>

This section presents the results of a study into mechanical seal failure mechanisms at a large U.K.petrochemical plant (Plant A). Plant A is described more fully in section 9.1. Plant A has a special group which collects mechanical seal failure data. Table 4.1 summarises the causes of 160 mechanical seal failures at Plant A which occurred during the period January 1988 to March 1990, and provide a good insight into the failure mechanisms behind the seal life distribution (see fig 9.2). All the seal lives are measured as installed life.

The seal life distribution (fig 9.2) clearly shows that premature failure mechanisms account for most failures of mechanical seals with a life below 10,000 hours. So about 60% of the mechanical seals at Plant A suffer premature failure. Table 4.1 confirms that the majority of the 160 mechanical seal failures were caused by premature failure mechanisms.

Random failure mechanisms account for most failures of mechanical seals with lives between 10,000 and 40,000 hours. The failure rate is very low (about 0.6% per 1000 hours). The life distribution (fig 9.2) suggests that around 6% of the mechanical seals at Plant A fail due to a random failure mechanism. This is confirmed by table 4.1.

The life distribution (fig 9.2) shows that over 30% of the seals had installed lives in excess of 24,000 hours. In the limited period of the study only 1% of the mechanical seals had appeared to fail in a long life wearout mode (table 4.1). The life distribution does not show when the wearout failure mechanism replaces the random failure mechanism - a longer study period is necessary (probably 40,000 hours).

4.5 Seal Face Materials and Their Modes of Failure

Mechanical seal face materials are of primary importance. At least two-thirds of the failures due to mechanisms internal to the seal are associated with the seal face materials. Flitney and Nau (23) have studied secondary seal failure mechanisms. This study will concentrate on the seal face materials and their modes of failure.

4.5.1 Properties of Mechanical Seal Face Materials

Introduction

The properties of mechanical seal face materials are particularly important because the seal faces operate in a mixed lubrication mode under normal operating conditions. The load at the seal faces is supported partly by a very thin fluid film, and partly through solid contact. The properties of the seal face materials profoundly affects the wear rate, deterioration of the faces through solid contact, and the formation of a thin fluid film between the faces. Mechanical seal life is highly dependent on the seal face materials. The best materials provide low wear rates, high heat dissipation, good corrosion resistance, and low seal face distortion (under high pressure and temperature). Table 4.2 summarises the properties of seal face materials used in the process industry.

Common Material Strategies

The stationary and rotary faces are not usually the same material. Generally a hard material (eg silicon or tungsten carbide) is paired with a soft material (eg carbon-graphite). This type of combination has good dry-running properties, excellent thermal shock resistance, good heat dissipation, and good wear characteristics (except in the presence of abrasives). A hard-hard material pair (eg silicon carbide and tungsten carbide) is the best combination to use if the sealed fluid contains abrasive solids. Soft materials are not usually paired.

Material Development

Mechanical seal faces have reflected the state of the art in material technology since the 1940's. Materials developed during the Second World War made mechanical seals commercially viable.

(a) Metal Alloys

The development of alloy cast irons (Ni-Resist) and hard alloys (Stellite) improved the chemical and wear resistance of mechanical seal faces. These materials could be used on a wide range of sealing duties. To reduce the cost of the faces, hard alloys were normally applied as a thin coating onto a stainless steel base. Unfortunately these hard alloys have poor heat dissipation properties so that dry-running or boundary lubrication conditions rapidly lead to seal failure. Stellite is normally paired with metallised carbon-graphite. Ni-Resist cast iron is normally paired with resin impregnated carbon-graphite.

(b) Carbon Composites

Carbon composites have had an enormous impact on seal face materials. Carbon-graphite composites are generally the first choice for one of the seal faces. A base grade formed by mixing carbon with natural or artificial graphite is held together with a pitch or resin binder. The mixture is baked at around 1000 degC and impregnated with resin or a metal (eg antimony), to form an impermeable material. Graphite gives the composite good self-lubricating properties, which reduces the risk of failure due to dry-running conditions. Carbon imparts mechanical strength to the composite.

The choice of carbon grade and type of impregnation (ie resin or metal) is highly dependent on the sealing duty. Fig 4.3 compares the surface profiles before and after operation, of four different grades of carbon paired with the same counterface material. The wear rates and surface profiles vary considerably between the four carbon grades. Carbon composites are heterogeneous materials. Cheap carbon composites are more susceptible to failure of the material itself (33), as the less expensive manufacturing techniques allow the carbon granules to break away from the resin matrix more easily (by differential thermal expansion), due to less uniform material properties.

Carbon composites are suitable over a wide range of temperatures (eg from cryogenic duties, to over 450 degC with special grades). They also have reasonable chemical resistance, tolerate minor imperfections in seal face geometry, possess good self-lubricating properties, and are generally inexpensive. Carbon composites can be paired with a wide range of materials. These characteristics account for the extensive use of carbon composites as a mechanical seal face material.

Carbon seal faces are not suitable on duties with abrasive or crystallising liquids, because of excessive wear rates. At high pressures carbon composites distort excessively so hard alloy, ceramic, or carbide materials are preferred.

(c) Aluminium Oxide

Alumina ceramics (aluminium oxide) were the first non-metallic materials to be used as a mechanical seal face material. Different grades are available, but 99.5% aluminium oxide is specified for maximum chemical resistance.

Alumina has excellent wear resistance, and chemical resistance (dependent on the grade). If alumina is paired with carbon, excellent running properties are obtained on water and aqueous solution duties. Alumina has poor thermal conductivity, poor self-lubricating properties, and poor thermal shock resistance. Consequently mechanical seals with an alumina seal face will fail rapidly under dry-running conditions.

(d) PTFE

PTFE has excellent self-lubricating properties (it is used for dry running bearings), but very low strength. The strength can be increased by adding chopped glass fibre, but is still inferior to carbon. PTFE is only used on very mild duties (near ambient temperature and pressure), in which carbon would be chemically attacked. PTFE is normally paired with alumina. This combination of materials has a very poor tolerance of dry-running conditions, because high heat generation causes severe deformation of the PTFE face.

(e) Tungsten Carbide

The application of tungsten carbide as a mechanical seal face material is relatively new. A significant increase in mechanical seal life outweighs the high material cost of tungsten carbide on many sealing duties. At first tungsten carbide faces were only used on the most severe duties, but seal manufacturers are now turning to tungsten carbide and silicon carbide faces over the whole range of sealing duties (in preference to hard alloy or ceramic materials).

Hard carbide particles are bonded together by a ductile metal (normally nickel or cobalt). The ductile metal provides toughness, chemical resistance, and tensile strength. The carbide particles give the material extremely good wear resistance properties. Tungsten carbide is usually paired with resin impregnated carbon-graphite, which produces a combination with good wear characteristics, and a good tolerance to dry-running. This combination is also more resistant to thermal shock than a ceramic-carbon pair. Tungsten carbide is paired with silicon carbide or another tungsten carbide face, on abrasive or crystallising liquid duties. These hard-hard material combinations are very resistant to wear, but have a poorer tolerance of thermal shock or dry-running conditions.

Most tungsten carbide grades are limited to pH > 6. Grades using a cobalt binder phase are restricted to pH > 7. Grades using a nickel binder phase are suitable on water and aqueous solutions where the pH > 6. Special grades are available for use on duties with a pH > 2, but they are very expensive.

(f) <u>Silicon Carbide</u>

Silicon carbide is the other new super-hard seal face material. Like tungsten carbide, it is becoming used on a wide range of sealing duties, because extended seal life outweighs the material cost penalty.

Three forms of silicon carbide are used for mechanical seal faces. The sintered alpha form contains no free silicon and has the best chemical resistance, lowest fracture toughness, and better friction characteristics than tungsten carbide. The reaction bonded form contains no free silicon. Reaction bonded silicon carbide has the best friction characteristics of all the seal face carbide materials. The converted form contains a carbon-graphite base with the surface converted to silicon carbide. This form of silicon carbide seal face is economically attractive on less severe duties, and is superior to ceramic.

Silicon carbide has the same basic material characteristics as tungsten carbide. Compared to tungsten carbide, silicon carbide is cheaper, about five times less dense, has better friction properties, and its hardness does not deteriorate with rising temperature. The disadvantages are lower toughness, lower strength, and chemical attack by strong alkalis. Silicon carbide has a very high maximum operating temperature of around 1400 degC. On abrasive duties the best wear and friction properties are achieved by pairing reaction bonded silicon carbide with tungsten carbide. This combination is also very good on severe duties where carbon faces would suffer excessive wear and distortion. Silicon carbide can be paired with carbon-graphite on a very wide range of duties to extend seal life. This material pair has an excellent resistance to thermal shock, and dry-running conditions.

4.5.2 Selection of Seal Face Materials

Mechanical seal operating experience suggests that poor seal selection is a major cause of premature seal failure (table 4.1). The seal face materials are critical to the performance and life of a mechanical seal. Excluding failures of the secondary seals (metal bellows or elastomeric seals), most failure mechanisms internal to the seal (4.4.1) are caused by the wrong choice of seal face materials.

Seal face materials are usually selected by the seal manufacturer, using details of the duty supplied by the pump operator. This is probably where the root of the problem lies. Detailed and accurate operating data is required to make a good material selection. Pump and plant design data is often used to provide the details of the seal duty. This is certainly true when a plant is first commissioned. Under real operating conditions the pump duties will vary from their original design. Off-design conditions may be caused by operating the plant beyond its design throughput, aging of the equipment, or later plant modifications. All seal face materials exist in several grades (4.5.1), which are suited to different operating conditions. Different grades of the same basic material can radically alter seal life. The first few mechanical seal failures on any new pump should be analysed in detail. If seal life is less than 2 years, a change of seal face material should be considered. Over several seal lives the material selection is adjusted to suit the real operating conditions. Seal face material selection should be viewed as an iterative process, where the first selection is based upon the design conditions, and subsequent material selections are based upon a detailed analysis of the previous seal face.

At present most process plants replace failed seals with an identical selection. This philosophy is a result of a poor understanding of the causes of mechanical seal failure, and a lack of time and/or manpower to carry out an investigation. Operating experience clearly demonstrates that many seal selections are not ideal (table 4.1). A re-selection of the seal face materials is often only made when a pump suffers a series of very short seal lives (ie 20-50 days). A much closer link between the seal manufacturers and seal users is vital. Each seal failure should be investigated at the plant, and the findings made available to the seal manufacturer so that the suitability of the material or seal type can be assessed.

4.5.3 Modes of Failure in Seal Face Materials

Seal face materials can fail in a large number of ways. These can be divided into mechanical, thermal, and chemical failure mechanisms. Often several failure mechanisms are active (eg abrasive wear results in the formation of wear particles which cause scratching and grooving of the seal faces).

Mechanical Failure Mechanisms

(a) Abrasive wear.

(b) Adhesive wear.

(c) Scratching and grooving - caused by particles in the sealed fluid, wear debris form (a) or (b), or the formation of hard crystalline solids.

(d) Chipping. This normally occurs at the edges of the seal face, and may be caused by poor handling during installation or storage, pump cavitation, of fluid film vaporisation (in the latter two cases, the faces slam together causing chipping).

(e) Gross cracking. Many seal face materials are brittle (especially carbon-graphite, and ceramics), so poor handling during storage or installation can easily cause the materials to fracture.

Thermal Failure Mechanisms

(a) Differential expansion. This mechanism is particularly associated with cheap heterogeneous seal face materials (22,33). The material can peel, crack, or flake as a result of different thermal conductivity and expansion coefficients, in the granules and binding matrix.

(b) Thermal cracking. Thermal cracking is a consequence of excessive thermal stresses in the material. It is usually caused by dry-running conditions, where the seal faces overheat, or during vaporisation of the fluid film between the faces (severe thermal shock is generated by the transition between a liquid and vapour film).

(c) Coking. It occurs on the atmospheric side of the seal, on high temperature hydrocarbon duties. High temperature hydrocarbon leakage during normal seal operation, carborises on the atmospheric side of the seal. The coke particles build up and eventually seize the floating seal face. Steam quenches are used to reduce the rate of coke buildup.

(d) Deposits. If the fluid film vaporises, solidified residue is deposited on the seal face. The deposits increase abrasive wear, and cause scratching and grooving of the faces.

Chemical Failure Mechanisms

(a) Corrosion.

(b) Flaking, peeling, or blistering of hard face coatings. This is caused by a defective coating process, or chemical attack of the bond between the base metal and coating.

(c) Sludging and bonding. These mechanisms occur in heterogeneous materials (eg carbon-graphite), when particles are pulled from the faces due to excessive shear stresses. High shear stresses can occur if the sealed product is too viscous, or if the sealed fluid has been allowed to solidify between the faces (eg after shutdown of a bitumen pump).

4.5.4 Post-mortem Analysis of Mechanical Seal Faces

The examination of failed seals is the most direct form of seal failure analysis. A post-mortem provides the physical evidence of the condition of the seal after it failed. A careful post-mortem procedure (29) ensures that important information is not lost. The reason for failure may be obvious (eg a broken spring, or fractured seal face) but often the condition of the seal faces provides the only clues.

The appearance of seal faces after a period of operation is highly dependent on the seal face material. The super-hard materials (ie silicon and tungsten carbide) are less likely to develop a wear track (fig 4.5) than softer materials (eg stellite and carbon), which have lower wear resistance (fig 4.4). Thermal shock is a common failure mechanism on duties sealing fluids with high vapour pressure and low viscosity. Thermal shock will cause thermal cracking in some seal face materials. Thermal cracking describes the formation of fine radial cracks on the seal faces. Thermal cracking can also be caused during dry-running if the seal faces overheat. Thermal shock occurs if the seal face fluid film vaporises. The liquid and vapour phases have very different heat dissipation properties, causing excessive thermal stresses in the material. Sealed products with high viscosity (especially at high temperatures and low pressures) often fail due to coking, sludging, or bonding.

Plant B provides several examples of stellite seal faces (fig 4.4) from failed mechanical seals. The stellite faces all exhibit deep circumferential scoring in the well developed wear track. This provides evidence of dry-running, probably during pump startup or shutdown. The plant pump operating procedures should be checked. Some of the faces show signs of thermal cracking. The seal faces may be overheating due to insufficient cooling, incorrect seal balance, fluid film vaporisation, or dry-running operation.

Plant B also provides some examples of tungsten carbide faces from failed mechanical seals (fig 4.5). The tungsten carbide faces appear very different from the stellite seal faces (fig 4.4). There is virtually no wear track, and the contact area is at the inner edge of the seal face. Heavy contact at the inner edge, combined with edge chipping, suggests that there was excessive thermal distortion of one of the seal faces. The seal face pair (a hard-hard combination) exhibit radial flaking within the contact zone. This may be fatigue due to seal face "chatter", or a type of thermal cracking. Deep circumferential scoring suggests that large abrasive particles are present in the sealed fluids. An improved seal flush system with a cyclone separator ,may prevent this type of damage.

The examples above demonstrate the speculative nature of post-mortem analysis. It is not always possible to give a precise reason for the failure, but the appearance of the faces does give evidence of the actual operating conditions. A post-mortem can highlight types of seal damage which could be avoided in the future by different operating procedures, seal face materials, or changes to the auxiliary seal connections (ie cooling, steam quench, better flush, etc).

Talysurf profiles show that there are considerable changes in the seal face profiles during operation (33). The change in the seal face profile is highly dependent on the material combination. In fig 4.3 four different grades of carbon exhibit very different wear and deposit characteristics when paired with a single grade of alumina. This clearly demonstrates that the depth of grooves and thickness of deposits on seal faces is not a good indicator of running life.

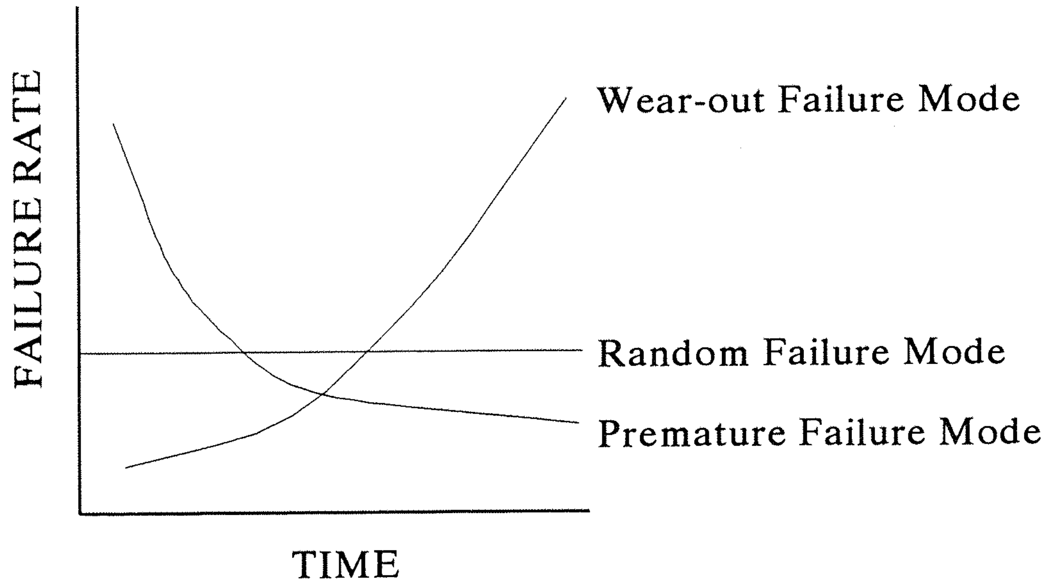


Figure 4.1 : Fundamental Modes of Failure

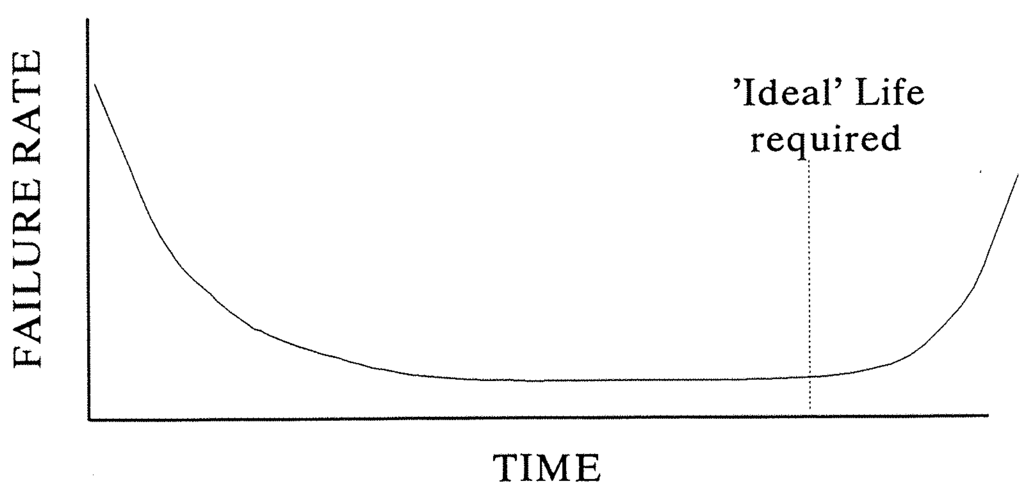


Figure 4.2 : The Bathtub Curve

Failure Mechanism	Cause	No. of Failures	Percentage of Failures
INTERNAL TO THE SEAL			
Selection and specification	Abrasive wear	30	18.7
	Hang-up	25	15.6
	Incompatible materials	9	5.6
	Chemical attack	2	1.2
	Erosion	1	0.6
Wear-out	Long running life	2	1.2
		69	42.9
EXTERNAL TO THE SEAL			
Installation	Incorrect seal setting	13	8.1
	Misalignment	12	7.5
Vibration	Cavitation or vibration	7	4.4
External Failure	Bearing failure	12	7.5
Operator error	Dry-run	30	18.7
		74	46.2
UNKNOWN		17	10.9

Table 4.1 : Mechanical Seal Failure Mechanisms at Plant A

	Carbon-graphite resin impreg-nated	*Carbon-graphite antimony filled*	*PTFE 25% glass*	*Stellite 1*	*Ni-Resist*	*Aluminium oxide 99.5%*	*Tungsten carbide Co binder*	*Tungsten carbide Ni binder*	*Silicon carbide reaction bonded*	*Silicon carbide sintered*
Density (kg/m^3)	1800	2500	2250	8690	7300	3870	14700	14700	3100	3100
Youngs modulus (GN/m^2)	23	33	—	248	96	365	630	600	413	390
Bending strength (MN/m^2)	65	90	—	—	—	320	1750	1700	500	450
Tensile strength (MN/m^2)	41	48	12–20	618 (UTS)	200	—	—	—	—	—
Thermal conductivity (W/mK)	9	20	0.4	15	40	30	80	70	200	70
Hardness	90–100 Shore A	85–95 Shore A	70–75 Shore D	600 HV	150 HV	1800 HV	1500–1600 HV	1300–1500 HV	2500–3500 HV	2500 HV
Thermal expansion coefficient (per °C $\times 10^{-6}$)	3.0	3.5	44–92	11.3	19.0	6.9	5.1	4.8	4.3	4.8

Table 4.2 : Typical Physical and Mechanical Properties of Commonly Used Face Materials (29)

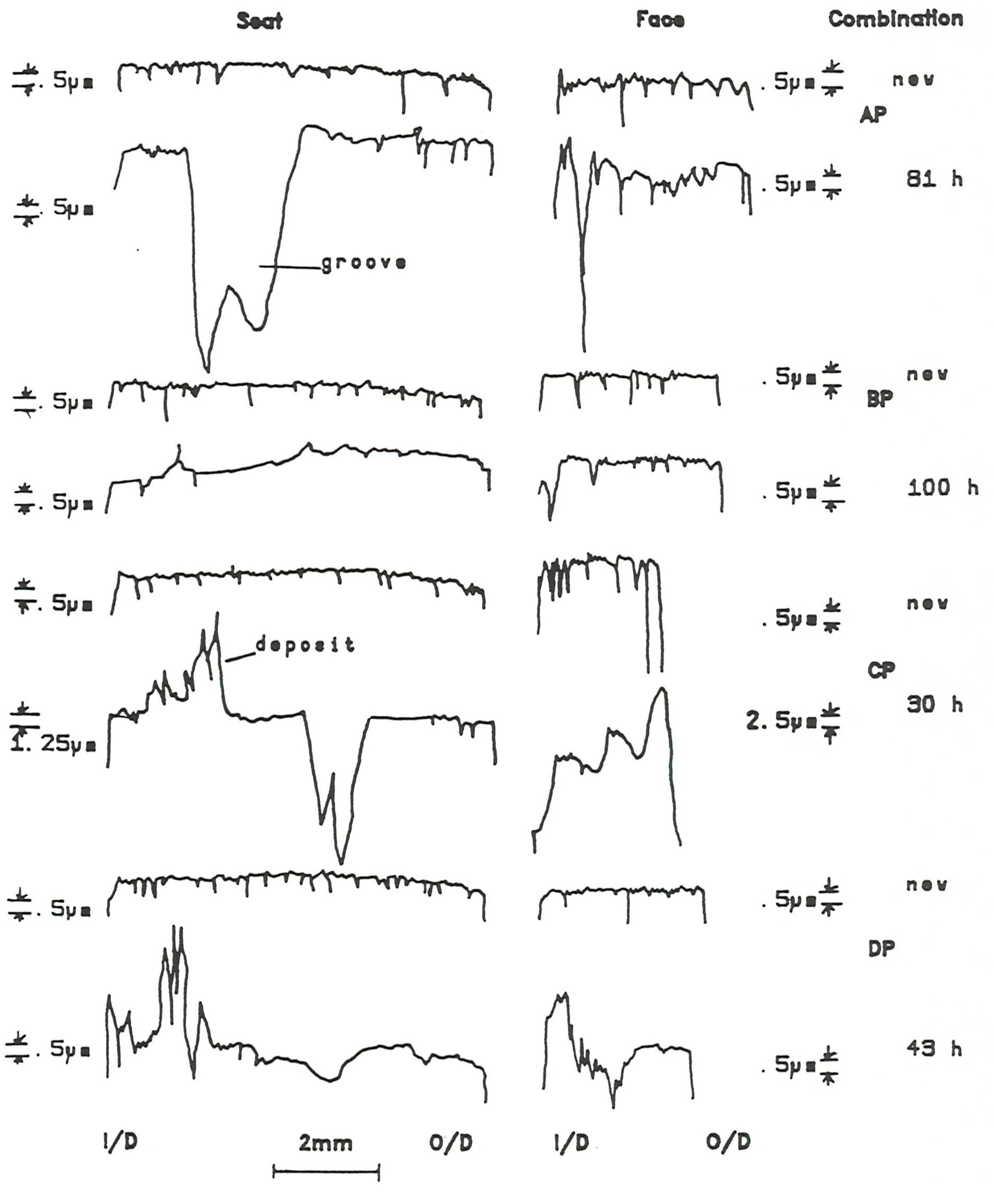
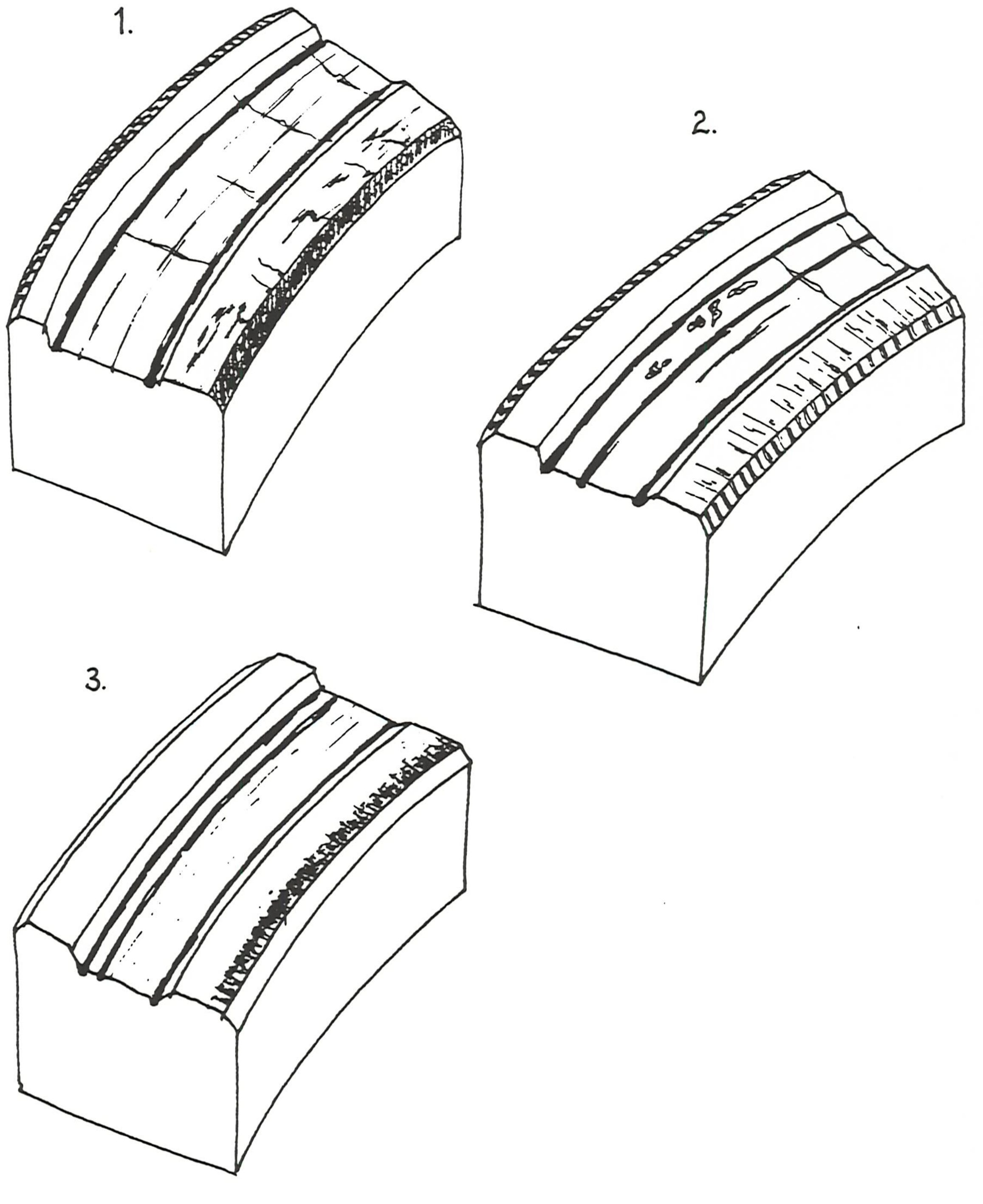


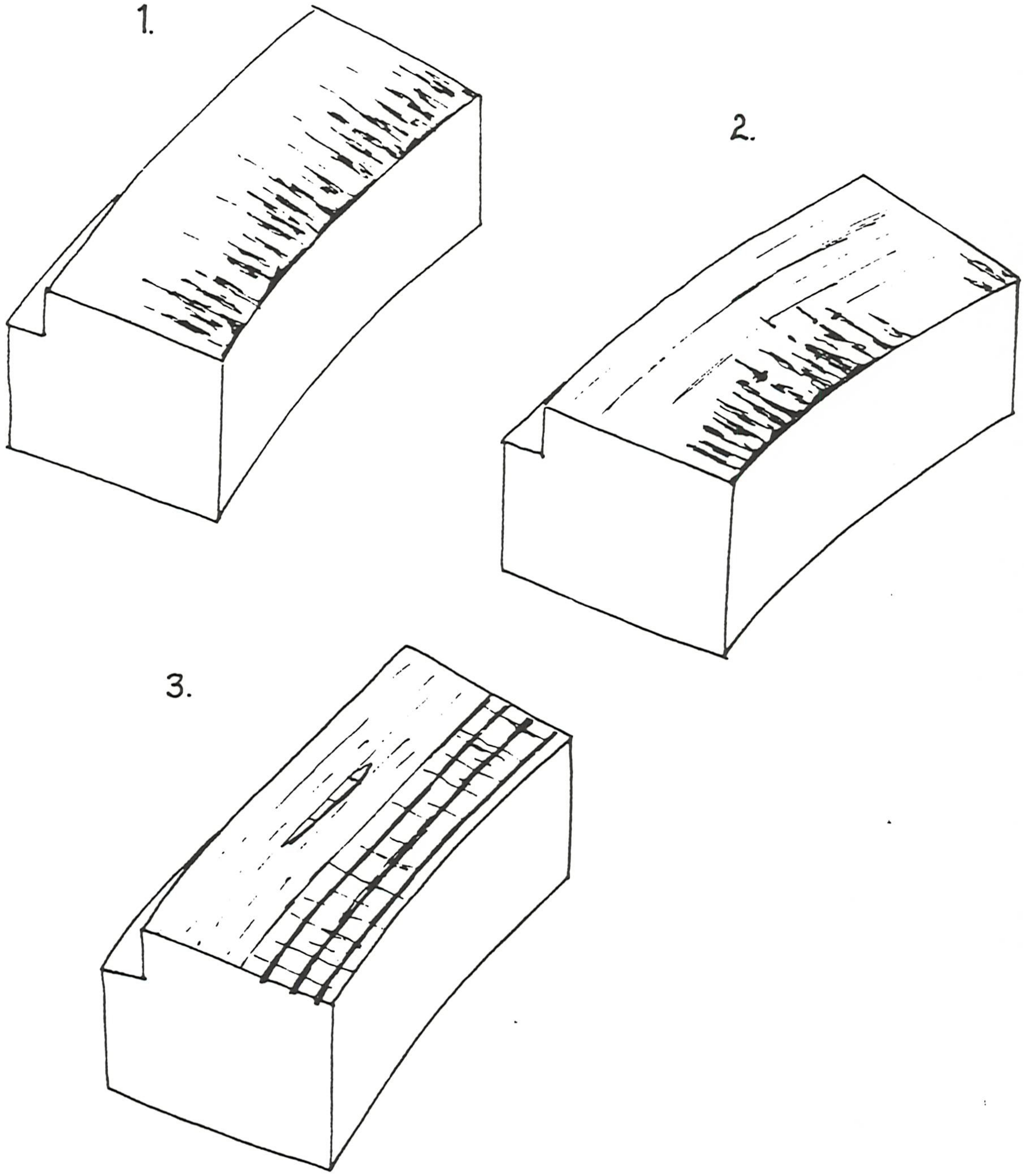
Figure 4.3 : Talysurf Profiles for Four Seal Face Material Combinations (33)



FEATURES:
Deep central wear track.
Thermal distress (small radial cracks)
Circumferential scoring.
Vaporisation/erosion marks at the inner and outer edges of faces 1 & 2.

MATERIAL: No.1 stellite faces on 18/8 stainless steel backing rings
SEALED FLUID: Light hydrocarbons (eg. naphtha, and gasoline)
DETAILS: All: rotary faces, inner seal dia. 60mm, face width 5mm.

Figure 4.4 : Stellite Faces from Failed Mechanical Seals at Plant B



FEATURES:
Contact zone at inner edge.
No wear depth
Deep circumferential scoring, and thermal distress in the contact zone of face 3.
Radial damage (faces 1 & 2) in the contact zone is fatigue or 'chatter'.

MATERIAL: Tungsten carbide faces on 18/8 stainless steel backing rings
SEALED FLUID: Fuel oil
DETAILS: inner seal dia.60mm, face width 5mm, faces 1 & 2 were a running pair, 1 was a rotary face, 2 & 3 were stationary faces.

Figure 4.5 : Tungsten Carbide Faces from Failed Mechanical Seals at Plant B

5.0 DIMENSIONLESS GROUPS FOR ANALYSING MECHANICAL SEAL FAILURES

Dimensionless groups have been successfully applied to the analysis of heat flow, fluid flow, and elasticity. Mechanical seal life is a function of all these processes, and the stability and cleanliness of the sealed fluid. The complex interaction of these mechanisms makes it unlikely that a single dimensionless group could completely describe the behaviour of mechanical seals. However several dimensionless groups, each dealing with a particular property of the system, could be applied simultaneously.

5.1 Principles of Dimensionless Groups

Many physical processes defy the establishment of precise quantitative relationships between the system variables, due to their inherent complexity. The exact conditions or interaction of variables may be unknown or unmeasurable in reality. A mathematical model would require some major assumptions. Dimensional analysis can provide a qualitative solution without making the assumptions necessary for a mathematical model. Subsequent experiments based on the qualitative solution from dimensional analysis can lead to a complete solution of the real process.

Dimensional analysis is based on the principle that two sides of an equation must be dimensionally equal for a true relationship to exist. The numerical value of a dimensionless group is independent of the system of units. Dimensional analysis uses dimensionless groups to describe the relationships between variables, and enables the experience of one situation to be applied to a similar situation in which the numerical size of the variables is different. Dimensionless groups describing a particular seal design could be extended to all sizes of that design.

5.2 The Stability Factor

A mechanical seal goes unstable if the forces keeping the seal faces closed fall to zero. The faces move apart causing excessive leakage. The seal face pressure (Pf) can be expressed quite simply.

$$Pf = Pp\,(b-k) + Psp \quad , \text{ where } 0<k<1$$

There will be a discontinuity in the value of Pf if the fluid film between the faces vaporises, due to the phase change. In unbalanced seals ($b>1$) the seal face pressure (Pf) cannot go to zero. Balanced seals ($b<1$) can go unstable if the pressure gradient factor (k) falls below the balance ratio (b). This can occur through changes in the seal face profile during normal operation. The pressure gradient factor has its maximum value ($k=1$) when there is a convergent fluid film between the faces (in the direction high to low pressure), and the contact is at the edge of the face. Under this condition the sealed fluid pressure at which the seal goes unstable then has its minimum value (Pcrit).

$$Pcrit = Psp \,/\, (1-b)$$

The critical pressure can be raised by increasing the spring pressure or increasing the balance ratio. However there is a limit related to the strength of the seal face material, wear rate, and heat generation through increased friction.

The stability factor indicates the margin between the critical seal face pressure (Pcrit), and the normal operating seal pressure (Pp).

$$\text{Stability Factor} = Pcrit \,/\, Pp$$

Roos (26) used the seal face pressure (Pf) as one of several parameters he calculated to check seal suitability for particular applications, and as a trouble-shooting tool. Roos found that Pf should lie within the range 0.5 to 3.5 bar. Roos assumed that there was a convergent leakage path (k=1) with full fluid pressure between the seal faces, and concluded that the Pf-value gave a reliable method of checking mechanical seal suitability. Buck (4) found some correlation between the stability factor and seal life. The stability factor makes no allowance for shaft speed, properties (other than pressure) of the sealed fluid, or the effect of fluid film vaporisation.

5.3 Thermal Stress Factor

Thermal deflections arise from stresses set up by the non-uniform temperature distribution within the seal. The greatest temperature gradient is at the seal faces. Thermal deflection modifies the shape of the leakage path between the seal faces. This alters the pressure gradient across the seal faces (and changes the pressure gradient factor (k)), and can cause instability in balanced seals (see 5.2). Thermal stresses can cause the seal face material to fatigue and crack. Thermal cracking is caused by excessive thermal stresses, generated by dry-running, or inadequate seal face cooling. Thermal shock can cause cracking in some materials. Thermal shock is generated by the phase change, if the fluid film between the faces vaporises. The yield stress of the face material provides an indication of the seal face resistance to thermal deformation. The thermal stress factor compares the thermal stress during seal operation to the yield stress of the seal face material.

Buck (4) found that the thermal stress factor and stability factor gave a similar correlation with seal life. This is not surprising since both are governed by the pressures and forces at the seal faces. The thermal stress factor is much more difficult to measure than the stability factor.

5.4 Delta-T Factor

The Delta-T factor is a measure of the possibility of fluid film vaporisation between the seal faces. The fluid film is essential to reduce wear and heat generated through friction, to an acceptable level. Although the mechanisms which maintain the fluid film are not fully understood, it is certain that the film is important for lubrication and the reduction of friction and wear. The lubrication properties of a fluid are poorer in the vapour phase than in the liquid phase. Vaporisation of the fluid film is detrimental to the performance of the seal faces, due to poorer lubrication and the deposition of solids. Vaporisation often causes thermal cracking in the seal face material, and excessive wear due to the poorer lubrication properties of the fluid film and higher seal face temperatures.

T[a] represents the rise in seal face temperature (above the sealed fluid temperature) at a given operating pressure. The maximum operating temperature of the sealed fluid is T[a] below the vaporisation temperature of the sealed fluid (fig 5.1), at the minimum seal face pressure. Vaporisation will take place at a lower temperature and pressure than the sealed fluid in the stuffing box, because there is a pressure drop across the seal faces. T[b] is the temperature difference between the vaporisation temperature and sealed fluid temperature, at the minimum seal face pressure.

Delta-T factor = T[a] / T[b]

The Delta-T factor should be less than unity to ensure that the fluid film does not vaporise under normal operating conditions. T[a] can be calculated by equating the heat generated and dissipated in the seal (29). This calculation includes variables associated with heat flow, pressure, geometry, material properties, and the fluid film.

Dolan et al (9,13) used T[a] to measure the suitability of mechanical seals prior to installation on oil production platforms. They found that the theoretical value of T[a] gave a very accurate reliable operating envelope on particular sealed fluids, if the minimum seal face pressure was assumed to be 10% of the sealed fluid pressure (fig 5.2). Buck (4) used a dimensionless group similar to the Delta-T factor, and found this group (liquid design factor) gave the best correlation with mechanical seal life. Roos (26) applied T[b] as one of several parameters (see 5.2) for trouble-shooting and checking the suitability of new mechanical seals.

5.5 The Duty Parameter

A balance between boundary lubrication and a hydrodynamic film is required at the seal interface, for long life and low leakage rate. Boundary lubrication alone, is associated with a very thin fluid film. The seal faces make solid contact, causing high friction (heat generated may cause the fluid film to vaporise) and excessive wear of the seal faces. A fully developed hydrodynamic film is associated with a thick fluid film between the seal faces. There is no solid contact between the faces, zero wear, and friction is low. A thick fluid film between the faces results in significantly higher leakage rates.

The duty parameter can be used to indicate the severity of the contact conditions at the seal face. The lubrication mode in the contact region is indicated by the relationship between the friction coefficient and the duty parameter. (fig 5.3). The value of the duty parameter (G) decreases as the severity of the contact conditions increases.

$$G = f * V * B / W$$

There are three distinct lubrication modes (fig 5.3):

(1) Very high friction at low G. This corresponds to boundary lubrication conditions, and typically occurs at start-up.

(2) Decreasing friction coefficient as G increases. Mechanical seals should operate in this mixed lubrication regime to achieve maximum life.

(3) Increasing friction coefficient as G increases. This corresponds to conditions in which a full hydrodynamic film

exists between the seal faces.

Studies (9,14,29) have tried to assess mechanical seal suitability (to increase life) by using the duty parameter (G). Success has been limited by the sensitivity of the friction coefficient to small changes in the operating conditions, because the seal faces distort through thermal and pressure loading.

5.6 PV Factor

The PV factor is not a dimensionless group, but it is included because it has been applied in a similar way. A seal operates best when the seal face load is supported both mechanically and hydrodynamically. The operating limit may be wear rate or the breakdown of the seal face materials under excessive load. These limits are dictated by the proportion of the seal face load that is supported mechanically.

The PV factor is used as an indicator of the severity of the seal face contact conditions. Limiting PV values have been established for various material combinations and sealed fluids, to define operating limits (pressure and shaft speed) for adequate seal life (29).

$$PV \text{ factor} = Pp * V$$

The PV factor is a very poor indicator of seal life under normal operating conditions, because there should be very little solid contact between the seal faces. The PV factor is more relevant where a high proportion of the seal face load is by solid contact. This situation occurs during dry-running conditions. There is some merit in using the PV factor to measure the tolerance of mechanical seal face materials to dry-running conditions. The main problem is the sensitivity of the limiting PV value, to slight differences in the sealed fluid properties.

Studies (2,4) suggest that the PV factor has very little relationship with mechanical seal life. Buck (4) found very poor correlation with seal life. Bauer (2) discusses the idea of a critical surface temperature at which severe wear (welding) occurs. If a critical temperature does exist, then the PV could be considered a constant. The seal geometry is a variable, and when combined with other heat loss effects it is, "doubtful that the PV factor has an all-encompassing significance" (2).

5.7 Comparison of the Methods

Long mechanical seal life and low leakage rate is enhanced by seal stability, absence of fluid film vaporisation, and low seal face friction. Studies (9,14,26,29) suggest that dimensionless groups can be found which can improve seal selection methods. The greatest attraction of applying dimensionless groups is the creation of seal selection criteria, based upon operating experience, which can be applied to a wide range of seal types, sizes, and duties.

Buck (4) compared several dimensionless groups, and found that they all contained one or more variables that were difficult to measure in practice. This poses the greatest practical problem to applying most dimensionless groups. It is not economically viable to install expensive instrumentation on all mechanical seals (eg seal face thermocouples, and pressure transducers). The duty parameter (5.5) and Delta-T factor (5.4) are the best dimensionless groups established so far. The duty parameter correlation with seal performance requires a measurement of the seal face friction coefficient. This could be achieved by installing a torque measuring device. The Delta-T factor requires an accurate measurement of the minimum seal face pressure, and seal face surface temperature. Thermocouples and pressure transducers would need to be fitted to the mechanical seals. A research program might be the best way to establish relationships between the more easily measured variables (ie sealed fluid temperature and pressure), and the conditions at the seal faces.

Dimensionless groups can be used to assist the seal selection process. The most promising groups require further application and possibly research work, to produce relationships which are practical to apply in the process industry.

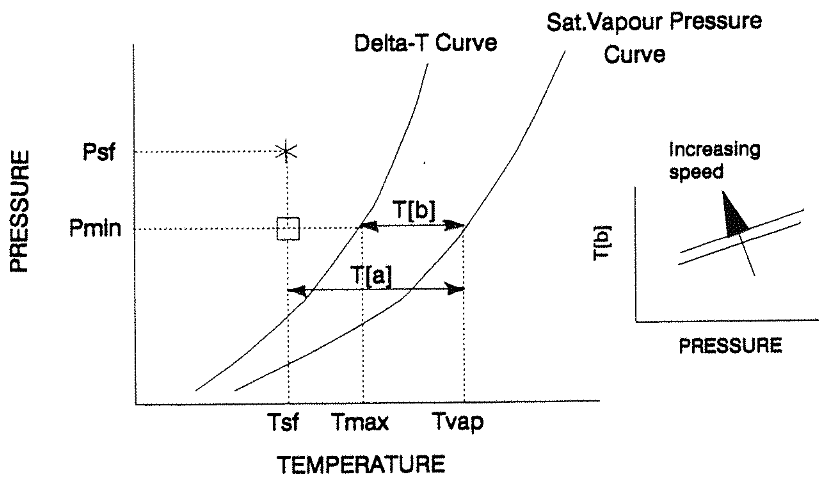


Figure 5.1 : The Delta-T Seal Operating Envelope

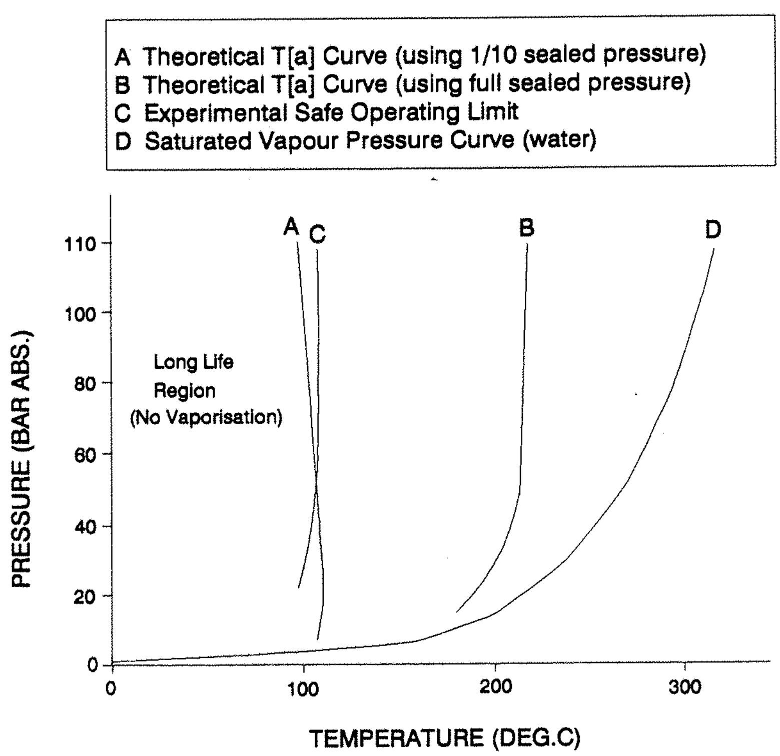


Figure 5.2 : Predicted Seal Operating Envelope for Water (13)

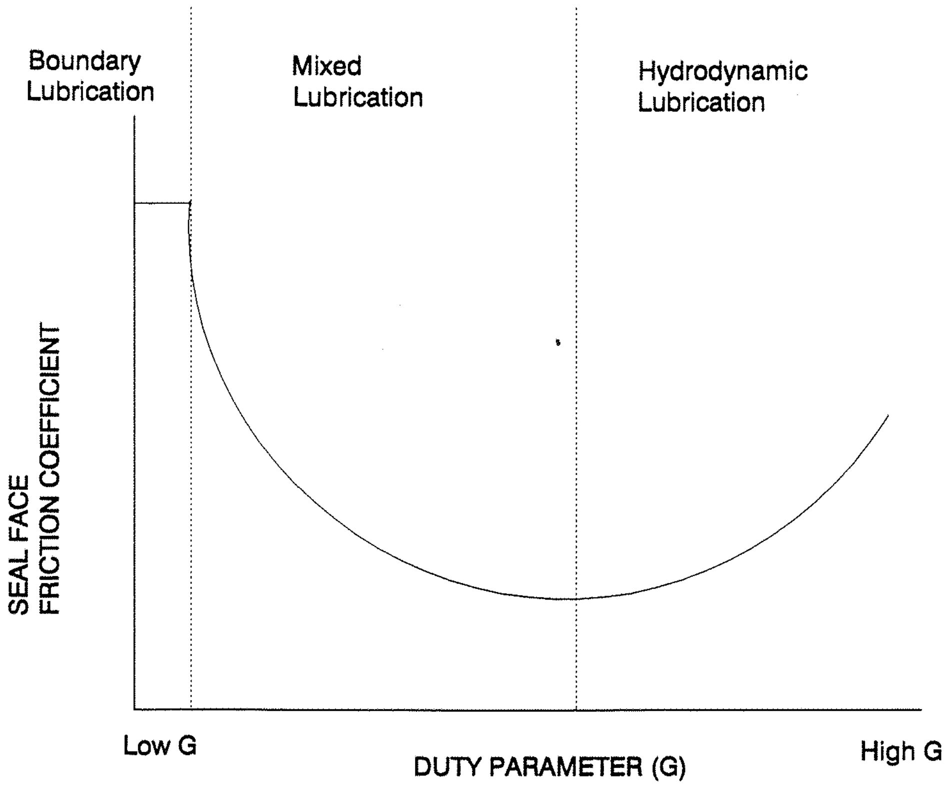


Figure 5.3 : The Relationship Between Lubrication, Friction Coefficient, and the Duty Parameter

6.0 STATISTICAL METHODS FOR ANALYSING MECHANICAL SEAL FAILURES

6.1 Mean Time Between Failures

Mean time between failures (MTBF) is probably the most common statistical term used to describe the average life of a group of mechanical seals.

$$MTBF = [\ L(1) + L(2) + \ldots + L(n)\ ]\ /\ n$$

L(x) are the individual seal lives, and n is the total number of seals in the group. MTBF provides a quick and simple calculation that can be used to compare the life of mechanical seals on different duties, over a number of failures.

In practice MTBF can produce a misleading picture. Time based trends are lost and cannot be revealed by MTBF. All lives are given an equal weighting. In a real historical database (chapter 7), long seal lives are more prone to errors resulting form missing failure data (ie a long seal life is more likely to be the sum of two short lives). MTBF is a good simple method of describing seal life on individual installations. This was demonstrated in the cost analysis (3.4) where MTBF was used to measure seal life on 15 pumps over a period of 10-15 years. Table 6.1 illustrates a situation in which a simple comparison of seal life at two different process plants was required. A rough value of MTBF could be calculated by multiplying the mid-value of each installed life range with the number of seals in that range. The life ranges (table 6.1) reflect a closer interest in the "problem" seals (ie short life). A calculation of MTBF based upon the mid-range life would be particularly unsuitable in this case, because the largest calculation and data errors occur at long seal lives. MTBF will give the long seal lives the same weighting as the short seal lives. This example demonstrates the need for care when using MTBF to compare large failure distributions, and the recognition that all lives are given an equal weighting.

6.2 Weighted Life Index

The simple MTBF statistic has limitations when it is used to compare large failure distributions. The weighted life index provides a method of applying variable weighting, to enhance particular parts of the distribution. Variable weighting has been used to enhance the short life distribution of mechanical seals at two process plants (table 6.1). Variable weighting can overcome some of the limitations of the simple MTBF statistic. The effect of missing data (chapter 7) can be reduced by lowering the weighting value of long seal lives. Particular areas of the failure distribution can be enhanced through higher weighting values, to allow a more accurate comparison of different seal life distributions.

The weighted life index provides a quick method of comparing seal failure distributions, with more meaningful results than a basic MTBF statistic. The method used in table 6.1 is shown below.

(1) Express the number of seals in each seal life range as a percentage of the total number of seals.

(2) Calculate the cumulative percentage failed for each seal life range.

(3) Identify which seal lives contain the L10 (ie 10% failed), L30, L50, L70, and L90 lives.

(4) Sum the weighting values associated with Lx lives, to give the weighted life index of the distribution. In this case the weighting values have been chosen to enhance the short life distribution, and reduce the effect of missing data.

Installed Life (Hours)	Weighting value	Cumulative Failures(%) Plant A	Plant B	Lx Weighting Plant A	Plant B
0 - 500	100	10.3 (L10)	9	100	
500 - 1000	80	18.7	17 (L10)		80
1000 - 1500	60	25.1	25		
1500 - 2000	50	30.7 (L30)	30 (L30)	50	50
2000 - 2500	40	35.4	35		
2500 - 3000	30	38.7	39		
3000 - 3500	20	41.8	42		
3500 - 4000	15	44.6	45		
4000 - 4500	10	46.7	48		
4500 - 5000	8	48.5	50 (L50)		8
5000 - 10000	6	59.4 (L50)	67	6	
10K - 30K	4	65.4	88 (L70)		4
over 30K	2	100 (L70,90)	100 (L90)	2+2	2
		Weighted Life Index =		160	144

Table 6.1: Comparison of the mechanical seal life distribution at two plants using a weighted life index to enhance differences in the short life distributions.

6.3 Regression Analysis

Regression is a mathematical method for evaluating the relationships between system variables. In a graphical sense, regression is a curve-fitting technique. A linear regression equation describes the best-fit line through a series of data points, using the optimum linear combination of known variables. Both curves and straight lines can be described by a linear regression equation.

The linear regression equation relates the mean value of the dependent variable (Y), to a linear combination of independent variables (Xi). Two examples are shown below:

Y = a + b(1)*X(1) + b(2)*X(2) + ...

Y = a + b(1)*X(1) + b(2)*X(1)^3 + b(3)*X(2) + ...

Each independent variable X(i) has a weighting factor b(i) called a regression coefficient. The regression coefficient is a measure of the independent variable's effect on the dependent variable Y.

When regression analysis is applied to a real system the variables may be collinear (not independent). The regression coefficient of a collinear variable will measure a combined effect. The true effect of a collinear variable can be evaluated using additional equations (appendix 2)(38).

6.3.1 Features of Regression Analysis

(1) Determines which variables have the greatest effect on the dependent variable.

(2) Eliminates variables with little or no effect on the dependent variable.

(3) Study and expense can be applied where it is most effective.

(4) Simple to apply with the aid of a proprietary computer program.

(5) Extrapolation outside the data ranges used to establish the regression equation should be avoided.

(6) Data should approximate to a normal distribution about the regression line, to validate confidence level calculations.

(7) Time trends can be isolated.

(8) The quality of the data is critical to the results of regression analysis (see chapter 7). Careful interpretation of outlying points is essential (3), since they will affect the regression equation very significantly.

6.3.2 A Method for Applying Regression Analysis to Mechanical Seal Failure

It is desirable to implement a method which minimises the collection of data which is difficult to measure or obtain, especially if it subsequently proves to have no connection with the particular variable of interest.

A failure analysis diagram (fig 6.1) makes the task easier. Although the failure mechanisms are complex, it is possible to group types of failure and their likely causes. Referring to fig 6.1, data is easiest to obtain at level A, and becomes increasingly difficult towards level E.

A regression analysis strategy can be implemented, with seal leakage (ie failure) as the dependant variable. The first regression analysis will use the four level B groups (fig 6.1) as the independent variables. The level B data could be obtained from a quick post-mortem (see 4.5.4) of the failed seals, by a non-expert. The regression analysis might show one group to be much more influential than the others (ie a higher regression coefficient). A second regression analysis could be carried out by collecting level C data (fig 6.1) associated with only the most significant level B group. This principle could be continued to obtain the fundamental causes of mechanical seal failure (at level E).

6.3.3 Studies of Mechanical Seal Performance Using Linear Regression Analysis

Nau and Flitney (24) used a multiple linear regression analysis to show the effect of eleven factors on mechanical seal life. The analysis was performed on data from 100 mechanical seals. Fig 6.2 shows the regression coefficients of the eleven factors. It is not made clear whether these are partial coefficients (ie the true effect of each factor, allowing for collinearity) or simply the coefficients from the regression equation. A positive coefficient indicates that an increase in the factor produces an increase in seal life. Conversely, a negative coefficient indicates that an increase in the factor produces a decrease in seal life. The regression analysis shows a strong relationship between vibration, corrosive nature of the sealed fluid, and poor seal life. High discharge pressure, and expensive seals seem to improve seal life.

This study illustrates how regression analysis can improve the understanding of environmental factors, on mechanical seal performance. It is simple to see whether an individual factor has a positive, negative, or zero effect on seal life. Appropriate action could be taken to improve seal life; reduced vibration limits, closer attention to material choice on corrosive duties, and better quality seals.

Rowles, Reddy, and Nau (27) conducted a regression analysis on test-rig data, to study the effect of shaft vibration on mechanical seal performance (ie leakage rate). The study was initiated by the earlier study described above (24). This is an example of regression analysis using the method described in 6.3.2. Five variables were considered: leakage (Q), rotational speed (N), seal chamber pressure (P), stationary seal face temperature (T), and angular vibration (A). The study produced the regression equations below.

$$Q = 1.34 * (N \;.\; P \;.\; T \;.\; A) \quad \text{ml/h}$$

If face temperature (T) was excluded (being the most difficult to measure), the regression equation became:

$$Q = 1.55 * (N \;.\; P \;.\; A) \quad \text{ml/h}$$

The correlation coefficients are tabulated below.

	[Q]	[P]	[T]	[N]	[A]
[Q]	1.00	0.70	0.66	0.39	0.29
[P]		1.00	0.47	0.08	-0.03
[T]			1.00	0.41	0.77
[N]				1.00	0.30
[A}					1.00

This study shows a strong relationship between seal leakage (Q), seal chamber pressure (P), and seal face temperature (T). Due to the strong collinearity between angular vibration A) and seal face temperature (T), the regression equation changes very little when seal face temperature is excluded. This collinearity can be explained by the high temperatures generated by friction induced by angular vibrations.

Prediction of seal leakage rate using the regression equations above met with variable results. Regression analysis can get around the problem of defining all the individual variables (by collinearity), but clearly the four measured variables in this study do not include all the factors affecting mechanical seal leakage. This regression analysis has successfully showed the relative significance of four factors on mechanical seal leakage, and the relationships between them.

6.4 The Discriminant Function Technique

6.4.1 Principles of the Discriminant Function

An analogy provides a simple way of explaining the principles and features of the discriminant function technique. We want to determine from a body of data whether a person is a city stockbroker, or is not a city stockbroker. If only one measure is considered, for example height, it would be difficult to assess which group the person belongs to as the measure does not discriminate city stockbrokers and other types of people. Other measures (eg salary, workplace, number of television monitors, etc) would discriminate much more successfully.

Grouping people on the basis of several measures will improve the group discrimination. The technique identifies people by comparing their features with measures associated with a group. In the example, people who are city stockbrokers would tend to have the following features: work in the city of London, high income, expensive house, and retire by the age of 35. The number of measures required to give a confidence limit in group discrimination, is dependent on the discriminatory ability of each measure. The discriminant function technique can be used to rank measures according to their discriminatory ability. An analysis can then be carried out to find the minimum number of measures to give a required confidence level. The most discriminatory measures reflect the most characteristic features of the group.

6.4.2 The Linear Discriminant Function

The value of a linear discriminant function (S) is formed form a linear combination of weighted variables (Xi). Variables are weighted according to their discriminatory ability, by maximising the overall ratio of group mean separation to within group variation. A different discrimination function will exist for each group.

$$S = a(1)*X(1) + a(2)*X(2) + \ldots\ldots + a(n)*X(n) + C$$

There are comprehensive mathematical treatments of the discriminant function in the literature (1,16,19,28).

The discriminant function technique can be used to predict future performance using data from past performance. A set of discriminant functions are set-up using the data from the past. Each function could be used to identify a particular failure mode. When a failure occurs, data from the failed component is applied to the discriminant functions. The failure would belong to the failure mode whose discriminant function produces the highest numerical value. This is the basis for data classification.

The discriminant function can also provide valuable information on the relative importance of each variable to a grouping. The coefficients in the discriminant function equation are not immediately comparable with respect to their individual influence on the overall discrimination. A special form of factor analysis (28,37) is necessary. Standardised coefficients allow a numerical ranking of the importance of each variable.

6.4.3 Features of the Discriminant Function

(1) Suited to multivariable analysis.
(2) Classification of data.
(3) A method for prediction.
(4) Numerical ranking of the significance of a large number of variables.
(5) Measurements can vary in both dimension and scale.
(6) A minimum number of variables for a specified confidence limit.
(7) The variables can be collinear.

6.4.4 Studies of Mechanical Seal Performance Using Discriminant Functions

Sayles (28) demonstrated the technique on data from mechanical seals and lip seals.

(1) Application to Data Classification

Weibull analysis (see 6.5) was used to classify mechanical seal failure distributions as exponential or bi-modal. A discriminant function was calculated for the two groups, using sealed fluid pressure, sealed fluid temperature, and seal diameter as the variables. Sealed fluid pressure was the most discriminatory variable. This is in agreement with other studies (6.3.3. and fig 6.2) using different statistical methods. Only 2 out of 18 seal failure distributions were classified incorrectly, when the values of the three variables (for each distribution) were put into the discriminant function. Despite the small number of variables (and chosen for convenience - easy to obtain), a good classification of the data was achieved.

(2) Application to the Prediction of Failures

The discriminant function technique was used to give a positive classification of premature failure in lip seals, using surface measurements of the unworn seal topography. Nine measures were used to define the discriminant function. Similar coefficients provided the possibility of good discrimination with a reduced number of variables. Discrimination plots were drawn (fig 6.3), using the best combination of variables. Excellent discrimination was achieved with only four out of the nine original variables. This discriminant function would allow a good prediction of leakage, by measurements of the lip seals before service.

A recent study (33) of small narrow-faced mechanical seals indicates that measurements prior to service can predict the performance of the seal in service. A discriminant function could be used to form a practical technique for assessing mechanical seal duties before service. This study (33) is particularly suitable because there are fundamental similarities (ie contact area) between narrow faced mechanical seals and lip seals.

6.5 Weibull Analysis

6.5.1 The Principles of Weibull Analysis

Carter (5) provides an extensive explanation of the principles and methods of applying weibull analysis to real data. The simplest form of failure analysis involves plotting a failure distribution. The form of the distribution will reveal the general mode of failure (see fig 4.1). Weibull analysis enables more detailed information to be extracted from a failure distribution.

The weibull distribution is an empirical function which can represent a wide range of real failure distributions (fig 6.4).

$$R(t) = \exp\left[- ((t - To)/h)^{B} \right]$$

To = origin of the failure mode
h = characteristic life (63% of the population has failed).
t = life at failure
B = weibull index

Some distributions cannot be adequately described by an empirical function. The failure of a finite proportion of weak components is one such case. A modified weibull distribution has been developed to describe this form of failure (5).

Substituting R(t) = [1-F(t)] and taking natural logs, reduces the weibull function to a straight line relationship on suitable graph paper.

$$\ln [- \ln [1-F(t)]] = B*\ln [t-To] - B*\ln(h)$$

Failure mechanisms are associated with different values of B, h, and To (table 6.2). Reference (41) provides a useful summary of Weibull plot characteristics with indicative plots of F(t), the instantaneous failure rate z(t), and the frequency distribution. This reference also discusses other types of failure distribution.

A real failure distribution normally contains several failure mechanisms. When the distribution is displayed on a weibull plot, the data will not plot as a straight line. The gradient of the line will change if there is a change of failure mechanism. Discontinuities in a weibull plot mark the change between two failure mechanisms. By drawing a best-fit line through the whole distribution, valuable information on the different failure modes is ignored. The weibull parameters (B, h, and To) should be calculated for each identifiable failure mode. A substantial number of data points (50 at the minimum) are necessary for a worthwhile weibull analysis of a failure distribution. BS-5760: Part 3 (42) provides examples of applying weibull plots in the assessment of reliability. The examples show single best-fit lines drawn on somewhat non-linear plots. As explained earlier great care should be taken in assuming a single best-fit line, since a marked change in failure mode will cause a marked change in gradient on the weibull plot. BS-5760 (41,42) provides cautions in the use of weibull plots where a significant proportion of the seals have not failed, and suggests the use of confidence limits to avoid a misleading sense of accuracy.

6.5.2 Features of Weibull Analysis

(1) Simple graphical method, based upon the failure distribution alone.

(2) Identifies modes of failure from the failure distribution.

(3) Provides useful statistical values (B, h, and To) which can be used to compare different failure distributions. These are more informative than a simple MTBF (6.1) or weighted life index (6.2).

6.5.3 Studies of Mechanical Seal Failure Using Weibull Analysis

Weibull analysis has been the most popular method of statistical analysis applied to mechanical seal failure data (11,12,28,30,31,35,36).

Gu and Wang (12) have made the most extensive use of weibull analysis. They concluded that mechanical seals have a failure distribution that is characterised by the seal construction, operating conditions, phase state of the fluid film between the seal faces, and seal face friction. The weibull index provides a measure of the failure distribution, so a mechanical seal will have a fixed weibull index unless there is a phase change in the fluid film, or a change in the seal face friction (assuming the seal construction and operating conditions are constant). Gu and Wang applied this principle quite successfully to trouble shoot individual seals on an oil refinery. Seal face friction was modified by improving the seal face flush (less particles), or changing the seal face materials. A fluid film phase change was achieved by heating or cooling the seal. This method did improve seal life on individual seal duties.

In most of the studies a single best-fit line is drawn through the points on the weibull plot. The most attractive feature of weibull analysis has been ignored - identification of changes in failure mode. Chapter 8 contains a comprehensive weibull analysis of mechanical seal data from Plant A and Plant B. Appendix A1 contains weibull plots for mechanical seals at Plant B, on seven different sealed fluids. Appendix A2 contains weibull plots for mechanical seals at Plant A, with six different seal face material combinations. Individual failure modes have been identified, and these are discussed in Chapter 8.

6.6 Comparison of the Statistical Methods

Regression analysis is a good technique for establishing the effect of a large number of parameters on mechanical seal life. The positive, negative ,or zero effect of each parameter can be determined. Unfortunately regression analysis is highly sensitive to poor data, or systematic errors. With good data, regression analysis provides a useful method of "homing in" on the most important parameters.

The discriminant function is suited to the analysis of mechanical seal data. Existing failure data could be used to set up discriminant functions for the most common failure mechanisms (eg thermal shock, misalignment, coking, etc). The discriminant functions would contain variables relating to the seal face materials, seal design, and operating conditions. These functions could then be used to select new mechanical seals, and improve seal life. An alternative approach could be to set up discriminant functions to classify seal lives. A series of functions could be generated to classify seals over the whole range of lives. New mechanical seal selections could then be classified using these discriminant functions, to predict operating life.

Weibull analysis has become the most common statistical method of analysing mechanical seal failures. Studies have shown that weibull analysis can be applied to real data, and improve the life on individual seal duties. Weibull analysis is very simple to apply since only life data is required. The most powerful feature of weibull analysis is the ability to identify between different failure mechanisms in the same failure distribution. Mean time between failures (MTBF) is a very simple statistical measure of seal life. MTBF should not be used to compare large failure distributions (eg failures at two different plants), as there can be large errors caused by missing or poor data. MTBF is ideal for averaging the lives of seals on individual duties. The weighted life index adds variable weighting to the basic MTBF statistic. This enables a more accurate comparison between large failure distributions. However a weibull analysis is a better statistical method, more informative, and simple to apply.

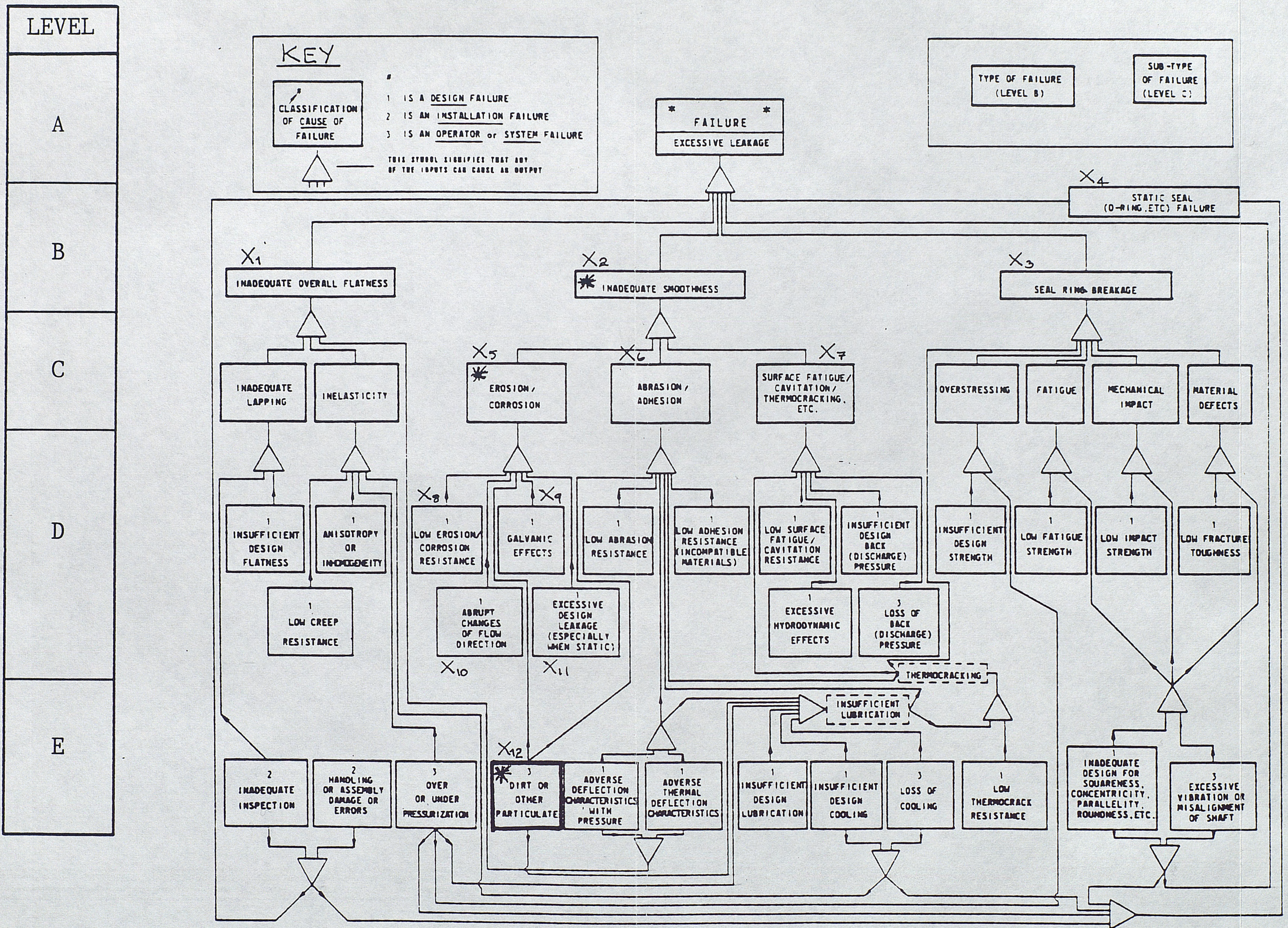
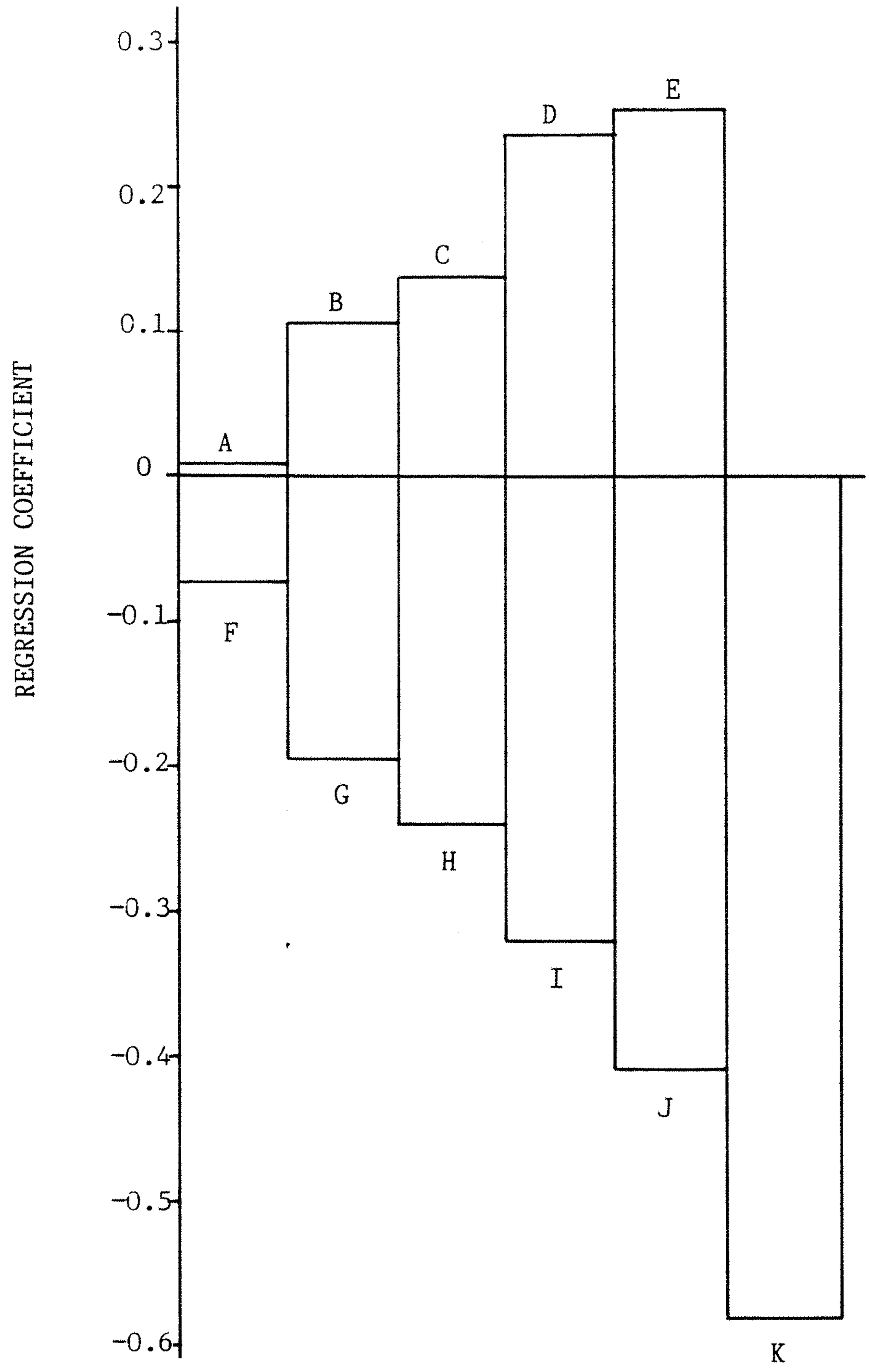


Figure 6.1 : A Mechanical Seal Failure Analysis Diagram (18)



SYMBOLS					
A	size	E	disch. pressure	I	suction pressure
B	solids	F	sealed fluid viscosity	J	pH (corrosive value)
C	sealed fluid b.p.	G	shaft speed	K	vibration
D	seal cost	H	sealed fluid temperature		

Figure 6.2 : Results of a Regression Analysis to Establish the Relative Effect of Eleven Parameters on Mechanical Seal Life (24)

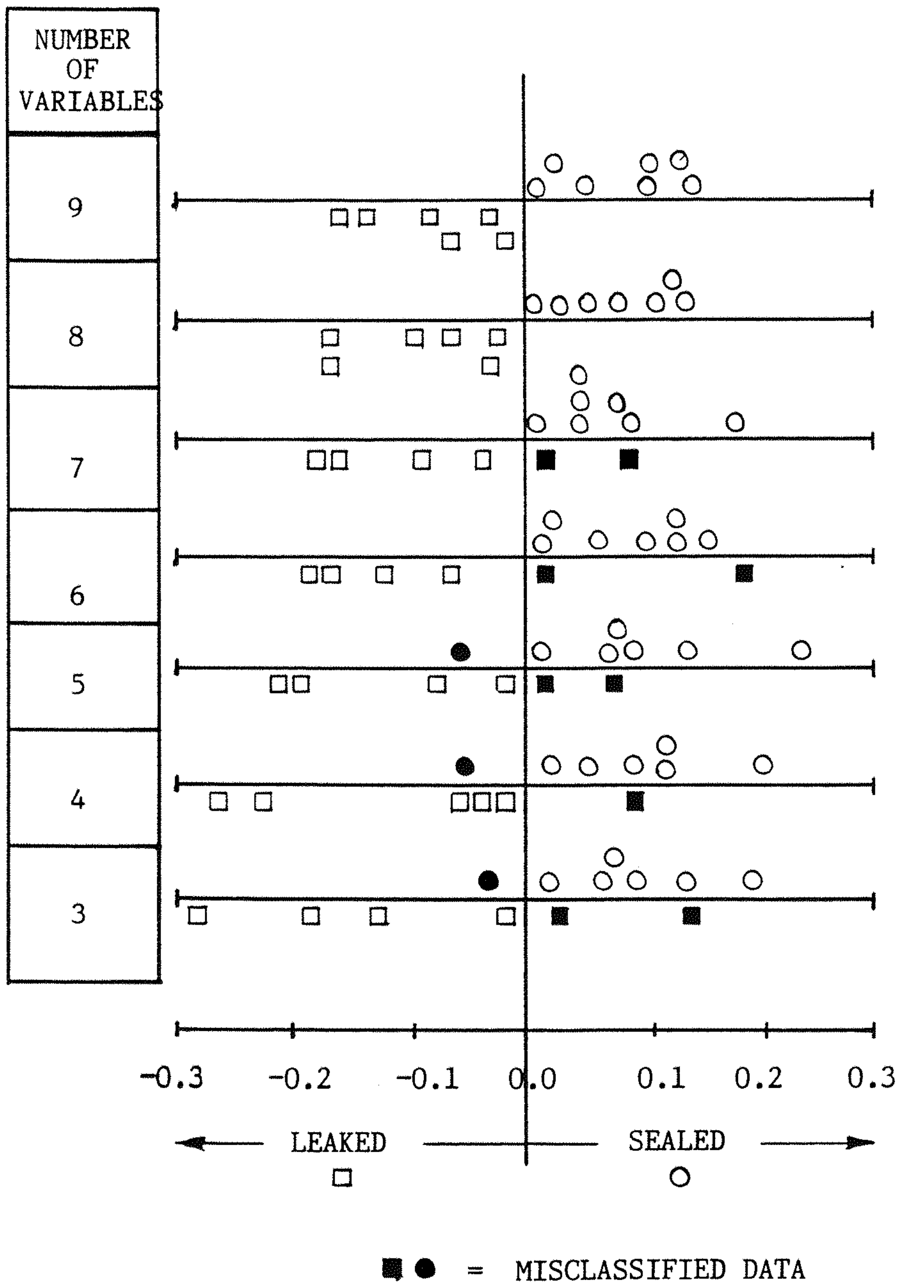


Figure 6.3 : Discrimination Plots Using Lip Seal Data (28)

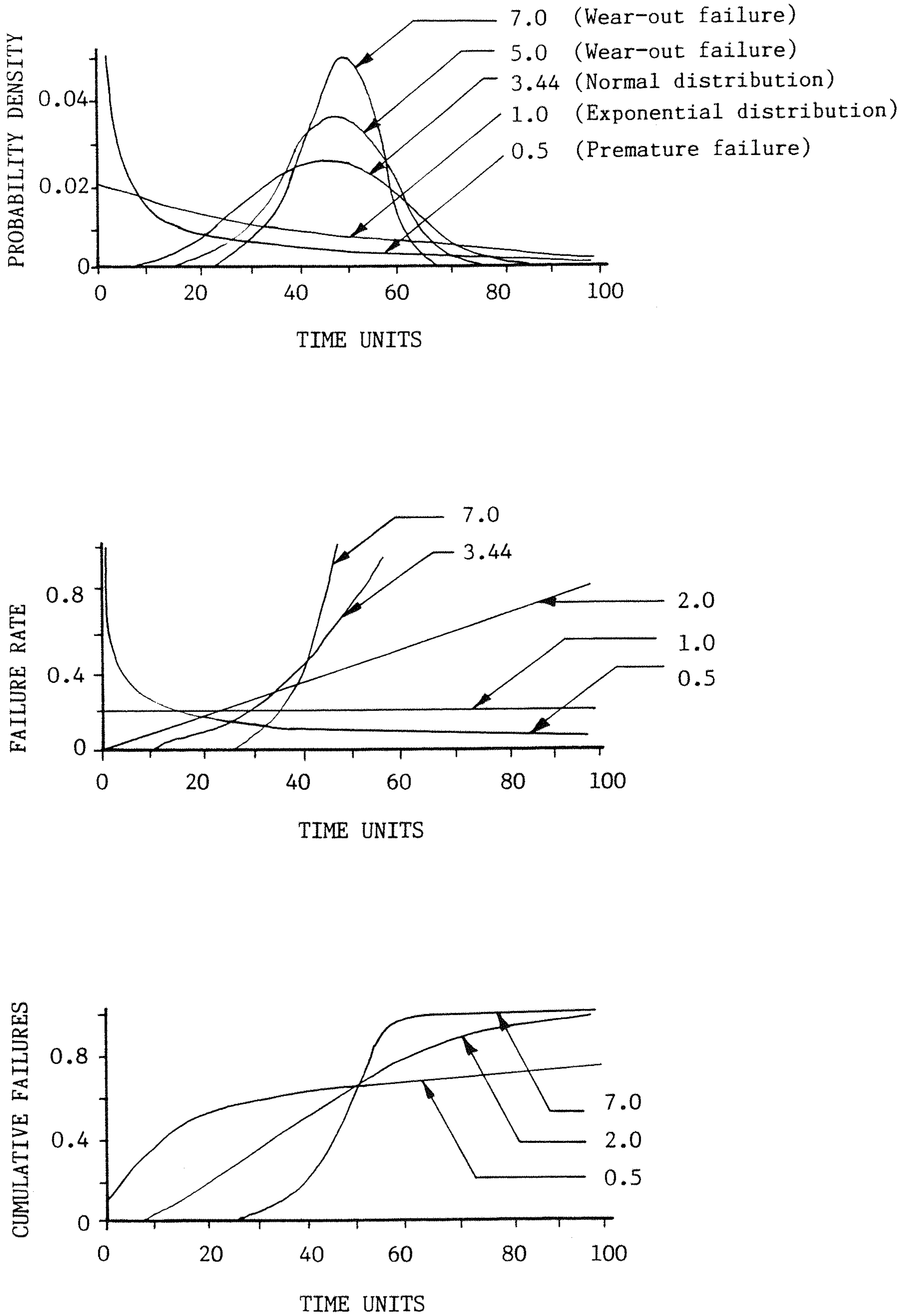


Figure 6.4 : The Effect on the Weibull Distribution of Varying the Weibull Index

The Normal Weibull Distribution			
To	B	Failure Mode	Possible Causes
To=0		Failure mechanisms operate as soon as the component is in service	
	B<1	Falling failure rate - never reaches zero. A premature failure mode, typical of "infant mortality".	Low safety margin (eg stress rupture), overloaded seal faces
	B=1	Constant failure rate. Random failures only. There is no dominant failure mechanism.	External component failures (eg bearings), random operational mishaps.
	B>1	Increasing failure rate. A wear-out failure mode from the start of service. This indicates a condition of premature wear-out. Wear-out failures should not occur until the design operating life is achieved.	Gradual wear in service.
To>0		Components are intrinsically reliable until time To.	
	B<1	Falling failure rate after a sudden increase at To. An acceptable wear-out failure mode if To is in excess of the components design life. However there is a very sharp drop in reliability if components are operated beyond time To.	Fatigue (eg metal bellows), fatigue of seal faces through excessive thermal cycling (eg thermal cracking).
	B>1	Increasing failure rate, starting at time To. The "ideal" wear-out failure mode if To is in excess of the components design life. Reliabiity falls gradually after timeTo.	Erosion, corrosion, build up of deposits - leading to hang-up.
To<0		The failure mechanism operates before the component is in service	
	B<1	Falling failure rate, with failures starting prior to service. A premature failure mode.	Damage during transit, damage during manufacture
	B>1	Increasing failure rate, with failures starting prior to service. A wear-out failure mode.	Limited shelf life, degradation of components due to storage conditions prior to service.

The Modified Weibull Distribution			
To=Tc	B<1	Increasing failure rate until time Tc, after which no failures occur. A premature failure mode.	Excessive wear
	B>1	Falling failure rate, reaching zero at time Tc. A premature failure mode.	Installation and assembly errors.

Table 6.2 : Interpretation of Weibull Distribution Parameters

7.0 CONSIDERATIONS WHEN COLLECTING DATA ON MECHANICAL SEALS

7.1 Relevant Data

Three types of design and performance data provide a comprehensive service history of a mechanical seal.

7.1.1 Design Data: criteria which determine the seal selection.

(a) Pump Data

- Operating environment (eg sealed product, pressure, temperature, shaft speed, etc) .
- Pump type.
- Material specification.
- Design tolerances and assembly clearances.

(b) Seal Data

- Seal type.
- Material specification (seal faces, secondary seals, and other elements).
- Auxiliary features (eg flush, cooling, and quenches)

7.1.2 Condition Monitoring: details of the operating environment and performance of the seal in service.

- Leakage rate.
- Vibration monitoring.
- Pump cavitation.
- Fluid film vaporisation.
- Pump starts and shutdowns.

7.1.3 Maintenance

- Pump modifications and overhauls.
- Modifications to the seal materials or seal type.
- Record the condition of the pump and seals.

7.1.4 Failure Data: details recorded after a seal failure

- Post-mortem of the seal.
- Pump condition.
- Any unusual operating conditions.
- Visual and audible evidence (eg vaporisation, pump cavitation, "squealing" noise from the seal, type of leakage, etc).

7.2 Implications of Incomplete Data Records

Missing data can have a very significant effect on the results of a seal failure analysis (section 6.6). Overall failure trends might not be affected too badly, but the records must be accurate and complete to produce quantitative results that allow comparisons between different data sources.

A fictitious scenario (fig 7.1) demonstrates the danger of using incomplete data records to assess the cause of failure. The scenario shows that it is very difficult to deduce the correct cause of failure if the data records (section 7.1) are incomplete.

Scenario (fig 7.1)

The diagram and table explain the processes leading to a seal failure, and the data records that could have been made. It is interesting to see what causes of failure would be deduced if only part of the data was recorded.

(a) Maintenance Records

The seal failure may be due to excessive vibration prior to the motor bearings being replaced. Poor alignment after the motor change may be the cause of seal failure. The seal failure may be unrelated to the motor bearing overhaul. There is insufficient information to establish the cause of failure. It is not even possible to say whether the seal failed as a consequence of the motor bearing failure - since there are no records of leakage before the bearing replacement.

(b) Maintenance and Operations Records

The seal failed due to excessive leakage after a period in service of 9 days. Details of the personnel who installed the motor appear unnecessary. The records suggest a sudden seal failure related to the motor overhaul. However the actual cause of the seal failure remains open to speculation.

(c) Maintenance, Operations, and Condition Monitoring Records

With the full data records it is clear that there was a problem in the way that the motor was reinstalled after the bearing failure. The condition monitoring record provides the most important information in this case. High vibration indicates that there is a problem with the pump-motor alignment. The misalignment has caused a rapid deterioration of the mechanical seal. Further investigation of the reason for poor alignment reveals poor training as the cause. So poor training was the actual cause of the mechanical seal failure. This could only be established with the aid of comprehensive data records.

7.3 Data Accuracy

There are several factors which can have a significant effect on the accuracy and quality of mechanical seal data:

- source or origin.
- experience, skill, and interest of the people recording the data.
- accuracy of instrumentation.
- system of data recording (ie logbooks, or computer records).

The accuracy of data is often overlooked in the results of subsequent study. The origin or source of service data on mechanical seals is often unknown, so the quality of the data is uncertain. This makes the results of quantitative studies less reliable.

The author carried out a study of mechanical seal life (6) whilst working at Plant B. This provided first hand experience of the limitations of using historical data. A pump maintenance log (containing information on mechanical seals) had been kept from 1967 to 1983, by the same rotating equipment engineer. At the start of 1984 a computerised system of maintenance records was introduced. There was a significant effect on the number of recorded seal failures (fig 7.2). A period of low recorded failures occurred immediately after the introduction of the new computerised record system. By mid-1985 the new system had become fully established, and the number of recorded seal failures almost doubled compared to the pump maintenance log. This suggests that data is missing for the period upto mid-1985. There is no evidence of a radical change in management philosophy, working practice, or plant modifications which could have caused a sudden deterioration in seal life. It is most probable that maintenance records were only kept on the more critical pumps upto 1984. With a computerised record system it was no more difficult to keep records on all pumps. So, the manual records upto 1984 are believed to be accurate, but apply to a smaller population of pumps than the subsequent computerised database.

Recorded life data is probably accurate to within 1-2 days, except where data is missing (ie the recorded seal life is actually two or more seal lives). An accuracy of 1-2 days would apply equally to a seal life of 10 days or 1000 days. However it is worth remembering that recorded seal life (ie time between failures) <u>includes</u> the seal repair and recommissioning period; this period is much more significant for the 10 day seal life.

The working relationship developed between the people recording and analysing seal failure data, can influence the quality of the data. A study of mechanical seals would take a minimum of 1-2 years. The data would normally be recorded by operations and maintenance personnel. An internal investigator (eg plant maintenance engineer) is able to keep in daily contact with the personnel recording the data. The investigator can quickly detect changes in the way data is being collected. An external investigator is much more remote, and probably relies on periodic visits to the plant. The quality and source of data is less certain if an external investigator is employed. An in-house study will produce more accurate data, as there is greater contact between the investigator and the personnel recording the data.

One of the greatest problems in selecting the correct mechanical seal is establishing accurate service conditions. When a pump is first commissioned, the seal is selected on the design conditions. Section 4.5.2 discusses the implications of inaccurate service conditions, on the selection of seal face materials. Instrumentation on the pump may enable a revised assessment of some service conditions seen by the seal. Some modern process plants (refineries in particular) have central computer controlled instrumentation which can display real-time schematics of the process conditions (eg temperature, pressure), and will retain a memory of these conditions for a period of days or weeks. The accuracy of other service data (eg sealed fluid viscosity, solids content, pH value, boiling point, saturated vapour pressure) is much poorer; especially hydrocarbon duties (section 8.2.2) which are defined within a range of properties.

7.4 Types of Database

7.4.1 An Events Database

Basic historical records are normally in the form of an events database. These records contain details of events in the running of the plant, over a period of time. Workshop records, the operations log, and condition monitoring reports would all be classified as events data.

7.4.2 A Generic Database

To make the basic events data more useful for analysis purposes, it is desirable to create a generic database (20) .Proprietary computer programs (ie spreadsheets) make it simple to sort large databases in a generic way. The events database could calculate the MTBF for each seal duty. A generic database could be formed by sorting the events data into a large number of generic groups (eg balanced seals, double seals, seals operating on water duties, etc). The average MTBF for seals in each group would be entered into the generic database. The group type will determine the size of the group. A small generic group (eg seals operating on HF acid) will provide a more accurate indication of performance than a large group (eg balanced seals). A feature hierarchy (fig 7.3) provides a useful guide to the probable size of different generic groups. The larger groups are to the left of the diagram. It is possible to develop a more complicated generic database by combining the basic generic groups (eg balanced seals, on HF acid duties). A generic database can be used to isolate particular groups of seals which are a problem.

Record Type	Day	Record Details
Workshop and maintenance reports	1 11	Renew motor bearings Renew mechanical seal
Condition monitoring	3 6 9	High vibration Seal dripping occasionally High vibration Seal dripping continuously High vibration Continuous flow from the seal
Operations log	1 2 11	Motor removed to workshop Motor reinstalled Permit to work shows alignment by J.Bloggs (Contractor) Pump back on line Product spraying from the seal, so pump shutdown.

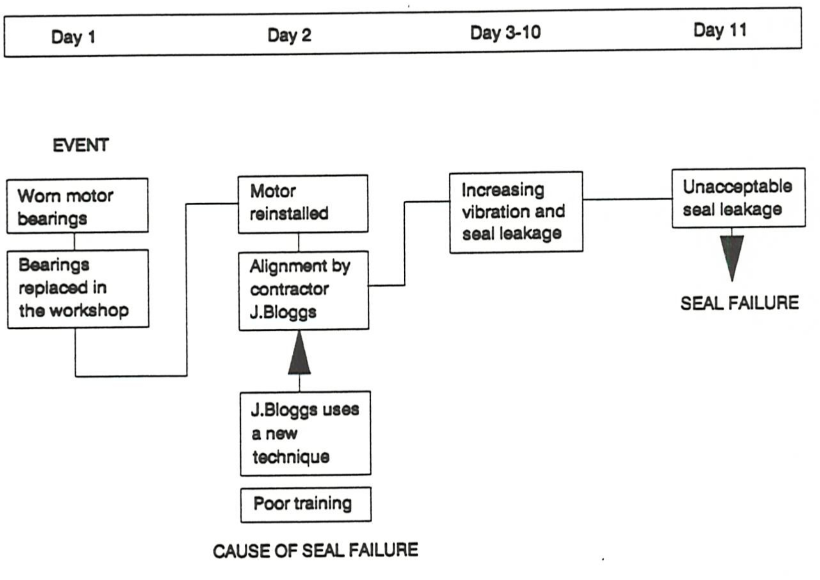


Figure 7.1 : Seal Failure Scenario

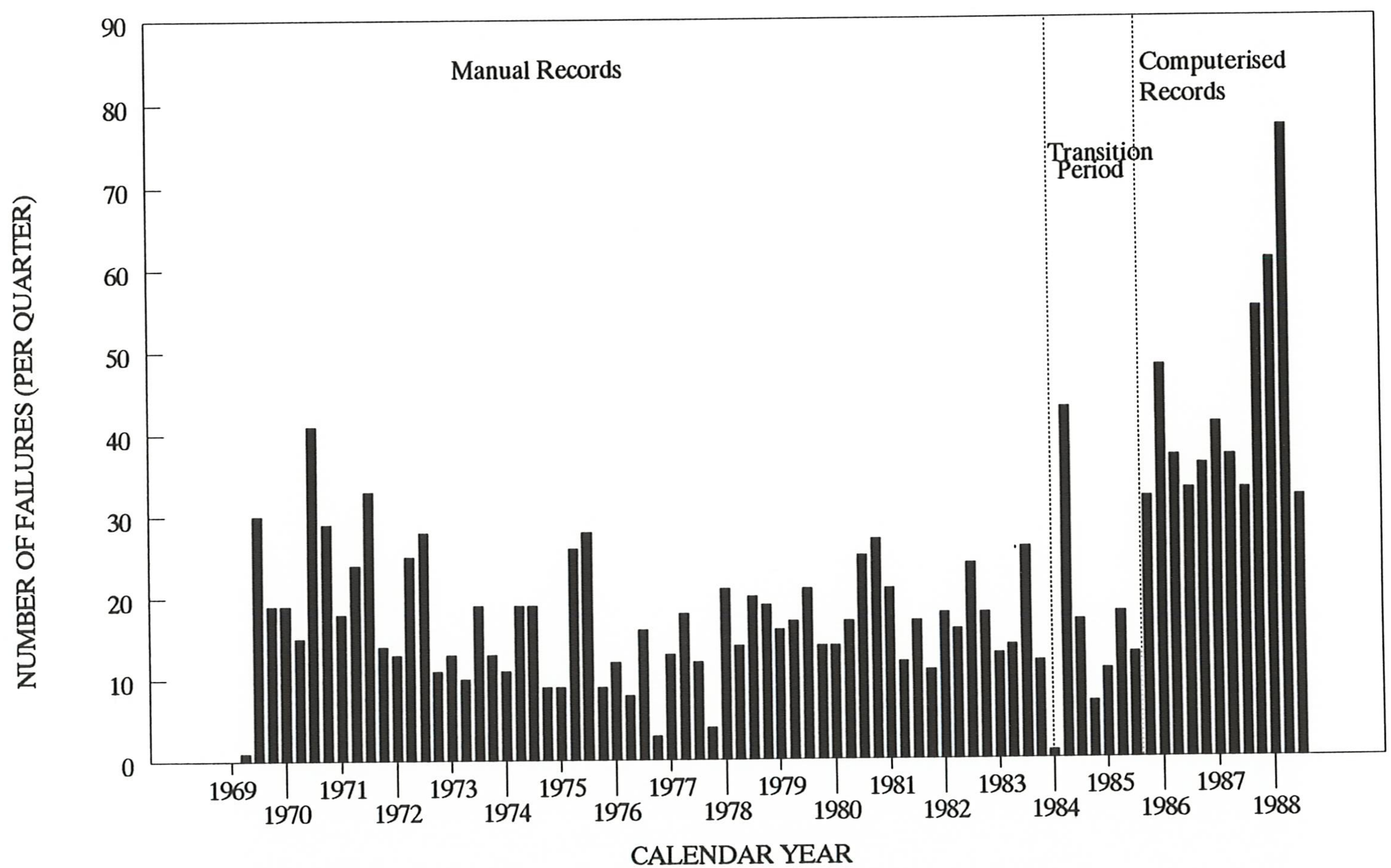


Figure 7.2 : Recorded Mechanical Seal Failures at Plant B

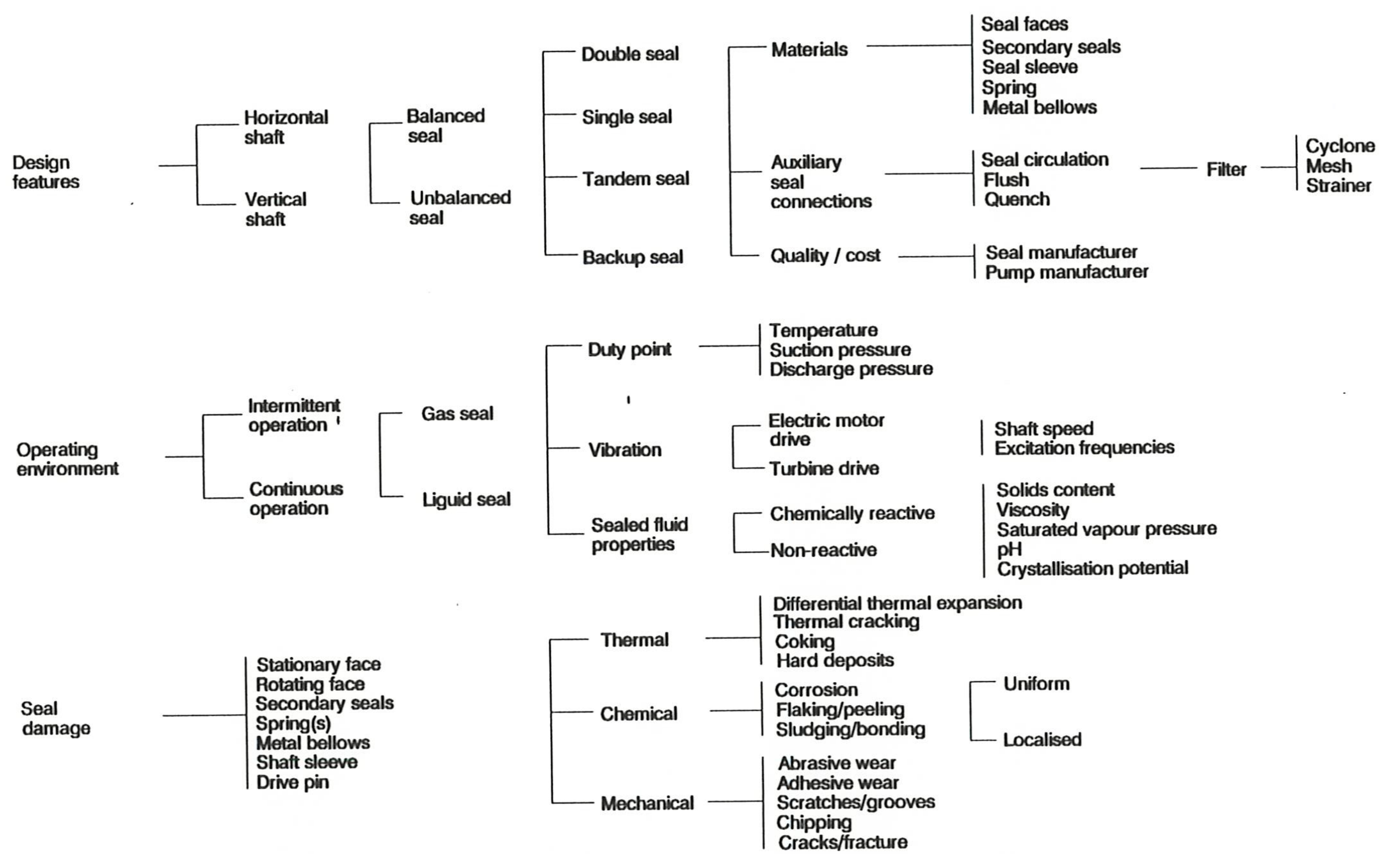


Figure 7.3 : A Feature Hierarchy for Mechanical Seals

8.0 IMPROVING AND PREDICTING MECHANICAL SEAL LIFE

All the costs in this chapter have been equated to 1990 pounds, and all references to seal life refer to installed life.

8.1 Cost Benefits

The primary reason for improving equipment life is to reduce maintenance costs and avoid disruptions to the plant processes. Centrifugal pump failures cause a large proportion of the mechanical maintenance in process plants. Pump failure seldom causes a plant to shutdown because most duties have a spare. However it is not uncommon for the spare pump to have a lower rating than the primary pump (it may be worn, an old design, or transferred from an obsolete duty), so there may be a reduction in plant throughput.

Surveys of petrochemical plants and oil refineries (6,23) have shown that upto 75 % of centrifugal pump failures are caused by mechanical seal failure. Chapter 3 provides a quantitative assessment of the cost of operating mechanical seals at Plant B (a medium sized oil refinery).

Indirect costs (ie lost production, standby equipment, and inventory) account for about 65 % of the total seal operating cost (fig 8.1) at Plant B; about £1,250,000 pa (£1785 pa, per pump). The direct costs associated with mechanical seal maintenance (ie labour, and new seal components) contribute the other 35 % of the total seal operating cost; about £630,000 pa (£900 pa, per pump). These are very significant costs, and have a severe impact on the operating profit margin of the plant.

Standby equipment is the largest item affecting mechanical seal operating costs (54 %). At Plant B standby equipment is costing around £1,000,000 pa (£1430 pa, per pump) in lost investment potential. Even if only the capital cost of the equipment is considered, this still amounts to nearly £525,000 pa (£750 pa, per pump)(see 3.4.2).

The present cost of operating mechanical seals is high. Improved seal life and reliability would reduce the number of standby pumps, size of the inventory, and direct maintenance costs. The cost analysis in Chapter 3 suggests that the direct seal maintenance cost would fall from £1170 to £290 pa (per pump), if the average seal life (MTBF) is increased from 1 year to 4 years. There is a very strong economic argument for improving seal life and reliability.

The cost of the mechanical seal inventory at Plant B is around £360 pa (per pump). Until recently this figure was probably representative of the majority of process plants. However there is an increasing trend towards the concept of consignment stock, which drastically reduces the capital investment in the inventory (see 3.3.3). Stock is supplied to the inventory at the plant, but not paid for until it is actually used. Suppliers like this arrangement because it reduces their own storage/warehouse requirements, and provides a guaranteed buyer of the stock.

Tougher environmental legislation will increase the cost of mechanical seal components. It will probably become necessary to install dry-running backup seals, tandem seals, or double seals on the majority of process duties to meet future restrictions on leakage and vapour emissions. The cost analysis at Plant B found that in mechanical seal maintenance, component costs are about 1.5 times the labour cost. This relationship was based upon a sample of pumps in which 66 % of the seals were simple single seals. New legislation will significantly increase the ratio between labour and component costs.

To increase plant efficiency and reduce costs, most process plants have regular shutdowns to undertake maintenance work. Pressure vessels must have statutory inspections at two year intervals, so most process plants make this the minimum period between shutdowns. The new Pressure Systems Regulations (which come into force in July 1994) will remove the two year statutory inspections. This may lead to longer periods between shutdowns, if pressure systems are well maintained and can be safely operated for extended periods. Between shutdowns the plant retains a small maintenance staff to deal with sudden failures. On many process plants a shutdown provides the only opportunity to carry out maintenance on many pieces of equipment. Maintenance is planned so that the majority of plant equipment is overhauled to ensure trouble-free running between shutdowns. At present mechanical seal maintenance is unplanned because of the unpredictability of seal life. In addition many mechanical seal duties have a MTBF significantly less than two years (ie the common shutdown frequency). The cost of replacing a mechanical seal during a routine shutdown is a fraction of the cost when the plant is on-line. During a routine shutdown there are no complications with isolating the pump from the plant process, and many pumps are overhauled anyway (eg bearings, wear rings, etc).

The mechanical seal failure data from Plant A and Plant B (see Chapter 9) show that less than 40 % of the seals have lives in excess of 3 years, and around 35 % of the seals have lives shorter than 4 months. These statistics are comparable with other studies (23,36). Improving seal life and reliability (so that seal life is more predictable), will enable a considerable reduction in seal operating costs. Considerable improvements in mechanical seal performance are necessary to achieve the goal of trouble-free running between shutdowns. If this goal can be achieved it is probably economical on many duties to risk lost production rather than install standby equipment.

8.2 An Analysis of Mechanical Seal Data from Plant A and Plant B

Description of the Sources of Data

Chapter 9 contains a detailed description of all the mechanical seal data obtained from Plant A and Plant B. The descriptions that follow describe the particular data used for the analyses in section 8.2.1, 8.2.2, and 8.2.3.

Plant A is a large UK Petrochemical and oil refining plant (refining about 260,000 barrels of crude oil per day). The plant operates 2600 mechanical seals in centrifugal pumps. Mechanical seal failures have been recorded in a computerised database since

01/01/88. This data set records the failure of 2457 mechanical seals, and trouble-free operation (upto 01/04/91) of a further 1300 mechanical seals. The data used in the weibull analysis (section 8.2.3) consists of 98 mechanical seal failures, from the period 11/08/89 to 18/06/90. Copies of failure record sheets were available for these 98 mechanical seal failures. These failure record sheets were used to assess the cause of each failure (section 8.2.1).

Plant B is a medium sized oil refinery (refining about 90,000 barrels of crude oil per day). The plant has about 700 centrifugal pumps fitted with mechanical seals. The data used in the weibull analysis (section 8.2.2) covers mechanical seal failures during the period 1967 to 01/01/88. The data set contains 1364 mechanical seal failures. The manual recording of seal failures during the period 1967 to 1983 is known to have missed some seal failures. Later seal failure rates (using a computerised record system) suggest that upto 50 % of seal failures were not recorded during the earlier period 1967 to 1983 (see 7.3, and fig 7.2). Some apparent lives will incorporate two or more seal lives. This is most likely to affect the long recorded seal life data. It is also probable that the recording of seal failures was restricted to certain pumps when the recording system was manual (ie 1967 to 1983). The introduction of a computerised maintenance record system made it far easier to record seal failures on every pump at the plant, since the failure records were now generated from work orders.

8.2.1 An Investigation of the Relationship Between Mechanical Seal Life and Cause of Failure

Plant A provided record sheets for 98 mechanical seal failures, with sufficient information to establish the probable cause of each failure. The same mechanical seal failures are used in 8.2.3 to investigate the relationship between seal life, and seal face materials. Twelve basic causes of failure were established from this data (fig 8.6). The data has been analysed to establish if there are any relationships between mechanical seal life, and the cause of failure. Trends do emerge despite the limited number of failure cases.

Misalignment of the seal faces and the shaft axis is the most significant cause of short life (ie. less than 50 days) mechanical seals (fig 8.4, fig 8.6). Misalignment can cause rapid fatigue failure of metal bellows, severe wear of secondary seals (causing leakage past the secondary seals), and uneven wear/overheating/dry running of the seal faces. This type of misalignment does cause failure on longer life seals, but this is much less common.

Embedded metal particles in the carbon face is the next most significant cause of short life failures (fig 8.4, fig 8.6). This cause of failure is associated with a particular sealed fluid (IPA) on seals with tungsten carbide and niresist counterfaces. The majority of these failures (63%) occur on seals with lives under 50 days. Other seals on the IPA fluid duty use ceramic counterfaces. The ceramic v carbon seals did not suffer from this cause of failure, which confirms that the metal particles do originate from the tungsten carbide and niresist counterfaces. This cause of failure clearly results from an incompatibility between the counterface material and the particular sealed fluid (IPA).

Hang-up (see 4.4.1) due to coking and crystallisation are a significant cause of failure in the seal life range 50 to 300 days (fig 8.4, fig 8.6). It is interesting to note that the two material combinations containing silicon carbide did not suffer **any** hang-up failures (fig 8.3). The excellent heat dissipation properties of silicon carbide have dramatically reduced the incidence of hang-up.

Failure of auxiliary seal systems (eg. quench, seal flush, and cooling lines) is an important cause of failure in the 0 to 50 days, and 50 to 300 days seal life ranges (fig 8.4, fig 8.6). The design and/or operation of the auxiliary seal systems is clearly a cause for concern at Plant A. This type of failure should be quite simple to "engineer" out. Auxiliary seal lines often have an orifice plate inserted into them, for pressure and flow measuring purposes. The poor position of this orifice is often the cause of blockages. The auxiliary seal lines should always be cleaned out and checked for blockage when a seal is overhauled.

Incorrect setting of the seal compression was an important cause of failure in the 50 to 300 day seal life range (fig 8.4, fig 8.6). If the seal spring is not preloaded correctly then the seal face pressure is wrong. This causes overheating and excessive wear if the spring is overcompressed. Instability (see 5.2) and the ingress of large particles can cause the seal faces to move apart (resulting in high leakage rates), if the spring is under compressed. This cause of failure is clearly an installation error, and should be quite simple to avoid through better training/procedures of fitters. It is interesting that an incorrect seal setting causes very few failures on seals in the 0 to 50 day life range (ie very short life), and no failures on seals with a life in excess of 300 days.

Dry-running is by far the most significant single cause of mechanical seal failure in the set of seals from Plant A (causing 20% of all the failures)(fig 8.6). The majority (70%) of dry-running failures occur in the 50 to 300 day seal life range (fig 8.4). Only 10% of these failures occur in the 0 to 50 day seal life range, so dry-running is not a particularly significant cause of start-up failures and very short seal lives. Dry-running is usually inferred from the condition of the seal faces; heavy wear, thermal cracking, chipping, overheating (blue colour of metallic components), and fractured faces. Dry-running failures are probably the most difficult to avoid. The data suggests that dry-running failures on start-up are not very significant; these are the most simple to avoid, through better pump start-up procedures. Most dry-running failures appear to occur as a result of the normal seal operating conditions. Since these failures mostly occur in the 50 to 300 day seal life range, there are similarities with the hang-up failure mechanisms (fig 8.4). This suggests that seals generally fail by prolonged exposure to dry-running conditions, rather than an isolated incident (eg. due to a plant upset). Some seal face material combinations have a greater resilience under dry-running conditions (see 4.5.1, tungsten carbide, and silicon carbide against carbon). As expected these seal face material combinations have the lowest proportion of dry-running failures (15 to 17%)(fig

8.3). Ceramic and stellite seal face materials have the poorest dry-running properties (see 4.5.1) and clearly have the highest proportion of dry-running failures (27 to 38%)(fig 8.3). Replacing stellite and ceramic seal faces with silicon carbide and tungsten carbide faces significantly improves overall seal life, because of a much greater tolerance of dry-running conditions.

Mechanical seals with life in excess of 300 days, are most likely to fail due to an external component failure (eg. bearings), shaft misalignment, dry-running, secondary seal failure (eg. embrittlement through prolonged exposure to high temperature), or an internal seal component failure (other than the seal faces, secondary seals, bellows, or spring(s)).

8.2.2 An Investigation of the Relationship Between Seal Life and the Sealed Fluid, Using Weibull Analysis

Weibull analysis has been used to compare the failure distributions of mechanical seals operating on a wide range of fluids at Plant B. The typical fluid properties are shown in Table 8.2. The main aim of this analysis was to determine whether there are any fundamental similarities between the failure distributions of mechanical seals operating on hydrocarbon, water, and chemical sealing duties. Mechanical seals on water duties are normally treated separately from mechanical seals on hydrocarbon and chemical duties.

If the failure distributions are similar, the most significant factors affecting the life and reliability of mechanical seals must be basically the same, whatever the sealed fluid. If the failure distributions are very different then the factors affecting the life and reliability of seals are closely associated with the properties of the sealed fluid.

It is important to remember that the seal type can affect the fluid "seen" between the seal faces (section 9.4). In a double seal the barrier fluid, rather than the sealed fluid, will exist between the seal faces of both inner and outer mechanical seals. In a tandem seals the sealed fluid will exist between the seal faces of the inner seal, and the barrier fluid between the faces of the outer seal. The barrier fluid is chosen to be non-hazardous, compatible with the sealed fluid, and provide the least arduous conditions for the mechanical seals. LPG and HF acid pumps normally use double or tandem seals due to the hazardous nature of the pumped fluid.

Hydrocarbon Sealing Duties

Liquefied petroleum gas (LPG) is made up of propane and butane (fig 8.2). LPG is a gas at atmospheric pressure. Due to the hazardous nature of LPG (highly inflammable), double, tandem, or single plus backup seals are used on LPG pumps.

Gasoline contains a blend of hydrocarbons (fig 8.2), so its properties have a broad spectrum. The wide boiling range of gasoline, due to the hydrocarbon blend, makes seal specification quite difficult. It can be difficult to avoid the vaporisation of light hydrocarbons in the gasoline blend, at the higher temperatures generated between the seal faces.

Naphthas are the major constituents of gasoline (fig 8.2), and generally need processing (Reforming) to make suitable quality gasoline. We would expect the performance of mechanical seals on naphtha duties to be very similar to the performance of seals on gasoline duties.

The term "heavy hydrocarbons" has been used to describe all the vacuum distillation products (fig 8.2). These include gas oil, and residue. Vacuum distillation takes place at high temperature (390 - 450 deg.C) and low pressure (0.03 - 0.05 bar abs.). Mechanical seals on heavy hydrocarbon duties generally suffer the most problems. Operating experience suggests that the most common failure mechanisms on these seals are hang-up and coking (see 4.5.3).

Lubricating oils are manufactured using some of the short residue produced by vacuum distillation (fig 8.2). The light hydrocarbons such as LPG, gasoline, and naphtha all have fairly poor lubricating properties. Operating experience suggests that the most common seal failure mechanisms on light hydrocarbon duties are chipping, excessive wear, and thermal cracking, of the seal faces. These failure mechanisms are all associated with poor natural fluid lubrication, overheating, and fluid film vaporisation.

Water Sealing Duties

Sour water is the name given to water containing dissolved hydrogen sulphide and ammonia. In a refinery the sour water system is normally alkaline. The sour water stripping process (to extract the hydrogen sulphide and ammonia from the water) is optimised at a pH of 9.5. Elsewhere in the sour water system the pH could vary from 9.5 to slightly acidic. Sour water is formed on a refinery wherever steam or water are used in a process. Sulphur and nitrogen in the oil react with the water or steam to produce ammonia and hydrogen sulphide.

Sweet water is the name given to water containing no dissolved hydrogen sulphide or ammonia. Boiler feed water and water which has passed through the sour water stripper would be classified as sweet water. Water is a poor lubricant, but has a small boiling range. Consequently sweet water duties are considered to be quite benign.

Chemical Sealing Duties

Hydrofluoric acid (HF) is an extremely hazardous chemical. HF is used in the alkylation process (fig 8.2) as a catalyst. Typically the process uses an acid concentration of between 83 and 92 % hydrofluoric acid (by weight), and less than 1 % water. Double, tandem, or single plus backup seals are used on HF acid duties. The process normally takes place at ambient temperatures, and at moderate pressures. A careful material specification is required to avoid corrosion or reaction problems with the seal materials.

Comparing The Weibull Distributions

The weibull analysis shows that mechanical seals on a wide range of fluids have very similar failure distribution characteristics (Table 8.1). There are two or more distinct failure modes for each distribution, which have a similar weibull index and proportion of the failures. The only parameter which varies significantly with the fluid being sealed, is the characteristic life (Table 8.3). So the distributions have been plotted using a dimensionless age parameter.

Dimensionless Age Parameter = t/h

t = Age when the failure occurs.

h = Characteristic life of the whole failure distribution.

This produces a weibull plot (fig 8.7) with a surprisingly narrow scatter of points. The weibull plot suggests that a single curve could be used to predict seal life and percentage (of the population) failed against time, once the characteristic life of the sealed fluid is known. Mechanical seals at Plant B almost fit a single curve over a range of hydrocarbon, water, and HF acid duties. A single shape of curve indicates that the **most significant** failure mechanisms are common to mechanical seals on a very wide range of sealing duties.

The data confirms that mechanical seal life is dependent on the fluid being sealed, but that the same failure mechanisms cause the majority of seal failures on a wide range of fluids.

First Failure Mode

The first mode on all the duties (Table 8.1) has a weibull index greater than 1.0, which indicates an increasing failure rate. This is unexpected since the classic bathtub curve has an initial mode in which the failure rate decreases (see fig 4.2). However the recorded seal life includes the time to overhaul and recommission the pump. Bertele (36) suggests that the calendar time between failures forms a poor basis for weibull analysis, because the actual time to overhaul and recommission the pump is unknown. The author's experience indicates that the time taken to overhaul and recommission a pump after a mechanical seal failure is between 1 and 3 days. The weibull distributions from Plant B (Appendix A1) show that the first failures are recorded in this range.

Seal lives upto about 50 days, on all the sealed fluids (Table 8.1), fall into this first failure mode (weibull index > 1.0). The data from Plant A (section 8.2.1, fig 8.4, fig 8.6) provides a valuable insight into the most significant causes of failure in this short seal life range. Poor shaft-seal alignment, and failure of the auxiliary seal systems (ie blocked) are probably the most significant failure mechanisms in this first failure mode. The weibull distributions show that the failure rate increases over the seal life range 0 to 50 days. This wear-out characteristic is consistent with the two failure mechanisms described above. The severity of the misalignment will affect the time to cause fatigue failure in metal bellows, and excessive wear of secondary (elastomeric) seals. Failure of auxiliary seal systems is most likely to occur on initial start-up, as the result of a blockage (eg. debris left in the piping, or hardened product due to poor pump shutdown procedures), or an incorrect start-up procedure by an operator. The data from Plant A also shows that hang-up due to crystallisation, is a significant cause of failure on seals with a life less than 50 days. Crystallisation can occur on fluid sealing duties that contain dissolved solids. Sour water and HF acid are the two fluids (in those studied at Plant B) most likely to cause crystallisation under suitable conditions (ie evaporation).

The first failure mode accounts for between 4% and 20% of seal failures on the different duties (Table 8.1). Naphtha and gasoline seals have an identical weibull index (as expected) but 20% of gasoline seals fail in the first mode compared to only 4% on naphtha duties. There appears to be a problem with several gasoline seal duties which is resulting in an abnormal number of early failures. The first mode lasts longest (lives upto 80 days) on sweet water duties, where the failure rate is almost constant (weibull index 1.06). Hang-up failures are unlikely on sweet water duties, because coking and crystallisation are not associated with this type of fluid. Hang-up will may occur as a result of elastomeric secondary seal degradation.

The two most significant causes of mechanical seal failure (based upon data from Plant A) are independent of the sealed fluid properties (ie. misalignment, and auxiliary seal system failure). The weibull distributions (Appendix A1) all have a distinct first mode, which has a weibull index greater than 1.0, and seal lives in the range 0 to 50 days. The next most significant cause of mechanical seal failure (based upon data from Plant A) is dependent on the sealed fluid (ie hang-up due to crystallisation). The two duties most likely to encounter crystallisation (ie sour water, and HF acid) have the highest (but very similar) weibull index (Table 8.1). The first distinct failure mode on all the weibull distributions from Plant B (Appendix A1), can be explained in terms of the three most significant short life failure mechanisms from Plant A. The weibull plots indicate that this first failure mode has an increasing failure rate characteristic; this is consistent with the characteristics of these three failure mechanisms. The weibull plots show that this first failure mode exists for seal lives upto about 50 days, on hydrocarbon, water, and chemical fluid sealing duties. There is a very distinct change of slope on all the weibull distributions after this first failure mode.

<u>Failure Modes in the Mid-life Range</u>

The weibull distributions (Appendix A1) show that there is an important change in the failure distribution of mechanical seals on all the sealed fluids, at seal lives around 20 to 50 days. This signifies the end of the first failure mode. The mid-range failure mode(s) have a weibull index less than 1.0 (Table 8.1). This indicates a decreasing failure rate with time.

The majority of the seals on hydrocarbon duties are in a single mid-range failure mode (80 to 96 %), and have a weibull index of 0.72 to 0.84.

The sweet water duty (Table 8.1, Appendix A1 - fig A1.5) has two failure modes which correspond to the second mode on the hydrocarbon duties. These modes have a weibull index in the range 0.62 to 0.88, and account for 90% of the seals failing on sweet water duties. There is very little difference between the failure distribution characteristics of sweet water and the hydrocarbons.

The sour water duty (Table 8.1, Appendix A1 - fig A1.6) has three failure modes which correspond to the second mode on the hydrocarbon duties. These modes have a weibull index in the range 0.63 to 1.11, and account for 88% of the seals failing on sour water duties. Once again there is very little difference in the failure distribution characteristics of this water duty and the hydrocarbon duties.

The weibull distributions for the water duties are probably more complicated than the hydrocarbon distributions due to considerably fewer data points. This highlights the need for a large number of data points to carry out a useful weibull analysis. This analysis shows that 50-60 seal lives are an absolute minimum. With around 200 data points (Appendix A1 - fig A1.1, fig A1.3, fig A1.4) a fairly smooth weibull plot is obtained, in which discontinuities (ie changes of failure mode) are distinct.

The HF acid duty (Appendix A1 - fig A1.7) has two failure modes which correspond to the second failure mode on the hydrocarbon duties. The first of these modes has a weibull index of 1.17. This indicates an almost constant failure rate (slightly increasing) after initial start-up. This implies that there are a relatively high number of random type errors during the early life running of these seals. The slightly increasing failure rate suggests that premature wear-out failures are significant. Special protective clothing is required when working in the presence of HF acid; this reduces the mobility and vision of operators and fitters. As a consequence there are probably more mechanical seal failures due to operator or fitting errors. The second of these modes of failure accounts for HF seals with lives over 400 days (23%). The weibull index is 0.89. This indicates a slowly decreasing failure rate with time. The general characteristics of the seal failure distribution for HF acid is very similar to the failure distributions for water and hydrocarbon duties.

<u>Long life Failure Modes</u>

We would expect mechanical seals to conform to the classic "bathtub" distribution (fig 4.2). The failure modes described above suggest that almost all the seal failures at Plant A fail on initial start-up or through premature failure mechanisms. Some of these failures will be due to random failure mechanisms (see 4.4.2), however none of the distributions have a region in which the failure rate is constant (weibull index of 1.0). Following a period of low constant failure rate (random failures) the "bathtub" distribution predicts a final region in which the failure rate increases. This is the long life wear-out mode (section 4.3).

A long life wear-out mode is evident on the LPG distribution (Appendix A1 - fig A1.1). This mode has a weibull index of 1.29, starts around 1600 days, and accounts for 10% of the LPG seal failures. The other distributions show vague signs of a long life wear-out mode starting between 5000 and 8000 days, but accounting for less than 2% of the failures in each distribution.

8.2.3 An Investigation of the Relationship Between Seal Life and the Seal Face Materials, Using Weibull Analysis

The data from Plant B covers the period 1967 to 1988. Few of the seals had silicon carbide seal faces. A large proportion of the seals had stellite v carbon or niresist v carbon seal face material combinations. The data from Plant B could not be arranged to give the age distributions of seals with each combination of face materials. Plant A provided record sheets for 98 mechanical seal failures, with details of the seal face materials. The two plants are owned and operated by the same company.

Weibull distributions have been plotted for six seal face material combinations (Appendix A2) at Plant A. Due to the limited number of data points for each distribution (6 to 39 data points), only the general slope of each distribution has been calculated.

The Stellite v carbon and niresist v carbon material combinations have the most data points (16 and 39 respectively). The weibull indices are 0.87 and 0.92. This agrees closely with the weibull parameters at Plant B (Table 8.1), where a large proportion of the seals had these material combinations. The weibull distributions for the other material combinations are significantly different (Table 8.4, Appendix A2 - fig A2.1). So it would appear that the shape of the weibull distribution is dependent on the seal face materials.

The confidence attached to the weibull parameters in this analysis is not as high as for the previous analysis (ie Plant B), due to the small number of data points for each weibull distribution. However even a few data points can provide a general indication of the failure distribution characteristics. The silicon carbide v silicon carbide distribution has very little scatter (Appendix A2 - fig A2.1a), an almost random failure characteristic over its whole range (weibull index 1.07), and the longest characteristic life (350 days). There is no evidence of a higher failure rate at initial start-p and during the first 100 days; unlike all the failure distributions at Plant B. Clearly the SiC v SiC material combination is far less sensitive to failure mechanisms which cause seal failure on start-up, or soon after. Two hard face materials are normally only used on particularly abrasive fluids (see 4.5.1). Silicon carbide has excellent heat dissipation and thermal shock resistance properties. There is often doubt about the resistance of two hard faces to dry running conditions. The data from Plant A suggests that the SiC v SiC combination is no more susceptible than any of the other materials.

Silicon carbide v carbon is generally considered to be the best seal face material combination on a wide range of sealed fluids. Some mechanical seal manufacturers are now making this material combination standard for almost all sealing duties (ie silicon carbide replacing alumina, stellite, and niresist). Unfortunately there are only six data points for this weibull distribution. The weibull distribution (Appendix A2 - fig A2.1c) suggests that using silicon carbide v carbon for the seal faces does not significantly improve the failure distribution characteristics. The weibull index (0.86) is very similar to the stellite v carbon and niresist v carbon distributions, and the characteristic life is short (120 days). However a close look at the individual causes of failure show that the six failures in this data set provide a very misleading picture.

Alumina v carbon has generally been superseded by tungsten or silicon carbide in new mechanical seals. The weibull distribution (Appendix A2 - fig A2.1d) confirms the operating experience that this material combination suffers a high number of start-up and short life failures. This material combination has by far the lowest weibull index (0.49), which indicates a very strong premature failure mode. Poor thermal shock resistance and thermal conductivity are resulting in a high proportion of the seals failing on start-up, and failing before they can run-in. It is interesting that there are then no more failures until after 400 days; seals with alumina v carbon face materials have good lives, if they do run-in.

Tungsten carbide v carbon is generally less common than silicon carbide v carbon, because tungsten carbide is more expensive and is not regarded as giving a better seal life. There are sealing duties where tungsten carbide cannot be replaced by silicon carbide (see 4.5.1). The data from Plant A is interesting because there is a distinct discontinuity in the distribution at around 110 days (Appendix A2 - fig A2.1e). The distribution upto 110 days has a very small scatter and weibull index of 1.69. There are then no more failures until 1300 days, which suggests a very sharp reduction in the failure rate after 110 days, and a distinct change in the failure mechanisms. These failure characteristics are similar to the distributions at Plant B.

The main difference between the failure distributions at Plant A and Plant B is the absence of a distinct first failure mode for seal lives under 50 days at Plant A (except for the tungsten carbide v carbon distribution, fig A2.1e). A smaller number of data points for the Plant A distributions makes it difficult to define individual failure modes - the scatter of the data points is too great. The niresist v carbon distribution (Appendix A2 - fig A2.1f) does appear to have a discontinuity at around 60 days, which would correspond to the end of the first failure mode found with the distributions from Plant B; but the discontinuity may just be a phenomena of the limited number of data points.

8.3 Creating a Mechanical Seal Database

A database is only as good as the quality of the data in it (at best!). Chapter 7 explored the requirements for collecting mechanical seal data to create a good quality events database. The purpose of any database should be clearly defined before any data is collected. It is a waste of time and effort to collect data which has no purpose in subsequent analysis. There is a tendency to record everything possible on the principle that "it might be useful". This is a poor approach which often causes the database to become so unmanageable that nothing useful can be extracted.

In section 8.2 data from Plant A and Plant B was analysed using weibull analysis, to examine the relationship between seal life, cause of failure, and the sealed fluid. The data from Plant A was in the form of standard mechanical seal failure record sheets (29). Plant A does have a computerised database containing design data (section 7.1.1), maintenance data (section 7.1.3), and failure data (section 7.1.4). The author was able to use some analysis results from this computerised database (section 4.4.3 and 9.2). A computerised database has a tremendous advantage over a paper based system, because it is very simple to interrogate the database and obtain statistics such as mean time between failures (MTBF), number of failures over a specified time period, and create graphs of the failure distribution. These common statistics are often the only way that a database is analysed; maintenance managers like to monitor the plant equipment reliability. Although these statistics provide a picture of the plant performance, they do not provide a means for **improving** mechanical seal life.

A worthwhile mechanical seal database will lead to improved seal life and a better understanding of the reasons for seal failure. Section 7.1 describes the relevant data that can be collected for a mechanical seal database. The detailed analysis of field plant data (section 8.2) and other studies in the literature provide important indications of the data most worth recording in a mechanical seal database.

For maintenance purposes a mechanical seal database would be used to establish "problem seals" (ie those failing most often), and whether seals are failing due to the way that pumps are maintained. This type of database should contain seal data (section 7.1.1), maintenance data (section 7.1.3), and failure data (section 7.1.4). The weibull analysis (section 8.2) clearly showed that the alignment of the shaft to the seal faces has a critical effect on seal life. Study (24,25,27) has clearly demonstrated the strong link between vibration, misalignment, and short seal life. Maintenance should place a greater emphasis on eliminating vibration and misalignment at the mechanical seal. To facilitate this process the mechanical seal database for maintenance purposes should contain the following dimensional information (29): seal working length, squareness of seal faces to shaft axis, concentricity of seal faces to shaft axis, shaft end play, shaft run-out, shaft deflection, shaft balance, and bearing clearances. By analysing the database it will be possible to identify pumps with poor dimensional tolerances, and evaluate the effect on mechanical seal life; there are recommended limits (29), but no published correlations with mechanical seal life.

A mechanical seal database could be used for evaluating the life of particular types of seal (eg tandem, double, etc), to find the best type of seal for a duty. This type of database is looking at the design and specification of the mechanical seal. Design data (section 7.1.1) and failure data (section 7.1.4) is most relevant to this type of database.

There has been very little inter-coupling of databases to date. This is partly due to the "secrecy" which seems to surround mechanical seal failure statistics, and more importantly the lack of a recognised database format. The National Centre of Systems Reliability (21) have a limited amount of data on mechanical seal reliability in their databank. This data has been collected from a number of different companies, and forms an insignificant central database. As the most critical factors affecting mechanical seal life emerge, there is a greater possibility of creating common database formats so that inter-coupling can take place. The weibull analyses (section 8.2) demonstrate the need for a large database to obtain meaningful results (ie 100+ data points for each weibull distribution).

8.4 Using Dimensionless Groups

Dimensionless groups have been used in many fields of engineering, to obtain better generality for comparing systems which are very complex. Studies (see chapter 5) have looked at several dimensionless groups to provide general relationships to describe mechanical seal behaviour. Success has been mixed. Two dimensionless groups have shown a good ability to predict seal suitability on particular duties; the duty parameter (section 5.5) and the delta-T factor (section 5.4). Unfortunately both of these dimensionless groups contain variables which are very difficult to measure accurately in the field without additional instrumentation. Other dimensionless groups, relying on external seal features, have a poorer correlation with seal life; the stability factor (section 5.2), thermal stress factor (section 5.3), and PV factor (section 5.6).

Dimensionless groups are most suited to **predicting** the performance of a seal, assuming specified design conditions. This is how most dimensionless groups are applied at present. The delta-T factor provides a measure of the temperature difference between the vaporisation temperature and sealed fluid temperature. A temperature margin similar to this is widely used for selecting mechanical seals. The principles behind the stability factor (eg critical face pressure, etc) are also widely used for selecting seals. There seems little hope of applying any of these dimensionless groups to monitor seals running in the field, because of the difficulty of accurately measuring some of the variables.

8.5 Using Statistical Methods

Statistical values such as mean time between failures (MTBF), failure rate, and number of failures over a specified period are all common measures of mechanical seal performance. These statistics, used by management to monitor equipment reliability, are frequently the only analyses performed on a mechanical seal database (section 8.3). Statistical methods are the standard way of analysing a database. Five methods were examined in Chapter 6, to examine their application to mechanical seal failure analysis.

Mean time between failures (MTBF) is a very simple statistical measure of seal life. It is very simple to apply, and provides an indication of the general seal performance. This explains its widespread use in the context of comparing the performance of different seal life distributions. However MTBF cannot reveal time based trends, and there can be large errors caused by missing or poor data (section 7.3). A weighted life index (section 6.2) can reduce the influence of errors caused by missing data, and "enhance" the comparative differences in seal life distributions, in a specified life range. The MTBF statistic can lead to improved mechanical seal life by highlighting individual "problem" duties; those duties with unusually low MTBF. The definition of a "problem" duty will probably depend on the average performance of seals at a particular plant. However a sealing duty with a MTBF less than 50 days is certainly a "problem" duty. Each "problem" duty is studied in detail to establish the causes of failure, and the seal specification (ie materials, seal design, and auxiliary seal systems) is modified to overcome the problem.

MTBF is the most convenient statistical measure of the life of a group of seals (and is the most commonly used). One problem with MTBF is that it includes the time for replacing and recommissioning the seal after a failure. This can lead to misleading conclusions regarding short lives, as Bertele (36) has suggested (section 8.2.2). The difficulty of recording seal running life has lead to the universal use of installed life for measuring mechanical seal life in plants. Installed life is usually measured as the calendar time between failures. MTBF is the logical statistic for measuring the average life of a group of seals.

A generic database (section 7.4.2) can be used to provide a means of analysing a database according to elements with common features. The feature hierarchy (fig 7.3) illustrates the types of common features which could be used to form generic groups. Characteristic life (ie the life at which 63% of the population has failed) or MTBF would be used to characterise the seal life for each generic group. A generic database is a particularly useful way of combining data from different sources, but there is a great danger of introducing misleading information due to time based errors, and missing data.

Regression analysis (section 6.3) has the ability of extracting the most influential factors on seal life, from a group of possible causes. The ability to rank the influence of a large number of variables, against mechanical seal life, makes regression analysis a very powerful statistical method for investigating the sensitivity of mechanical seal life to a wide range of operating parameters (eg alignment, seal quench, fluid vapour pressure, etc). Computer programs are readily available for carrying out multiple linear regression analysis. This makes regression analysis an easy method to apply in practice. Multiple linear regression analysis is the most simple statistical method to apply in practice, for increasing the understanding of which operating parameters have the greatest effect on mechanical seal life. The major drawback with the method is its sensitivity to systematic error (time based errors can be identified), and poor data (section 7.3).

The discriminant function method (section 6.4) is similar to regression analysis, and shares its sensitivity to poor data and systematic error. In addition computer programs are not readily available. Consequently, discriminant function analysis is considerably more difficult to apply in practice. However, discriminant functions do have important characteristics that do not appear in regression analysis. For example, existing failure data could be used to set up discriminant functions for the most important seal face failure mechanisms (section 4.5.3). The discriminant functions would be made up of variables relating to the seal face materials, seal design, auxiliary seal systems, and sealed fluid. These discriminant functions could be used to predict the failure mechanisms of new seals. Mechanical seal life could be improved by using this method. Some failure mechanisms are more desirable than others (ie they are associated with longer seal life (section 8.2.1)). The discriminant functions could be used to select and assist the specification of new mechanical seals. For example, it should be possible to see the effect of changing the auxiliary seal system specification, on the probable failure mechanism. Insufficient data (in a suitable form) was available for the author to test this method of data classification.

Weibull analysis has been extensively used in this (section 8.2) and previous (section 6.5) analyses of mechanical seal data. It is interesting to note that the characteristic seal lives at Plant A and Plant B are far longer than in the other major mechanical seal reference (12). Gu and Wang (12) provided a detailed picture of seal performance at an oil refinery, but rather an unhappy one. Some of the earlier papers were based upon rather small mechanical seal populations.

Weibull analysis is a simple, graphical method of extracting the maximum information from a failure distribution. The analysis of data from Plant A and Plant B (section 8.2) demonstrates that weibull analysis can clearly identify different failure modes. However, information on the causes of failure and seal duties needed close examination, to explain the characteristics of each weibull distribution. Weibull analysis is the best method for comparing seal failure distributions, although it is more complicated than a MTBF or weighted life index method. A weibull distribution will clearly show any differences in the short life failure distribution (ie duration, and severity), and allow a good comparison of the short life, mid-range, and long life characteristics. This information enables a much more informative comparison to be made between different sets of data.

Weibull analysis can only go part of the way in analysing mechanical seal data. Regression analysis and discriminant functions provide the best statistical methods for increasing our understanding of the importance of particular operating parameters on mechanical seal life. The weibull analysis (section 8.2) suggests that the most important seal failure mechanisms are common to a wide range of sealed fluids (ie water, hydrocarbons, and chemicals). This important finding indicates that the results of regression analyses, and discriminant functions, would be applicable to the behaviour of mechanical seals in a wide range of process industries.

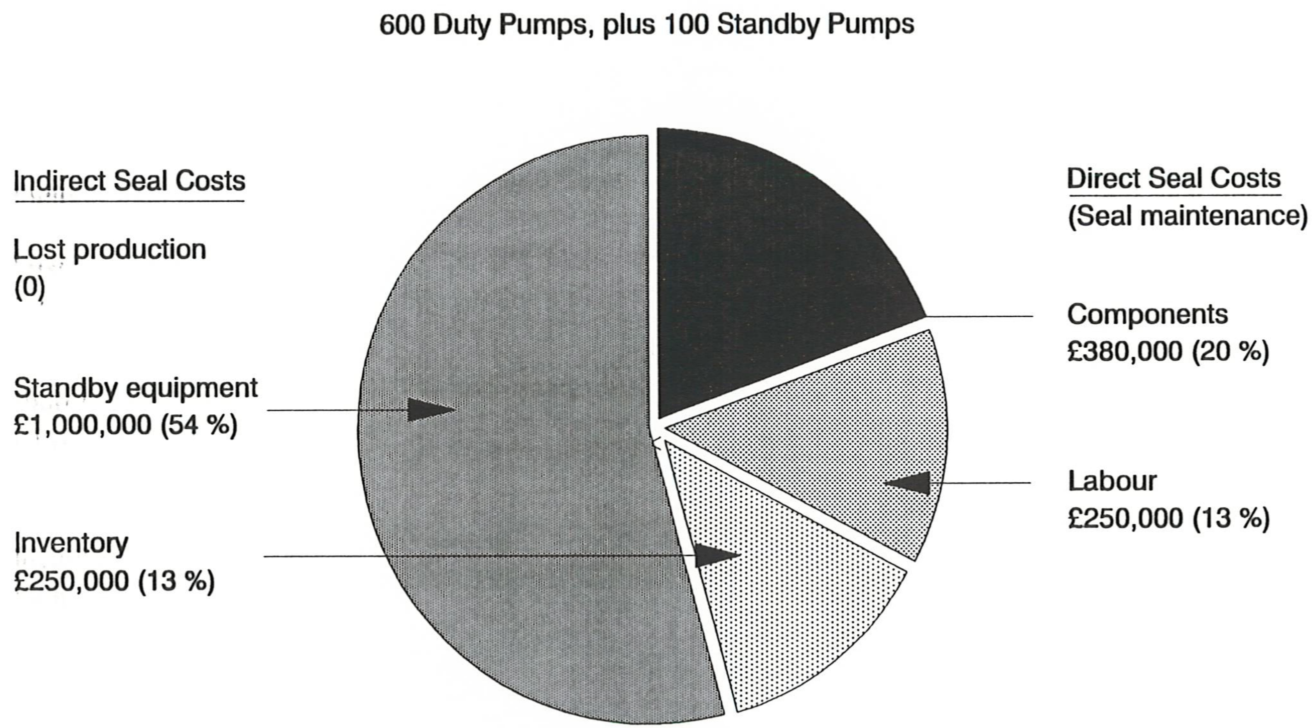


Figure 8.1 : Mechanical Seal Operating Costs at Plant B (1990 Costs)

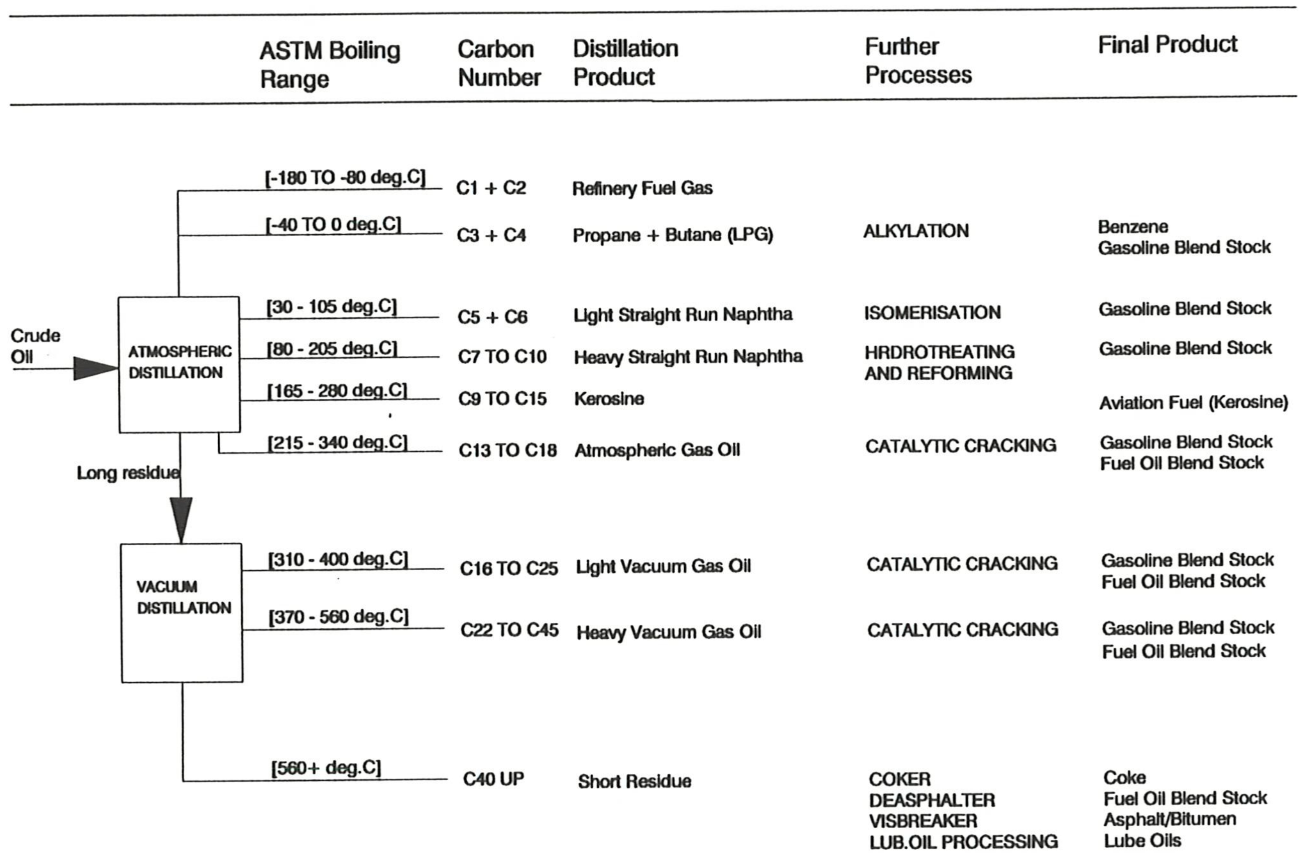


Figure 8.2 : Typical Hydrocarbon Refining Processes and Products

Sealed Fluid	Failure Modes															
	First Mode				Second Mode				Third Mode				Fourth Mode			
	N	To	B	h	N	To	B	h	N	To	B	h	N	To	B	h
LPG	11(6)	0	2.61	18	160(84)	20	0.84	327	20(10)	1640	1.29	2300				
Gasoline	18(20)	0	1.28	37	72(80)	50	0.72	453								
Naphtha	7(4)	0	1.28	12	172(96)	15	0.77	506								
Heavy Hydrocarbons	39(18)	0	2	21	177(82)	30	0.75	298								
Sweet Water	7(11)	0	1.06	50	20(32)	80	0.88	145	36(57)	245	0.61	910				
Sour Water	6(12)	0	3.45	24	6(12)	25	0.63	45	16(32)	65	1.11	122	22(44)	160	0.81	864
HF Acid	12(8)	0	3.27	17	97(69)	25	1.17	176	32(23)	400	0.89	975				

N = Number/(percentage) of seals   To = Age at which the failure mode begins (days)   B = Weibull index   h = Characteristic Life (days)

Table 8.1 : Weibull Distribution Parameters for Mechanical Seals at Plant B

Sealed Fluid	Temperature (Deg.C)	Vap. Pressure (Bar. abs.)	Boiling Temp. @1 atm. (Deg.C)	Specific Gravity (15/15 Deg.C)	pH
LPG	38	55 to 345	-40 to 0	0.300 - 0.356	
Gasoline	38	0.34 to 13	-40 to 68	0.507 - 0.664	
Naphtha	38	0.004 to 1.03	30 to 200	0.631 - 0.734	
Heavy Hydrocarbons	38	< 0.004	> 230	> 0.734	
Sweet Water	38	0.065	100	1.00	7
Sour Water	38	0.065	100	1.00	6 to 9.5
HF Acid	38	1.86	19.4	0.88	< 1

Table 8.2 : Typical Properties of the Sealed Fluids

Sealed Fluid	Overall Characteristic Life (Days)	MTBF (Days)
Sweet Water	530	545
Naphtha	457	550
Gasoline	420	535
LPG	413	500
Sour Water	325	420
HF Acid	270	340
Heavy Hydrocarbons	248	305

Table 8.3 : Overall Failure Distribution Parameters

Seal Face Materials			MTBF (Days)	Weibull Distribution			
				N	To	B	h
Silicon Carbide	v	Silicon Carbide	290	14	0	1.07	350
Silicon Carbide	v	Carbon	91	6	0	0.86	122
Tungsten Carbide	v	Carbon	290	13	0	1.69	75
Stellite	v	Carbon	250	16	0	0.87	180
Alumina	v	Carbon	233	11	0	0.49	170
NiResist	v	Carbon	190	39	0	0.92	120

Note: N = Number of mechanical seal failures
To = Age (days) at origin of the distribution
B = Weibull Index
h = Characteristic life (days)

Table 8.4 : Failure Statistics for Various Seal Face Material Combinations at Plant A

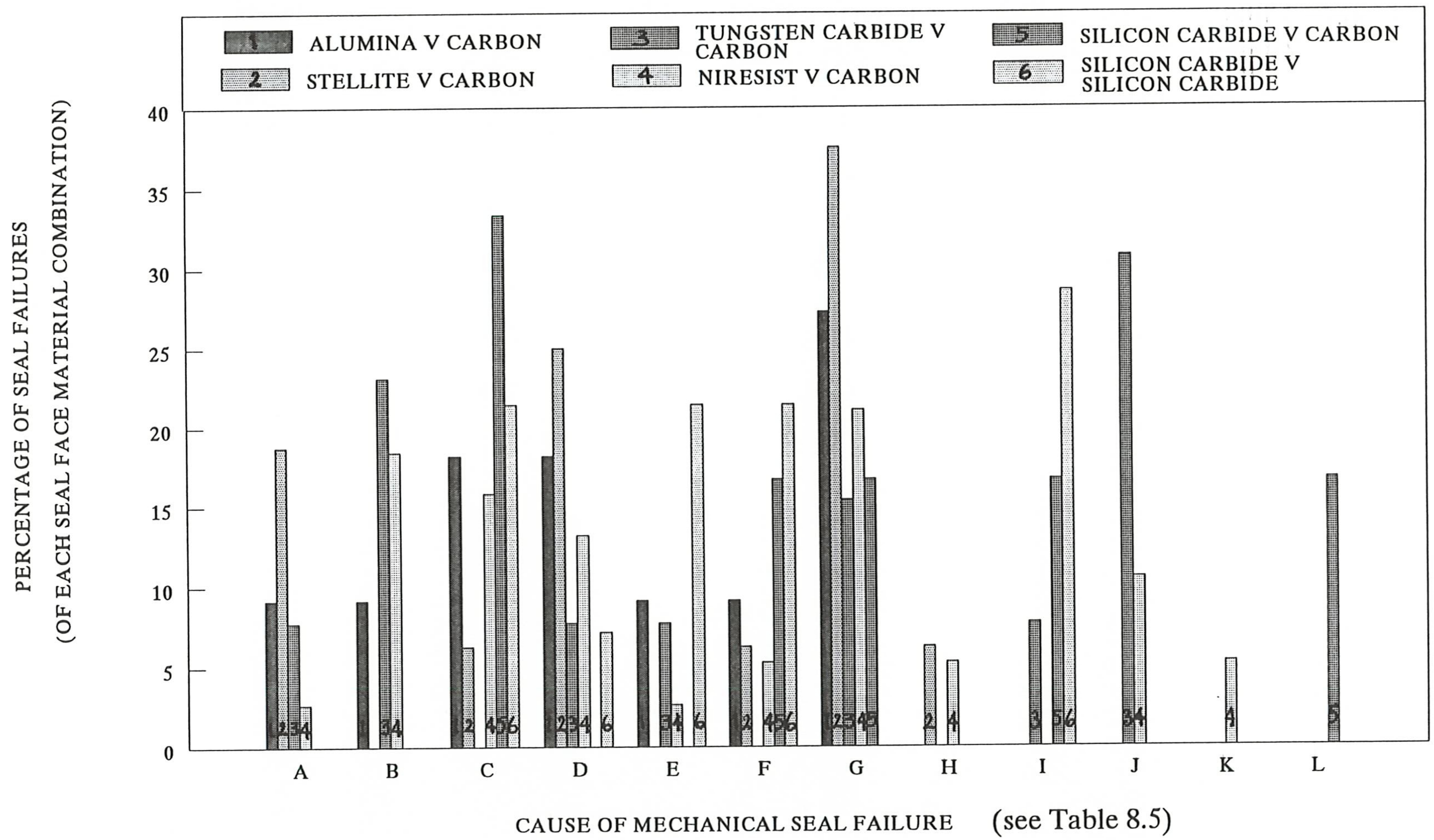


Figure 8.3 : The Relationship Between the Cause of Failure and Seal Face Materials at Plant A

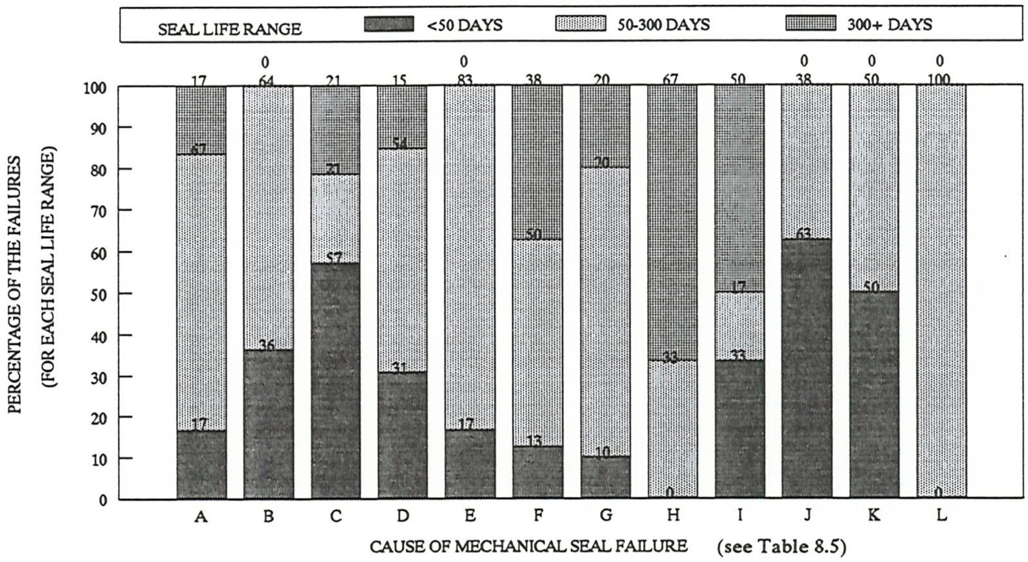


Figure 8.4 : The Relationship Between the Cause of Failure and Seal Life at Plant A

Codes for Causes of Seal Failure	
A	Hang-up (coking)
B	Hang-up (crystalisation)
C	Shaft and seal face plane misaligned
D	Auxiliary system failure (ie quench, cooling, recirc., flush)
E	Incorrect seal setting (ie wrong spring compression)
F	External system/component failure (eg bearing failure)
G	Dry-running
H	Secondary (elastomeric) seal failure
I	Seal component failure (other than the faces, sec.seals, bellows)
J	Embedded metal particles in the carbon face
K	Highly worn carbon (>4mm)
L	Chemical attack of the carbon

Table 8.5 : Codes for Causes of Mechanical Seal Failure at Plant A

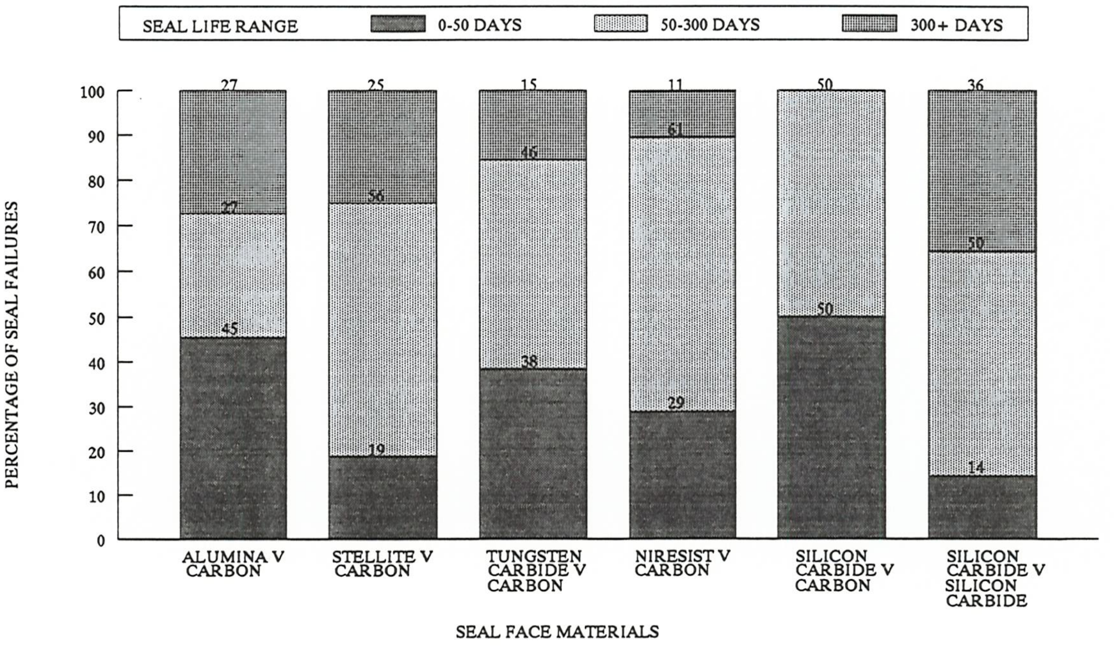


Figure 8.5 : The Relationship Between Seal Life and the Seal Face Materials at Plant A

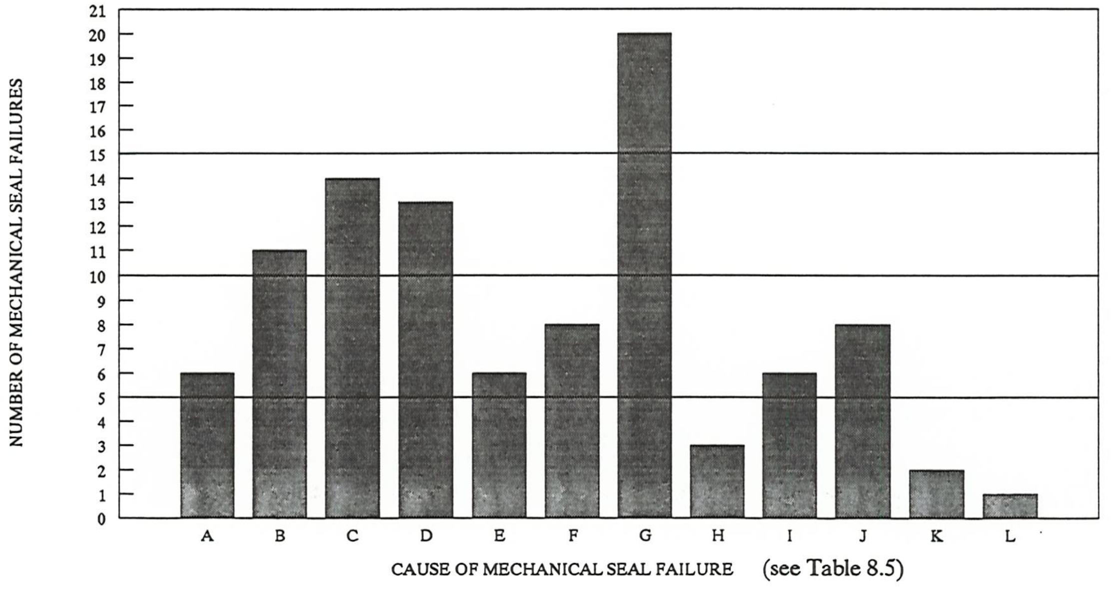


Figure 8.6 : Relative Number of Seal failures by Cause of Failure

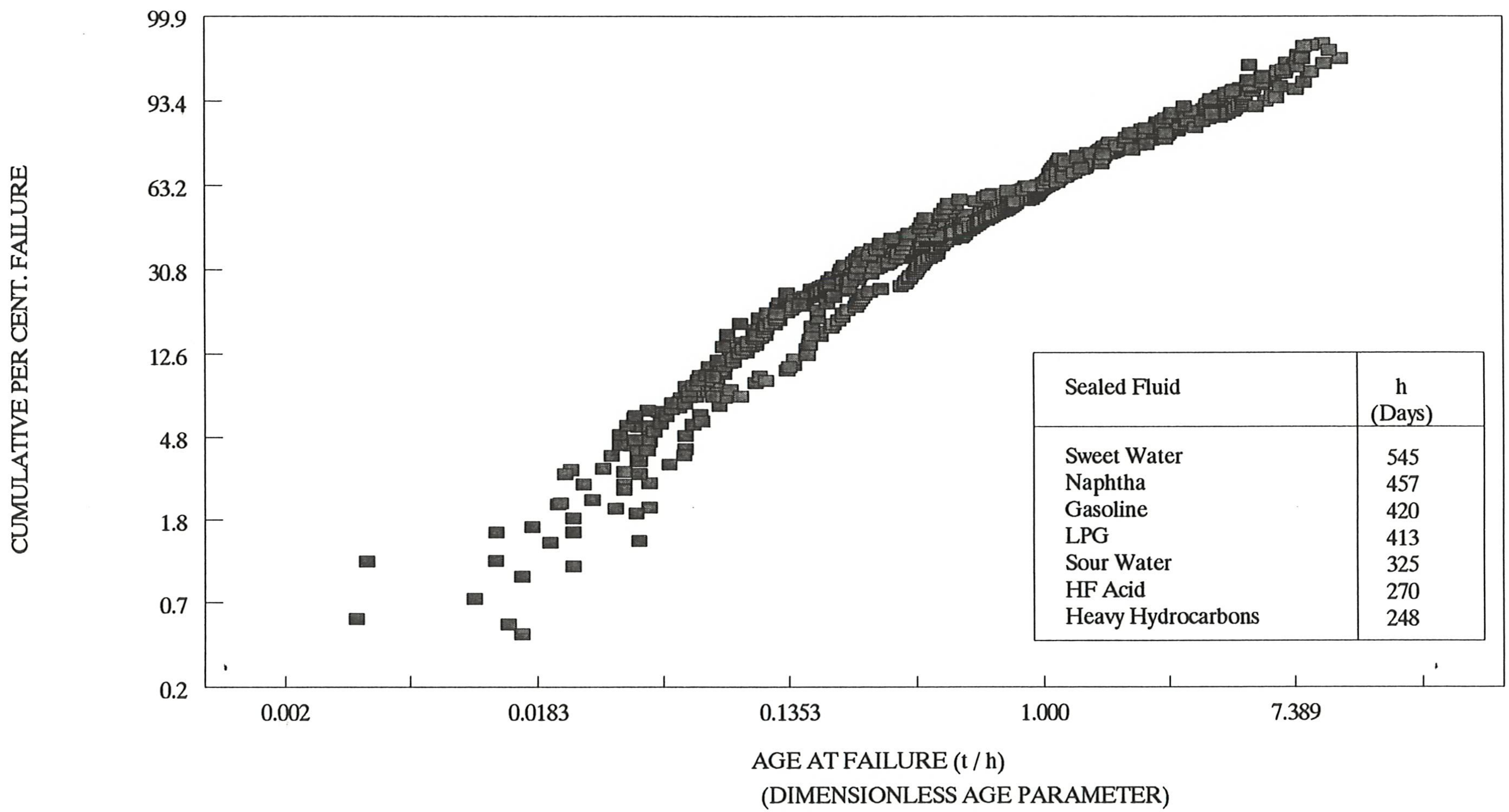


Figure 8.7 : Weibull Distributions for Mechanical Seals on Different Fluids at Plant B Using a Dimensionless Age Parameter

9.0 CASE STUDIES: AN ASSESSMENT OF MECHANICAL SEAL LIFE IN REFINERIES AND PETROCHEMICAL PLANT

9.1 Description of the Data Sources

Two new sources of mechanical seal data (Plant A, and Plant B) have been used to provide field data in section 3.4, section 4.4.3, section 4.5.4, section 7.3, section 8.1, and section 8.2. These data sets are completely separate, and have not been used in any other published literature. The BHRA carried out an extensive survey (23) of mechanical seal performance; published in 1987. This is the only published reference paper containing an extensive database of field operating data, from mechanical seals in process plant. The reference (23) presents the results of a survey of mechanical seals on centrifugal pumps, in three oil refineries and five chemical plants. However 800 of the 1000 seal failures reported in the survey, came from the three oil refineries. The survey found that the performance of mechanical seals in refineries is generally more uniform than in chemical plants, and seal life is generally shorter in chemical plants. The BHRA survey (23) provides a suitable reference to compare with the data from Plant A and Plant B.

Plant A

Plant A is a large petrochemical plant and oil refinery (refining about 260,000 barrels of crude oil per day). Equipment at Plant A has a wide age range. Some units and equipment are over 30 years old, but considerable investment during the 1980's has resulted in a large amount of new equipment. The plant operates 2600 mechanical seals in centrifugal pumps. Access was given to data on the Plant's computerised maintenance record system. Mechanical seal failures have been recorded on this computerised database since 01/01/88. The database contains details of the date and cause of each seal failure, as part of the overall maintenance record system. The quality of the seal data is high, because failures are analysed and recorded by a professional engineer, who is part of a team dedicated to rotating equipment reliability.

This data set contains the failure of 2457 mechanical seals, and records the trouble-free operation of a further 1300 mechanical seals. The data covers the period 01/01/88 to 01/04/91.

Plant B

Plant B is a medium sized oil refinery (refining about 90,000 barrels of crude oil per day). The Plant has had little recent investment in new plant and equipment; the majority of the Plant is at least 20-25 years old. Plant B is operated by the same parent company as Plant A.

Plant B has 700 centrifugal pumps fitted with mechanical seals. The data set contains 1364 mechanical seal failures, and covers the period 1967 to 01/01/88. The manual recording of seal failures during the period 1967 to 1983 is known to have missed some seal failures. Later seal failure rates using a computerised record system (after 1984), suggest that upto 50% of mechanical seal failures were not recorded during the period 1967 to 1983 (section 7.3). Some apparent seal lives incorporate two or more seal lives. This is most likely to affect the long recorded seal life data. With a manual record system it is probable that seal failures were only recorded on certain pumps. The introduction of a computerised maintenance record system (in 1984) made it much easier to record seal failures on every pump at the plant; this would explain the higher number of reported failures using a computerised record system. The author compiled the mechanical seal data set, from the manual and computer maintenance records, whilst working full-time at Plant B.

Standby Pumps

Plant A and Plant B operate a similar policy. A pump has an installed spare, if its failure would result in lost production (ie all critical process duties). Generally if a duty has a spare pump, only one pump is run at any one time, and either pump can be the duty pump. Consequently the installed life of most mechanical seals at Plant A and Plant B is about twice the running life. The BHRA survey (23) found a similar result at the three oil refineries and five chemical plants.

9.2 Inter-plant Comparisons

Figures 9.2 and 9.3 compare the seal life distributions for all duties at Plant A and Plant B, for the whole measured populations. The data from Plant A (fig 9.2) was measured over a period of 40 months (30000 hours), wheras the data from Plant B (fig 9.3) was measured over a period of 20 years. This is reflected in the time-scales used in the two figures.

Figure 9.1 shows that there is little difference between Plant A, Plant B, and the BHRA survey (23), for the distribution of seal lives upto 5000 hours (though the BHRA survey has slightly higher rates). At Plant A 48% of the seals failed within 5000 hours, and at Plant B, 50%. There is a marked difference between the three, for seals failing after 5000 hours. Plant A is significantly better than Plant B, with the BHRA survey (23) indicating the most pessimistic seal life.

Figure 9.2 indicates that 35% of the seals at Plant A had lives in excess of 24000 hours. Actually 1300 of the installations required no seal maintenance during the total period of assessment (30000 hours). This indicates that there were very few failures between 24000 hours and 30000 hours. In fact (fig 9.2) there were very few failures after 15000 hours (within the 30000 hour assessment period). At Plant B (fig 9.3) only 12% of the mechanical seals ran over 30000 hours. This indicates considerably fewer long life seals than at Plant A. However, even this is much better than the 3% in the BHRA survey (23). Figure 9.1 clearly shows the great difference in long seal life (over 30000 hours) from Plant A, Plant B, and the BHRA survey. Section 9.3 discusses whether this phenomenon is a time trend.

It appears that the BHRA survey provides a pessimistic view of mechanical seal life. Both Plant A and Plant B have a considerably higher percentage of seals with lives in excess of 30000 hours. The long seal life data from Plant A is reliable and accurate. The long life data from Plant B is open to a little more doubt (section 9.1). Von Bertele (36) found that 60% of the mechanical seals in a chemical plant had lives in excess of 30000 hours. The data from Plant A, Plant B, and (36) all indicate that mechanical seal life is significantly better than indicated by the BHRA survey (23).

Von Bertele (36) has criticised other studies, because they have been based upon investigations of seal failure, rather than seal life. Von Bertele argues that this approach has caused people to confuse the seal life experienced with known "problem" seals, with the actual seal life achieved on the majority of mechanical seals. I disagree with Von Bertele's criticism since it is natural and logical to investigate the failure of short life seals, because they offer the greatest financial reward if their life can be improved (fig 3.4). However he makes a valid point, which was demonstrated at Plant A. Plant A recorded 2457 mechanical seal failures during the period of assessment (30000 hours). However 1300 of the 2600 pumps fitted with mechanical seals required no mechanical seal maintenance during this period. This confirms Von Bertele's point, that most mechanical seal failures are concentrated within a small group of the total population of seals at a plant. The data from Plant A and (36) is comparable in terms of seal size, pump size, temperature, and range of sealed fluids.

Figure 9.4 shows a comparison of the failure rates from different mechanical seal manufacturers. It is interesting that Plant A and Plant B have very similar failure rates for each manufacturer (except D - because Plant B did not have any seals from this manufacturer). It has not been possible to determine the extent to which this reflects purchasing practice for different applications, different types of seal, or different seal quality. It is common for manufacturers to be favoured for specific sealing duties. This raises the whole question of selection practice. Often selection is influenced by historical precedent (ie in the past a particular manufacturer has supplied a successful seal for the particular duty). It is also difficult to change the inventory and purchasing instructions at a large plant. All these reasons have a major influence on the manufacturer chosen to supply mechanical seals to a plant. It is common for a manufacturer to be chosen for particular types of seal (eg double seals, tandem seals, single seals with metal bellows, etc), because they are believed to perform the best. The author was unable to establish whether Figure 9.4 reflects real differences in seal quality, or the effect of selection practices.

9.3 Time Trends

Section 9.2 described the differences between the failure distributions at Plant A, Plant B, and the BHRA survey (23). Plant A has the longest seal lives, followed by Plant B, then the plants in the BHRA survey. These three data sources provide an opportunity to examine whether there is any time trend to these distributions.

Data from Plant A was obtained during the period 01/01/88 to 01/04/91, and represents the most up to date measure of seal performance. The BHRA survey (23) was published in 1987, but collected data during a three year period after 1981. Data from Plant B was obtained over the period 1967 to 01/01/88, and clearly contains the oldest data on mechanical seal performance. Figure 7.2 (chapter 7) shows the number of recorded failures each quarter at Plant B, from 1969 to 1988. This graph indicates that there was a gradual decrease in the failure rate upto 1978. Then there is a small step increase which is maintained steady upto 1984. The cause of the large step increase from mid-1985 onwards has already been discussed (section 9.1). The step change in 1978 probably reflects the expansion of the refinery with new process units; increasing the total number of mechanical seals. Figure 7.2 also indicates that the failures from 1980 onwards represent about 70% of the total recorded failures from Plant B.

The BHRA survey (23) focused a lot of attention in the process industry, on the poor reliability and life of mechanical seals. Consequently the performance of seals indicated by the BHRA survey (23), represents performance before mechanical seals were given a high priority in many plants. As we might expect (and hope!), the

BHRA survey gives the most pessimistic account of mechanical seal life. Plant B recognised mechanical seal failure as the most important cause of pump outage around 1987. It was at this time that the author first became involved in the subject. So the data from Plant B largely reflects seal performance prior to a dedicated programme to reduce mechanical seal failures. Plant B shows better performance than the overall performance of all plants in the BHRA survey. However seal performance at Plant B was comparable to the three oil refineries in the BHRA survey. Plant A began to look closely at mechanical seal failures several years earlier than Plant B, and employed a special group to concentrate on improving the life and reliability of rotating equipment (including pump seals). The data from Plant A is the most recent, and reflects mechanical seal life at a large plant which is actively trying to improve seal life and reliability. Plant A has the highest proportion of seals with long seal life.

The data indicates that efforts to improve mechanical seal life in recent years have been partly successful. The proportion of seals with lives in excess of 30000 hours has increased dramatically. But the proportion of seals failing within 5000 hours is unchanged. This situation highlights an important weakness in MTBF as a measure of seal performance. The MTBF has increased, but the number of short life failures is unchanged, resulting in very little improvement in reliability. The efforts at Plant A had little or no impact on the short life failure rate.

At Plant A the efforts to improve mechanical seal life have been directed at improving the seal face materials, verifying the seal selection, and installing auxiliary seal systems (eg quench, cooling, etc) if applicable. The data indicates that this approach does increase the number of seals lasting over 30000 hours. But conversely this approach has had little effect on the number of seals failing within 5000 hours (ie premature failures). Section 8.2.1 investigated the cause of failure of a sample of the seals from Plant A (fig 8.4). It appears that fitting errors (eg incorrect seal setting), and poor installation (eg shaft and seal face plane misaligned, and other misalignment) are significant causes of short life failures. Better procedures for fitting mechanical seals (and overhauling pumps), and greater emphasis on eliminating misalignment would certainly have an impact on reducing the number of seals failing within 5000 hours. The high number of failures due to crystallisation and dry running (fig 8.6) could be reduced by investing money in auxiliary seal systems for the relevant pumps, although Plant A also has a problem with the way that it maintains the existing auxiliary seal systems. To summarise, Plant A has very clearly demonstrated that seal life can be significantly extended by close attention to mechanical seal selection and quality, but a reduction in the short life failure rate requires an improvement in fitting and alignment procedures.

9.4 The Environment Between the Seal Faces

Fluid Film Properties

A narrow fluid film between the seal faces is generally believed to be essential for long seal life. If the film is too thick leakage is unacceptably high, and the seal is deemed to have failed. If the fluid film is too thin the faces will be in physical contact, causing high friction and wear. High friction usually results in the fluid film vaporising, causing thermal cracking, and physical damage as the faces slam together.

The Duty parameter (section 5.5) has been used with limited success as a method of predicting the the environment between the seal faces, in terms of lubrication mode. Accurate measurement of the seal face friction coefficient poses the greatest difficulty.

Figure 9.5 compares the overall MTBF of seals on 17 different fluids at Plant B. There is a 6:1 ratio between the best and worst fluids. Figure 8.2 indicates where most of these fluids fall into the hydrocarbon range. Medium hydrocarbons like medium residue, and kerosene have the longest average seal life. This supports the idea that the lubrication properties of the fluid between the seal faces is very important; lubricating oils are manufactured from medium /heavy hydrocarbons. Chemically passive fluids, such as sweet water, also produce long average seal life. The shortest average seal life occurred with bitumen. Bitumen is the "bottom of the barrel" produced by the crude oil refining process. Bitumen is pumped at high temperature (to reduce its viscosity sufficiently for pumping), and often contains abrasive particles. The worst five fluids in figure 9.5, have high fluid viscosity, and high sealing temperature in common. Fluids which are chemically reactive (eg HF acid, and sour water - compared to sweet water) experience considerably shorter average seal life. These findings are in complete agreement with the results of a regression analysis (24) carried out by Flitney and Nau (section 6.3.3); there is a strong relationship between poor seal life ,corrosive nature of the sealed fluid, and vibration.

Seal failures at Plant A indicate that most short life failures are due to poor seal and pump installation procedures (section 9.1). Experience at Plant B confirms that many of the seal failures on bitumen pumps are due to blocked auxiliary seal lines, and incorrect pump start /shutdown procedures. These errors lead to rapid seal failure due to clogged bellows or springs, hang-up, and seized faces.

Weibull analysis (section 8.2) was used to examine the failure distributions of mechanical seals on several sealed fluids. These fluids (ie water, light hydrocarbons, heavy hydrocarbons, and HF acid) have a very wide range of fluid properties. When the failure distributions were each normalised to their characteristic life (fig 8.7), they produced almost identical Weibull distributions. This indicates that seal failure mechanisms are common to a wide range of sealed fluids, although there is a time-shift for different fluids. This provides further evidence that a limited range of the fluid film properties affect mechanical seal life. The data indicates that the lubricating properties, and corrosive potential are the most important fluid properties.

Seal Face Materials

The seal face materials have a large influence on the environment between the seal faces. To avoid excessive leakage the fluid film between the faces must be thin enough to allow some asperity contact of the seal faces. The properties of the seal face materials will dictate how much friction heat is generated, and how easily this heat is conducted away from the fluid film.

Under dry-running conditions there is no fluid film between the seal faces, so the environment between the seal faces is solely dependent on the seal face material properties. Data from Plant A (section 8.2.1) indicates that dry-running is the most significant single cause of seal failure on longer life seals. The data also shows that seals using silicon carbide or tungsten carbide faces are the least susceptible to dry-running failures, even on fluids with poor lubricating properties and containing abrasives.

<u>Seal Type</u>

The seal type will have an effect on the environment between the seal faces. In a single mechanical seal the faces are lubricated by the sealed fluid. Tandem and double seals both use a barrier fluid. In a double seal both the inner and outer mechanical seals are lubricated by the barrier fluid. In a tandem seal the inner mechanical seal is lubricated by the sealed fluid, and the outer by the barrier fluid.

The fluid between the faces has a significant effect on the seal life. In tandem and double seals it should be possible to select a barrier fluid which has good lubricating properties, and low corrosive properties. There are limitations since the barrier fluid must be compatible with the sealed fluid, and not provide a source of contamination.

Most HF acid and LPG pumps use double seals due to the highly hazardous nature of these fluids. In figure 9.5 LPG and HF acid fall about halfway between the fluids with best and worst average seal life. Relating this back to the Weibull analysis (section 8.2), it is not surprising that the HF acid and LPG seals have similar characteristics to the better natural lubricants; there is a barrier fluid between the seal faces, rather than the sealed fluid.

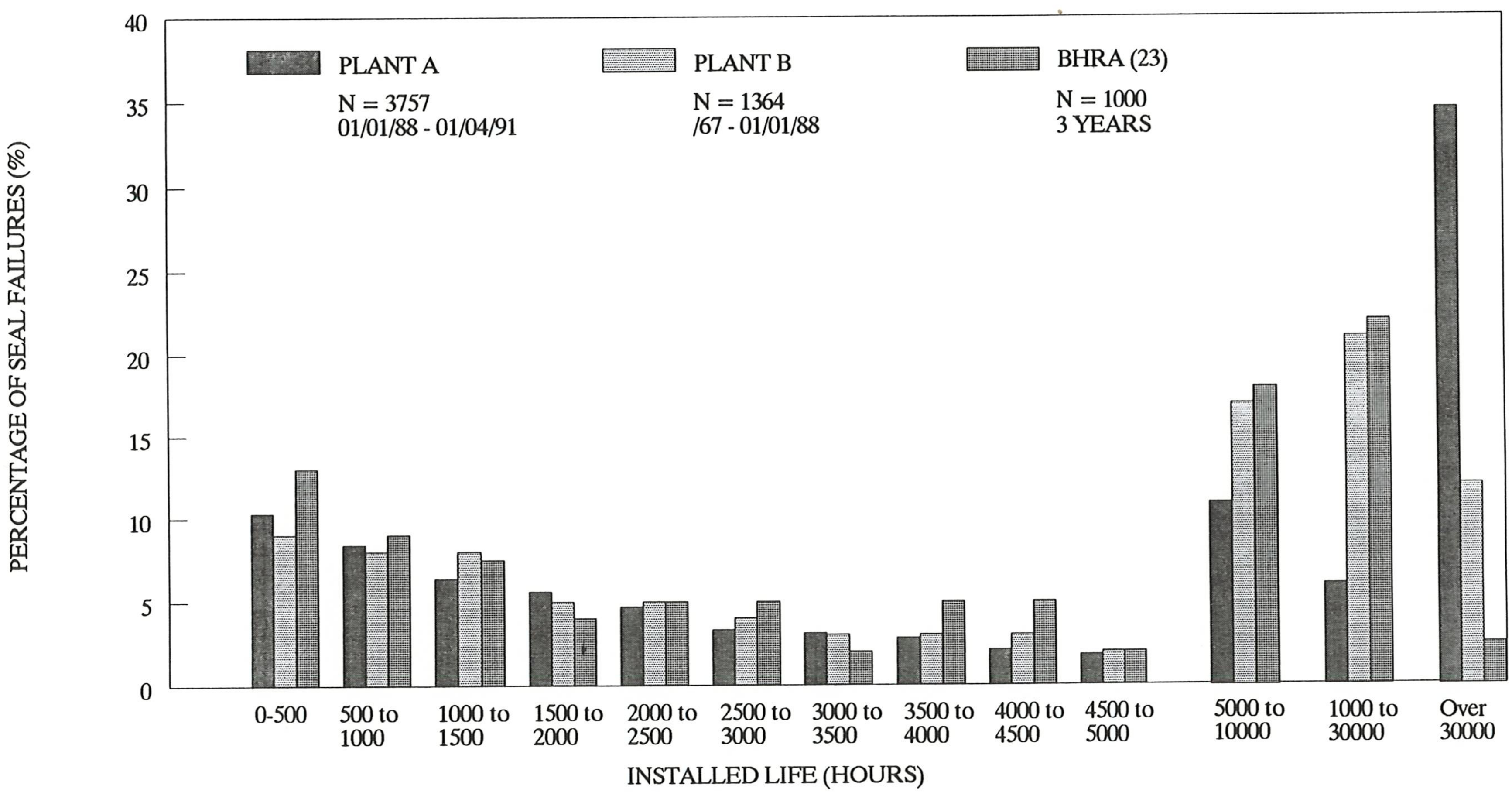


Figure 9.1 : Comparing Mechanical Seal Life Distributions From Three Sources

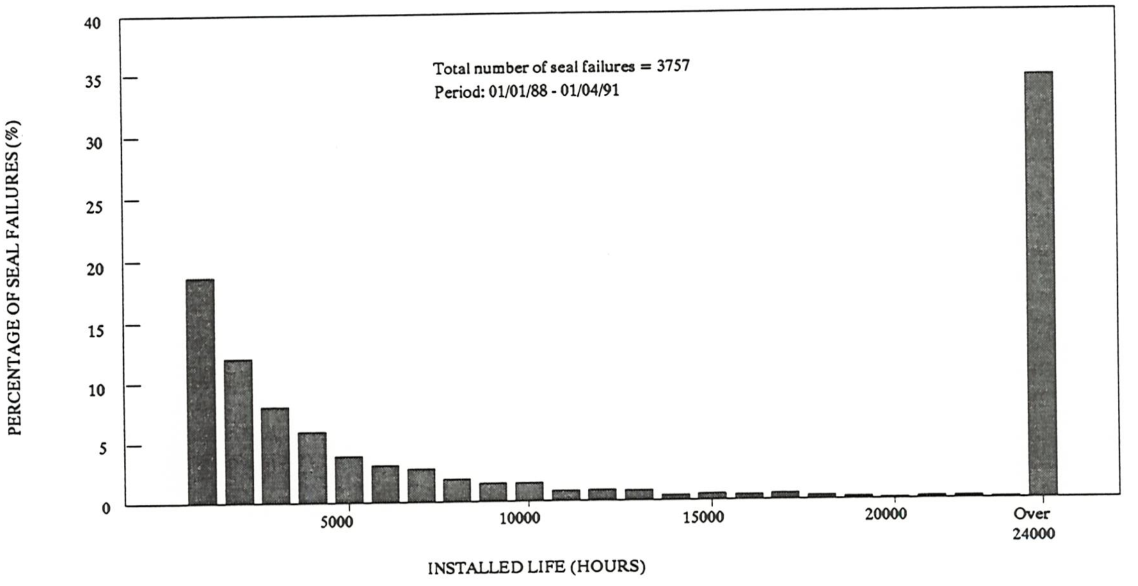


Figure 9.2 : Mechanical Seal Life Distribution at Plant A

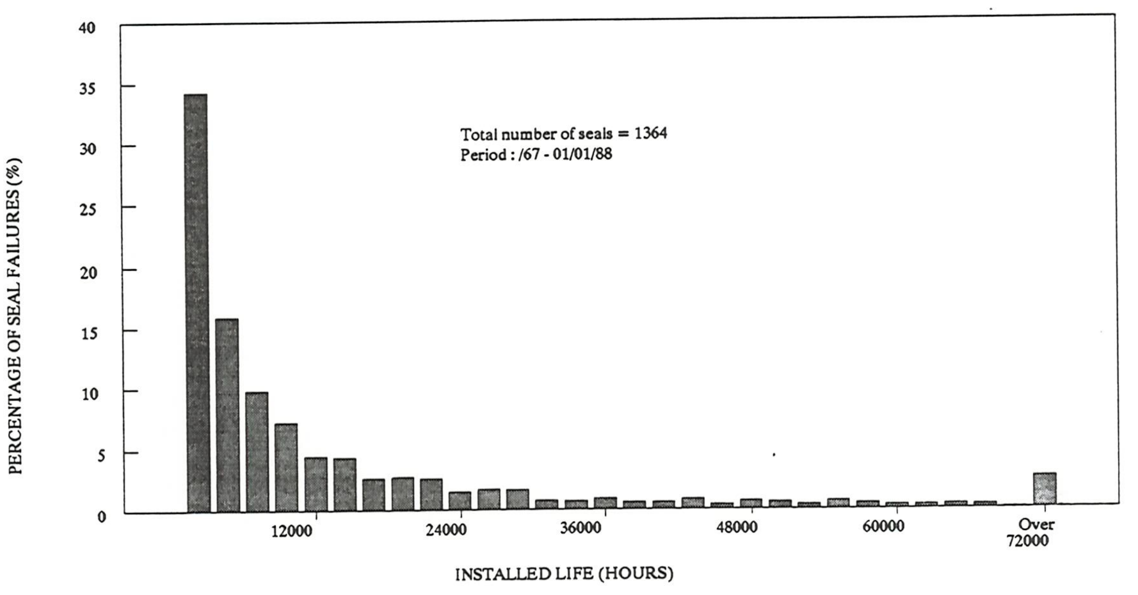


Figure 9.3 : Mechanical Seal Life Distribution at Plant B

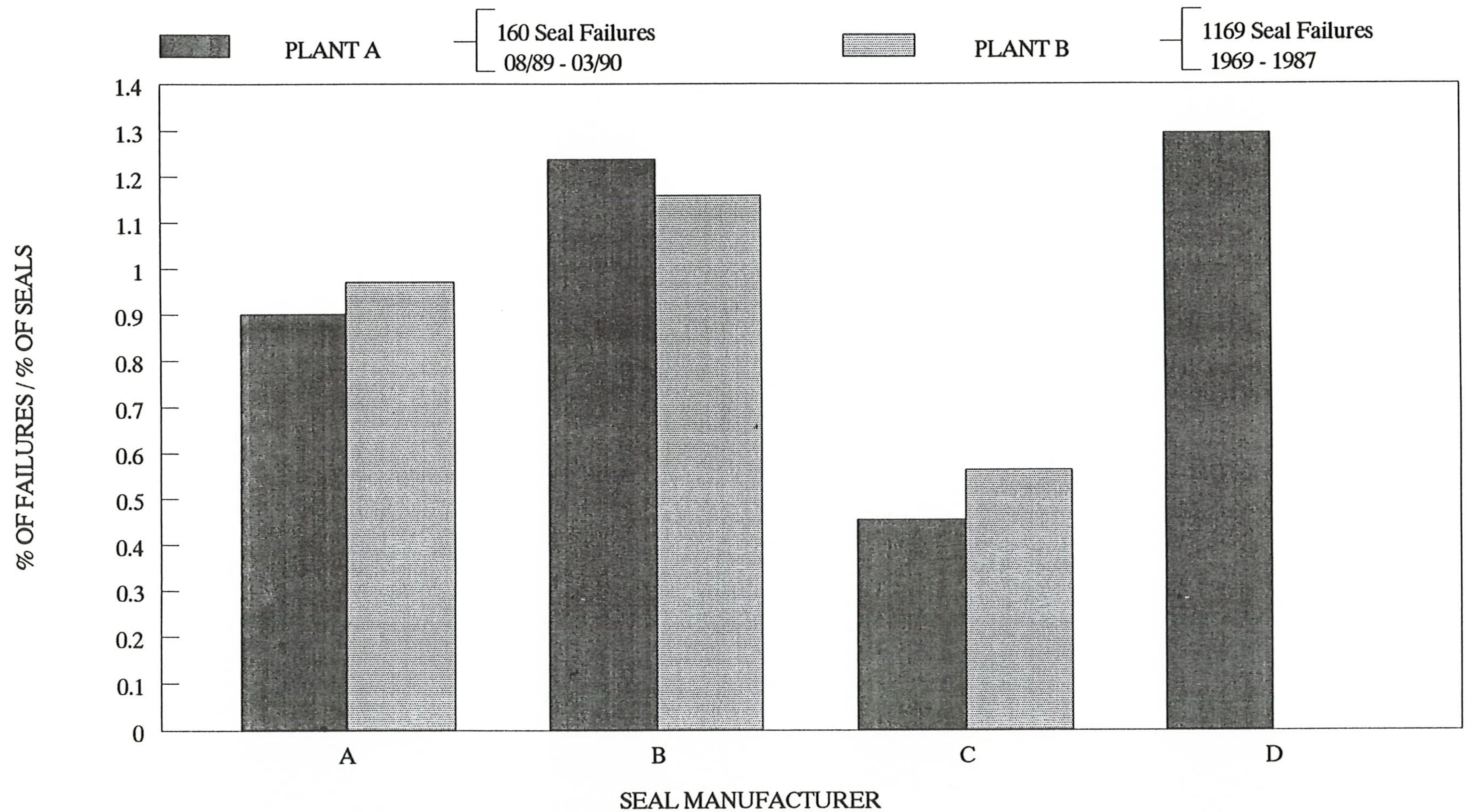


Figure 9.4 : A Comparison of Mechanical Seal Failure Rates From Different Manufacturers

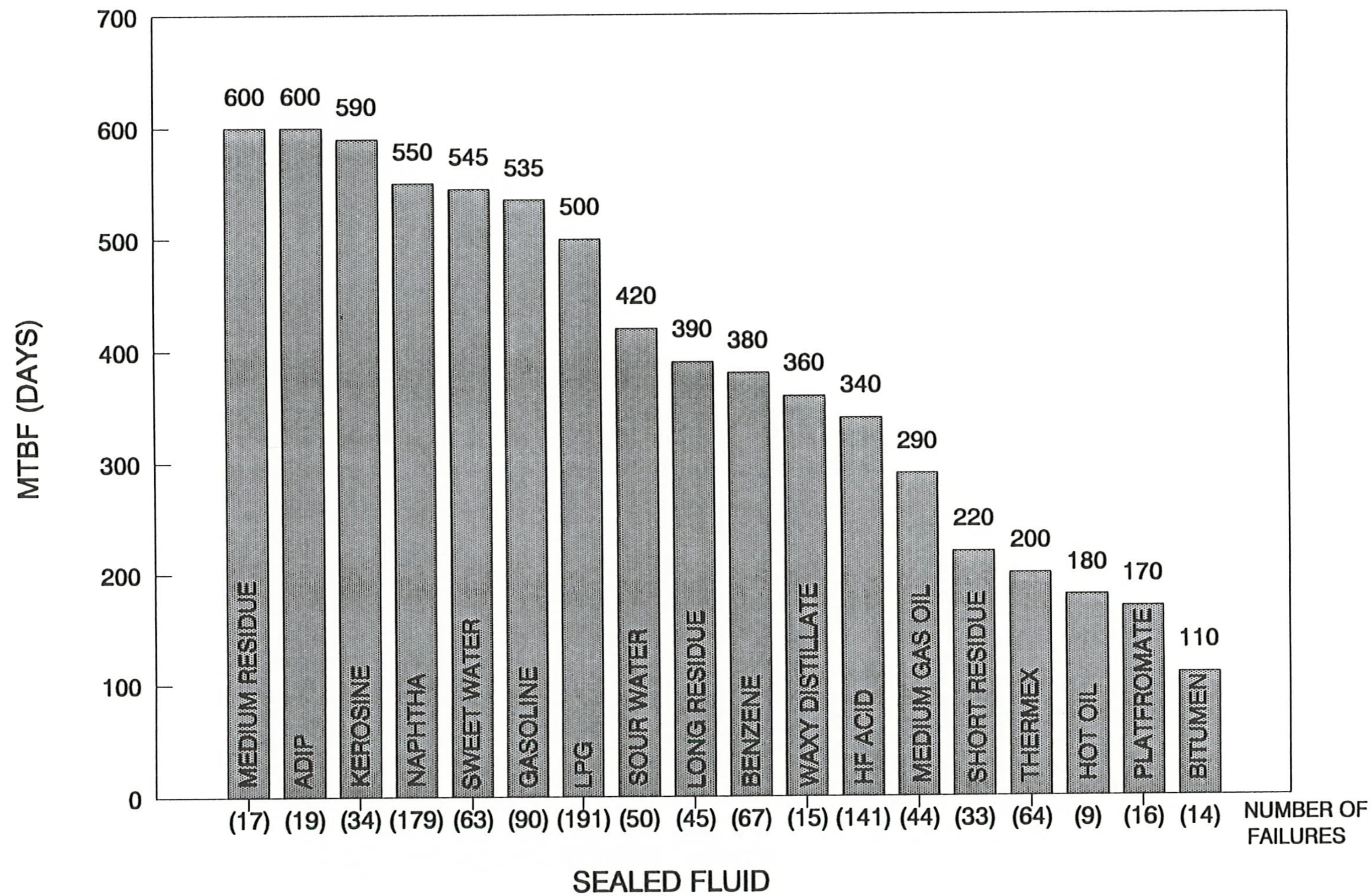


Figure 9.5 : Sensitivity of Seal Life to the Sealed Fluid

10.0 CONCLUDING COMMENTS

In the concluding comments there are references to group 1, group 2, and group 3 mechanical seals. Group 1 refers to single mechanical seals with elastomeric secondary seals. Group 2 refers to more expensive single mechanical seals with a metal bellows secondary seal. Group 3 refers to the most expensive type of seal; the double seal.

10.1

Quantified information presented on costs at Plant B, in terms of 1990 pounds, indicates an average total seal operating cost of £2,690 per pump, with an average seal life of 16 months. Direct costs (ie seal replacement costs) account for only 25% of the total seal operating cost.

10.2

In terms of 1990 pounds, a direct seal cost (ie labour and renewed component costs) of £1200 pa (per pump) relates to an average MTBF of 1 year. The direct seal costs fall to £300 pa (per pump) with a MTBF of 4 years. Longer seal life will significantly reduce direct seal operating costs.

10.3

In terms of 1990 pounds, the indirect cost of operating mechanical seals is £1860 pa (per pump). Standby equipment contributes 80% of this cost. At present it is cheaper to install standby equipment, than suffer lost production or reduced throughput. Longer seal life will only have an effect on the indirect seal operating costs, if there is a corresponding increases in seal reliability. Reliability is associated with the number of short life seals.

10.4

Component costs were generally 50% higher than labour costs, for mechanical seal maintenance at Plant B. Tougher environmental legislation will further increase the cost of mechanical seal components. It will be necessary to install tandem seals, double seals, or back-up seals on many sealing duties in the process industry; to restrict fugitive emissions and leakage in the event of a seal failure.

10.5

Group 1 mechanical seals can be reliable and run trouble-free, for as long as more expensive types of mechanical seal (fig 3.4). However a poorly suited group 1 seal (ie the duty is too arduous) will cost considerably more to operate than a more expensive type of seal. There is an attractive cost benefit in optimising seal group selection on individual sealing duties.

10.6

Seal life can be extended considerably, by improving seal face materials (ie use tungsten carbide and silicon carbide seal faces), checking seal selections, and installing auxiliary seal systems. Plant A has used this approach, and 35% of the seals now have lives in excess of 30000 hours, compared to only 3% in the BHRA survey (23) in the mid-1980's. However, almost 50% of the seals fail within 5000 hours, which is virtually unchanged from the BHRA survey (23).

10.7

A large proportion of seals failing within 50 days are incorrectly installed, and exposed to excessive vibration through poor pump, coupling, and shaft alignment. Plants must concentrate much more effort on reducing the acceptable levels of vibration and misalignment on centrifugal pumps. Plant A indicates that fitting errors (eg incorrect seal compression), and poor installation of the pump and/or seal (eg shaft and seal face plane misaligned) are a very significant cause of premature seal failures. There would be a marked improvement in mechanical seal reliability (ie fewer short life seals) if seal installation and pump overhaul/ installation procedures were tightened up.

10.8

Seal reliability could be improved by using cartridge seals, since they remove the possibility of incorrect seal compression. The technology is available (ie CNC machines) to produce pumps and seals with small dimensional tolerances and accurate alignment. Balancing machines, and laser alignment equipment is readily available to facilitate high degrees of alignment. With stricter procedures it is quite feasible to reduce pump seal vibration and misalignment to a very low level.

10.9

Analysis of seals at Plant A indicates that failure mechanisms external to the seal (ie where failure occurs as a direct result of an operating malfunction, which puts the seal under conditions for which it was not designed) account for almost the same number of failures as mechanisms internal to the seal (ie where failure occurs as a direct result of the seal selection or specification, under normal operating conditions).

10.10

The seal surveys at Plant A and Plant B do indicate that there has been considerable progress in reducing the burden of process pump mechanical seal failures, when compared to other similar published information, eg the BHRA survey of nearly a decade ago.

10.11

Dimensionless groups are best applied to mechanical seals as selection criteria. Limiting values of the dimensionless group for long seal life can be established from past seal performance. The duty parameter and delta-T factor are the most useful dimensionless groups. The main barrier to using dimensionless groups, is the difficulty of measuring some of the parameters. Indeed, on existing seal installations these parameters cannot be measured. Further research on test rigs, or specially instrumented seals in the field could be used to establish empirical relationships between easily measured parameters (eg seal chamber pressure and temperature) and the dimensionless group parameters.

10.12

Multiple linear regression analysis is an excellent method for establishing the relative importance of a large number of operating parameters on mechanical seal life. Good quality data is required, because regression analysis is sensitive to systematic errors and inaccurate data. Unlike most statistical methods, regression is able to identify time trends. Computer programs are readily available, which makes multiple linear regression very easy to apply in practice.

10.13

Discriminant functions provide an interesting technique for classifying data. It is possible to specify a minimum confidence in the classification, and then minimise the number of parameters used to achieve this classification. This method provides a useful statistical tool for mechanical seal selection, with the added attraction of selectively reducing the number of measured parameters. Discriminant function analysis is more difficult to apply in practice, because computer programs are not readily available.

10.14

Weibull analysis is a simple graphical method for extracting the maximum information from a failure distribution. Weibull plots provide a simple method for comparing and quantifying the characteristics of different failure distributions. Weibull distributions of mechanical seal failures clearly isolate different failure modes. The transition between different seal failure modes (a change in the slope of the weibull distribution) is very distinct, although 50-100 data points (minimum) are required to give confidence that transitions are not simply due to scatter.

10.15

As expected, different sealed fluids have a marked effect (upto 6:1 ratio) on average seal life. However a weibull plot of 930 mechanical seals on seven sealed fluids (LPG, gasoline, naphtha, heavy hydrocarbons, sweet water, sour water, and HF acid) using a dimensionless age parameter (ratio of actual life to characteristic life) shows that the distributions are virtually identical over their whole range.

This indicates that mechanical seals have the same failure modes on a wide range of sealed fluids, ie. the most significant seal failure mechanisms are common to a wide range of sealed fluids.

10.16

Weibull analysis indicates that there are distinct transition characteristics between infant mortality, premature failure, and wear-out failure, on a wide range of sealed fluids.

The first failure mode (infant mortality) accounts for seals with a dimensionless age parameter < 0.15 (typically a seal life less than 50 days). The first failure mode has an increasing failure rate, ie. weibull index > 1. Plant A indicates that failure mechanisms resulting from incorrect seal installation and misalignment are very significant in this first failure mode.

The second failure mode (premature failure) accounts for seals with a dimensionless age parameter > 0.15, and has a decreasing failure rate, ie. a weibull index < 1. Plant A indicates that hang-up and dry-running are the most significant failure mechanisms in the second failure mode. This failure mode contains about 75% of all mechanical seal failures.

There is some evidence of a third failure mode (wear-out failure) starting at a dimensionless age parameter > 5.0 (typically a seal life in excess of 5 years). This corresponds to the wear-out region of the bathtub curve. Plant A indicates that secondary (elastomeric) seal failure, and external component failure (eg bearings), are the most likely causes of failure on long life seals. Less than 1% of the mechanical seals at Plant A failed due to wear-out.

10.17

Since mechanical seals on a wide range of fluids exhibit the same failure characteristics, perhaps only a small subset of fluid properties are of real significance to mechanical seal life. This subset of fluid properties determines the rate at which the failure mechanisms operate.

Longest life is achieved on sealed fluids with similar properties to sweet water (eg naphtha). Seal life deteriorates as the fluid vapour pressure, boiling temperature, and corrosive potential (eg pH) move away from the properties of sweet water. Low fluid vapour pressure is more critical than high vapour pressure, and a high fluid boiling temperature is more critical than a low boiling temperature. Specific gravity appears to have little effect on mechanical seal life. Corrosive potential has a very significant effect on seal life (eg sweet water, and sour water).

11.0 <u>FUTURE WORK</u>

11.1

Establish a better understanding of seal face lubrication conditions; in particular, variations in seal face friction coefficient. Apparently identical seals (eg a like-for-like seal replacement) often exhibit different seal face lubrication conditions. There is a need to establish whether these differences are a function of quality control during seal manufacture, material variability, or some other cause.

The lubrication conditions between the seal faces certainly have a major influence on seal life. A better understanding of the factors affecting seal face lubrication, will provide an improved likelihood of establishing conditions for maximum seal life. The Duty Parameter has shown an excellent correlation with seal life , if an accurate seal face friction coefficient can be determined.

11.2

Work should be carried out in the field of sound and vibration analysis, to see whether these techniques can be used to indicate the condition of the seal. Vibration monitoring is a well established technique at many plants, so the equipment is already available. Very little work has been carried out in this field, in the mechanical seal application. The main argument against these techniques has been that noise and vibration from other components (eg bearings, gears, etc) will drown out any vibrations/noise from the seal. However there may be vibration characteristics , of a much smaller amplitude which are "thrown away" at present. Seal vibrations would correspond to the mechanical seal operating conditions (eg lubrication mode between the seal faces).

This may prove a practical way of measuring the seal face lubrication conditions (see 11.1), on mechanical seals in real service conditions.

11.3

This study and others before it, have shown clear evidence that vibration and misalignment have a very significant effect on mechanical seal life. The statistical data in this study has indicated that a large percentage of short life seals fail due to poor seal and pump installation. Poor installation results in high vibration levels, misalignment, and incorrect seal compression. The data presented in this study indicates that the number of short life seals could be significantly reduced through better pump and seal installation procedures. At a practical level, we need to quantify the relationships between seal life, types and degree of misalignment, and levels of vibration. Seals of a cartridge design should be compared to non-cartridge seals, to quantify how well they remove some of these installation problems. There may be threshold values of vibration and misalignment, below which seal life is little affected. If so, these would then give a good guideline for improved seal and pump installation procedures, leading to fewer short life seals.

A major programme will be necessary, to produce enough statistical data to form guidelines with confidence. This may justify a cooperative effort from industry and research establishments.

11.4

In chapter 8 this study established a dimensionless Weibull plot for mechanical seals on a wide range of sealed fluids at Plant B. This Weibull plot suggests that a single curve can be used to predict seal life and percentage failed against time, based upon the characteristic life of the sealed fluid. Corroboration is necessary from other plant databases, to establish if a similar dimensionless Weibull plot is produced at other plants.

If the dimensionless Weibull plot does apply at other plants, then a statistically based seal life prediction method will have been established. The characteristic Weibull life for a wide range of sealed fluids, from a large database, could be tabulated. These values could then be used to establish a predicted seal life (for a specified probability of failure) on different sealed fluids, by simply reading from the dimensionless Weibull plot.

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APPENDIX A1

Mechanical Seal Weibull Distributions For Various Sealed Fluids at Plant A.

Associated With Chapter 8, Section 8.2.2

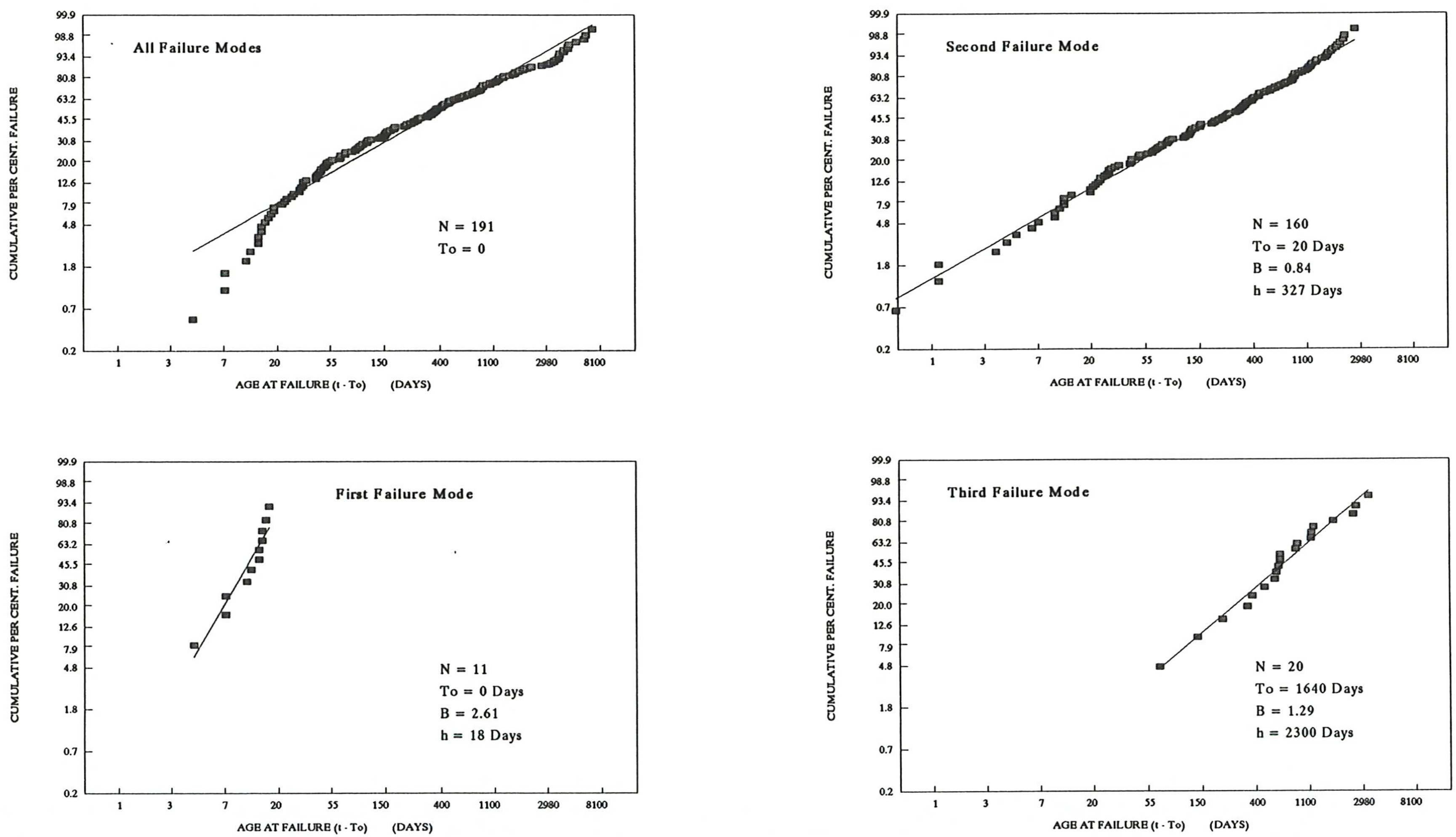


Figure A1.1 : Weibull Plot of Mechanical Seals on LPG Duties at Plant B

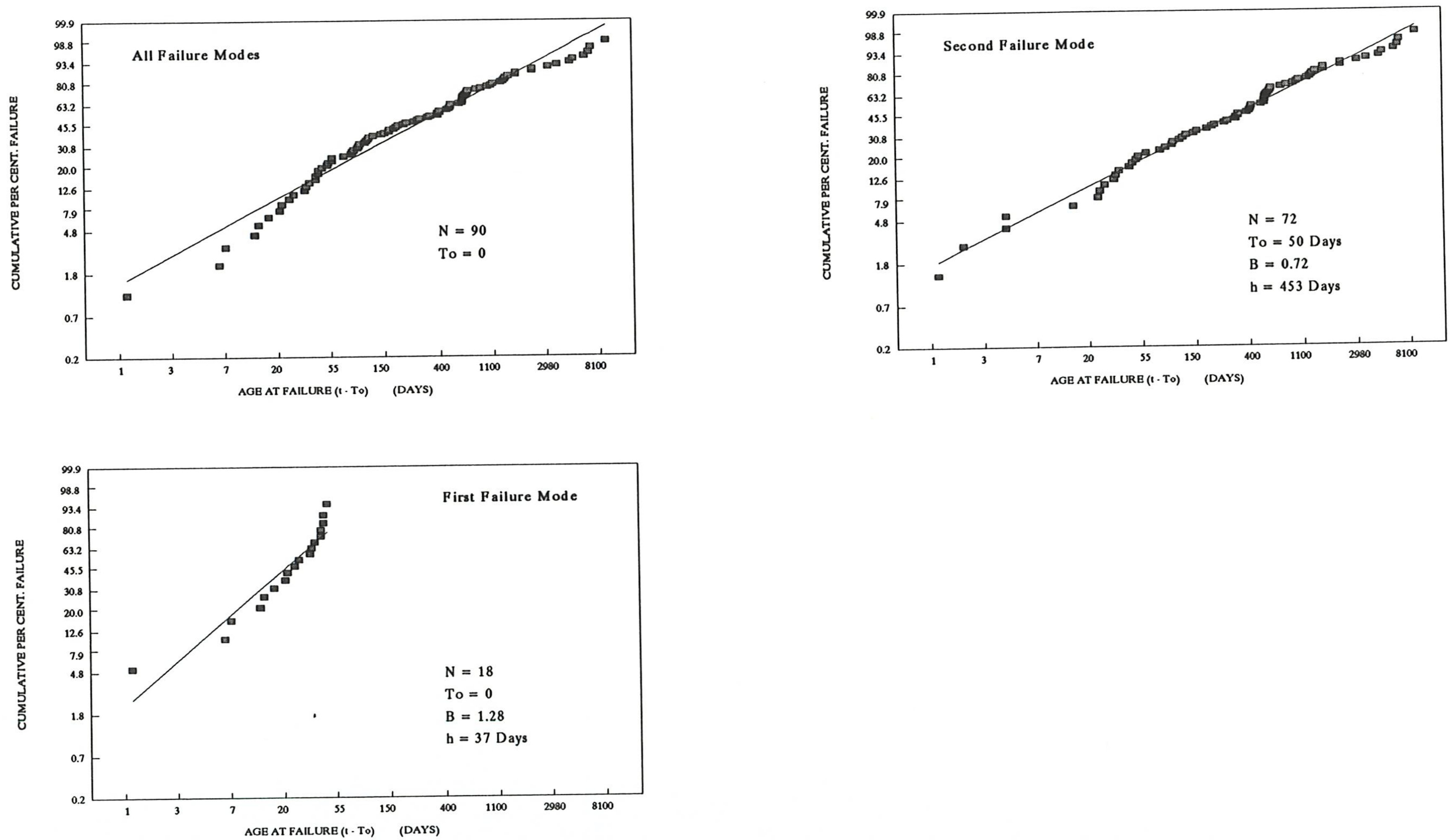


Figure A1.2 : Weibull Plot of Mechanical Seals on Gasoline Duties at Plant B

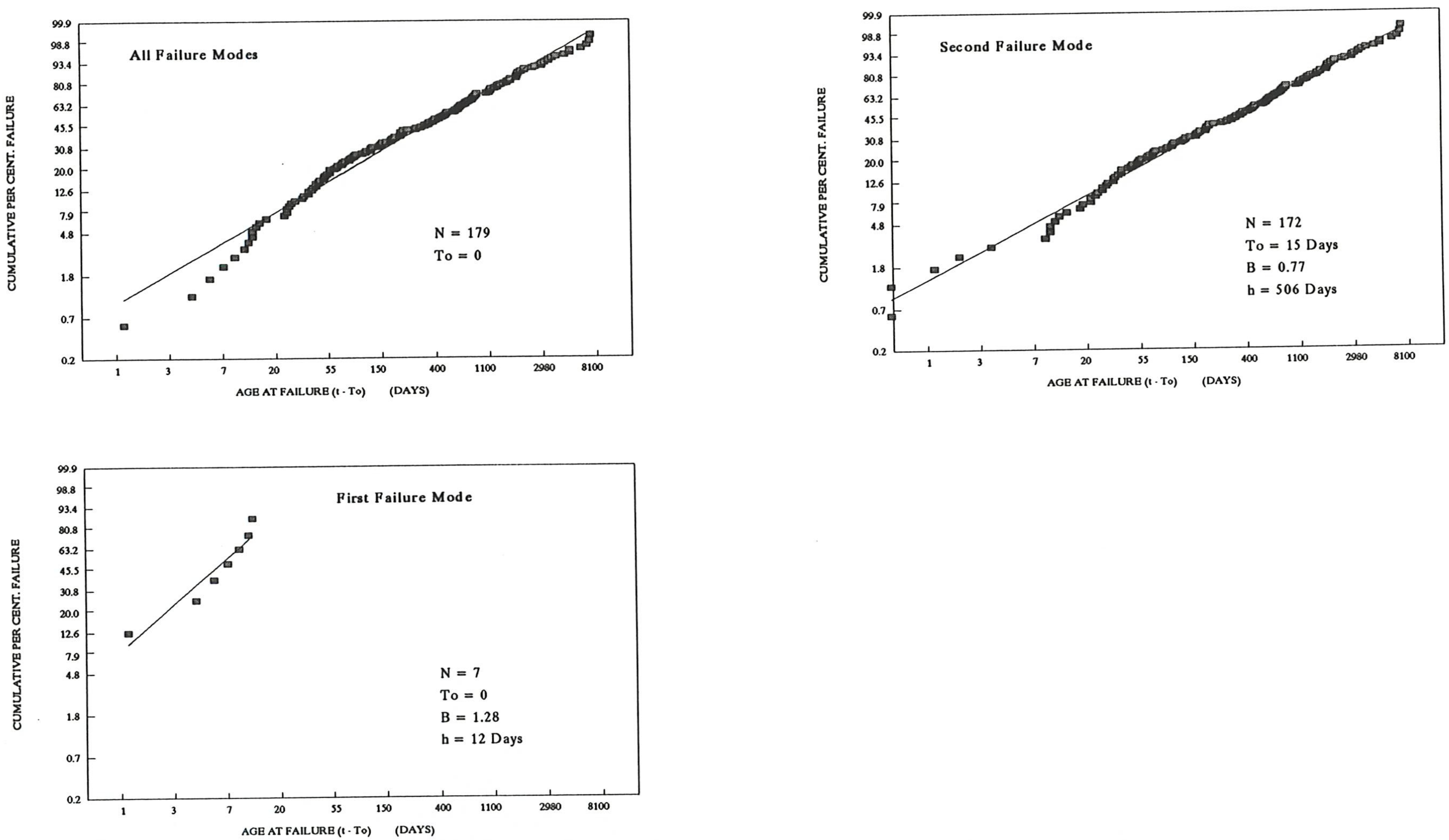


Figure A1.3 : Weibull Plot of Mechanical Seals on Naphtha Duties at Plant B

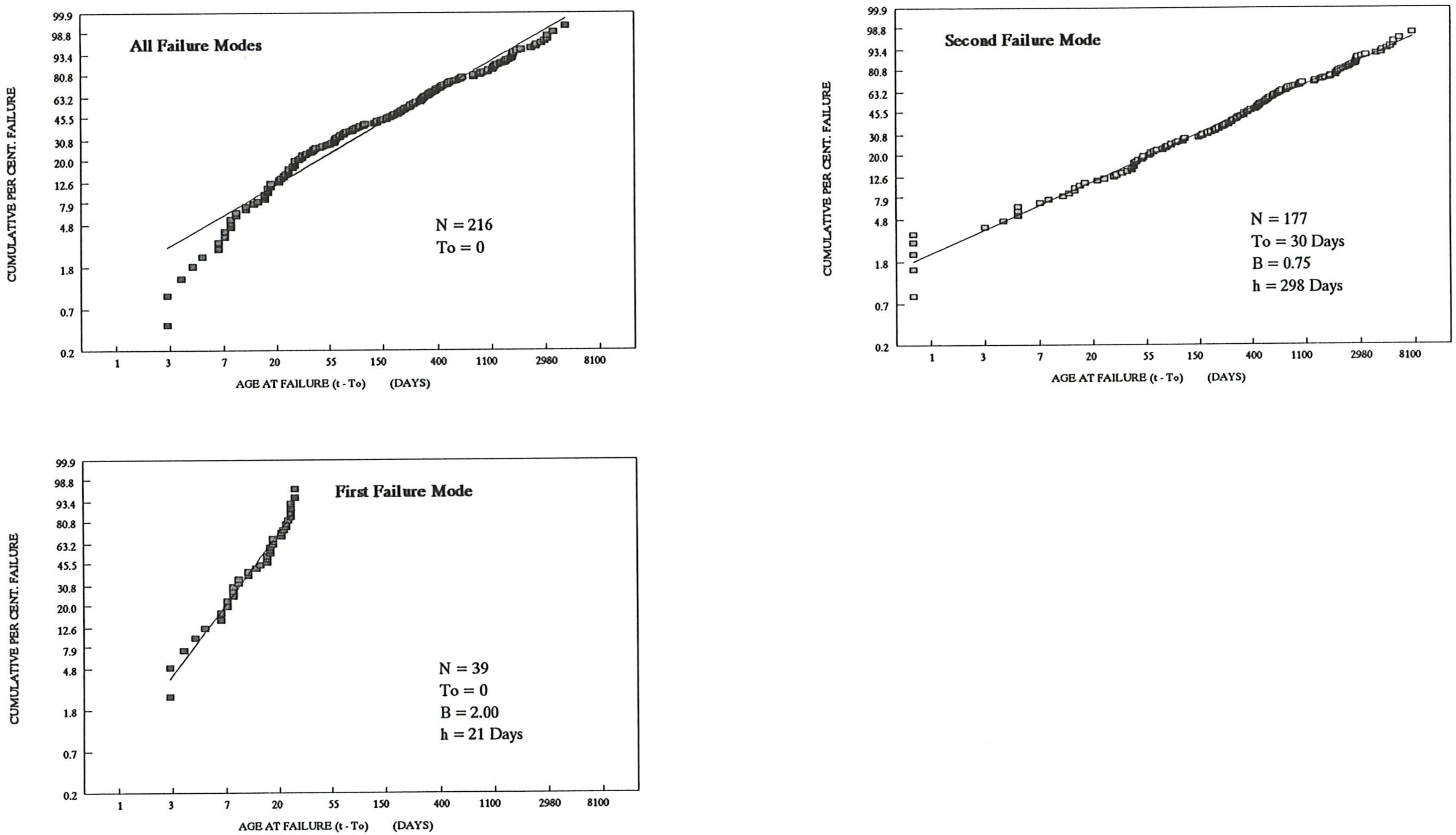


Figure A1.4 : Weibull Plot of Mechanical Seals on Heavy Hydrocarbon Duties at Plant B

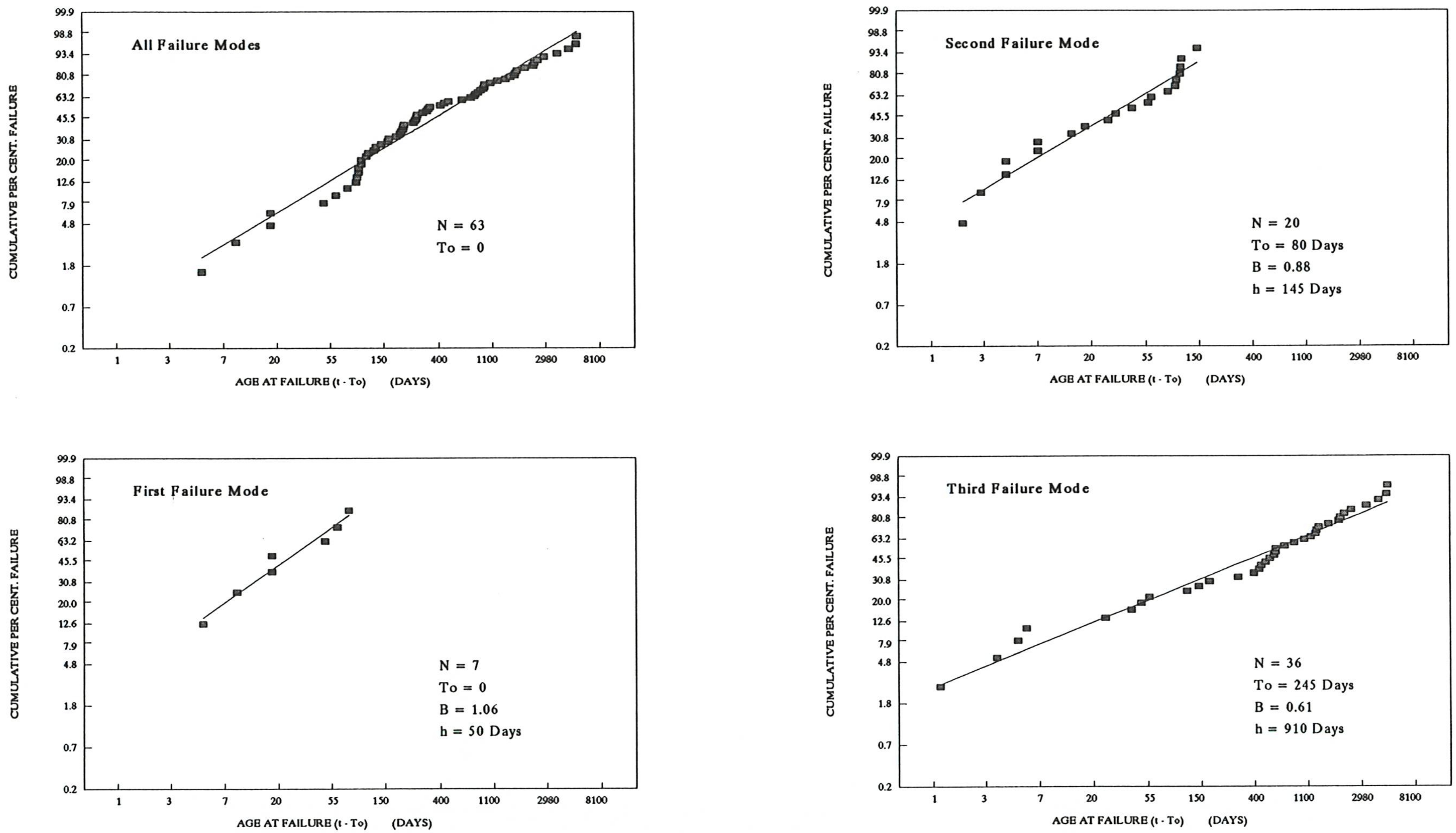


Figure A1.5 : Weibull Plot of Mechanical Seals on Sweet Water Duties at Plant B

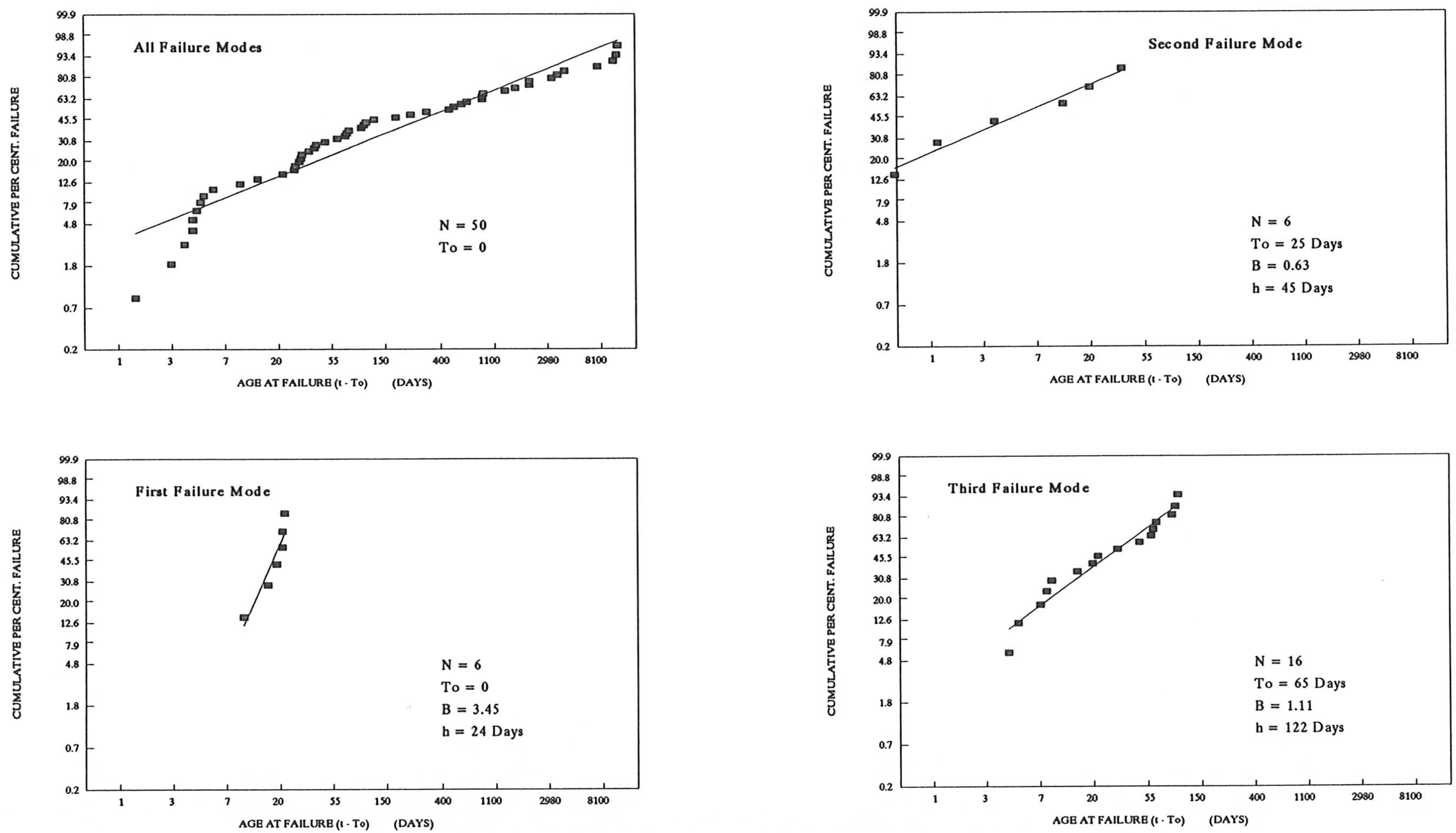


Figure A1.6 : Weibull Plot of Mechanical Seals on Sour Water Duties at Plant B

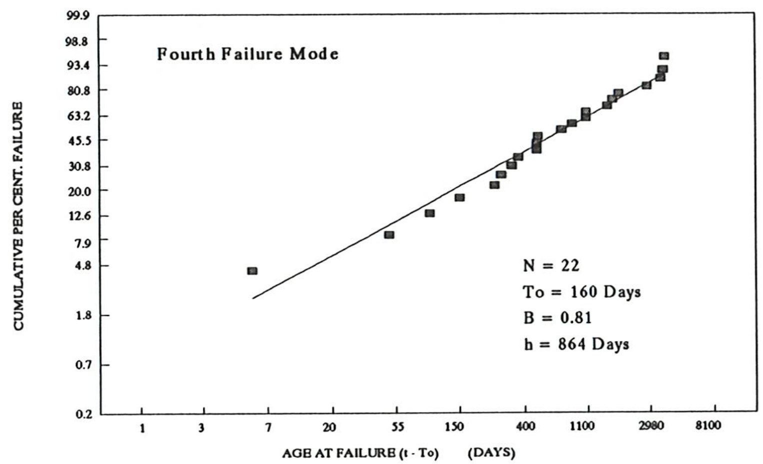


Figure A1.6 (Cont) : Weibull Plot of Mechanical Seals on Sour Water Duties at Plant B

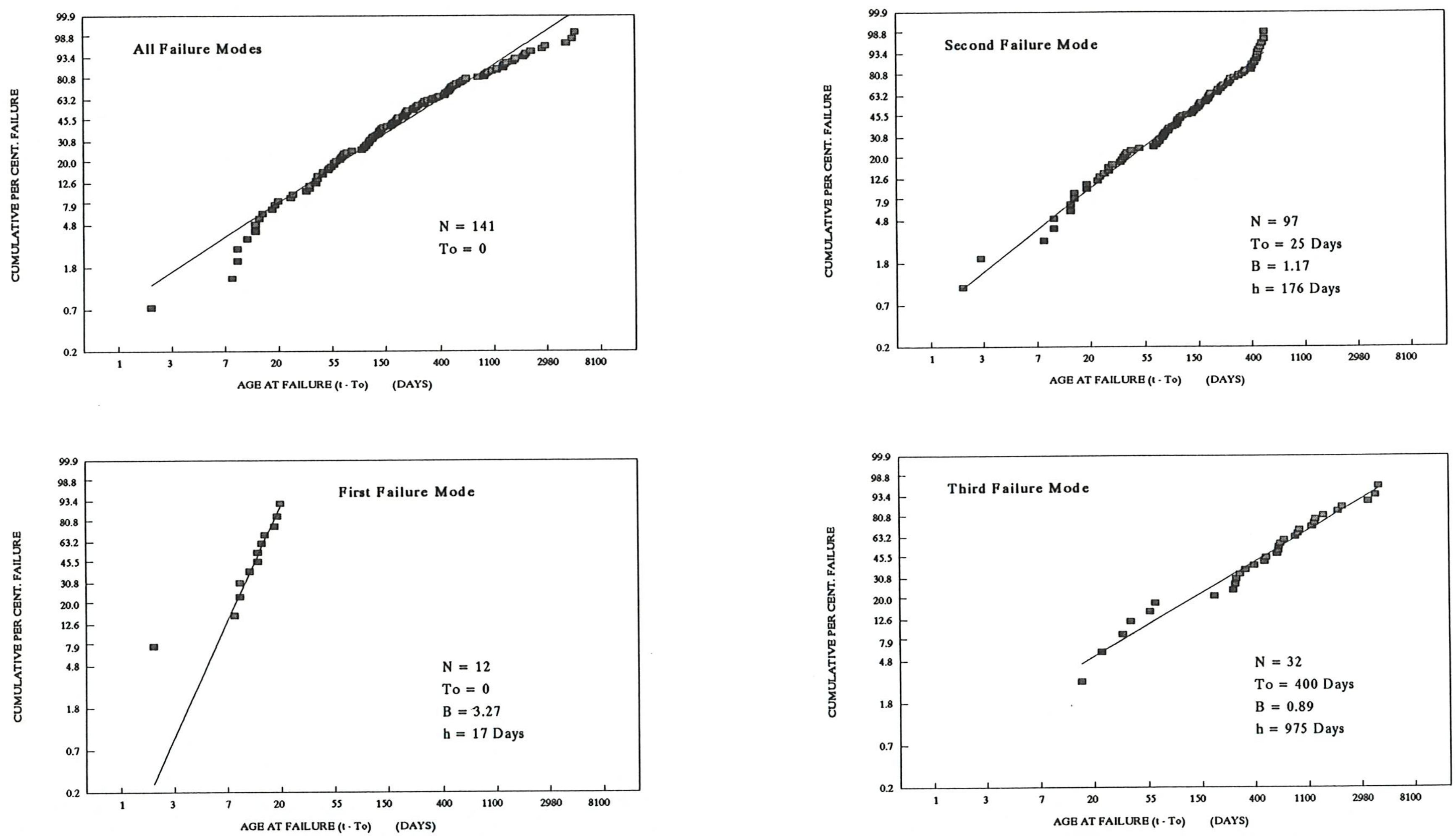


Figure A1.7 : Weibull Plot of Mechanical Seals on HF Acid Duties at Plant B

APPENDIX A2

Mechanical Seal Weibull Distributions For Various Seal Face Material Combinations at Plant A.

Associated With Chapter 8, Section 8.2.3

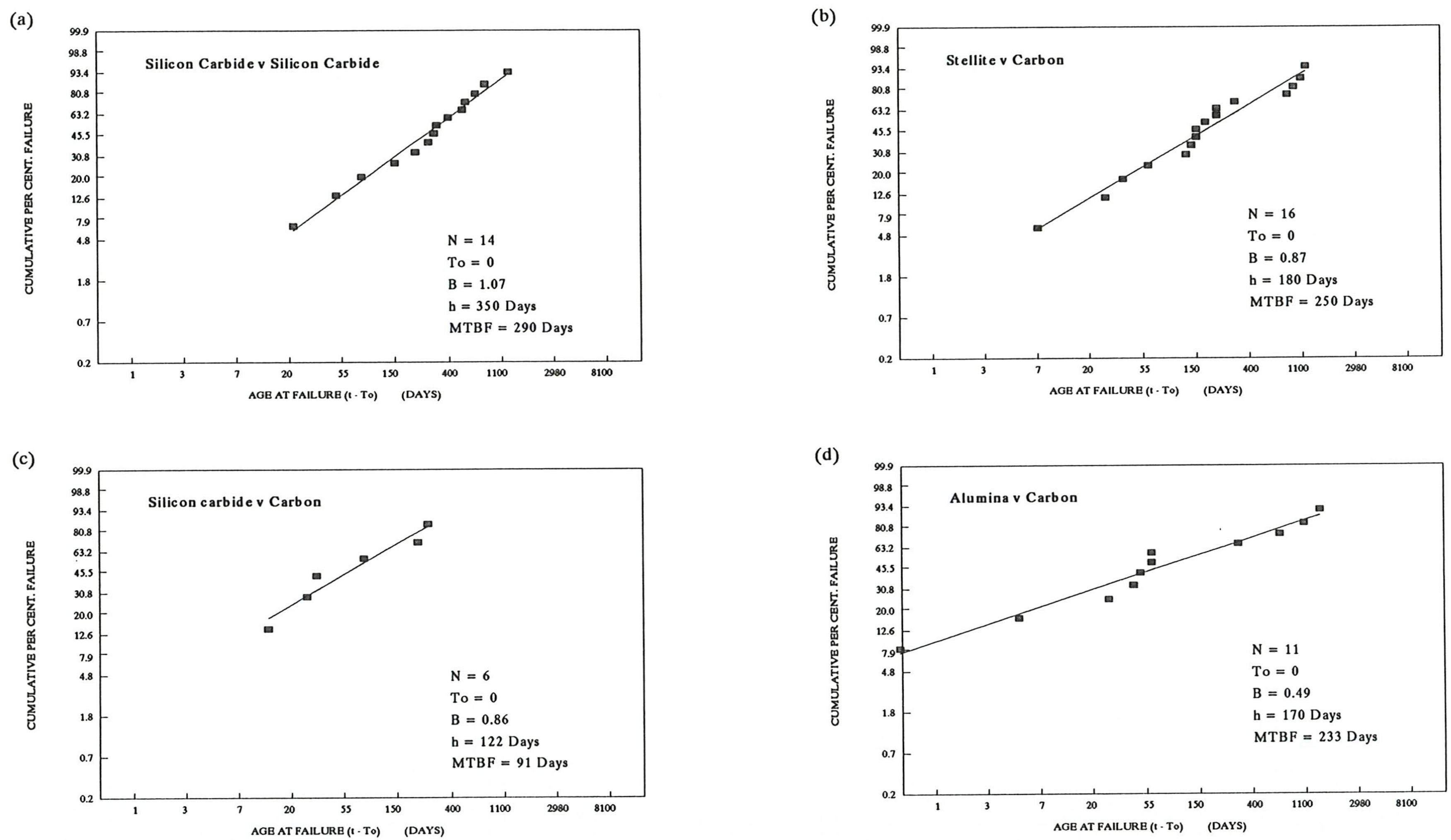
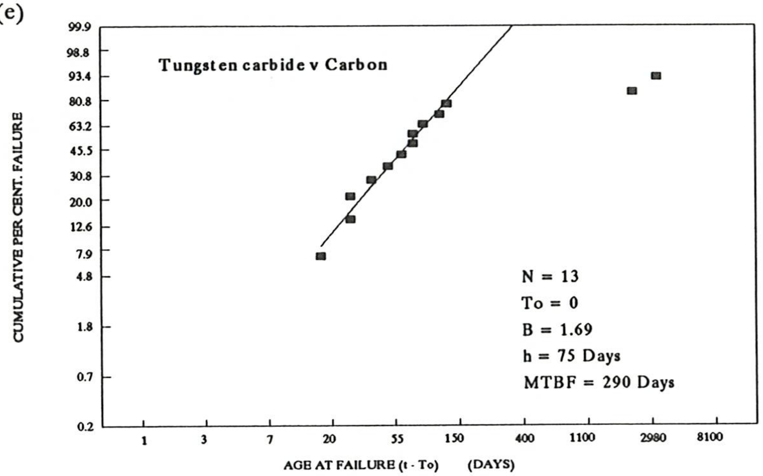


Figure A2.1 : Weibull Distributions for Seal Face Material Combinations at Plant A

(e)



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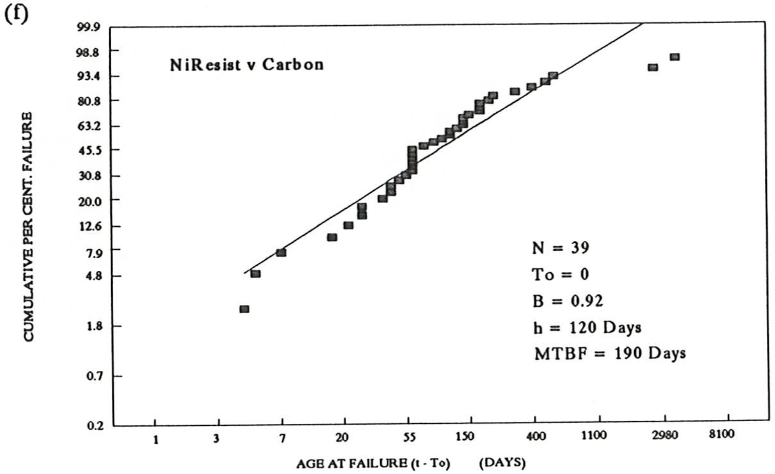

Figure A2.1 (cont) : Weibull Distributions for Seal Face Material Combinations at Plant A