AN INVESTIGATION OF ENERGY-BASED PLANNED MAINTENANCE OF OFFSHORE DRILLING MUD PUMPS

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ABSTRACT

Mud pumps used on offshore installations for drilling operations have been known to experience unpredictable breakdowns, including during critical stages of drilling. The fluid end has been identified as requiring more maintenance work due to component failure, compared to the power end. The most common maintenance strategies in use include breakdown maintenance, time-based maintenance and condition monitoring. Time-based maintenance, based on running hours, is the most commonly preferred method by most, if not all, mud pump operators. However, the nature of drilling operations require pump performance with variable loads (pressures), variable speed characteristics and time-based maintenance would not be able to account for the different operating conditions within any identical time frames.

To address this shortcoming, this research looked at the postulation that material wear loss is related to the energy expended and developed a dedicated reciprocating wear test system to identify and investigate the effect of operating variables on the wear loss of piston rubbers, which was considered to be the most problematic of the fluid end components. Experiments were run on the wear test system which represented the fifth complexity level of tribo-testing approaches. The operating variables which have significant effect on wear included sand content, load, temperature and mud viscosity. The combined effects of sand and load, and load with temperature were found to result in the most volume loss and the higher specific wear rate. Material wear loss (V_m) was found to be related to the energy expended (E) via the form : $V_m = aE^b$. The coefficients 'a' for each of the variables were constant while the value of the exponent 'b' increased proportionally as the value of the parameters. Since each of these variables were considered to affect the wear process

independently and synergistically, this rendered very difficult to establish a single coefficient 'a' and exponent 'b' which were representative of all the four parameters, especially given that the latter was found to vary for different values of the parameters.

Analysis was done on pump records collected from three offshore installations for six mud pumps covering a period of four years. The analysis used 'pressure-strokes' (PS) in the similar analogy as the relative unit 'pressure-velocity' (PV) which were adapted for use by researchers. The results of the analysis agreed with earlier experimental work about the debilitating effects of sand on component wear, and permitted the definition of a PS 'failure band' which allowed the forecasting of cumulative strokes at the 'failure point' at which component require replacement. Although the results of this analysis are only applicable for piston swab sizes of 152mm and 165mm diameter, it has indicated its feasibility and further work would be required to compile a complete list of 'failure bands' for all the swab sizes currently on offshore drilling installations.

Preliminary work on the design and development of a mud flow loop were completed and only require further industrial collaboration to bring them to fruition. The flow loop constituted the next stage of the research and offered the facilities to explore other techniques and technologies which enhance the reliability and maintainability of mud pump systems.

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Finally, this research would not have been possible without the love and support of my wife, Eunice, who has sacrificed a lot, endured tremendous difficulties and contributed in no small part to making this a success, and Yi-Lin and Yuh-Ming for just being my children and providing heaps of joy and fun.

DEDICATION

This thesis is dedicated to my family, especially to my wife, Eunice for her loving support and patience, and who has been my main source of inspiration.

NOMENCLATURE

a	acceleration (m/s^2)
$a_{(l)}$	coefficient for load effect
$a_{(s)}$	coefficient for sand effect
$a_{(t)}$	coefficient for temperature effect
$a_{(wm)}$	coefficient for mud viscosity
$b_{(l)}$	exponent for frictional work due to load effect
$b_{(s)}$	exponent for frictional work due to sand effect
$b_{(t)}$	exponent for frictional work due to temperature effect
$b_{(wm)}$	exponent for frictional work due to mud viscosity
d	diameter of loose wear
d_f	degree of freedom
E	energy expended/frictional work done, in (Nm)
$\acute{E}^{w}{}_{x}$	input power (Nm/s)
$\acute{E}^{w}{}_{y}$	use-output power (Nm/s)
$\acute{E}_z$	loss-output energy rate (Nm/s)
$\Delta\acute{E}_s$	stored energy (Nm)
$\acute{E}^{wt}$	thermal energy transformed from mechanical work (Nm)
$(\acute{E}^{w})_x$	input work (Nm)
$(\acute{E}^{w})_y$	use-output work (Nm)
$(\acute{E})_z$	loss-output work/energy (Nm)
$(\acute{E})_t$	thermal loss due to mechanical work (Nm)
F	ANOVA statistical value
F'	feed size
F_l	con-rod load (N)
Fp	force due to discharge pressure (N)
g	gravitational acceleration (m/s^2)
h^1	hardness
l	con-rod length (m)
M	mass of piston (kg)

N	cumulative strokes
Nr	reaction load
P	product size
P_p	pressure (N/m^2)
r	crank radius (m)
rad/s	radians per second
v	linear velocity (m/s)
V_{disp}	volumetric displacement (m^3)
V_w	volumetric wear loss (mm^3)
W	net energy of input (Nm)
WD	work done (Nm)
W_{ab}	energy of adhesion/surface energy (Nm/m^2)
Wi	work index (Nm)
x	distance placement (m)
$Y_{(1)}$	estimated average response (volumetric wear)
$Y_{(2)}$	estimated average response (volumetric wear)
Z_m	material loss-output
Ω	angular velocity
Σ	summation
S	system structure
{A,P,R}	system elements, properties of elements, relations between elements

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Chapter 1

INTRODUCTION

1.0 Introduction

In offshore oil well drilling, the operating environment presents conditions which cause severe wear problems in equipment which contacts the drilling fluid (called 'mud' in the drilling industry). The mud performs several important functions for the drilling operation [Appendix 1], amongst which it is required to lubricate and cool the drill string and bit; transmit hydraulic power to the drill bit; transport cuttings from the bottom of the hole; exert sufficient hydrostatic pressure to prevent well blowouts and support the walls of the hole; and also to provide information about the formations penetrated. The drilling mud operates in a 'closed' system where the mud is recirculated from the supply tanks through the derrick and drill string; out through the drill bit; up the hole carrying the drilled materials; into mud cleaning and conditioning equipment; and back to the supply tank. A typical mud system is shown schematically in Fig 1.1. The mud may be a mixture of water and/or oil; clays to vary the viscosity; materials to increase the mud weight; and chemicals to facilitate compatibility with the formations drilled. During drilling operations, the drill cuttings and formation solids are picked up and carried in suspension in the mud. These particles may not be completely removed by the mud conditioning equipment and tend to remain in the mud which makes its way back to the mud pump for

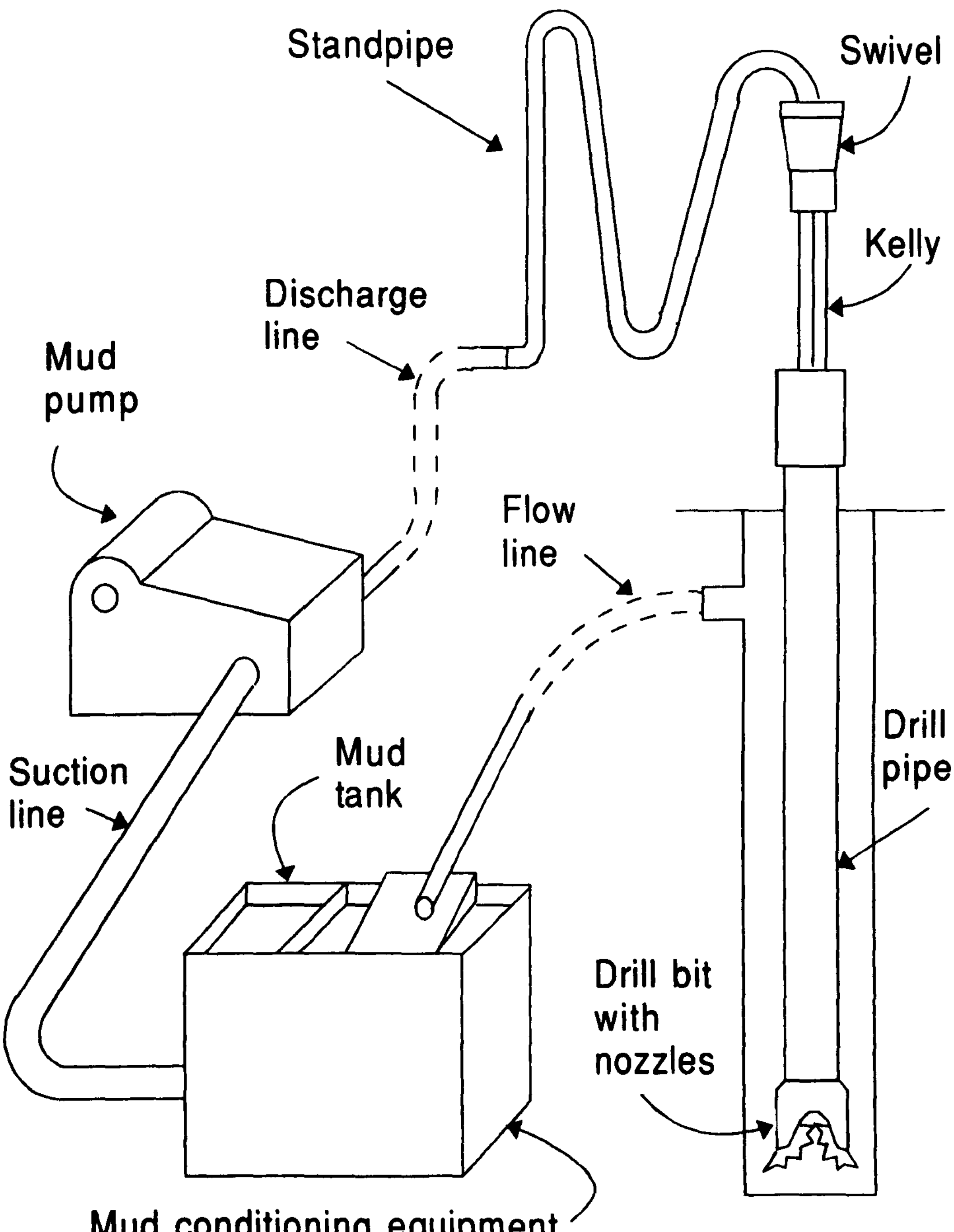


Fig. 1.1 Typical Mud System

recirculation. These materials make the fluid very damaging to the system components because of their abrasivity and chemical activity.

The mud pumps used offshore are usually of a reciprocating positive-displacement design with single- or double-acting pistons. An earlier investigation [1.1] had concluded that mud pump downtime represents a cost burden to drilling operations, equivalent to 1.55% of time spend on drilling phase activities, after excluding freak failures. Fluid end failures account for between 73% and 100% of mud pump downtime on offshore installations. The most common source of failure are fluid end expendables such as piston rubber and liners, valves and valve seats. Most of these failures tend to result in a small amount of downtime per failure but the total time spent on maintenance had a significant influence on the overheads expenditure of drilling operations. Mud pump downtime may have safety implications especially when the drilling operation has encountered a critical situation and well formation pressures need to be stabilised by drilling mud with the appropriate mud weights. Although there are usually at least two mud pumps in any offshore drilling installations, the high probability and frequency of pump downtime have given cause for concern for oil operators and drilling companies. The wear of fluid end components in reciprocating pumps is an accepted phenomena in the drilling industry. It is the unpredictable failure and subsequently equipment downtime which are considered unacceptable. Moreover, the nature of drilling operations with the presence of abrasive sand particles in the mud have complicated and accelerated the wear processes and increased the frequency of pump downtime for component replacement.

Another complicating factor is the maintenance strategies adapted by the mud pump operators and/or oil companies. A survey of the maintenance strategies of oil companies and drilling companies [Appendix 2] has concluded that most companies follow a maintenance strategy either dictated by the client oil companies or according to its own operational capabilities and economic pragmatism. The strategies generally include run-to-failure and time-based maintenance. Companies who practice planned maintenance generally use running hours for their maintenance scheduling. The nature of drilling operations, with the potential of unpredictable formation pressures, dictates that mud pumps are run on a variable speed, variable load basis. This mode of operation places irregular and uneven stresses on the fluid end expendables and result in wear which cannot be measured or predicted on the basis of time (i.e. running hours). Condition-based monitoring is deemed not suitable for the same reasons. Another example is the wire-rope carrying the block and drill pipe in the derrick, which is subject to varying loads. The loads may vary from the weight of the travelling block and swivel or elevators alone, to that plus a complete drill or casing string, as it travels up and down within the derrick. A time-based maintenance strategy would not seem suitable since the demands made upon the components are widely varying with respect to time. The current practice is to base maintenance intervals on the work done by the wire-rope, calculated by the cumulative products of hook load (plus travelling block weight) and the distance travelled up and down inside the derrick, i.e. force x distance, expressed in ton.miles. The planned preventive maintenance (PPM) intervals expressed in ton.miles are given for various wire-rope types in the American Petroleum Institute standard, API RP9B [1.2].

Most mud pump operators change out the fluid end expendables before the commencement of a new drilling phase, 'just to be sure' and to reduce the possibility of pump downtime at the wrong time. The competitiveness of the drilling industry has resulted in the optimisation of drilling equipment. This means operating at higher speed and pump pressures. Higher pump pressures may translate into greater fluid end component wear and hence higher maintenance cost. However, the higher pressures also imply greater rate of penetration (ROP) and hence favourable drilling rates and faster well completion. This translates into earlier return on investment from faster oil production. In this context, the maintenance cost of mud pump fluid end consumables is considered negligible when the overall situation encompass minimum drilling downtime and earlier oil and gas production. Conversely, one mud pump fluid end failure may mean a mere one hour or up to five days of drilling downtime, depending on the criticality of the hole section. The situation may be complicated when performing directional drilling. This may be so because when mud circulation stops during directional drilling, the drill cuttings suspended in the mud tend to start settling. The inclination of the hole means the cuttings have a shorter distance to settle, subsequently resulting in a build-up of cuttings around the drill pipe, especially round the bends. When this happens, the drill pipe may become stuck and expensive downtime incurred to free the drill pipe before drilling can resume. This situation may be minimised by more efficient hole cleaning through better mud circulation. This calls into question the reliability of mud pumps and the ability to run efficiently with minimum downtime.

It is in this context that this project has been initiated. A more reliable means of maintenance scheduling is desired to ensure maximisation of component service life and hence reduced maintenance cost. But more importantly, the ability to predict the cumulative wear and failure of the fluid end components. The proposal is to use the energy expended rather than running hours as the basis for maintenance scheduling and wear prediction. The research is focused on the piston rubber-liner configuration as this is one of the most problematic of the fluid end failures, and a survey of offshore mud pump operators [Appendix 3] have revealed that the piston rubber is, on average, replaced three times for every change of the liner.

1.2 Objectives of the project

The objectives of the project are :

1) To develop a novel technique for predicting wear in variable-speed, variable-load reciprocating machine components. This technique is based on predicting component wear via cumulative energy consumption.

2) To generate information necessary for the development of a planned maintenance strategy.

3) To determine the factors which would maximise the service life of fluid end consumables in reciprocating pumps, particularly the piston rubber and liner of mud pumps.

4) To explore the possibility, through analysing mud pump field service data, of creating a new unit of work done, suitable for practical application in offshore field service.

1.3 Literature Survey

Much has been written about the merits of a maintenance programme. Danahy [1.3] postulated that maintenance is the single activity offshore that can have the most impact on productivity, safety and downtime. Similarly, Bloom [1.4] pointed out the key to reliable slush pump performance is a thorough pump maintenance programme. Failure to establish and adhere to sound maintenance practices can result in higher operating costs, increase pump downtime and generally inefficient and undependable slush pump service. He concluded that it is much easier and frequently less costly to schedule parts replacement which are convenient, rather than to lose drilling time due to unexpected component failures. A good preventive maintenance programme can reduce pump operating costs and improve rig efficiency. Rizzone [1.5] also emphasised the need for proper maintenance and operation to derive the optimum performance from reciprocating pumps. Components of reciprocating pumps, in particular the fluid end [1.1] are expected to wear. As Arnett [1.6] claimed, 'controlled wear is the very purpose of fluid end expendables'. The pulsating nature of reciprocating pumps typically give rise to a host of problems not encountered by other pump configurations [1.7-1.8]. To compound the situation, Babaev et al [1.9] found that sudden slush pump failures occur owing to the failure of fluid end components by 'hydroabrasive erosion' and concluded that poor reliability of these pumps are due, primarily, to the short service life of fluid end components. Pump downtime is a routine of mud pump operation but what is not desirable is the occurrence of unexpected and unscheduled failures and the apparently high frequency of downtime. To address this problem, there are maintenance strategies which are commonly employed :

1) Run to failure/breakdown maintenance
2) Planned or preventive maintenance
3) Predictive maintenance

Run to breakdown maintenance would often be preferred in situations where there is either a standby system with automatic change over to relieve the failed unit, or where the failure is apparent but not critical [1.10]. But there are obvious disadvantages, which included :

a) the failure component can cause damage to others resulting in very high consequential cost;

b) the failure may occur at any inconvenient time, ie it is unpredictable;

c) the failed equipment may affect production, especially if critical machinery or process.

Preventive maintenance would appear to be an improvement as the complexity and criticality of machineries increase and there is demand for better reliability in the machinery systems. Pratt [1.11] outlined the step-by-step approach to the practical implementation of an effective maintenance enhancement program, through condition based maintenance. But this is not the optimum method because engineering components do not fail at regular intervals. Some disadvantages include :

a) failures will continue to occur between overhauls and may be unexpected and inconvenient;

b) during overhaul, many components which may be in good condition will be stripped and inspected unnecessarily, and any mistakes on reassembly can make the condition worse than before the overhaul;

c) the overhaul process involve the time-consuming examination of many components and may contribute to loss of profitable production.

A predictive maintenance program seeks to minimise unscheduled machinery failure, maintenance costs and loss of production. This is accomplished by regularly monitoring the status of all critical plant items, identifying and quantifying incipient problems, quantifying the severity of each problem and scheduling maintenance to prevent failure. Another new concept is condition-based maintenance, where condition monitoring (CM) is used to determine the plant condition, and it is then stopped for repair just before a fault occurs. However, CM is only suitable for components which fail by progressive failure, as contrasted with sudden failure mechanism. The progression of the failure gives the lead time required to investigate the necessary corrective actions. This may not be very suitable for critical components which do not fail in a relatively slow progressive manner. The monitoring system must monitor each component regularly and reliably, waiting to highlight the moment when random failure occurs. There are many publications which explain in detail the various methods of condition monitoring, including references [1.10-1.15].

Both regular preventive and condition-based maintenance assume a reasonably uniform operating environment, to justify the regular overhaul interval, and the concept of regular condition monitoring. For machinery which is subjected to random overload situations, or even just variable loading conditions (as in reciprocating mud pumps), the tendency to fail at random intervals may tend to occur at greater frequency than the progressive failures. To claim that there is the tendency of failure at random intervals is not altogether accurate because there exist the rationale that machinery fail as a result of cumulative stresses or loads, or rather the failure occurs when the cumulative stress has reached the point of material failure. There does not appear to be any indications that maintenance has been scheduled on the basis of cumulative loads or stresses. A car manufacturer [Appendix 4] has developed a computerised system which have a variable planned maintenance (PM) schedule based on the drivers' driving habits, distance covered and the time elapsed. The distance and time may be pre-determined but the engine speed varies and depend on driving conditions and habits, load, road conditions etc. These factors are correlated by the computer which then determine the PM schedule.

Condition monitoring can be successful for components with progressive failure modes which can be monitored by direct measurements. There are components which fail without a prior indication from current monitoring methods. These components are dealt with differently in terms of a 'safe life' after which the component has to be discarded [1.10]. The determination of safe life is very difficult as it requires the assessment of the likely variation in material properties between one component and another. If the operating conditions are variable, the contribution to the consumption of the safe life period by each operating phase has to be determined. The challenge lies in the question of what form and in what units.

Performance monitoring involves considering the functions of a component or system and the application of methods to measure these functions. There may usually be two related conditions associated with a component or system and a comparison of the relationship between the two can indicate its condition. The pressure/flow relationship of pumps is one example. However, the pressure in a reciprocating pump is dependant on the rotational speed of the crankshaft. Since there are no standard operating conditions of the mud pump, the highly variable loads add to the difficulty of determining the extent of 'safe life' remaining after each operating phase. Consequently, this research project addresses this aspect by attempting to determine the safe life of fluid end components prior to failure, by measuring the cumulative work done or energy expended. Other aspects such as the effects of abrasives in the mud and temperature are also considered.

Recent developments have seen the introduction of systems which enhance the on-line and off-line monitoring capabilities in detecting changes in performance characteristics. Parks [1.16] reported the use of data-acquisition (DAC) systems in conjunction with condition-based preventive maintenance programmes to lower operating and repair costs on rotating equipment. Recent improvements in measurement while drilling (MWD) technology have increased drilling efficiency by allowing the driller to steer the bit with real-time formation evaluation measurements during drilling [1.16]. Similarly, the Management Drilling System (MDS) system developed by Sedco Forex [1.17] facilitates three main functions combined in an integrated network: the real-time monitoring, event detection and alarms; off-line analysis and reporting of drilling data; and rig management; allowing faster access to more accurate and reliable information. The PPA (pump pressure analyzer) is reported to provide an accurate measurement of mud flow into the well and also gives advance warning of fluid-end parts that are starting to fail. However, though the system allows the driller to spot trends in the mud pump standpipe pressure, it does not detect and measure the deterioration in the fluid end module thus making it difficult to quantify and predict when the fluid end parts would fail. Hence though the MDS system can spot early trends and prevent further pump damage, if this happens during a critical drilling phase, then the risk of pump downtime is still there. There is no facility to forecast possible component failure.

Downtime in mud pumps may be caused by failure of either the power or fluid end. McKay [1.1] has concluded that the majority of failures are caused by fluid end components. The major causes can be traced to the wear caused by the reciprocating effect of pumping highly abrasive mud. A list of the failure modes of fluid end components [1.18] has been compiled in a fault-tree format [Appendix 5]. A functional analysis of the pumping cylinder in mud pumps, comprising the liner and piston, by Lewis [1.19] attributed the piston failures to the deterioration of the piston rubber (which is also called piston 'swab' in the drilling industry) due to the effects of heat, abrasion and rubber extrusion where the high fluid pressure forces the piston rubber material through the gap between the liner and the piston, causing 'nibbling' of the extruded rubber.

Component failures tend to be associated with wear and friction, considered sub-areas of tribology. Since its inception in 1966 [1.20], the term *tribology* has been regarded as a general concept embracing all aspects of transmission and dissipation of energy and materials in mechanical equipment including the various aspects of friction, wear and lubrication. A glossary of terms and definitions for the field of tribology has been included as a key word index in the ASME Wear Control Handbook [1.21]. Friction is thus defined as 'the resisting force tangential to the common boundary between two bodies when, under the action of an external force, one body moves, or tends to move relative to the surface of the other.' Wear is 'the progressive loss of substance from the operating surface of a body occurring as a result of relative motion at the surface' [1.22]. Hence friction and wear are two different yet inherently linked entities. This is due to friction being essentially an energy dissipation process where the frictional work generates frictional heat which has an effect on tribological interaction which in turn result in wear processes leading to the generation of loose wear particles. A friction-induced energy dissipation model of friction has been compiled by Briscoe [1.23], as shown in Fig 1.2. The frictional work is considered to be dissipated in two separate regions in the interfacial region. The interfacial zone is thought of as being very narrow and corresponds to high rates of energy dissipation. The other process involves deformation within a larger volume of material and hence lower rates of energy dissipation. These two processes involved in the model can be characterised as :

a) plastic grooving, leading to microcutting;

b) viscoelastic grooving causing fatigue cracking and tearing with sub-surface heating and damage;

c) true interfacial sliding; high, effective rates of surface strain and heating;

d) interface zone shear; rupture within the polymer and transfer wear;

e) a subgroup of true interfacial sliding, the propagation of Schallamach waves.

There is every possibility that in practice, all these processes may overlap and occur simultaneously. Because friction is an energy dissipation process, an energy consideration of friction [1.22] may be useful where the 'loss' process of mechanical energy due to friction may be divided into different phases as compiled in Fig 1.3. Firstly, the mechanical energy is first introduced into the contact zone by the formation of the real area of contact. Secondly, a transformation of mechanical energy takes place mainly by the effect of plastic deformation, ploughing and adhesion. Finally, the dissipation phenomena include the effects of thermal dissipation, storage or emission.

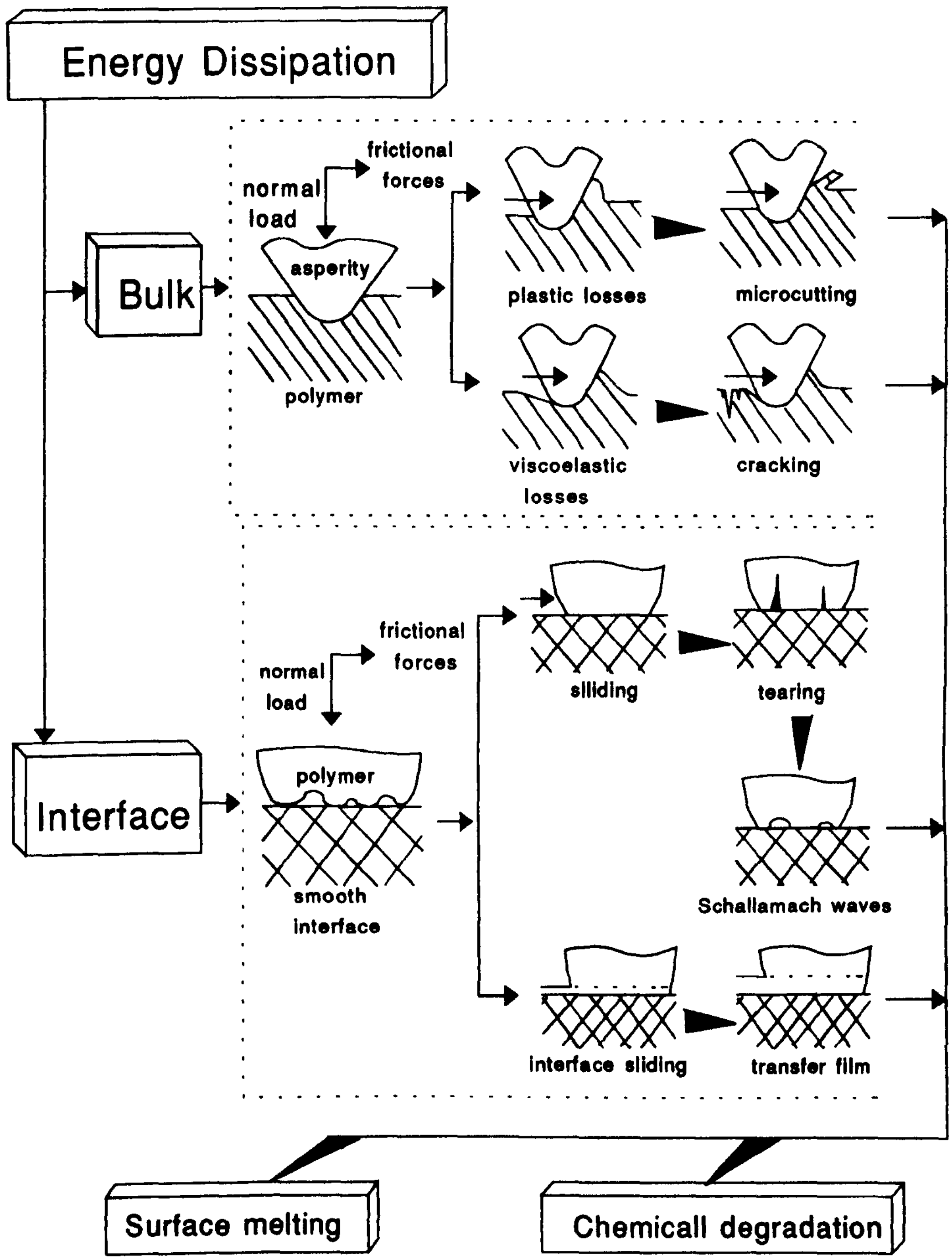


Fig. 1.2 Schematic of energy dissipation model of friction (adapted from ref.1.23)

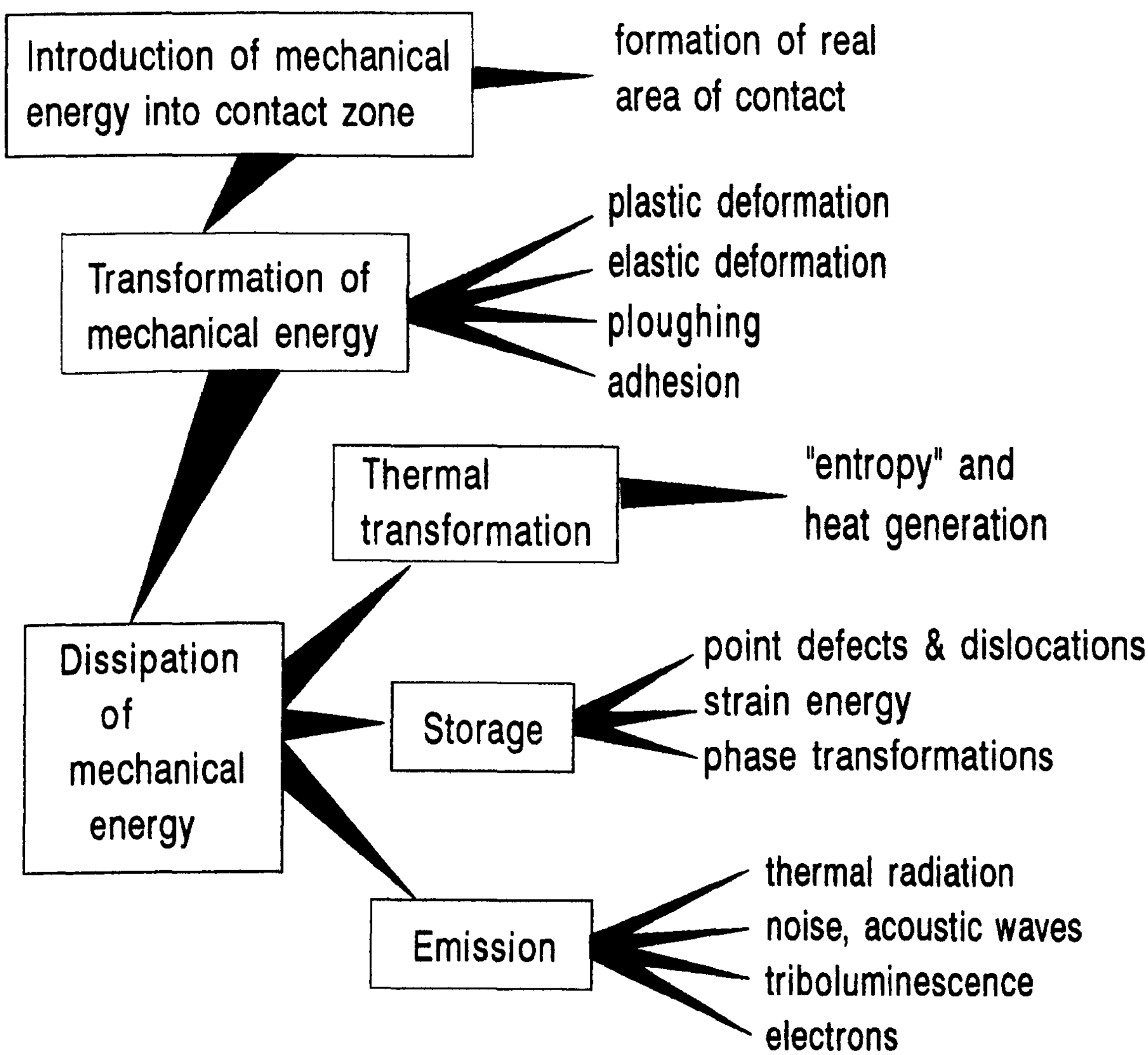


Fig. 1.3 Different phases of mechanical energy 'loss' process due to friction (adapted from ref. 1.22)

The great variety of types, modes, or processes of friction, lubrication and wear which determine the behaviour of any tribological entity, may necessitate a sub-classification of these terms, using criteria such as :

a) kinematics or type of motion (sliding, rolling, impact, etc),

b) type of material (solid, fluid, metals, polymer, etc),

c) type of interfacial tribological process (hydrodynamics, adhesion, abrasion, etc).

A discussion on wear process tend to include the complexity of the generation of loose wear particles. The two broad classes of tribological processes (Fig 1.4) that initiate the chain of events leading to the generation of wear particles and material removal from a given tribological system are :

a) *stress interactions*. These are due to the combined action of load, forces and frictional forces and lead to wear processes described broadly as surface fatigue and abrasion.

b) *material interactions*. These are due to intermolecular forces either between the interacting solid bodies or between the interacting solid bodies and the environmental atmosphere (and/or the interfacial medium) and lead to wear processes, described broadly as tribochemical reactions and adhesion.

A significant amount of research had been done on these four broad classes of wear processes. These included the 'delamination theory of wear' put forward by Suh [1.24,1.25], a contribution to the theory of surface fatigue wear mechanisms by Halling

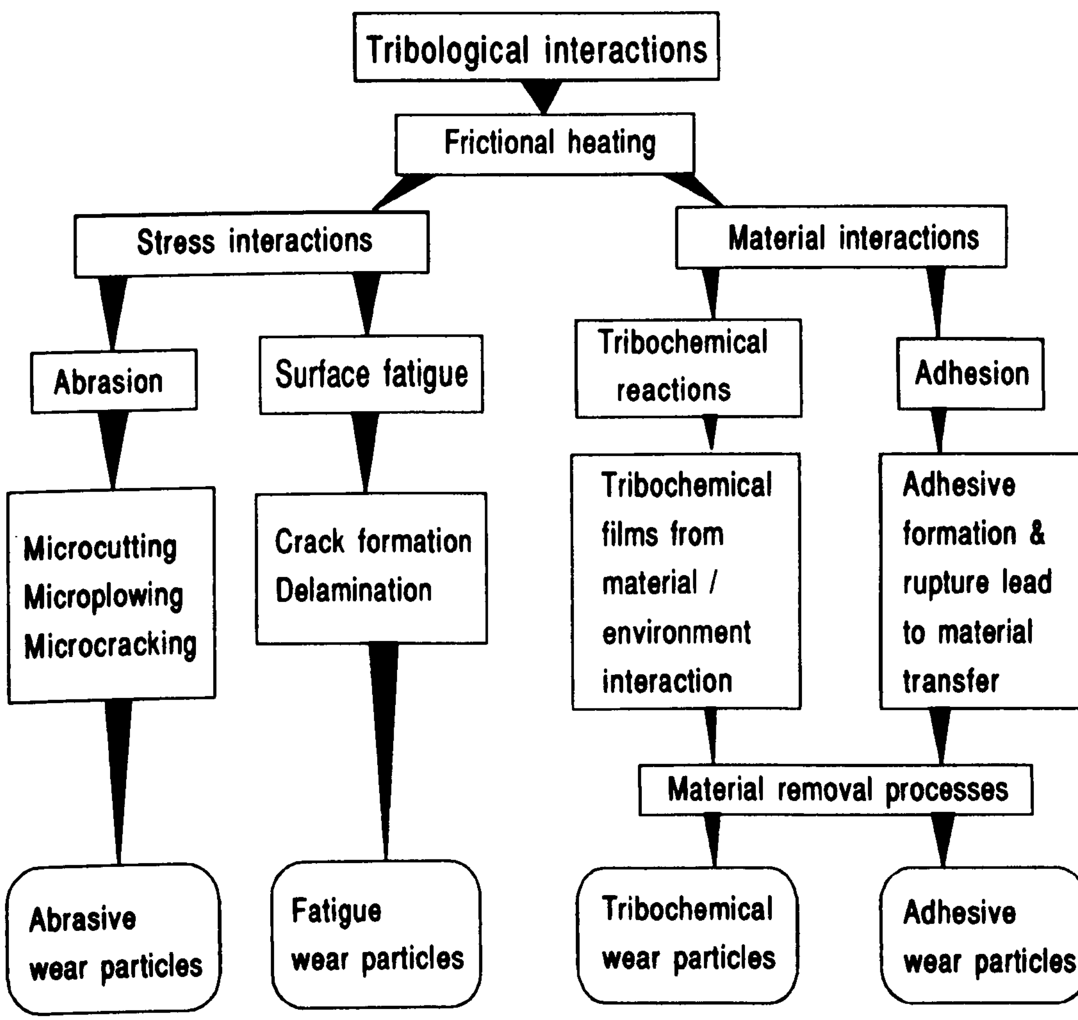


Fig. 1.4 Tribological interactions and wear mechanisms (adapted from ref.1.22)

[1.26], a tribochemical wear hypothesis on steel by Quinn [1.27], discussions on adhesive and abrasive wear by Bowden and Tabor [1.28], Kragelskii [1.29] and Rabinowicz [1.30]. Given the great variety of types, modes and processes of wear and its many influencing factors, it helps to classify the types of wear occurring within a tribological entity. Fig 1.5 summarises the classification of wear processes where the types of wear are named according to the materials involved in wear (solids, fluids etc); the type of kinematic tribological action (sliding, rolling, impact, etc) and the type of interfacial wear mechanism (adhesion, abrasion, surface fatigue, etc).

Friction and wear depend on so many influencing factors that the whole 'tribological system' (or 'tribo-system') may have to be considered. Quinn [1.31] defined a tribosystem as a system in which motion, energy and materials are transmitted, in various relative amounts according to the required functions of the system, from clearly prescribed inputs to desired outputs. Thus measured quantities of friction and wear, eg wear rate or friction coefficient, depend on the following basic groups of parameters [1.22] :

a) the 'structure' of the tribological system, ie the material components of the system and the tribologically relevant properties of the system's components;
b) the operating variables, including load (or stress), kinematics, temperature, operation duration, etc;
c) the tribological interactions between the system components.

The above form the fundamentals and methodology of a systems approach to tribology, as outlined by Czichos [1.22,1.32-1.34]. The analysis of a tribo-mechanical system can

System structure	Tribological action	Types of wear	Effective mechanisms: Adhesion	Abrasion	Surface fatigue	Tribo-chemical reaction
Solid — interfacial medium — solid	sliding, rolling, impact				✓	✓
Solid — solid (with solid friction, mixed lubrication)	sliding	sliding wear	✓	✓	✓	✓
	rolling	rolling wear	✓	✓	✓	✓
	impact	impact wear	✓	✓	✓	✓
	oscillation	fretting wear	✓	✓	✓	✓
Solid — solid and particles	sliding	sliding abrasion		✓		
	sliding	sliding abrasion (3-body abrasion)		✓		
	rolling	rolling abrasion (3-body abrasion)		✓		
Solid — fluid with particles	flow	particle erosion (erosion wear)		✓	✓	✓
Solid — gas with particles	flow	particle erosion (erosion wear)		✓	✓	✓
	impact	impact particle wear		✓	✓	✓

Fig. 1.5 Classification of wear phenomena (adapted from ref. 1.22)

be achieved with the following steps :

1) Systems function

   a) separate the system from its environment by the proper choice of a hypothetical system's envelope,

   b) compile all inputs of operating variables and, where possible, also the use-outputs,

   c) describe the functional input-output relations.

2) Systems structure

   a) identify the "elements" (or material components) of the system,

   b) characterise the interrelations and interactions between the elements (i.e. the contact, friction and wear processes),

   c) specify the relevant properties of the elements.

Fig 1.6 shows the structure of a tribo-system and the tribological interaction between system components. Seifert and Westcott [1.35] also recommended the use of a systems approach to understanding the tribological processes at the interface of moving surface in mechanical systems.

Wear prediction refers to making an estimate of how much wear will occur after a given amount of time or frictional work done on the tribological system. Because of the many factors which influence the rate of wear and the many conditions under which friction and wear processes occur, be it two- or three-body dry or wet abrasion, there is no single

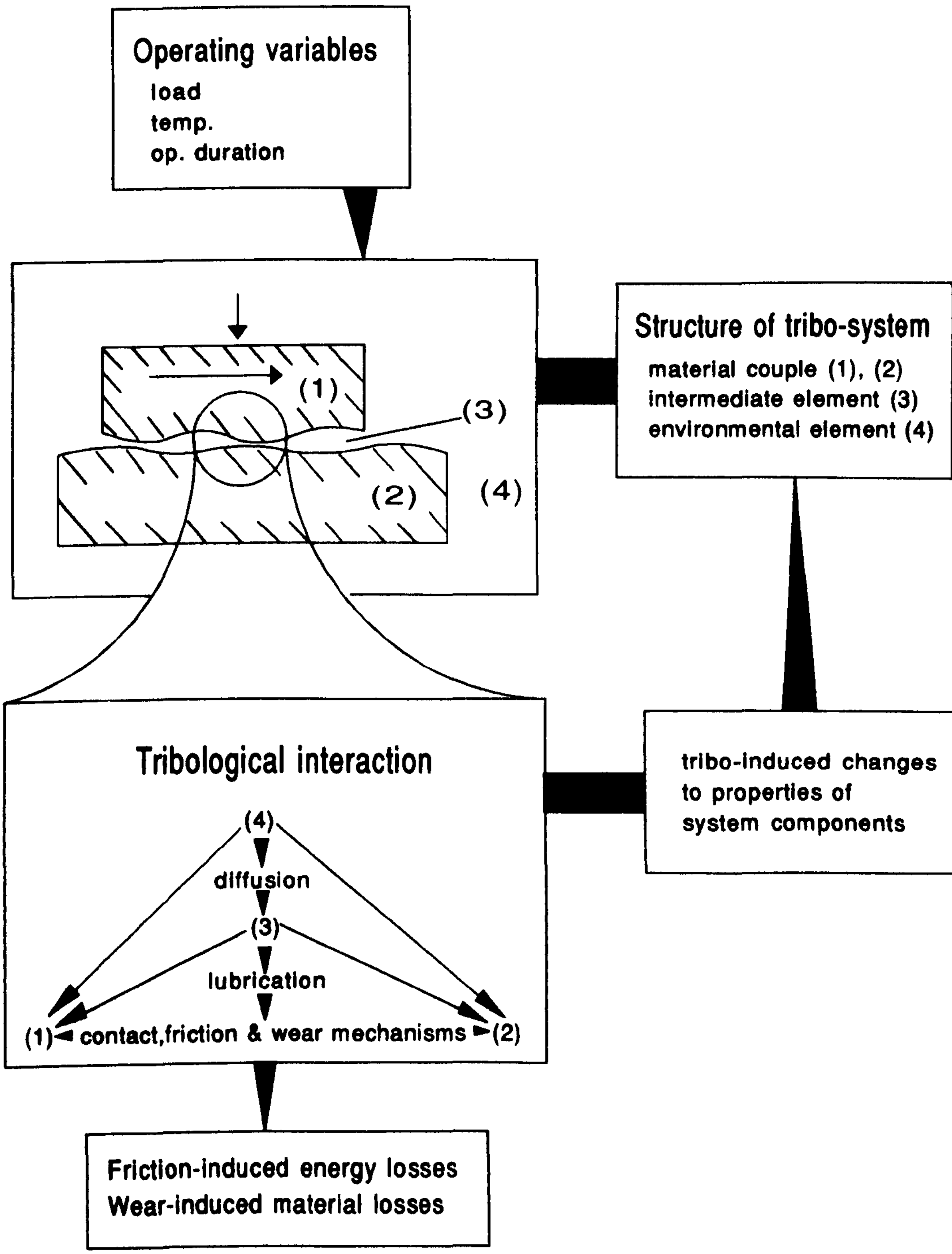


Fig. 1.6 Structure of tribo-system and the tribological interaction between system components (adapted from ref. 1.22)

solution to the prediction of wear. Nevertheless, attempts have been made by researchers to develop models to attempt to predict wear, though mainly in dry sliding conditions. Dumbleton and Rhee [1.36] examined the applicability of the zero wear model to a polymer in sliding contact with a metal counterface. They observed that the model enable the number of passes to reach the zero wear limit to be predicted but that deviation between the theory and experiment occurred for large number of passes. Elkholy [1.37] obtained an analytical model for the prediction of the amount of wear in a slurry pump under specified running conditions. He observed that experimental and calculated results for cast iron as an example of a hard and brittle material are comparable on a specimen weight loss basis. Lin and Cheng [1.38] introduced a wear model to study the dynamic wear behaviour observed in practice. However, though the postulation in this model can explain the commonly observed running-in, steady-state, or accelerated wear phenomena, it appeared not to have considered the effects of factors other than sliding distance or time, such as wet three-body abrasion as is dominant in mud pump operation. Wang et al [1.39] proposed a different approach for predicting wear of unlubricated piston rings in gas compressors by using a new method to evaluate practical values of the PV factor for piston rings. Since gas compressors are axially-loaded rotating machinery which normally operates at a fixed speed for optimum performance, the parallel cannot be drawn for reciprocating pumps operating on a variable-speed variable-load regime. Rymuza [1.40] presented an analysis of wear in unlubricated and lubricated miniature journal bearings which concluded that the energetical state of the polymer material, which governs the wear process and the loss of mass of the polymeric element, is a function of the structure of the tribological system and the operating conditions. Muhr and Roberts [1.41] maintained in practice the wear of rubber remains difficult to predict. They suggested that though the main cause of abrasion is tearing or fatigue under the action of high local stresses caused by friction, wear is proportional not only to abradability but also to the amount of sliding suffered by a rubber article (eg rubber tires), and this is often an unknown quantity. In mud pumps, quantifying the volumetric wear of the piston rubber is equally difficult since wear occurs in chunks and by micro-tearing, as suggested by Lewis [1.42] in the form of rubber extrusion under high pressure during reciprocating sliding. Peterson [1.43] mentioned three different approaches to predicting wear in service :

a) analytical - where a simple wear equation is proposed and attempts made to account for significant variables, either directly or through the use of wear coefficients, obtained from bench tests or service experience.

b) component - where wear is measured under controlled conditions in component bench test or prototype tests and their results extrapolated to service usage.

c) service wear measurement - where wear is measured either directly or indirectly on components in service or those temporarily removed from service. The wear rate is usually determined as a function of operating time with little considerations being given to variables such as load, temperature etc.

The varied nature of mud pump operations in offshore drilling make wear measurement of fluid end components very difficult to achieve. The normally high pressure ranges of mud pumps, up to as high as 34.5 MPa (5000 psi), would necessitate the availability of special equipment robust enough to simulate high pressure cylinder pumping in order to enable wear to be measured. A combination of these three options in various modified forms may present the best means of predicting the wear of fluid end components in mud pump operations.

Several researchers have indicated the correlation between the wear of materials and the energy expended. Utez and Fohl [1.44] considered wear as an energy transformation phenomena whereby during a tribological process, the differences between the input and output energies was equivalent to the friction energy, ie the energy lost during the operation of the system (Fig 1.7). All three elements of the tribosystem, (1) base body, (2) counter body, and (3) intermediate matter, participated in this energy transformation. The main contribution to friction energy was the energy expended in plastic deformation and which appeared as thermal energy. Also considered important was fracture energy (surface energy) which was the energy required to generate new surfaces and thus form wear particles. An indication of the energy absorbing processes on pairs of metallic materials listed according to their estimated proportions is given in Fig 1.8. Some correlations between surface energy and characteristic wear phenomena have also been found by Rabinowicz [1.45]. The basic premise of the theory for dry wear proposed by Qiu and Plesha [1.46] was that wear was an energy-related process, ie the production of a certain amount of wear volume requires a specific energy. This energy was usually supplied by sliding work and dissipated by plastic deformation, heat generation and ultimate material rupture. Wear on a microstructural level was generally accepted as a result of one, or a combination, of several different and very complicated physical processes including subsurface cracking and delamination, adhesion, ploughing and thermal, environmental and chemical effects. Qiu and Plesha's approach to wear modelling did not explicitly account for any particular microstructural wear mechanism but rather included the actual wear mechanisms under the general umbrella of an energy-

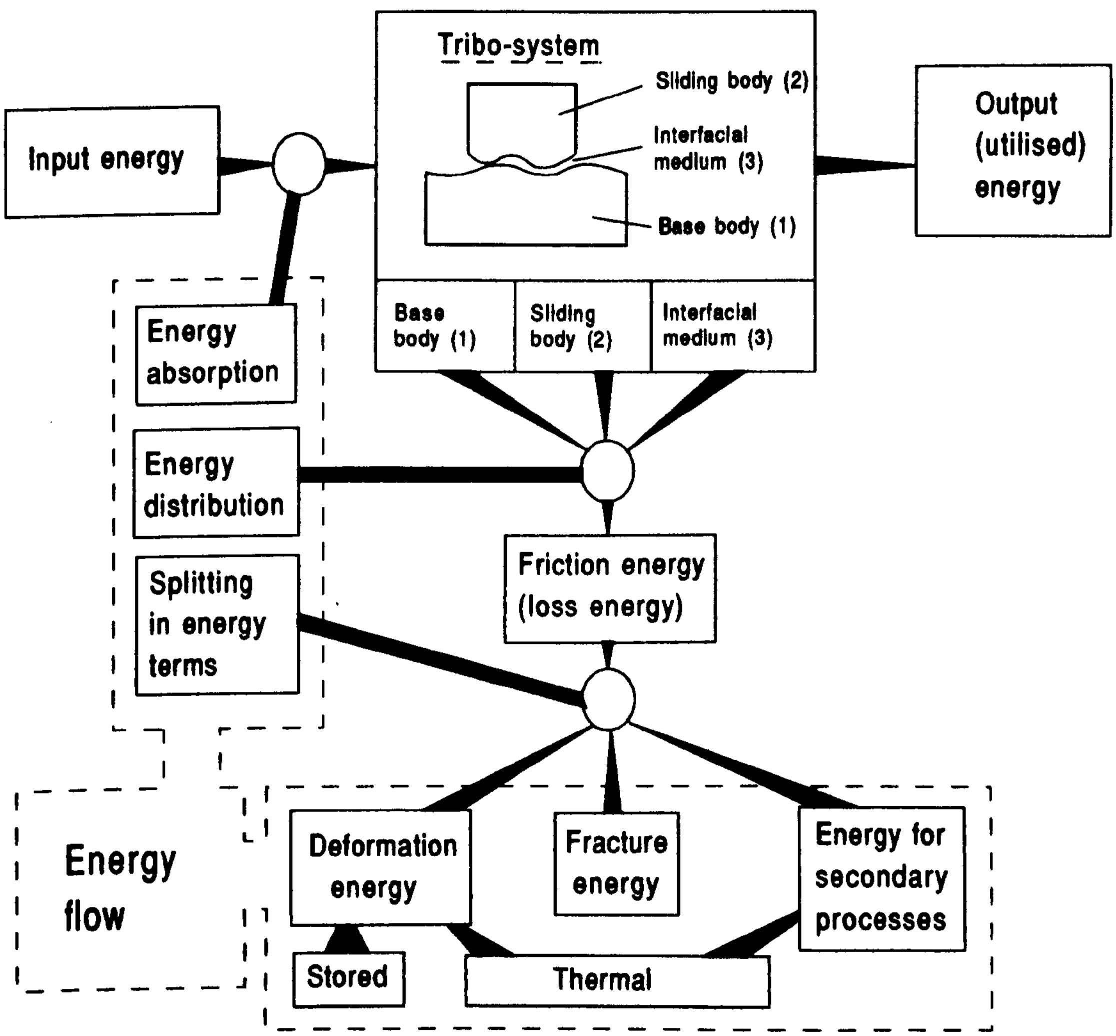


Fig. 1.7 Energy transformation during tribological interactions (adapted from ref. 1.44)

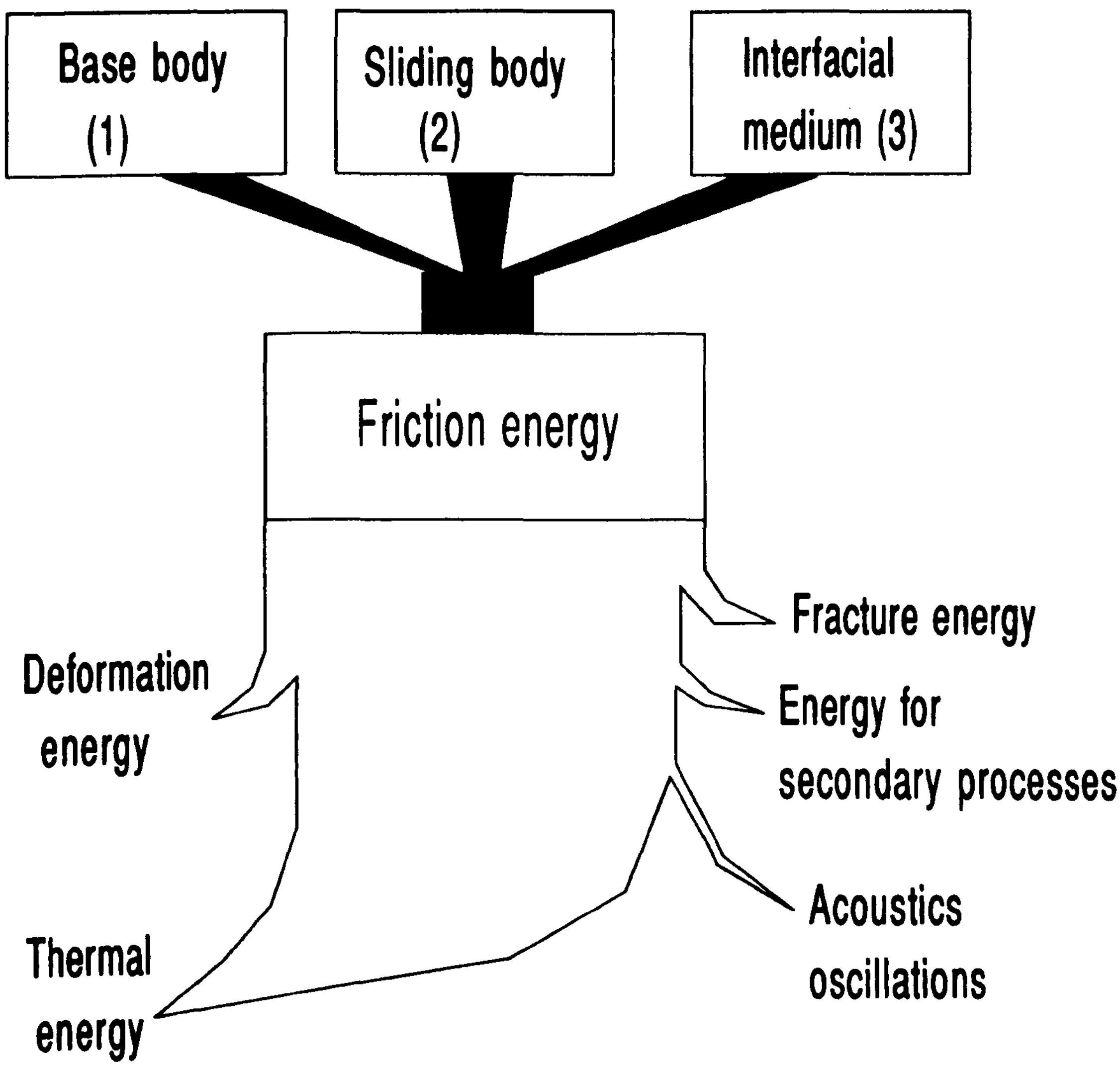


Fig. 1.8 Energy distribution among physical & chemical processes (adapted from ref. 1.44)

related process. In the energy-based friction model used by Heilmann and Rigney [1.47] to develop expressions for the friction coefficient, the main assumption was that the friction work performed was equal to the work of plastic deformation during steady state sliding and that the work was dissipated as heat. In his discussion on adhesive wear, Rabinowicz [1.30] described his condition for loose particle formation as :

$$d = 60\,000\,\frac{W_{ab}}{h^1} \tag{1.1}$$

where d = diameter of loose wear particles (cm)

h^1 = hardness (kg/mm^2)

W_{ab} = energy of adhesion (surface energy) (erg/cm^2)

Since this implies that the diameter of loose particles depended on energy of adhesion, it followed that generation of loose particles can only come about as a result of energy expended, ie wear particles are generated for every unit of energy expended. The link between wear and energy is further implied by Zhang [1.48] who found that the wear rates of polymers are increased with increasing normal load. Similarly, Puchugen [1.49] found the wear resistance of the rubber seal in the piston of a drilling pump was inversely proportional to the load per unit area. The concept of specific energy for wear was used by Jahanmir [1.50] who considered a specific amount of energy expended through the interaction of surfaces in any wear situation. He proposed that the worn volume V_w can by found by the equation :

$$V_w = \frac{Energy\ input}{Wear\ energy\ /\ unit\ volume} \tag{1.2}$$

where the energy input is the total energy expended in the surface interaction. For adhesion, delamination or abrasion this energy is equivalent to the external work done, ie (Tangential force) x (Sliding distance). He defined the specific wear energy as the energy expended in the removal of one unit of volume of material. The inverse of the specific of the specific wear energy is the specific wear rate, expressed in units of mm^3/J.

Bond's 'Third Theory' [1.51] related that the energy input required to crush or grind ore is directly proportional to the square root of a new surface area produced. His fundamental equation :

$$W = \frac{10Wi}{\sqrt{P}} - \frac{10Wi}{\sqrt{F}} \tag{1.3}$$

indicated the energy which would be required (in kWhr per short ton) to crush or grind ore from a given feed size F' to the desired product size P. Wi is the work index or kWhr required to reduce a ton from theoretically infinite size to 80% passing 100 μm. The work or energy input W is the net energy input to the crusher or mill. Norman [1.52] reported that abrasive wear in a specific crushing or grinding operation is in direct proportional to energy consumed. Hence a ball mill operation changed to produce a finer product would increase the kWhr per ton as predicted by the Bond equation. Bond correlated wear and energy consumption from a large number of crushing and grinding operations, and was able to derive an abrasion index in terms of pounds or grams per kWhr.

Czichos [1.32] introduced the concept of generalised energy balance where a system's net power equals zero if all important processes of storage and transformation of energy are summarised, ie $\Delta\acute{E} = 0$. A simplified description would be that the function of tribo-mechanical systems consists in converting the inputs eg motion, mechanical energy and materials, into outputs which are used. The functional cause-and-effect relations between inputs and outputs are accompanied by loss-outputs of mechanical energy and materials, denoted by friction and wear losses. It followed that 'input work must be equal to the use-output work + loss-output energy + energy stored in the system + energy transformed to other forms'. In other words a power balance, hence :

$$\Sigma\acute{E}^{w}_{x} = \Sigma\acute{E}^{w}_{y} + \Sigma\acute{E}_{z} + \Sigma\Delta\acute{E}_{s} + \Sigma\acute{E}^{wt} \qquad (1.4)$$

where $\acute{E}^{w}_{x}$ = input power

$\acute{E}^{w}_{y}$ = use-output power

$\acute{E}_{z}$ = loss-output energy rate

$\Delta\acute{E}_{s}$ = stored energy

$\acute{E}^{wt}$ = thermal energy transformed from mechanical work

Fig 1.9 shows a schematic view of a tribo-mechanical system with inputs and outputs. Loss-output energy may be translated from mechanical energy to tribo-processes of plastic deformation, ploughing, abrasion and adhesion, leading to material wear loss. This concept of generalised energy balance and transformation of energy through various forms further strengthened the notion that energy and wear are inter-related.

In the category of experimental methods, Kragelskii [1.29] divided the kinematic designs of all machines for wear investigations on small specimens into two types : (1) with

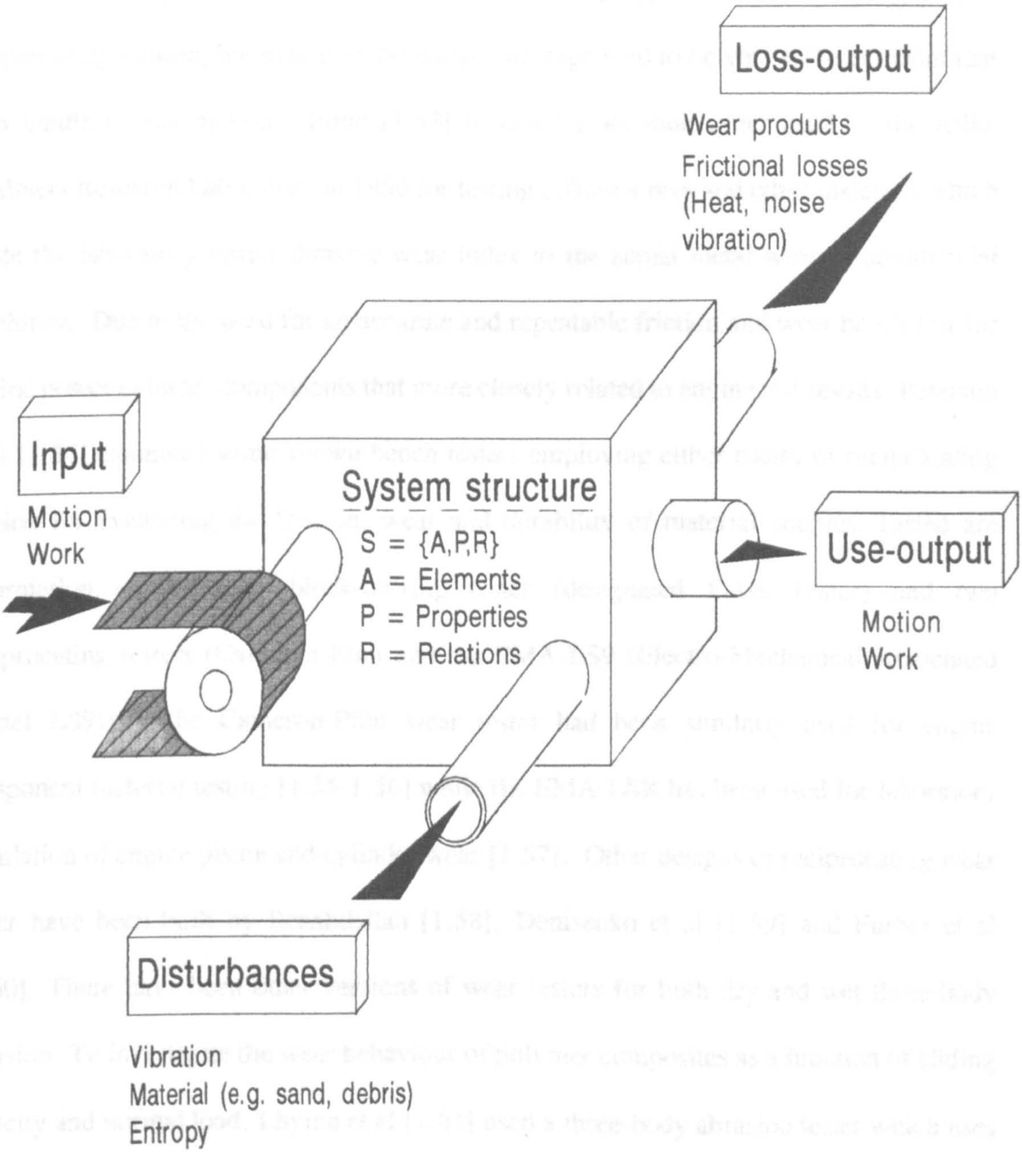


Fig. 1.9 Schematic view of a tribo-mechanical system with inputs and outputs (adapted from ref. 1.53)

unidirectional motion, (2) with reciprocating motion. He listed eight different types of machines into eight sub-divisions due of the differing types of surface damage. For reciprocating motion, the nature of the surface damage tend to be very different from that with unidirectional motion. Bond [1.53] reported a abrasion tester used by the Allis-Chalmers Research Laboratory in 1956 for testing different ores and other materials which relate the laboratory tested abrasive wear index to the actual metal wear in commercial machines. Due to the need for an accurate and repeatable friction and wear bench test for engine power cylinder components that more closely related to engine test results, Paterson et al [1.54] examined some known bench testers employing either rotary or reciprocating motion for evaluating the friction, wear and durability of material couples. Listed are information on a rotary block-on-ring tester (designated False Tester) and two reciprocating testers (Cameron-Plint and the EMA-LS9 (Electro-Mechanical Associated Model LS9)). The Cameron-Plint wear tester had been similarly used for engine component material testing [1.55-1.56] while the EMA-LSR has been used for laboratory simulation of engine piston and cylinder wear [1.57]. Other designs of reciprocating wear tester have been built by Benabdallah [1.58], Denisenko et al [1.59] and Furber et al [1.60]. There have been other versions of wear testers for both dry and wet three-body abrasion. To investigate the wear behaviour of polymer composites as a function of sliding velocity and normal load, Lhymn et al [1.61] used a three-body abrasion tester which uses a rotating rubber-lined wheel and sand particles as abrasives, expressing the specific wear rate as the volumetric wear per unit of frictional work. Fang et al [1.62] used a reciprocating wear tester and a pin-disc type tester for short-travel and long-travel three-body abrasion respectively. The wear tests were done on metallic specimens with sand particles as abrasives. The three-body abrasions tests so far outlined were conducted in dry sliding conditions. To determine the wear characteristics and relative effect of material properties in the abrasive wear resistance of O-rings, Burr and Marshek [1.63] used a special pin-disc-type testing machine which allowed both the specimen and the wear cylinder to be immersed in an abrasive mud of the type used for oil and gas well drilling. To investigate the effects of abrasive concentration on wear, Mayville [1.64] used a reciprocating wear configuration in which the ring specimen was stationary and the liner specimen moved with a reciprocating motion. A liquid-solid particle slurry was used as the abrasive. Another example of three-body abrasion experimental wear test machine was the pin-on-disc configuration used by Zhang [1.65] to investigate wet abrasion or hydroabrasive wear of polymers. This machine had a rotating steel disc with the test specimens running in a container of abrasive mud.

1.4 Outcome of literature survey

This survey has revealed that pump downtime is a routine of mud pump operation but there is a necessity to be able to predict the failure of fluid end components in order to reduce downtime frequency and minimise the safety risks during drilling. There are three types of maintenance in common use which include corrective, preventive and predictive measures. Though condition-based maintenance and condition monitoring are the more preferred techniques in modern times, they do not appeared to be very suitable for reciprocating pumps which operate on a variable-speed, variable-load basis. Customised systems developed by drilling companies offered the facilities of real-time monitoring, event detection and alarms, to spot early trends and prevent further equipment damage. But they lack the ability to forecast possible component failure. A maintenance strategy based on the energy expended appears to have practical implications for reciprocating pumps which do not have standard operating conditions. The differing oil well hydrostatic pressures encountered during offshore drilling would require varying mud pressures which are met by changing the mud pump reciprocating speeds. This would have an affect on condition monitoring, which are best suited for constant loading conditions, but would not affect an energy-based maintenance strategy since the energy consumption depend on the pressures and the cumulative strokes of reciprocation, irrespective of the running speed.

Although much work had been done by researchers on the field of tribology, two- and three-body wear, there appears to be a gap of knowledge in the category of wet abrasion or hydroabrasive wear, particularly for elastomeric wear on steel. There is also a need to research into reciprocating wet abrasion with abrasive mud as the interfacial medium. This is the case since a survey of mud pump operators have concluded that the sand content in the mud is one of the factors which have a direct effect on the wear of the piston rubber and liner. Most tribological studies tend to focus on the investigation of fundamental tribological processes, without relating the results to wider systems envelope which encompass the whole tribological system. A systems approach would allow a systems envelope to identify and resolve the different levels of complexity, where different operating variables and parameters can be taken into consideration. Different types of testing of tribological subjects can also be distinguished, ranging from field tests to fundamental laboratory friction and wear test and surface investigations. Applying the methodology of a systems approach to the development of an energy-based maintenance scheduling strategy is deemed appropriate in the context of mud pump fluid end components of which the wear processes would have to consider the effects of parameters like sand or heat. Since the piston rubber-liner material combination exhibit the highest change-out frequency compared to other fluid end components, it seems logical to focus the tribological studies on this material couple. Besides proving the viability of the energy-based maintenance concept, there is also the need to develop an energy processing unit (EPU) which would incorporate experimental and field data with a sand correction factor, to account for the abrasive effects of sand, to monitor the cumulative energy expended and hence provide a means to predict possible component failure.

After considering all the different designs and functions of wear testers, there appeared a need to design and build a customised reciprocating wear tester which, besides achieving reciprocating motion of the material specimens in the presence of drilling mud, must also facilitate repeatability of experiments in controlled conditions and have the facilities to measure the volumetric wear loss in addition to the energy expended.

Chapter 2

SYSTEMS APPROACH TO RECIPROCATING PUMP WEAR

2.1 Reciprocating pumps

The majority of mud pumps used for offshore drilling are reciprocating triplex single acting pumps for several reasons, among which are; ease of accessibility for maintenance, simplicity of design and higher pump pressures and speeds. It is inevitable that the design and operational nature of reciprocating pumps would present a unique host of problems and characteristics which include the following :

1) The 'overrunning' rotation of the crankshaft is such that the crank approaches the crosshead from the top of the rotation circle (Fig 2.1), causing the connecting rod force component to be directed downward together with the crosshead weight component. This result in wear of the underside of the crosshead and the sliding surface of the lower crosshead guide. Although more rigidity can be built into the lower crosshead guide to resist the normally high crosshead forces, more wear can be expected on the lower guide as compared with the upper guide. This mode of rotation, however, minimises crosshead slap or knock which tend to be more significant if the pump is run in the opposite ie 'underrunning' direction. When the wear in the lower crosshead guide becomes significant, mis-alignment may result,

causing the piston to dig into the liner resulting in piston and liner failure due to rubbing of the metallic flange of the piston body against the steel liner. This condition can cause sudden catastrophic failure.

2) During pumping, the elastomeric material of piston seals (called 'swab' in the drilling industry) tend to be forced into the clearance gap between the piston and liner, resulting in the original shape of the swab being deformed drastically. During the reverse stroke of the piston, a lowering of system pressure causes the clearance-gap seal to be momentarily broken due to the action of the reduced friction and the relaxation of the deformed swab when returning back to its original shape (Fig 2.2). This action causes the slight leakage of fluid, or mud, trapped around the swab. Any debris and sand trapped together with the mud would have a devastating three-body abrasive effect on the liner and swab when the next stroke reversal causes another pressure change. Miller [1.7] described this tendency of swabs leaking slightly during stroke reversals on reciprocating pumps as 'Jacoby leakage'.

3) During oil-well drilling, the drilling fluid that constitute mud tends to include the drill cuttings and suspended particles from the well. Though these particles are passed through mud conditioning equipment and 'desanded' before recirculated through the mud pump, there is the possibility of a small amount of sand or solids concentration remaining in the mud. In this respect, different drilling companies have different standards imposed on the 'limit' of sand percentage permitted. These can range from zero to 3 percent. An earlier survey [Appendix 4] concluded

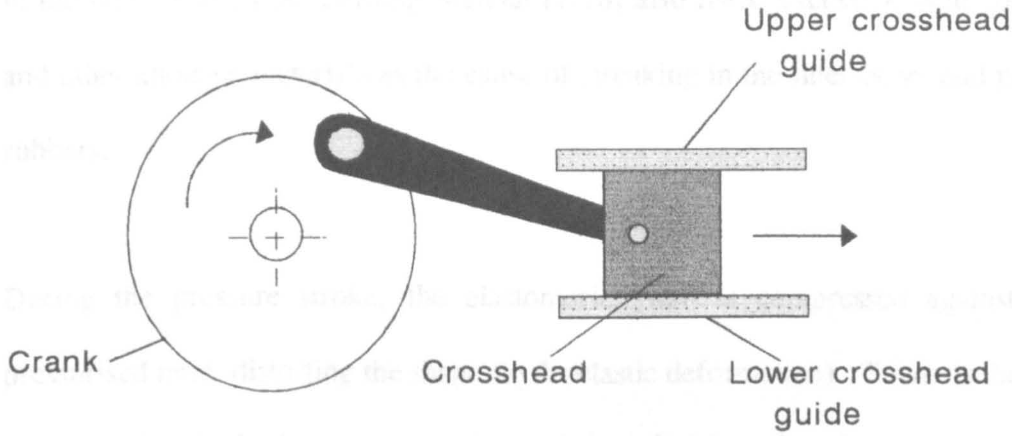


Fig 2.1 "Overrunning" rotation of crankshaft

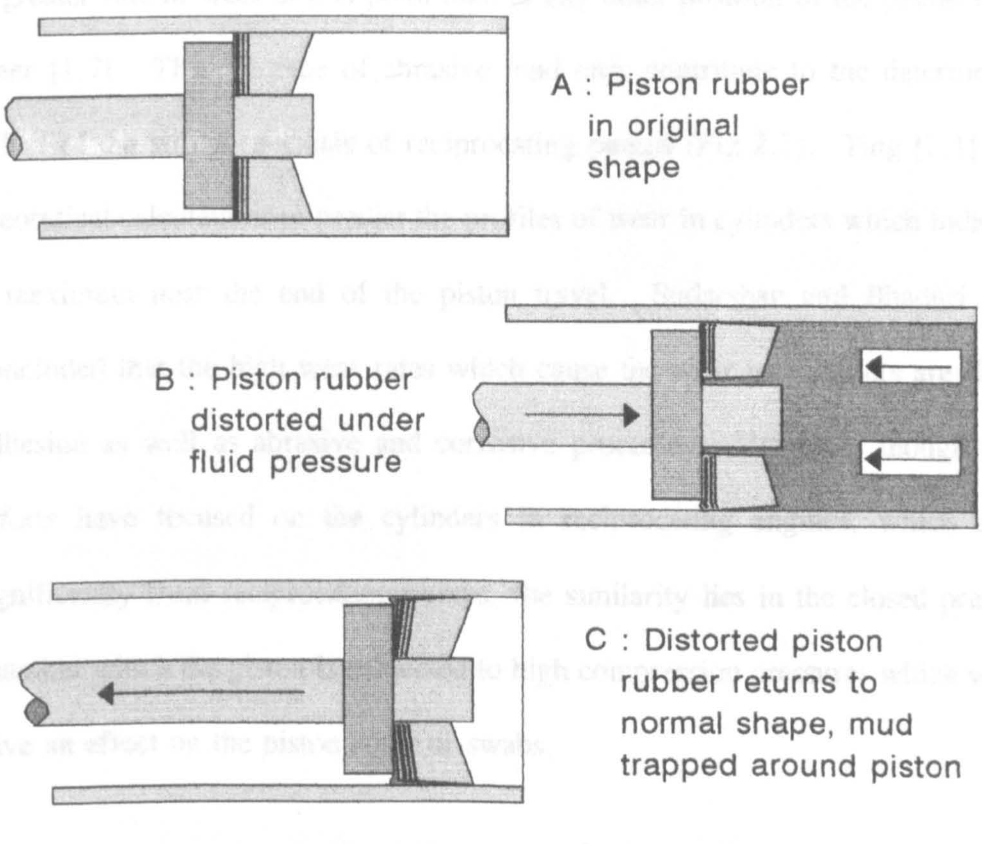


Fig 2.2 Jacoby leakage effect in reciprocating piston (Ref 1.7)

that one of the main reasons for swab failures is the abrasive sand content present in the mud. The IADC Drilling Manual [1.18] also listed excessive sand content and other abrasive materials as the cause of streaking in the liner bores and piston rubbers.

4) During the pressure stroke, the elastomeric swab is compressed against the pressurised mud, distorting the shape (hydroelastic deformation). Towards the end of the stroke, the high pressure on the swab is suddenly reduced to comparatively low pressure. The effect of the swab suddenly changing back to its original shape together with the abrasive sands trapped between the swab and liner wall generates a greater rate of wear at this point than at any other position of the piston in the liner [1.7]. The presence of abrasive mud only contribute to the deteriorating effect of the stroke reversals of reciprocating pumps (Fig 2.3). Ting [2.1] used theoretical calculations to predict the profiles of wear in cylinders which indicated a maximum near the end of the piston travel. Sudarshan and Bhaduri [2.2] concluded that the high wear rates which cause the wear in cylinders are due to adhesion as well as abrasive and corrosive processes. However, though these efforts have focused on the cylinders in reciprocating engines, which differ significantly from reciprocating pumps, the similarity lies in the closed pressure chamber which the piston is subjected to high compression pressures which would have an effect on the piston rings or swabs.

5) It is common knowledge that the piston-liner clearance has a direct effect on the life of the piston swabs. A larger gap would facilitate the extrusion of the

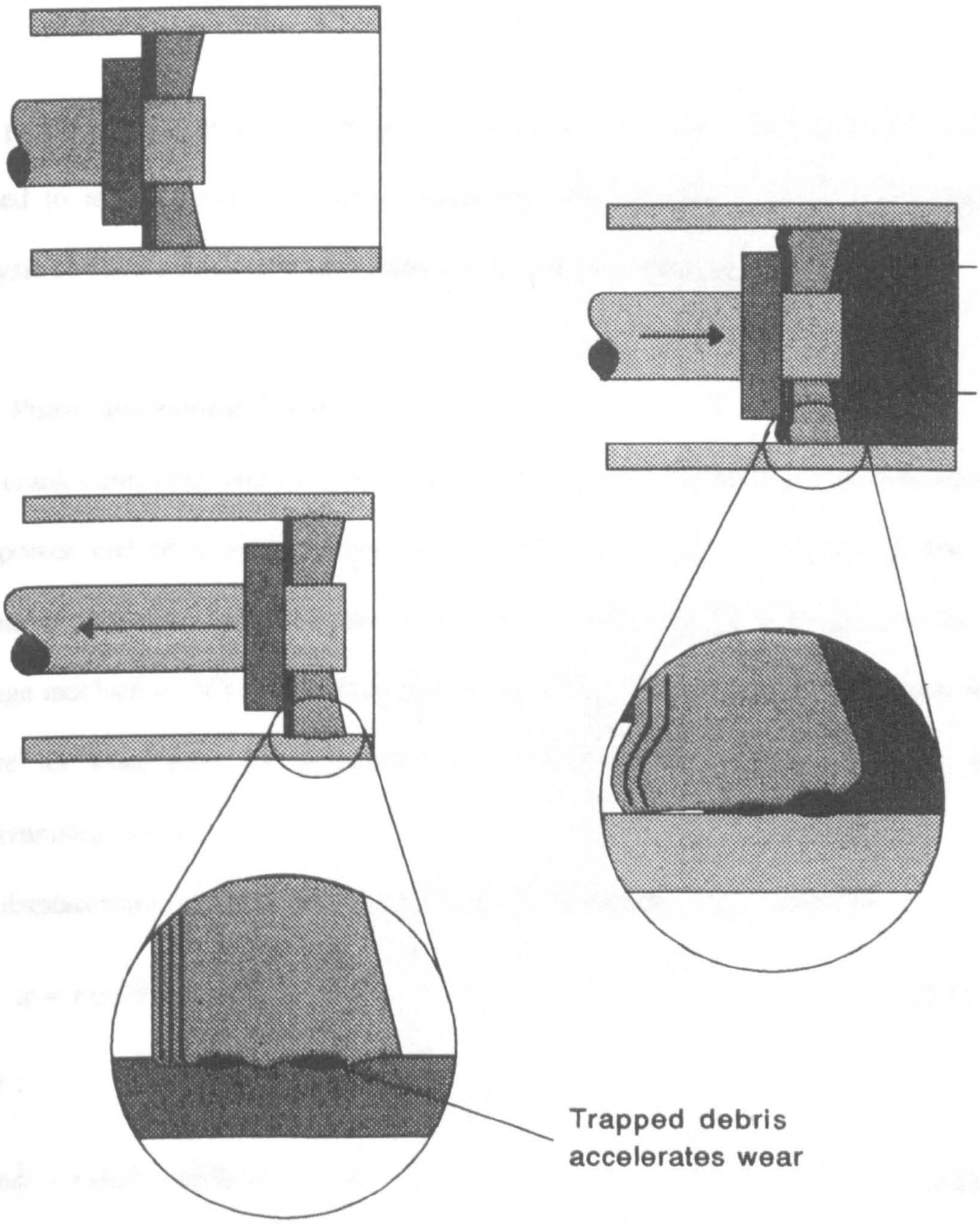


Fig 2.3 Effect of stroke reversals on wear (Ref 1.7)

elastomeric piston material due to the high pressures of the drilling mud. The higher the mud pressures, the lower the piston-liner gap tolerances needed to ensure the same wear life of the piston rubber.

The problems encountered by the mud pumps are varied and may be caused by factors related to maintenance procedures, operational and drilling conditions. A fault-tree analysis of failure modes of mud pumps is detailed in Appendix 6.

2.2 Pump mechanism forces

The crank-connecting rod-crosshead configuration forms the fundamental mechanism of the power end of a reciprocating pump. Fig 2.4 shows a schematic of the pump geometry. A mathematical model can be used to predict the forces involved in the crank linkage mechanism of reciprocating pumps. Fig 2.5a shows the pump mechanism forces, where the crankshaft, O, is rotating at constant angular velocity, N rad/s, in the 'overrunning' sense.

The displacement, x, of the piston, P, from the crankshaft, O, is such that :

$$x = r.\cos\theta + l.\cos\phi \tag{2.1}$$

since :

$$l.\sin\acute{\phi} = r.\sin\theta \; ; \; while \; \phi = \pi - \acute{\phi} \tag{2.2}$$

then

$$\acute{\phi} = \sin^{-1}(r\,/l.\sin\theta) \tag{2.3}$$

$$\acute{\phi} = \sin^{-1}(c.\sin\theta) \; , \; where \; c = r\,/\,l \tag{2.4}$$

Gorman and Kennedy [2.3] has shown that $\Theta = \Omega.t$ and

$$A1 = 1 \qquad (2.5)$$

$$A2 = c + 1/4.c^3 + 15/128.c^5 + \text{higher odd powers} \qquad (2.6)$$

$$A4 = -1/4.c^3 - 3/16.c^5 + \text{higher odd powers} \qquad (2.7)$$

$$A6 = 9/128.c^5 + \text{higher odd powers} \qquad (2.8)$$

Since in practical mechanism configurations, the ratio $c \approx 1/3$, all terms beyond c^5 are truncated off.

Algebraic manipulation would give

$$x = r\,(A1.\cos\theta + A2.\cos 2\theta + A4.\cos 4\theta + A6.\cos 6\theta) \qquad (2.9)$$

Considering a free body diagram of the piston (Fig 2.5b) where F_l = con-rod load, M = mass of piston, Nr = reaction load, Fp = force due to discharge pressure, the acceleration, a, of the piston is given by :

$$a = -\Omega^2.r\,(A1.\cos\theta + A2.\cos 2\theta + A4.\cos 4\theta + A6.\cos 6\theta) \qquad (2.10)$$

the linear velocity, v, is given by

$$v = -\,\Omega.r\,(A1.\sin\theta + A2.\sin 2\theta + A4.\sin 4\theta + A6.\sin 6\theta) \qquad (2.11)$$

Considering horizontal forces, where all vectors pointing to the right (→) are positive,

$$F_l\,.\cos\phi' - Fp = M.a \qquad (2.12)$$

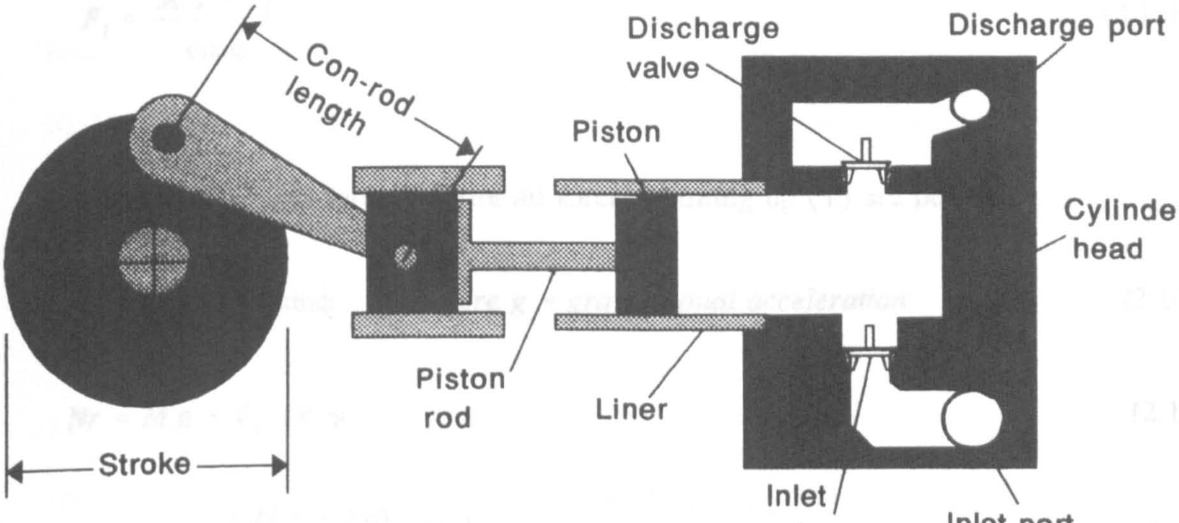


Fig 2.4 Pump geometry

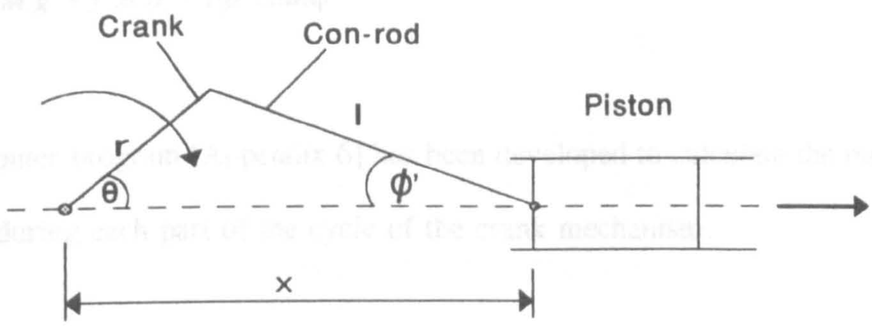


Fig 2.5a Pump mechanism forces

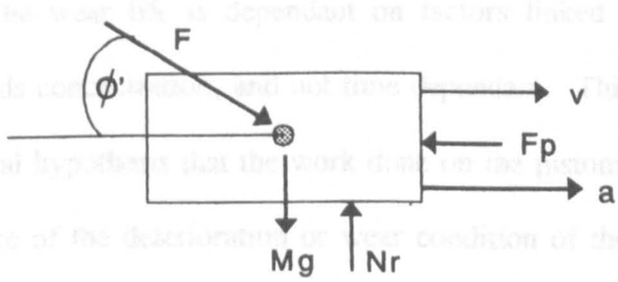


Fig 2.5b Free body diagram

$$F_l = \frac{M.a + Fp}{\cos\acute{\phi}} \tag{2.13}$$

Considering vertical forces, where all forces pointing up (↑) are positive,

$$Nr - M.g - F_l.\sin\acute{\phi} = 0 \text{ , where } g = \text{gravitational acceleration} \tag{2.14}$$

$$Nr = M.g + F_l.\sin\acute{\phi} \tag{2.15}$$

$$Nr = M.g + \frac{(M.a + Fp)}{(\cos\acute{\phi})} . \sin\acute{\phi} \tag{2.16}$$

$$Nr = M.g + (M.a + Fp).\tan\acute{\phi} \tag{2.17}$$

A computer program [Appendix 6] has been developed to calculate the pump mechanism forces during each part of the cycle of the crank mechanism.

From the discussion on the operational characteristics of mud pumps, it would appear that the effect of the pump stroke reversals on the wear life of the pistons and liners point to the fact that the wear life is dependant on factors linked to the cumulative strokes, pressures, solids concentration, and not time dependant. This seems logical and in-tune with the general hypothesis that the work done on the pistons and liners provide a more accurate picture of the deterioration or wear condition of the fluid end components. In contrast, a typical week of pump concentration may witness seven different pumping speeds as demanded by the drilling conditions. Tracking the amount of wear on the basis of the hours run by the pump in this instance would not be accurate, if not impossible.

The analysis of the pump mechanism forces indicate that the discharge pressure is directly related to the angular velocity of the crank mechanism and the rate of stroke reversals. And since the phenomena of stroke reversal induces liner wear through debris trapped by the swab when it returns to its original shape after the pressure stroke, it follows that the discharge pressure has an effect on liner and piston wear.

2.3 Concept of Systems Approach

The definition of tribology implied there are three main entities which are interrelated : friction, wear and lubrication. The resultant friction when two bodies contact in relative motion give rise to tribological processes which result in material loss or wear of the contacting bodies. The presence or absence of lubrication serve to reduce or increase the frictional forces and hence the wear of the bodies. In most, if not all, practical situations, the contacting bodies do not operate in isolation. There are usually other factors or variables which have an influence on the interaction of two bodies. Most materials which contact in relative motion would operate in the presence of lubricants, as exemplified by bearings or bushings. Then there are other variables which come into play such the debris or loose wear particles and the environment, which can contribute dust, vibration etc. A study of the tribological processes between any two materials without considering these 'external' factors would not be complete. One means of addressing this issue would be to use a systems analysis approach.

The concept of systems approach to tribology is not new and has been suggested by other researchers including Ku [2.4], Salomon [2.5] and Czichos [1.32-1.34, 2.6]. Loosely defined, a tribological system or 'tribo-system' is taken to encompass sets of elements in a system which are interconnected by the structure and function of the system. The main characteristics of a system include :

1) Structure of a system, as defined by the elements (A); properties of the elements (P); and the relations between the elements (R). This is generally represented as **S** = **{ A, P, R }**.

2) If each system can be separated hypothetically by a systems envelope from its environment, the connections may be classified as :

   (a) inputs (x) to the system,

   (b) outputs (y) from the system.

3) The function of a system is to transform the inputs into outputs. The transformation (T) of the inputs into outputs may be described either mathematically, verbally or as a physical analog. The general description of a system is detailed in Fig 2.6.

2.3.1 Mechanical system

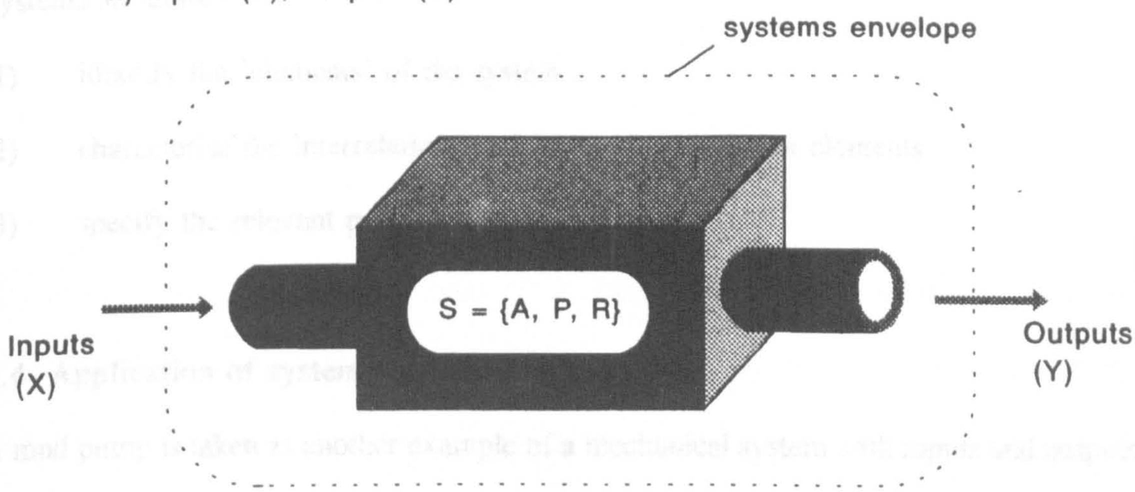
A mechanical system is designed to transform certain inputs into outputs for technical purposes ie the technical function of this mechanical system can be formally described as a transformation of the inputs into the outputs via a certain transfer function (T). The dynamic performance is accompanied by perturbations on the function and structure. Energy losses due to frictional resistances generally occur. The structure of the system changes relative to the action of tribological processes of friction and wear. Hence a

Definition : A system is a set of elements interconnected by structure and function

I) Structure $S = \{A, P, R\}$

a) Elements, A
b) Properties, P
c) Relations between elements, R

II) Inputs (X), Outputs (Y)



III) Function

Fig 2.6 General description of a system (Ref 1.32)

functional description of a mechanical system require to include a detailed study of the system and the influences of tribo-induced structural changes on the functional behaviour of the system. A detailed systems description would involve the following [1.32] :

Systems function -

(1) identify a systems envelope separated from its environment

(2) compile the inputs and outputs

(3) describe the functional input-output relations.

Systems structure -

(1) identify the 'elements' of the system

(2) characterise the interrelations and interactions between elements

(3) specify the relevant properties of the elements.

2.4 Application of system concept to mud pumps

A mud pump is taken as another example of a mechanical system with inputs and outputs. Though the dynamic piston swabs of mud pumps would invariably lead to regular maintenance, the unacceptably high frequency and unpredictability of pump downtime has prompted a closer look at the tribological processes occurring at the heart of the pump activity ie the piston-liner interface. But first, there is the need to identify the systems envelope.

2.4.1 Systems envelope

There are different levels of complexity of tribological processes. The different levels of tribo-testing approaches are shown schematically in Fig 2.7. In the first level, the system envelope is located broadly around the equipment in question ie a mud pump operating offshore under field conditions. The behaviour of the system tend to result from dynamic interactions of the pump (driven), engine or electric motor (driver), lithology of the well, the driller (the human factor) and the drilling conditions. At the next level, the system envelope would centre around the mud pump under bench tests. The third level may identify the fluid end as a sub-system. Next, the behaviour of basic components eg the piston-liner configuration may be studied. At the most fundamental level, the contact and the wear processes between two relatively moving (ie reciprocating) surfaces may be investigated on a suitable wear tester under controlled conditions in a laboratory environment.

2.4.2 Inputs, outputs

The technical function of mud pumps is the transmission of energy into mud with low suction pressure so that the resulting high pressure mud can perform a technical function such as providing hydraulic power to jet nozzles in drill bits. The operating variables which constitute the inputs may include pressure, pump speed, temperature, mud weights, particle concentrations. Determining which of these variables contribute significantly to the deterioration of the piston-liner material couple would be the object of specific wear tests as outlined in Chapter 3.

The outputs may include high pressure mud (use-output), frictional energy (eg heat, vibration, noise, power) and loose wear material, all of which come under the heading of 'loss- output'.

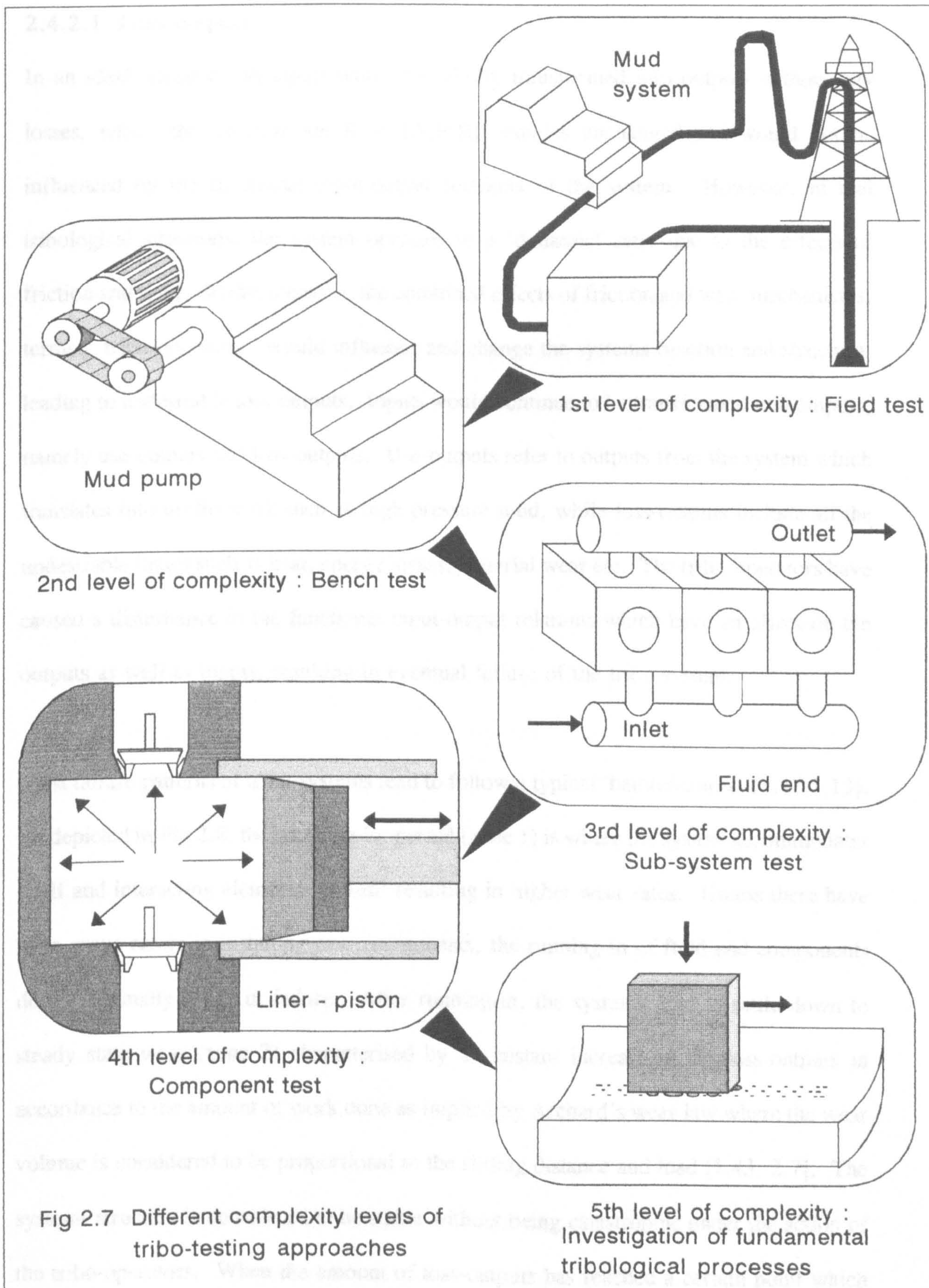


Fig 2.7 Different complexity levels of tribo-testing approaches

2.4.2.1 Loss-outputs

In an ideal situation, all inputs would be wholly transformed into outputs without any losses, where the structure set S = {A,P,R} remains unchanged and would not be influenced by the functional input-output relations of the system. However, in real tribological situations, the system operates in a 'dynamic' state due to the effects of friction and wear. In this scenario, the combined effects of friction and wear mechanisms, termed 'tribo-operators', would influence and change the systems function and structure, leading to undesirable loss-outputs. Inputs would continue to be transformed into outputs, namely use-outputs and loss-outputs. Use-outputs refer to outputs from the system which translates into useful work such as high pressure mud, while loss-outputs include all the undesirable losses such as heat, energy losses, material wear etc. The tribo-operators have caused a disturbance in the functional input-output relations which have an effect on the outputs as well as inputs, resulting in eventual failure of the tribo-system.

Most failure patterns of tribo-systems tend to follow a typical 'bathtub curve' [1.12,1.13]. As depicted in Fig 2.8, the 'running-in' period (zone 1) is where the system accommodates itself and interacting elements 'bed-in' resulting in higher wear rates. Unless there have been grave errors in installing new components, the running-in of fluid end components do not normally result in failure. After running-in, the systems tend to settle down to steady state wear (zone 2) characterised by a constant increase in the loss-outputs in accordance to the amount of work done as implied by Archard's wear law where the wear volume is considered to be proportional to the sliding distance and load [1.43, 2.7]. The systems structure is considered to be stable without being catastrophic under the action of the tribo-operators. When the amount of loss-outputs has reached a certain point which would cause a significant change to the systems structure (zone 3) the resultant self-accelerating process would lead to rapid material loss (or wear-out) and subsequent failure of the system. The three modes of material loss-outputs may be schematically represented as in Fig 2.9 [1.32]. The maximum level of material loss-output, $(Z_m)_{lim}$ in Fig 2.9, indicated the limit where the system structure has changed, due to the material loss-output, to the extent that the functional input-output relations are affected severely enough to result in catastrophic system failure. This applies to dynamic and dissipative systems where kinetic energy is transformed through irreversible processes into heat or other energy forms.

2.4.3 Function of Tribo-mechanical systems

If a systems envelope is drawn on the fluid end of a mud pump, its main function is one of power transmission where the inputs of motion and energy are transformed into work for hydraulic transmission. The energy sources may be electrical (electric motor) or chemical (diesel fuel). The output is high pressure mud to; lubricate and cool the drill bit, protect the walls of the oil well and transmit hydraulic horsepower to the drill bit (Appendix 1).

2.4.4 Structure of Tribo-mechanical systems

If a wear process occurring in a piston-liner material interface is enclosed by a system's envelope, the four basic elements which participate in the wear process are :

(1) piston swab - nitrile rubber (moving tribo-element)

(2) liner - chrome steel (stationary tribo-element)

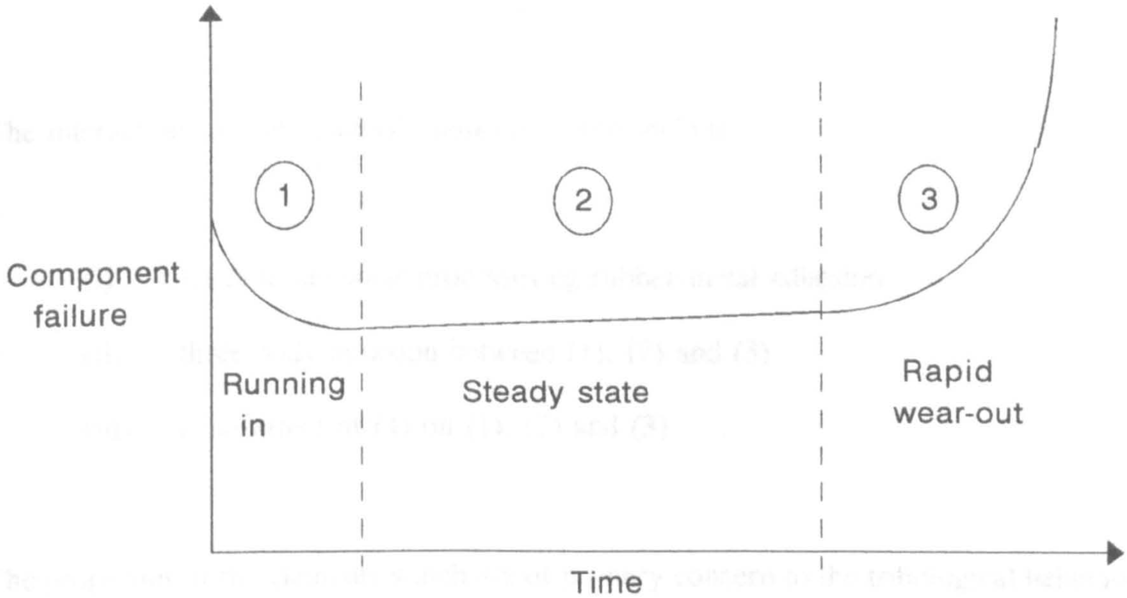


Fig 2.8 Classical failure pattern - Bathtub curve

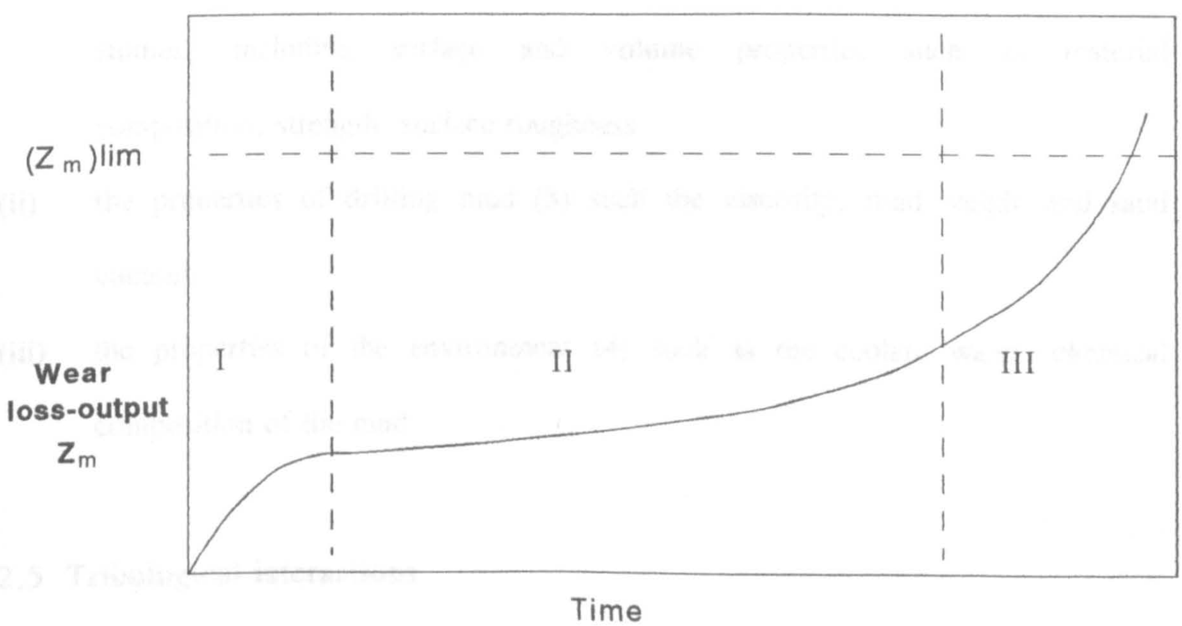


Fig 2.9 Material loss-output curves of a tribo-mechanical system (Ref 1.32)

(3) drilling mud and sand (interfacial medium)

(4) environment (air, water (coolant), chemical activity etc)

The interactions of the systems' elements would include :

(i) friction and wear processes eg rubber-metal adhesion

(ii) three-body abrasion between (1), (2) and (3)

(iii) tribo-effect of (4) on (1), (2) and (3)

The properties of the elements which are of primary concern to the tribological behaviour of the system include :

(i) the properties of piston rubber (1) and steel liner (2) relevant to tribological studies, including surface and volume properties such as material composition, strength, surface roughness.

(ii) the properties of drilling mud (3) such the viscosity, mud weight and sand content.

(iii) the properties of the environment (4) such as the coolant water, chemical composition of the mud.

2.5 Tribological interactions

If the terms 'inputs' and 'outputs' is taken to have implied the flow of quantities through a system, then the concept of quantity flows can be used to explain the different processes occurring within a system. Czichos [1.32] used three-dimensional conceptual planes to explain the transfer of inputs and outputs in a system. An adaptation of this concept has been used to explain the transfer processes occurring in the piston-liner component couple of mud pumps.

2.5.1 Frictional work plane

If work can be considered as a quantity flow, in the same context as a mass flow is taken as a flow of material quantity, then work is the result of the influence of one element upon another as the effects of motion and speed have on increasing the mud pressure during the pressure stroke of the piston. This work would have occurred during the transmission of mass (mud) through the system resulting in the generation of wear particles (mass transformation) from the wear processes. There is also the likelihood that the frictional work would have been transformed into heat (energy losses). Hence friction is said to have resulted in energy losses while wear processes have caused material losses. If these losses can be categorised, there may be three possible phases :

1) work is initiated into the contact zone (piston swab/inner liner wall), resulting in formation of contact areas,
2) work is transformed within the contact zone, resulting in plastic and elastic deformations of the swab, three-body abrasive and adhesive wear processes,
3) energy is dissipated via heat, friction, vibration, acoustics or stored in the system.

Considering an energy balance, the work input of a tribo-system must equal the useful work + loss-output energy + energy transformed to other forms (heat). The energy balance can be summarised in the equation (schematically in Fig 2.10) :

$$\Sigma(E^{W})_X = \Sigma(E^{W})_Y + \Sigma(E)_Z + \Sigma(E^{W})_T$$

where $\Sigma(E^{W})_X$ = input work

$\Sigma(E^{W})_Y$ = use-output work

$\Sigma(E)_Z$ = loss-output work/energy

$\Sigma(E^{W})_T$ = thermal loss due to mechanical work (friction).

2.6 Tribological processes

Not all of the energy supplied to a tribo-system are converted to work. The friction present in interfacial bodies implies that a certain amount of the total input energy is lost and dissipated through various forms eg heat or generation of loose wear particles. Briscoe [2.8] has shown that there is a correlation between friction and abrasive wear of polymers. He concluded that the greater the wear, the higher the friction. This implied the extent of wear is dependent on the applied load or pressure, given that the heat generated by the frictional work due to the load/pressure affects tribological interactions in wear processes which would result in wear particle generation. Two tribo-elements which contact in relative motion would result in the frictional work being dissipated through deformation of the softer elements and ultimately exit as heat from the system. In reciprocating tribo-systems, the repeated cyclic loading would also have an effect on the fatigue life of the material of the tribo-elements especially the piston swab. In their review on the sliding wear of polymers, Briscoe and Tabor [2.9] suggested that the energy input during sliding causes a certain degree of fatigue damage.

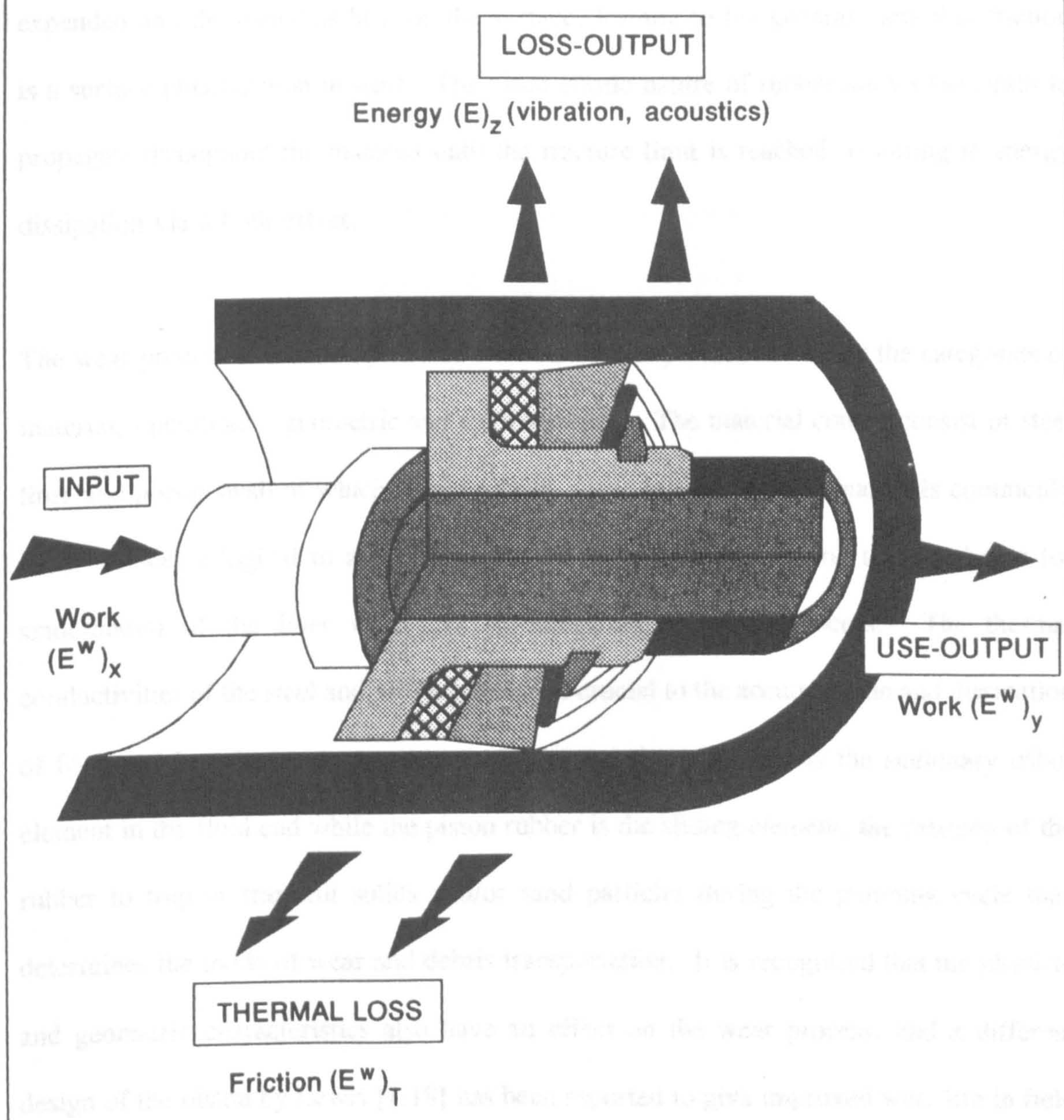


Fig 2.10 Work and energy transactions in piston-liner

The effect of surface forces on steel and rubber tend to result in different responses. The elastic-plastic nature of steel would first experience elastic deformation followed by surface plasticising when the yield limit had been reached. When this happens, energy is expended and dissipated as heat on the surface, leading to the general view that friction is a surface phenomenon in steel. The visco-elastic nature of rubber allows the strain to propagate throughout the material until the fracture limit is reached, resulting in energy dissipation via a bulk effect.

The wear process in a tribo-system is effected by many variables under the categories of material, operational, geometric and environmental. The material couple consist of steel liner and piston swab of which nitrile rubber is one of the synthetic materials commonly used. It seems logical to assume that the hardness of the steel and the roughness (or smoothness) of the liner walls has an affect on the wear process. The thermal conductivities of the steel and nitrile rubber are crucial to the accumulation and dissipation of frictional heat in the contact zone. Given that the steel liner is the stationary tribo-element in the fluid end while the piston rubber is the sliding element, the capacity of the rubber to trap or transmit solids and/or sand particles during the pumping cycle then determines the mode of wear and debris transportation. It is recognised that the physical and geometric characteristics also have an effect on the wear process, and a different design of the piston by Lewis [1.19] has been reported to give improved wear life in field service. However the continued use of the traditional piston design [1.7] has warranted a closer look at the operational variables which directly affect the wear process, such as solids content in the drilling mud, the sliding distance, total number of strokes, the pressure, piston velocity etc. The sliding velocity in combination with the pressure determines the frictional force which in turn affect the operating temperature and consequently the material strength and wear properties of the affected material ie rubber [2.10]. The operating temperature is affected by the generation of frictional heat, thermal conductivity of the steel countersurface, cooling system as well as the bulk properties of the rubber material. The other environmental variables such as ambient pressure and temperature, humidity, etc are beyond the scope of the pump operator just as the chemicals in the lubricant (ie drilling mud) are dependant on the drilling requirements and well conditions. Maintaining the solids concentration in the drilling mud to a minimum is a priority for the mud engineer as dictated by experience.

The wear process in a tribo-system involving a material couple with a rubber element can be divided into three phases [2.10] :

1) *Pre-detachment phase*. When exposed to wear processes, the material surface layer tend to be deformed and altered but may not always result in material loss. The dynamic load conditions cause gradual deterioration of the material properties due to chemical, thermal and mechanical exposure. Fatigue follows in the presence of cyclic intermittent loads due to the material being subjected to a wide range of strains and strain rates. The wear rate, as has been suggested by Ratner and Lure [2.11], depends on the applied load. Cracks have been shown to develop from naturally occurring flaws in the rubber [2.12-2.13]. Micro-tearing in the material are caused by voids which are formed near the surface when rubber is elongated and have been observed to increase with increasing strain [2.14]. Investigations have also shown there is a correlation between tearing energy and crack growth

[2.15-2.16].

2) *Detachment of surface material.* The detachment of material from a rubber surface during wear processes occur either by brittle fracture or smearing [2.10]. Brittle fracture occurs when microscopic strings of rubber are severely strained until tensile rupture. Micro-tearing and micro-flexing fall into this category. Smearing on the other hand are caused by the thermal, chemical and mechanical degradation of the elastomeric material, leading to a viscous layer on the rubber surface. Surfaces in severe wear conditions have been known to exhibit abrasion patterns described as Schallamach waves [2.17] or simply as ridges, ribs or bands [2.13, 2.18-2.20]. The continuous flexing and compression of the contact surface during sliding may have caused these abrasion patterns. The extent and appearances of these patterns depend on the sliding conditions and surface topographies and tend to occur perpendicular to the direction of motion, unlike in metals where the abrasion lines follow the direction of sliding. Flexing and deformation of the rubber surface generally precede detachment of wear particles which are caused by a combination of cutting, cracking, tearing, ploughing etc.

3) *Post-detachment phase.* The question as to where the wear particles end up after detachment depends on the sliding conditions, the properties of the liner countersurface and the piston rubber. The wear properties and rates of particle removal of tribo-systems are affected by the transportation mechanism of wear debris [2.18]. The detached material may be transported in three modes : (1) as loose debris; (2) transferred to the steel countersurface through particle

penetration and deposition, and smearing; and (3) back transfer to the rubber itself. This reinforces the phenomenon of Jacoby leakage and stroke reversal rates, as described in Section 2.0. The second and third modes result in the wear debris gouging and ploughing into the softer rubber surface as well as the chrome steel giving rise to 'streaking' in the liners as is common in field service.

2.6.1 Influence of tribo-processes on system structure

If the structure of a system does not change, then the system function will not be affected and would continue to perform as expected. Since the systems structure include elements and the interrelations between them, therefore any changes in the elements or its interrelations would obviously affect outputs of the system via its functions. Due to the effects of tribo-processes, the structures of tribo-systems would change during the functional performance of the system ie a tribo-system possesses systems structure and functions which are dynamic in nature. Changes in tribo-systems may include :

1) alteration of system elements, where the contact surfaces experience changes which give rise to friction-induced energy losses and wear-induced material losses, eg the generation of wear particles as third bodies which in turn can lead to three-body abrasion, gouging and ploughing.
2) changes in the relations between elements eg a transition in wear mechanism from abrasive to micro-tearing affects the wear rate of the piston rubber and the ultimate wear life of the component.
3) changes in the properties of the system elements eg the occurrence of streaking in the steel liner would result in grooves and surface damage which affects the

performance of the piston rubber. Other changes in the elements properties may include :

i) changes in surface topography. In tribo-systems where elements come into contact in relative motion, changes in surface topography are expected due to (a) plastic deformation and (b) wear processes [2.21-2.24].

ii) changes in surface composition. The presence of chemicals in drilling mud and abrasives serve to accelerate the deterioration of the surface through abrasion-corrosion wear eg if the chrome coating in a chrome-plated liner surface become worn off exposing the relatively soft base metal causing the piston rubber to be rapidly worn off by the sharp edges of the worn chrome plate.

iii) changes in surface strength. The effects of wear-induced heat generation have been found [2.25-2.26] to lead to the degradation of tensile strength and abrasion resistance, causing a significant change in the element properties of tribo-systems.

2.6.2 Influence of tribo-processes on system function

The function of the tribo-system, which comprise the inputs and outputs of the systems as well as the functional translation of the inputs into the outputs, are affected by the changes of the systems structure. As explained by Czichos [2.6], tribo-processes in a dynamic tribo-system give rise to 'disturbances' which affect the systems function and structure. Frictional processes affect the performance of the system function with resultant energy losses while wear-induced material losses are due to changes in the system structure. Fig 2.11 provides an overview of the effect of tribo-processes on the tribological system.

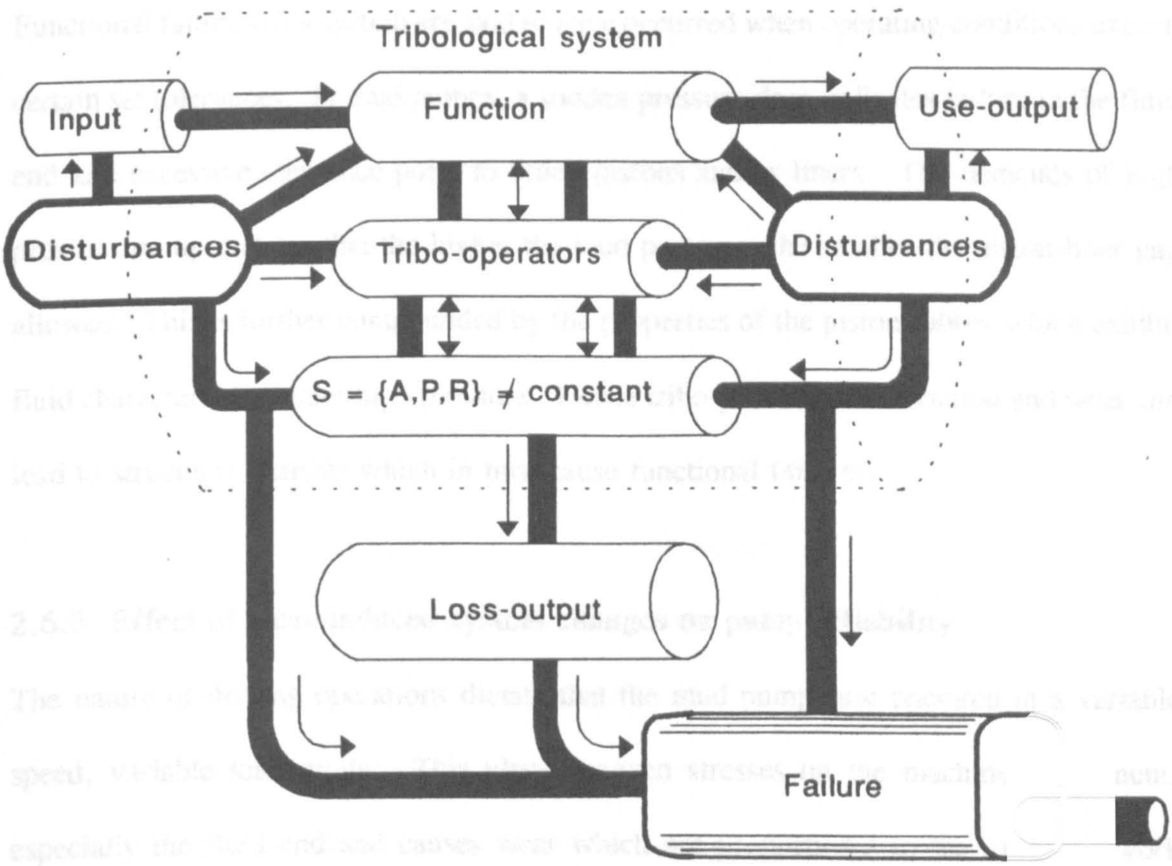


Fig 2.11 Overview of dynamic mechanical tribological system (Ref 2.6)

Frictional processes may give rise to disturbances such as vibration and 'stick-slip' motion which are intermittent jerks generated when asperities adhere and 'stick' during surface contact causing high friction which tend to zero when the continued external force forces the junctions to rupture and 'slip' [2.27-2.29].

Functional failures of a system are said to have occurred when operating conditions exceed certain set tolerances. In mud pumps, a sudden pressure drop indicates failure in the fluid end and excessive clearance point to failed pistons and/or liners. The demands of high pressure pump dictates that the higher the mud pressure, the smaller the piston-liner gap allowed. This is further compounded by the properties of the piston rubber which exhibit fluid characteristics under high pressure. Hence tribo-processes like friction and wear can lead to structural changes which in turn cause functional failures.

2.6.3 Effect of wear-induced system changes on pump reliability

The nature of drilling operations dictate that the mud pumps are operated in a variable speed, variable load mode. This places uneven stresses on the machine components especially the fluid end and causes wear which are proportional to the extent of work output of the pumps. The presence of abrasive sand in the mud lead to aggressive three-body wet abrasion with the sand, embedded in the piston rubber partly as a result of stroke reversals and partly as a result of the fluid nature of rubber under high pressure, causing substantial damage to the liner and piston from gouging, ploughing and 'nibbling' of the piston rubber. The liner surface would be scoured and start to deteriorate while the discharge pressure would fall if the piston rubber starts to develop any leaks due to excessive wear. The changes to the system structure caused by the wear hence affect the operational reliability of the pump. It seems logical to assume that the deterioration of the piston and liner are directly dependant on the amount of work transmitted from the tribo-mechanical system. This combined with the wear characteristics of the fluid end components leads to the difficulty of predicting the reliability of the mud pumps. Forecasting the wear life of the fluid end relative to the work transmitted from the pump system is then be more practical than correlating the inputs and outputs of the system. This is complicated by the unique characteristics of the drilling mud which, though lubricating the piston and liner, also contain highly abrasive solids which serve to enhance the wear processes. The surface contact between the liner and piston rubber is a combination of both dry and wet sliding contact.

2.7 Simulative tribo-testing

An investigation of the wear processes of a tribo-system may involve either the actual practical system or a test system in the laboratory. There are pros and cons in both approaches and attention is required in the testing of systems with an open system structure such as in drilling [2.30]. In particular the nature of drilling operations due to different rock formations and lithology mean the operating input to the tribo-systems in the fluid end are not constant. The solids concentration may change and correspondingly affect the wear rates of the piston rubber. Given these complications, tribo-test investigations would be more realistic if the type of motion and materials of the system elements are similar to those of practical systems. In the case of the piston and liner in the fluid end, this involves a test system which uses the actual liner and piston as used in offshore drilling mud pumps and involves reciprocation sliding in the presence of drilling mud with chemical composition and properties suitable for field applications. Suitable sand can be added to the mud to simulate solids concentration and drill cuttings. A suitable simulation criteria can then be selected which indicate a similarity of tribological processes in both practical and test systems, such as the appearances of worn surfaces. The operating variables in the test system can be adjusted until similar appearances of worn surfaces result. The results of such simulative testing can be used to determine the wear rates of the piston rubber under conditions deemed as similar to those of practical systems.

2.8 Discussion

The preceding sections have explained the use of a systems approach for a better understanding of the wear processes in tribo-systems and the procedures involved. It has been established that the operating inputs to the tribo-system in the fluid end of mud pumps are affected by the drilling conditions and the rock formations. The operating variables such as pressure and drill cuttings and solids concentration from the oil well have an effect on the wear life of the fluid end components in such a way which test systems cannot simulate. By selecting a suitable simulation criteria, test systems can establish the tribo-processes deemed to occur in a three-body wet abrasion sliding system and facilitate experimental results to determine the wear coefficient or particular wear rate which is defined as the ratio of the wear volume to the product of the load and distance ie the work done or energy expended. This wear coefficient can serve as a constant against which the work done in the fluid end can be used to predict the wear volume under similar operating conditions. The system envelope to achieve this involves the investigation of fundamental tribo-process at the fifth complexity level of tribo-testing approaches (refer to Fig 2.7). The wear tests are able to establish the effect of operating variables and determine the specific wear rates and the effect by variables such as sand content and temperature.

These experiments are detailed in the next chapter.

Another approach would involve analysing field data of mud pump fluid end maintenance records which are correlated with the sand content in the mud, considered a priority by most, if not all, drilling companies. These field data would reflect the true working conditions in offshore drilling environment. Analysis of these records may enable the investigator to pinpoint the stage where steady wear rate changes to rapid wear-out and under what operating conditions and what sand content. There is the potential to realise the objective of creating a new working unit which, using existing measurement parameters such as discharge pressure and cumulative strokes, can be used to assist in locating the transition point when steady wear turns into rapid wear. The ability to predict this 'failure point' would enable mud pump operators to forecast the failure of fluid end components such s piston rubbers and liners and plan drilling operations around such downtime events. The reliability of mud pumps and fluid end components in particular would then be a known entity. The analysis of these field data are detailed in Chapter Four.

Chapter 3

EXPERIMENTAL ANALYSIS

3.1 Introduction

This chapter details the wear test system and the experimental programme which has been used to identify and determine the operating variables which significantly affect wear. The system envelope selected for the test system corresponds to the fifth level of complexity of tribo-testing (see Fig 2.7) which involves the investigation of fundamental tribological processes of piston rubber sliding on steel. Given that the basic design of mud pumps has not changed significantly and the prevalence of fluid end components which are not significantly different from those used in the early days, the work here is focused on experimentation which provides a better understanding of the effect of operating variables on wear and generate information for the development of a planned maintenance strategy. In order to match the tribo-test investigations to be as realistic as to those of a field mud system, the type of motion and materials of the test system selected were similar to field operation, ie the actual steel liners and piston rubbers were used and involve reciprocation sliding in the presence of drilling fluid with the same properties and mud weight as used offshore. Sand was added to the mud to simulate the effect of drill cuttings and solids concentration. The simulation criteria selected would be the appearance of worn surfaces.

3.2 Experimental programme

The main thrust of experimentation included the determination of a dimensional wear coefficient (or specific wear rate) and investigating the effects of operating variables on the wear rate and wear processes. While it is recognised that there is not one single wear coefficient which is applicable for all the numerous operating conditions normally encountered in offshore drilling, it was envisaged that a wear coefficient adjusted with the relevant correction factors would be a good start for the adaptation of energy as the basis for monitoring wear loss. Attention was focused on the correlation of the energy expended and the volumetric wear loss. To this end, a special wear tester was designed and constructed, as detailed in Section 3.3, which allowed the measurement of the frictional force exerted per stroke and hence the energy expended for the duration of the test. Similarly, by monitoring the volumetric loss of the piston rubber sliding on the steel liner in the presence of drilling mud, the cumulative wear of rubber could be monitored and trended on a regular and periodic on-line basis while the tribo-test system was still running. From initial experiments rubber samples were examined by scanning electron micrography (SEM) and microphotography and compared with field samples of failed piston rubbers. This step was taken to validate the design of the experimental wear tester as achieving the simulation criteria chosen, which was the appearance of worn surfaces. Once this has been established, the remainder of the experimentation was focused on identifying the operating variables which significantly affect wear processes and determining the individual and combined effects of these variables on the wear coefficient or, alternatively, the specific wear energy. By running a matrix of tests under controlled operating conditions whereby other variables were kept constant while one variable such as sand content, was varied and monitored, it is possible to determine the effect of that operating variable on the wear of the piston rubber.

The experimental procedure was not simulated on an actual mud pump because the size and weight of the average mud pump used on offshore installations, weighing as much as 36000 kilograms and measuring 6 m by 2.9 m and 2 m in height, makes laboratory experimentation impossible. Also, given that there are usually only two mud pumps on most drilling installations, the competitive nature of the drilling business and criticality of the operational routine of the pumps would not permit any experiments to be performed nor any instrumentation to be installed which were deemed to affect the routine execution of the equipment. Besides the cost factor, requiring a capital outlay of more than £1.2 million [1.1] to install a mud pump system, a closed pressurised system would not facilitate the periodic and progressive measurement of the targeted piston rubber under experimentation.

3.3 Test equipment and procedures

In order to investigate the fundamental tribological processes of piston rubber sliding on steel, a dedicated wear tester was designed and built, shown in Fig 3.1(a) and (b). The wear tester was designed to have reciprocating motion which was provided by a modified Bran & Luebbe (B & L) metering pump (Fig 3.2). The sliding stroke of the machine was 50 mm. For the crank mechanism, the plunger shaft of the B & L pump was replaced with a rod which had been machined to accommodate a load cell, L (Fig 3.3), and connected to a cage, C (Fig 3.4), which was a custom built square-shaped lattice steel structure. The normal load, W (Fig 3.5), was also square-shaped to fit snugly into the cage and designed to move vertically within the cage. The close tolerances between the cage and normal load

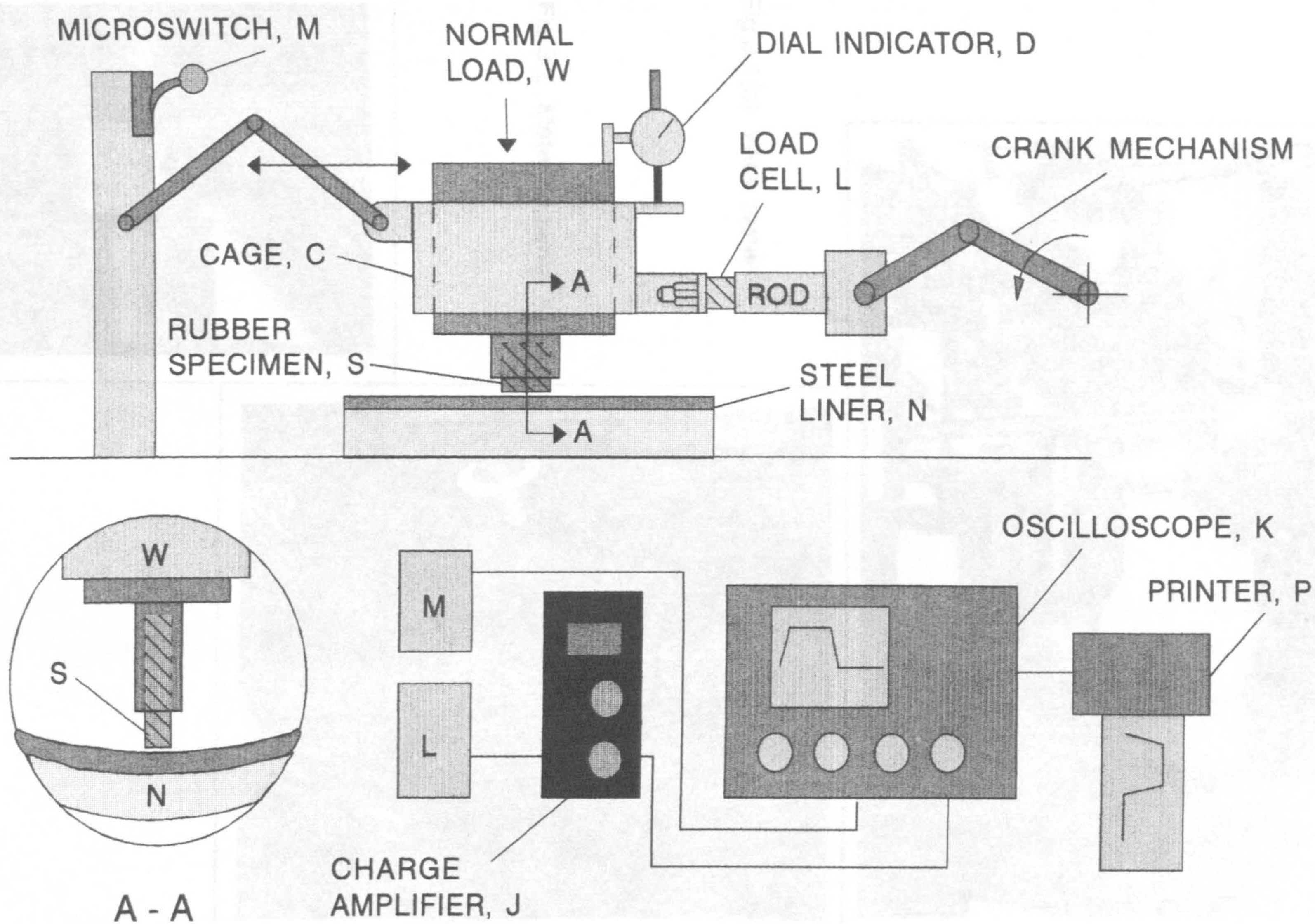


Fig 3.1(a) Schematic layout of wear tester

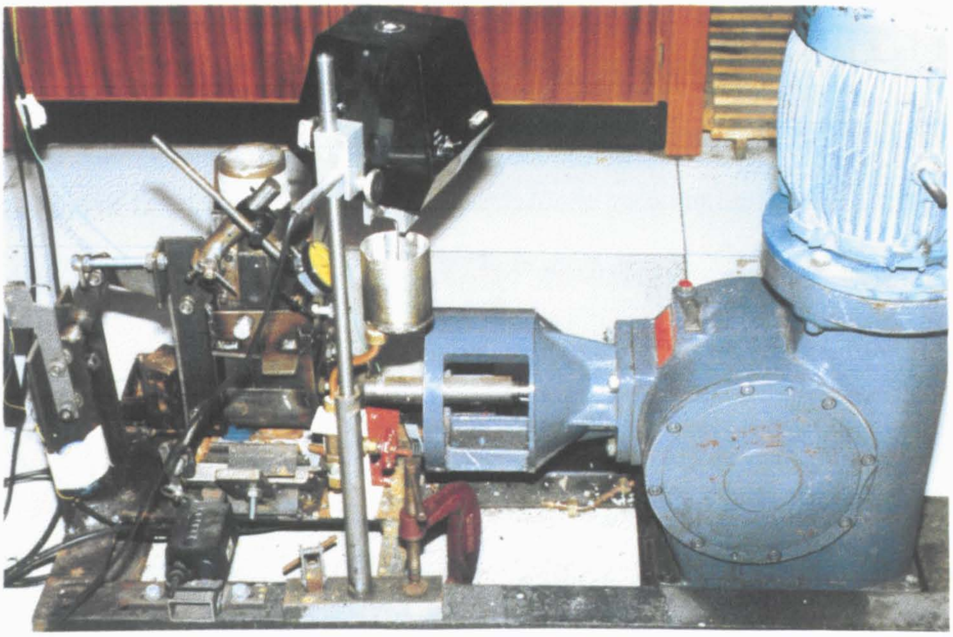


Fig 3.1(b) Wear tester

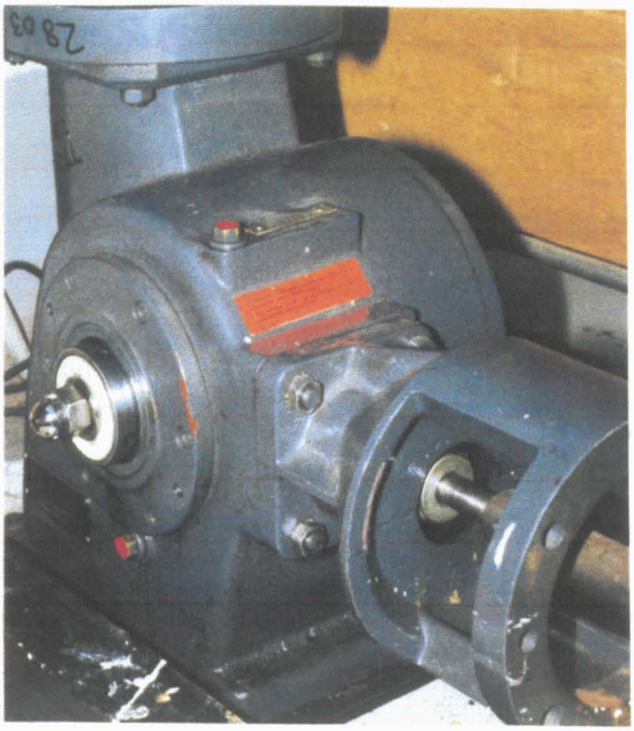


Fig 3.2 Metering pump

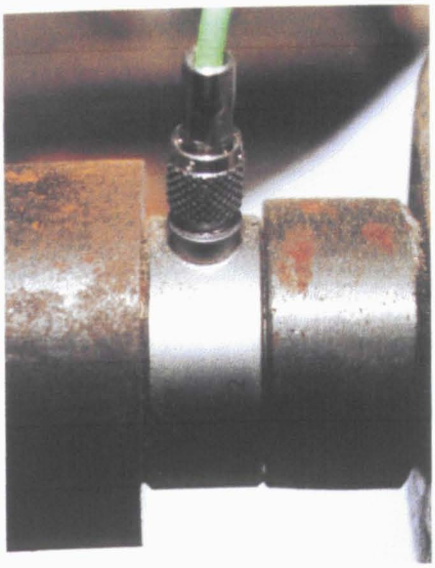


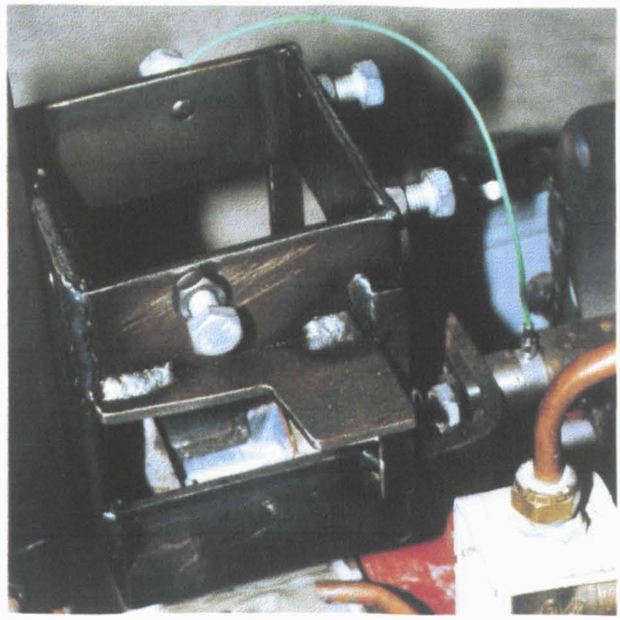
Fig 3.3 Load cell, L

ensure that the vertical travel of the latter was achieved with minimum or no horizontal wobble. This was essential to the dial indicator, D (Fig 3.6), which was specially mounted on the side of the normal load, W, and designed to provide on-line measurements of volumetric loss of the rubber specimen, S, undergoing testing. A micro-switch, M (Fig 3.7) was positioned to trigger at the end of the forward stroke, and provided the external trigger for an Gould Type 1425 oscilloscope, K (Fig 3.8), which captured and stored the force profile of the forward and return strokes. The load cell, (piezoelectric Kistler 9024A), preloaded when installed on the wear tester to enable both tensile and compressive forces to be measured, provided electrical signals, proportional to the mechanical sliding forces, which are sent to a Kistler charge amplifier Type 5007, J (Fig 3.9) to give a direct reading in NV^{-1} depending on the scale selected. These readings were sent to the oscilloscope where they were stored and digitised via a personal computer (PC) program for analysis (Appendix 7). These were then incorporated into a spreadsheet which calculated the frictional force, F_f, per angular turn of the crank. The frictional force, F_f, was obtained by deducting the (mass of the normal load, M x acceleration, A) from the horizontal force, F, measured via the load cell, ie $F_f = F - M.A$. The multiplication of the frictional force and the distance travelled during that angular turn of the crank gave the energy expended in Newton-metre (Nm). The work done per cycle gives the energy expended per stroke. By correlating the total volumetric loss and cumulative work done, a dimensional wear coefficient (or specific wear rate) could be found. A printer, P, was used to provide a plot of the force profile, an example of which is shown in Fig 3.10.

The other features of the wear tester include :

1) the Danfoss VLT Type 3006 frequency converter for the three-phase 4 kW

Fig 3.4 Cage, C



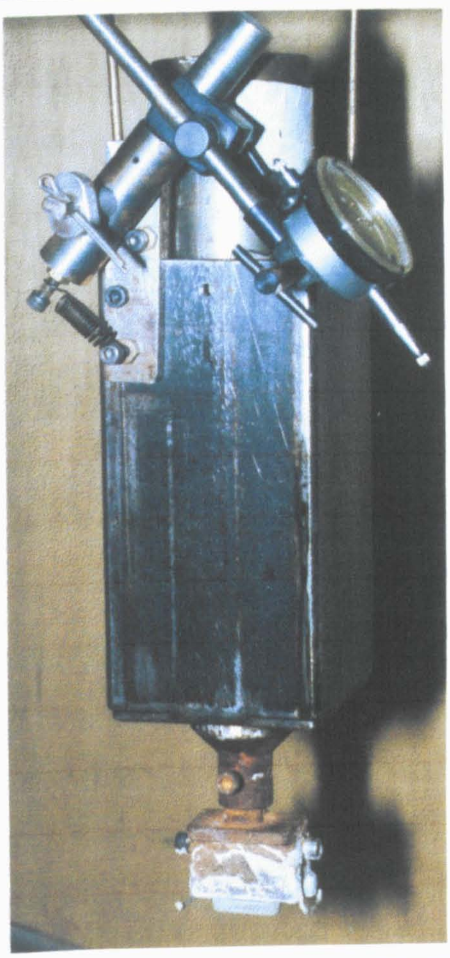


Fig 3.5 Normal load, W

Fig 3.6 Dial indicator, D

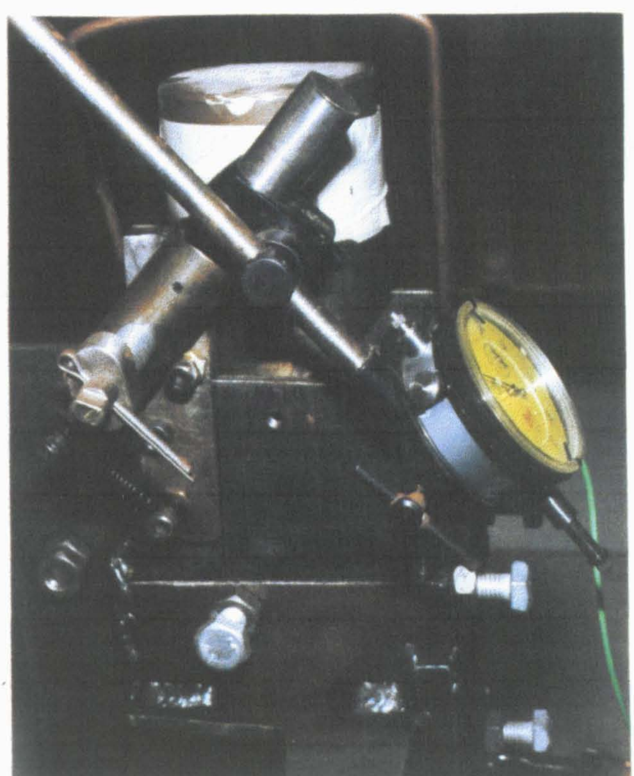


Fig 3.7 Micro-switch, M

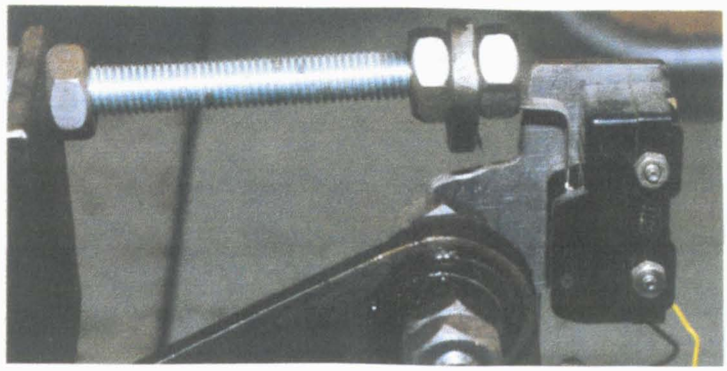


Fig 3.8 Oscilloscope, K

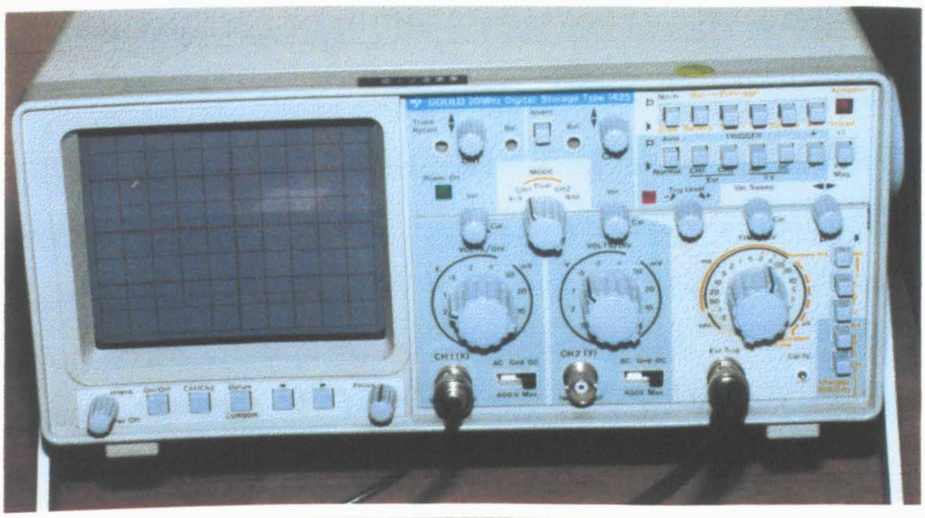
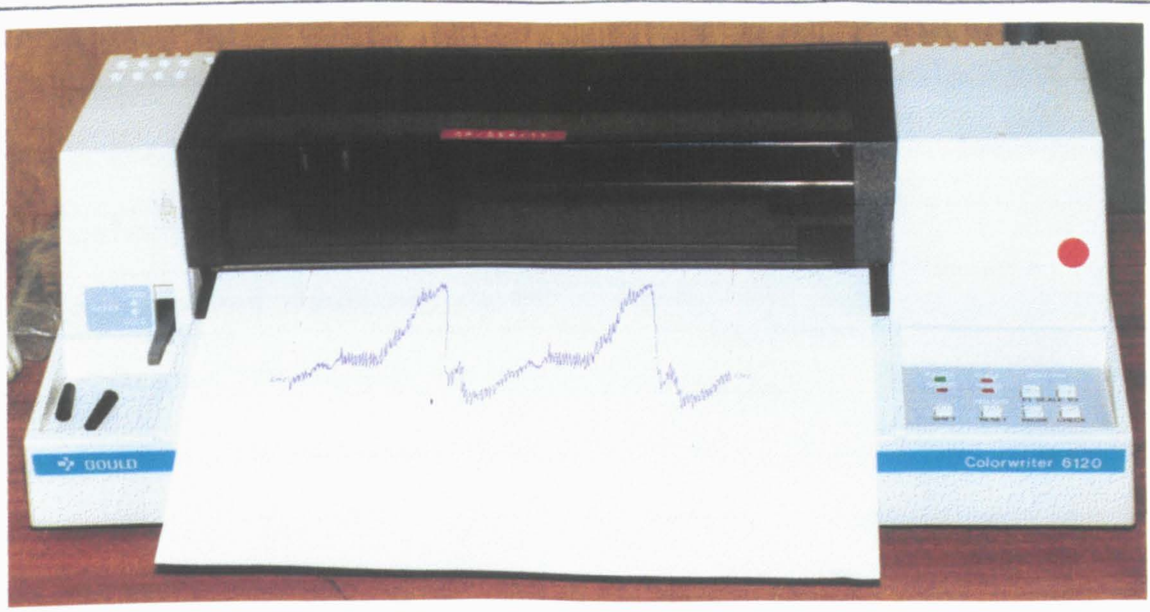


Fig 3.9 Charge amplifier, J

Fig 3.10 Printer, P, with plot of force profile



electric motor which provided variable speed drive (Fig 3.11).

2) a mud decanter which provided controlled metering of the drilling mud for wet sliding (Fig 3.12). The mud was adjusted to give a flow rate of 2 cc/min.

3) a stroke counter to total up the cumulative strokes run (Fig 3.13).

4) a silicone mat heater which heated up the steel liner as a means of simulating the working temperatures inside a pressurised liner during pumping (Fig 3.14). In actual pump operation, cooling systems are used to keep these temperatures to a level which do not cause the piston to fail prematurely. The mat heater was 'sandwiched' between the chrome sleeve and cast steel liner shell.

5) a temperature control loop and capillary rheostat which maintained the temperature of the silicone heater at any set level (Fig 3.15). The rheostat was mounted on and contacted the chrome sleeve being heated by the silicone mat and senses the temperature rise. The feedback from the rheostat in turn controlled the amount of current going to the heater coil embedded in the silicone mat.

6) steel liner, N, which was sectioned and cut from an actual mud pump liner (Fig 3.16). The steel liner is prepared and cut from a new 177.8 mm diameter (7 inch) liner with 28% Chrome steel sleeve. The chrome steel sleeve may have been centrifugally cast and machined and shrunk into the steel shell of mild steel or carbon steel or it may have been centrifugally cast into the liner shell, depending on the mode of manufacture. For the purpose of experimentation, a circular section of the liner, 150 mm long, was cut on the lathe and further divided into three equal segments. The steel shell segment was welded on a fixture and bolted onto the wear tester. The chrome sleeve was secured on the shell by means of a clamp which allowsthe sleeve to be cleaned after each experiment.

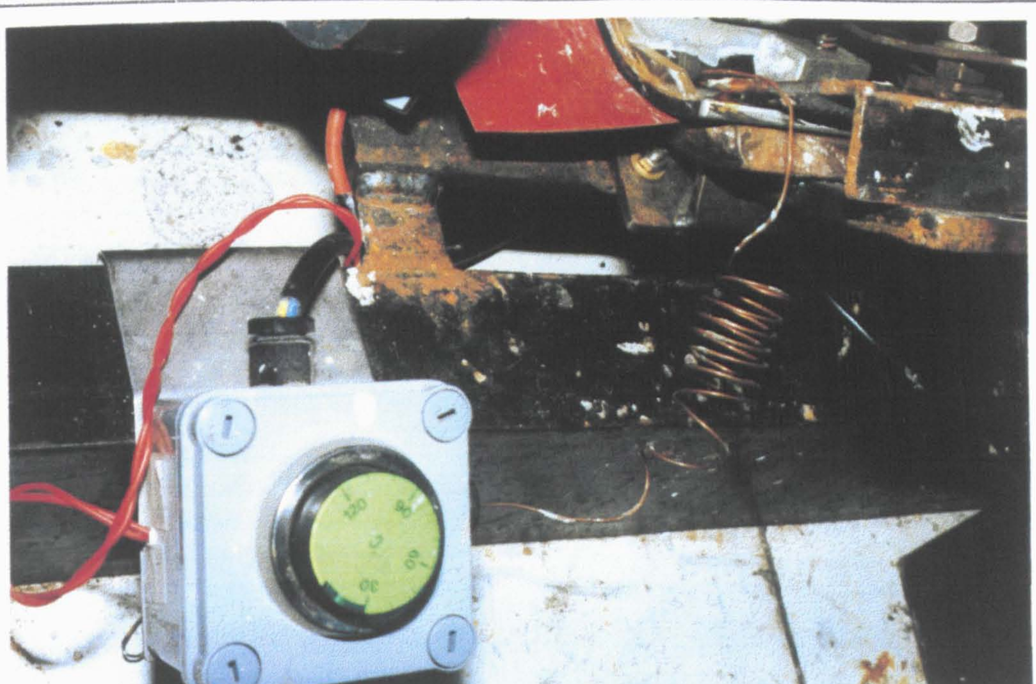
Fig 3.11 Frequency converter

Fig 3.12 Mud decanter

Fig 3.13 Stroke counter

Fig 3.14 Silicone mat heater

Fig 3.15 Temperature control loop

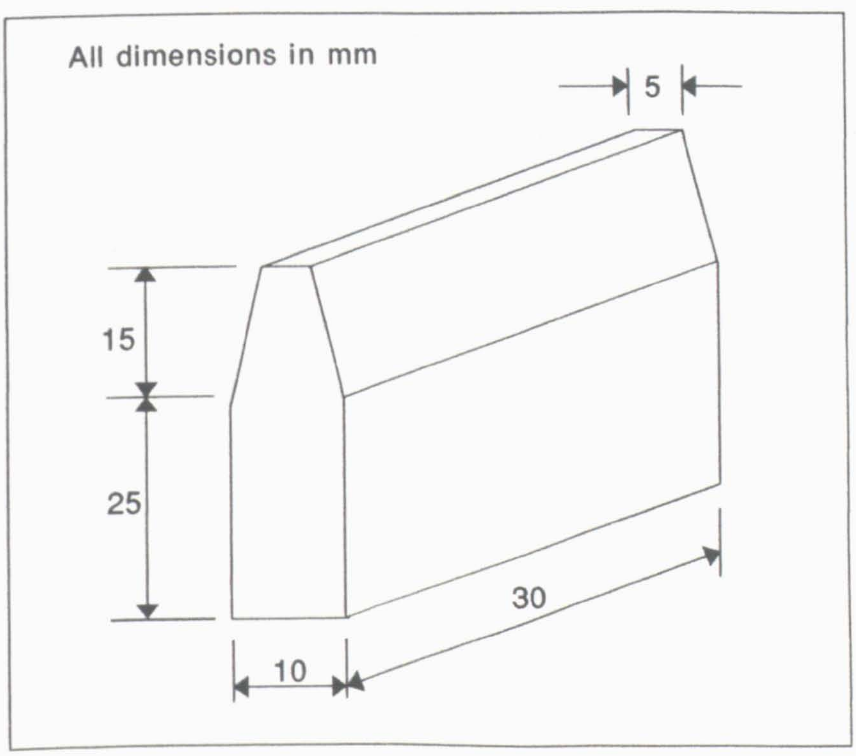


The rubber specimens were prepared from new actual sized piston rubbers designated for mud pumps, among which one of the common materials was Buna-N or nitrile butadiene rubber (NBR), with a hardness of 70 Shore hardness. The dimensions were detailed in Fig 3.17 and had been decided on after a series of trial and error experimenting to determine the minimum dimensions which can take the normal loads used in the experiments without undue deformation. The shape of the rubber specimen had been decided on the premise that pressure acts equally on the piston rubber which was considered to be made up of segments which form the circumference of the piston. The contact surface of the rubber specimen was prepared to assume part of a circumference profile with a diameter of 177.8 mm. Hence when the 6 mm wide rubber specimen contacted the steel liner, the contact surface was assumed to be a flat-on-flat geometry. The normal loads which the rubber specimen could reasonably support for the duration of experiment ranged from 70-150N.

Drilling fluids, both oil-based and water-based muds, were provided by mud companies and were prepared to the same specifications as those used offshore. Because of the new regulations which require the oil discharged from oil-based mud drill cuttings to be reduced from 10% to 1% by weight, it was envisaged that 70% of all future drilling operations would be using water-based mud [3.1]. Water-based mud typically contain barite, bentonite, caustic soda, chemical additives, emulsifiers, corrosion inhibitors and water. To simulate drill cuttings and debris, Finesse grade sand with grain size 0.01-0.5 mm and specific gravity 2.8, was used. These were added by volume or weight to the mud by using with a fluid mixer, Scovill 936-1S, for 10-12 minutes to ensure even mixing. The mechanical properties and chemical composition of the mud were listed in

Fig 3.16 Steel liner, N

Fig 3.17 Dimensions of test specimen



Appendix 8.

For each experiment, the work done per stroke was obtained. If the normal load and rotational speed was kept constant, it was found that the horizontal sliding force per stroke as registered by the load cell was very uniform over the duration of the test. The energy expended and the volumetric wear loss was computed for every 5,000 strokes of sliding. Each experiment was allowed to run for 1,000 strokes to allow sliding to 'stabilise' before measurements were taken for every 5000 strokes and continued for 50,000 strokes. Trials have established that the wear processes on the chrome steel sleeve resulted in only very slight scratches after more than 1.3 million strokes ie a sliding distance of 130 kilometers (Appendix 9). This rendered it very difficult to quantify the wear of the chrome sleeve after only 50,000 strokes (ie 0.5 kilometer sliding distance). Because field drilling experience has dictated, on average, a liner change for every three swab (piston rubber) failures, the decision to focus on quantifying the wear of the piston rubber was taken as being more objective in the initiative to establish a correlation between energy and wear.

After each experiment, the rubber specimens were cleaned and rinsed with water to rid of the mud, marked and put into envelopes and left overnight for them to dry. They were then examined the following day using microphotography and scanning electron micrography (SEM). Details of the experiments were explained in the following relevant sections.

3.3.1 Assumptions

The assumptions which have been taken while performing the experiments included :

a) the wear on the contact surface of the rubber specimens were taken to wear evenly and several trials have indeed indicated this was so in the presence of mud. Hence the measurements translated from the readings on the dial indicator were multiplied with the area of the contact surface to give volumetric material loss.

b) the main material used in these experiments had been Buna-N (nitrile butadiene rubber - NBR). The resultant dimensional wear coefficient or specific wear rate and the empirical wear equation obtained using this material with varying operating conditions was assumed to be representative similar elastomeric materials used for the piston rubber in mud pumps.

c) the drilling mud in the actual mud pump piston served a dual function of lubricating the rubber during sliding as well as being the pressure medium during pumping. The pressure in the pressure chamber was considered to act in all directions, with the result that the forces acting on the rubber material nearest the liner surface was taken to be normal to the liner surface, similar to the wear test system.

d) in any wear situation, a specific amount of energy is expended through the interaction of surfaces in order to generate a unit volume of wear particle. It is assumed that all of this energy is used in the generation of wear particles. The energy input, as computed via the load cell, would be less than the energy supplied to the electric motor drive due to mechanical efficiencies, and is taken as the total energy expended in the surface interaction ie the external work done and is equal to the (tangential force x sliding distance). The tangential force is measured by the load cell at every specified crank angle which along with the corresponding sliding distance, when totalled, would yield the work done per stroke. Hence the specific wear energy is defined as the energy expended in the removal of one unit volume of wear material.

3.4 Experimental results and analysis

3.4.1 Comparing experimental with field samples

To compare the appearance of worn surfaces of experimental rubber samples with those from the field, a series of experimental runs are performed with various loads and speeds with different sand contents. The experiments are run for a range from 20,000 to 50,000 strokes. The rubber samples are then cleaned and examined with SEM and micro-photography. Random samples of failed piston rubbers taken off mud pumps from various offshore drilling installations in different fields are similarly examined using SEM and microphotography. The comparisons between field samples with rubber specimens run on the wear tester are shown in Fig 3.18-3.25.

The horizontal line in the middle of in Fig 3.18 shows signs of the initiation of ploughing in a field specimen which has undergone 332 hours of sliding in a mud pump on an offshore installation. This could have been caused by drill cuttings and wear debris which are embedded in the rubber and subsequently become involved in the three-body abrasion process between the rubber and steel. The test specimen in Fig 3.19 also show signs of grooves being initiated although being subjected to less severe conditions of sliding loads and distance. The deep grooves and ploughing caused by the wear debris in Fig 3.20 may

Fig 3.18 Field specimen
Initiation of ploughing
by sand particles
(x 50)

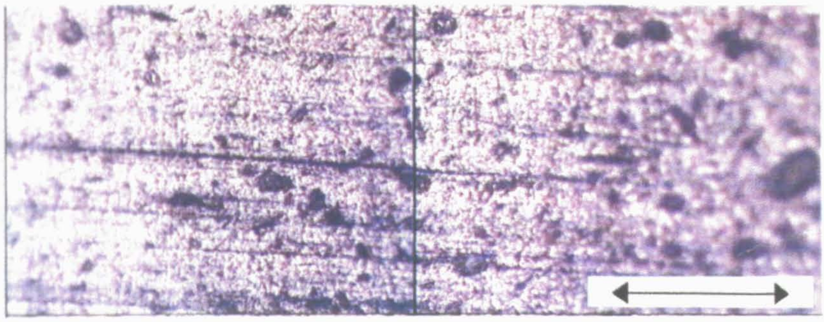


Fig 3.19 Test specimen
Start of ploughing by
sand particles
(x 50)

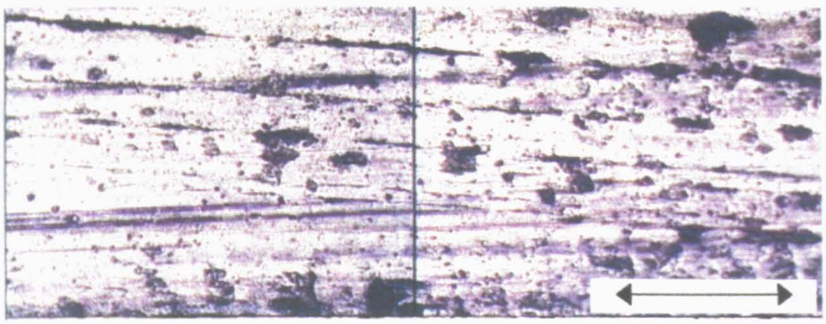


Fig 3.20 Field specimen
Deep ploughing and
grooves
(x 50)

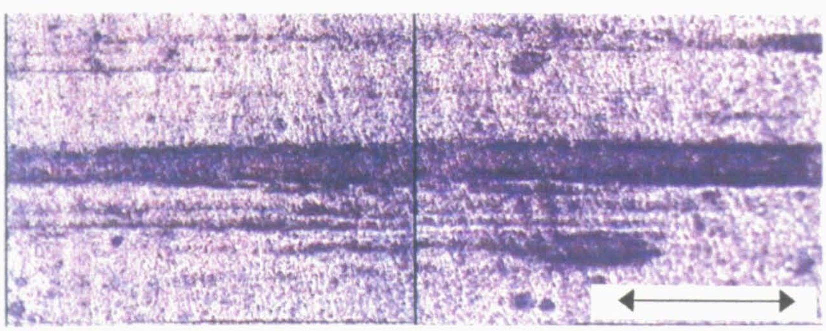
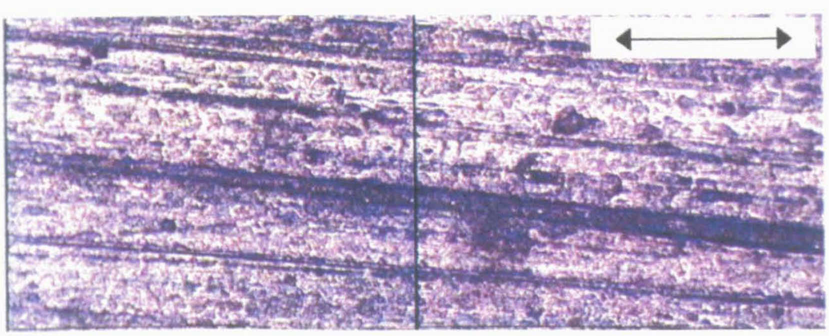


Fig 3.21 Test specimen
Ploughing by sand
particle
(x 50)



have been due to the high pressures (26.4 MPa) which cause the sand debris to 'burrow' into the softer rubber. The grooves exhibited by the test specimen (Fig 3.21) appear quite similar to those on the wear surface of field specimens in Fig 3.20 though not as deep and wide. This is the result of relatively lesser normal loads, 150N, being applied. Other wear characteristics which appeared common to both field and test specimens are abrasion patterns such as waves and ripples. The wear surface of test samples in Fig 3.22(a) & (b) showed such abrasion patterns which are similar to those described by other researchers as Schallamach waves [2.17] or simply as ridges, ribs or bands [2.18-2.20]. The ridges and ripples found in the wear surface of field specimens (Fig 3.23(a) & (b)) are much more rounded and flattened, and smaller compared with test samples. This could have been the result of higher pump pressures which has 'eroded' away the ridges.

Another interesting phenomena in the field sample of Fig 3.23 are the grooves which are diagonal to the direction of sliding and 'criss-cross' the wear surface. These diagonal grooves are similarly obvious in Fig 3.24 which form spirals round the circumferential surface of the piston, much like the spirals found on cylinder bores after a honing operation. This may have been due to the swirling action of the pressurised mud during pumping which force the fluid into the space between the liner and piston at angles diagonal to the sliding motion. Further evidence of the 'criss-cross' path of wear particles is facilitated in Fig 3.25 which shows a wear particle changing direction while ploughing through the rubber.

The similarity of the wear surfaces of test specimens when compared with field samples verifies the design of the wear tester as having achieved the simulation criteria mentioned

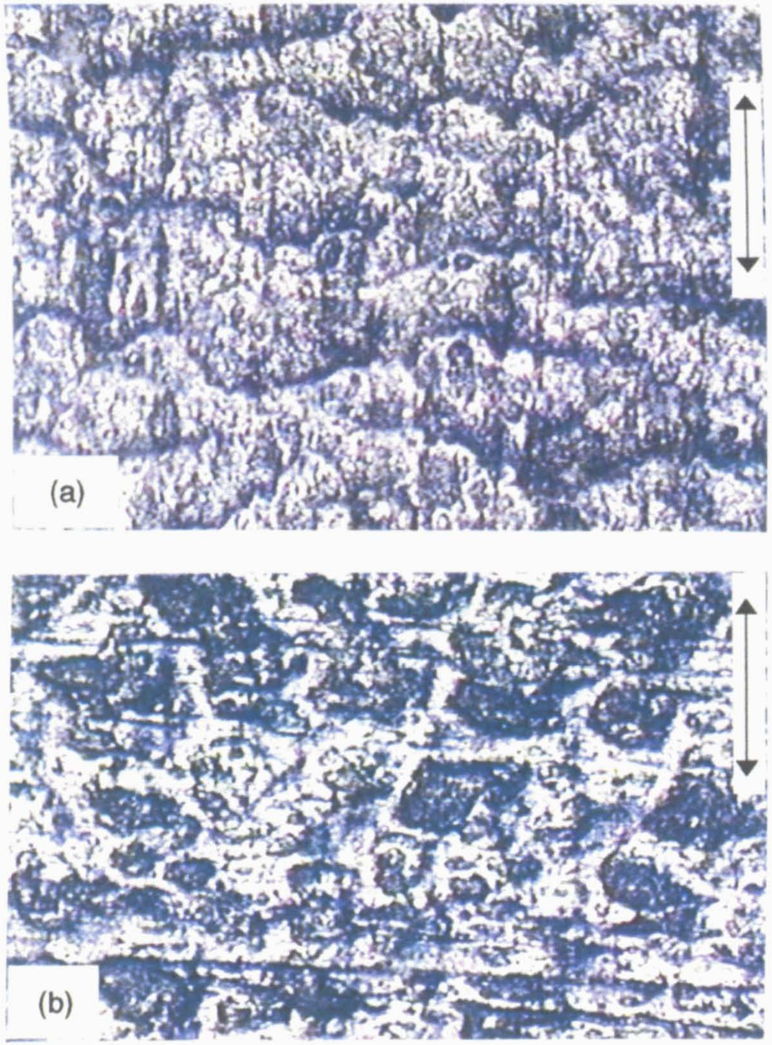


Fig 3.22 Test specimens
Waves & Ripples effect
(x 50)

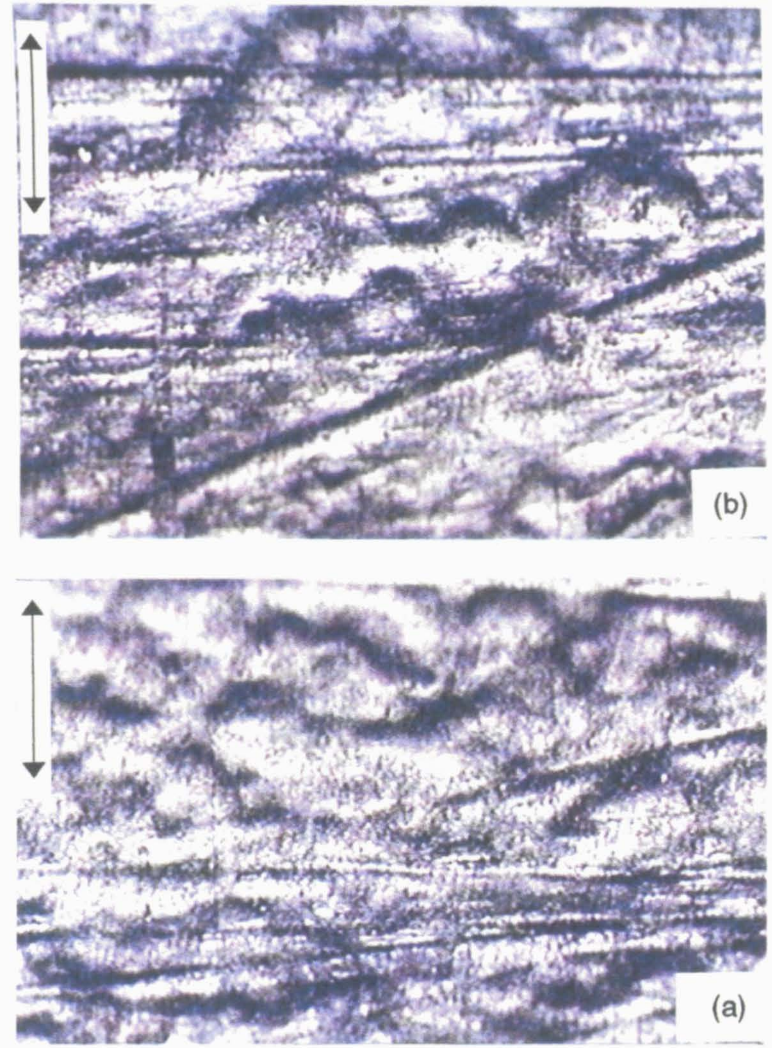


Fig 3.23 Field specimens
Ripples effect
(x 50)

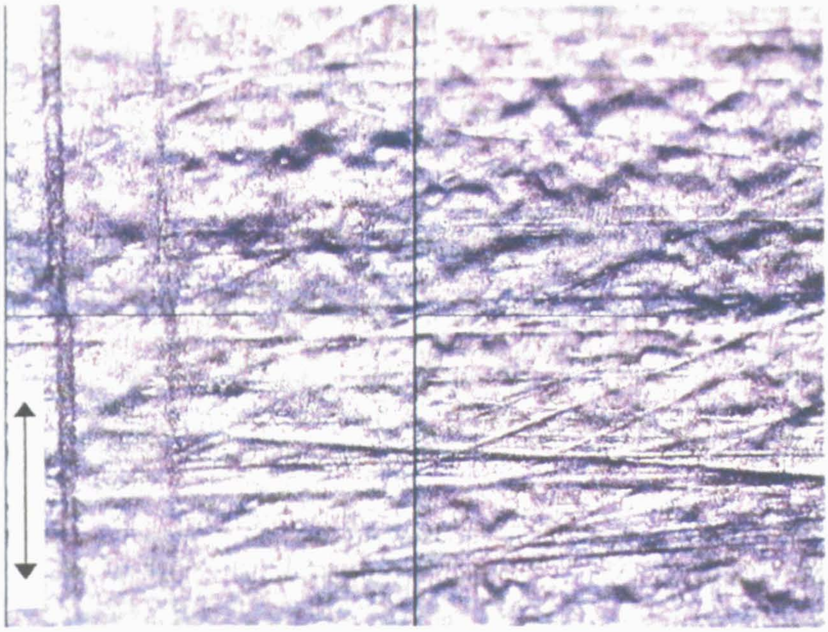
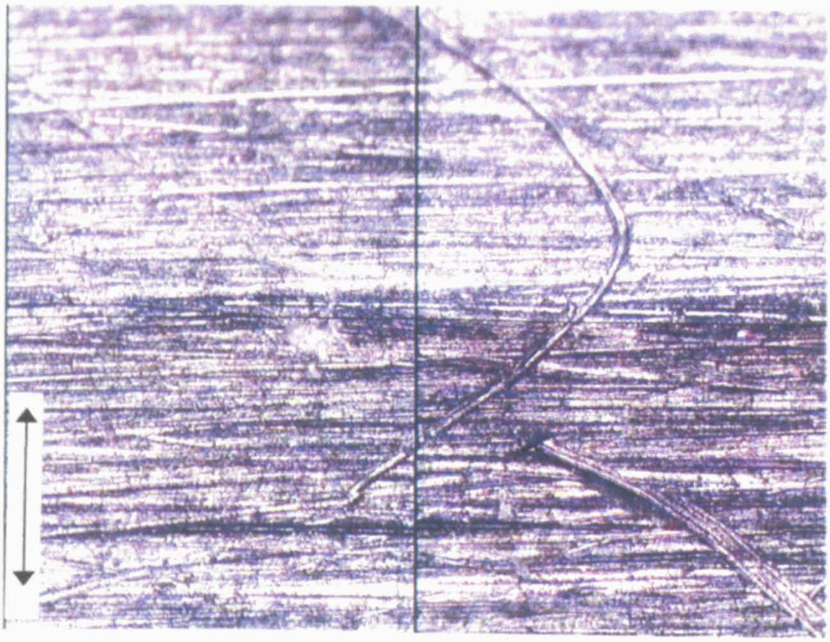


Fig 3.24 'Criss-cross' effect on field specimen
(x 50)

Fig 3.25 'Criss-cross' path of wear particle
(x 50)



in section 2.8 ie worn appearance, although the surface pressures exerted on the rubber specimens due to the normal loads are not anywhere near those in actual mud pump operational levels.

3.4.2 Identifying operating variables which affect wear

Many researchers among which included Peterson [1.43], Benabdallah [1.58], Schallamach [2.17), Anderson [2.24], Brodskii et al [2.26] and others [3.2-3.5] have showed that friction and wear depends on operating parameters such as applied load, sliding speed, roughness and contacting surfaces, temperature, test duration (ie sliding distance) and environmental conditions. Five operating variables are selected for their possible effects on wear ie normal load, sand content, temperature, speed, mud weight.

To determine which of the various operating variables have significant effect of the wear processes and at what extent, there are several routes that may be taken. The one-factor-at-a-time option represents a scientific approach but is very time-consuming and not cost effective, especially if capital equipment and expensive processes are involved. There is also the difficulty of finding any interactive effects. The full factorial approach explores every combination of the factors under consideration. However, since an experiment with, say, seven factors at three levels consists of over 2000 tests, the prohibitive cost would usually discourage this option. In practice interactions above two levels are not as common as generally assumed. A statistical approach popularised by Taguchi [3.1] makes it possible to perform a few well chosen experiments to give a high level of confidence that the experiments will be representative of all the conditions encountered. This approach could be used as an alternative to expensive prototype modelling, as outlined by

Pitts [3.2]. The Taguchi methodology uses orthogonal arrays which are often termed fractional factorial designs because they are a particular combination of trials from a full factorial. Only a small fraction of the possible test conditions need to be investigated as only the main effects of each factor are investigated. Interactions can be included in the test design. And as a result, a large number of factors can be investigated simultaneously. It is generally accepted that an orthogonal array is on average greater than 90% efficient as compared to running a full factorial experiment, and this loss in efficiency is more than justified in terms of resources and time-savings.

This experiment sets out to identify the effects of five operating variables at two levels over eight trials. To achieve this, it is decided to run two blocks of eight trials under the 16 fold-over experimental procedure to clear any doubts as to whether the effects are interactive or due to main factors [3.6]. The experimental set-up allows two sets of response values to be gathered over the duration of the experiments, namely; volumetric wear per stroke (mm^3/stroke) and energy expended per stroke (J/stroke). The response values is entered into response tables and their effects graphically displayed. An analysis of variance (ANOVA) is performed to ascertain the significant levels of effect. A list of the five factors and the experimental levels is shown in Fig 3.26. The design matrix for a 5-factor experiment in eight trials is detailed in Fig 3.27 whilst the design matrix for 16-run fold-over experiment is detailed in Fig 3.28.

3.4.2.1 Volumetric wear per stroke as the response value

Fig 3.29 and 3.33 show the two sets of observed response values (volumetric wear per stroke, $y \times 10^{-4}$ mm^3) respectively. As can be seen from Fig 3.30, it would

Fig 3.26 Experimental levels for 5-factor experiment

	Factor	Level 1	Level 2
A	Load	70N	140N
B	Sand content	1 % wt.	3 % wt.
C	Temperature	20 C	50 C
D	Speed	60 spm	120 spm
E	Mud Weight	12.8 ppg	14 ppg

Fig 3.27 Design matrix for a 5-factor experiment in 8 trials

	Main Effects				
Standard Order	A	B	C	D	E
1	1	1	1	2	2
2	1	1	2	2	1
3	1	2	1	1	2
4	1	2	2	1	1
5	2	1	1	1	1
6	2	1	2	1	2
7	2	2	1	2	1
8	2	2	2	2	2

Fig 3.28 Design matrix for 16-run fold-over experiment (Ref 3.6)

		FACTOR COLUMNS						
Run no.	Block	1	2	3	4	5	6	7
1	1	1	1	1	2	2	2	1
2	1	1	1	2	2	1	1	2
3	1	1	2	1	1	2	1	2
4	1	1	2	2	1	1	2	1
5	1	2	1	1	1	1	2	2
6	1	2	1	2	1	2	1	1
7	1	2	2	1	2	1	1	1
8	1	2	2	2	2	2	2	2
9	2	2	2	2	1	1	1	2
10	2	2	2	1	1	2	2	1
11	2	2	1	2	2	1	2	1
12	2	2	1	1	2	2	1	2
13	2	1	2	2	2	2	1	1
14	2	1	2	1	2	1	2	2
15	2	1	1	2	1	2	2	2
16	2	1	1	1	1	1	1	1

Fig 3.29(a) Response table for 5-factor experiment (Block1)

Random Order No.	Standard Order No.	Response Values ($y \times 10^{-4}$ mm^3)	A Load (N)		B Sand (% wt.)		C Temp. (C)		D Speed (spm)		E Mud Weight (ppg)		BC / DE		CD / BE	
			70	140	1	3	20	50	60	120	12.8	14				
			1	2	1	2	1	2	1	2	1	2	1	2	1	2
4	1	25.8	25.8		25.8		25.8			25.8		25.8		25.8	25.8	
6	2	25.5	25.5		25.5			25.5		25.5	25.5		25.5			25.5
2	3	18.6	18.6			18.6	18.6		18.6			18.6	18.6			18.6
8	4	21.6	21.6			21.6		21.6	21.6		21.6			21.6	21.6	
3	5	33.3		33.3	33.3		33.3		33.3		33.3			33.3		33.3
7	6	22.9		22.9	22.9			22.9	22.9			22.9	22.9		22.9	
1	7	40.5		40.5		40.5	40.5			40.5	40.5		40.5		40.5	
5	8	39		39		39		39		39		39		39		39
Total		227.2	91.5	135.7	107.5	119.7	118.2	109	96.4	130.8	120.9	106.3	107.5	119.7	110.8	116.4
No. of Values		8	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Average		28.4	22.9	33.9	26.9	29.9	29.6	27.3	24.1	32.7	30.2	26.6	26.9	29.9	27.7	29.1
Effect			11.02		3		-2.35		8.6		-3.6		3		1.4	

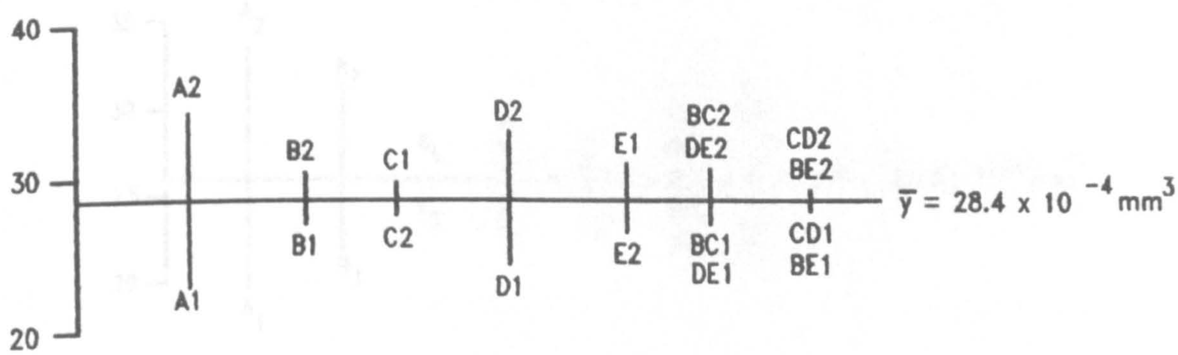


Fig 3.29(b) Graph of esitmated effects for volumetric wear per stroke

Fig 3.30(a) Response table for 5-factor fold-over experiment (Block 2)

Random Order No.	Standard Order No.	Response Values (y x 10^{-4} mm^3)	A Load (N)		B Sand (% wt.)		C Temp. (C)		D Speed spm		E Mud Weight (ppg)		BC / DE		CD / BE	
			70	140	1	3	20	50	60	120	12.8	14				
			1	2	1	2	1	2	1	2	1	2	1	2	1	2
4	1	43.8		43.8		43.8		43.8	43.8		43.8		43.8			43.8
6	2	36.3		36.3		36.3	36.3		36.3			36.3		36.3	36.3	
2	3	24		24	24			24		24	24			24	24	
8	4	26.4		26.4	26.4		26.4			26.4		26.4	26.4			26.4
3	5	20.4	20.4			20.4		20.4		20.4		20.4	20.4		20.4	
7	6	25.2	25.2			25.2	25.2			25.2	25.2			25.2		25.2
1	7	15.6	15.6		15.6			15.6	15.6			15.6		15.6		15.6
5	8	16.2	16.2		16.2		16.2		16.2		16.2		16.2		16.2	
Total		207.9	77.4	130.5	82.2	125.7	104.1	103.8	111.9	96	109.2	98.7	106.8	101.1	96.9	111
No. of Values		8	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Average		25.99	19.4	32.6	20.6	31.4	26	25.95	27.9	24	27.3	24.7	26.7	25.3	24.2	27.8
Effect			13.2		10.8		-0.05		-3.9		-2.6		-1.4		3.6	

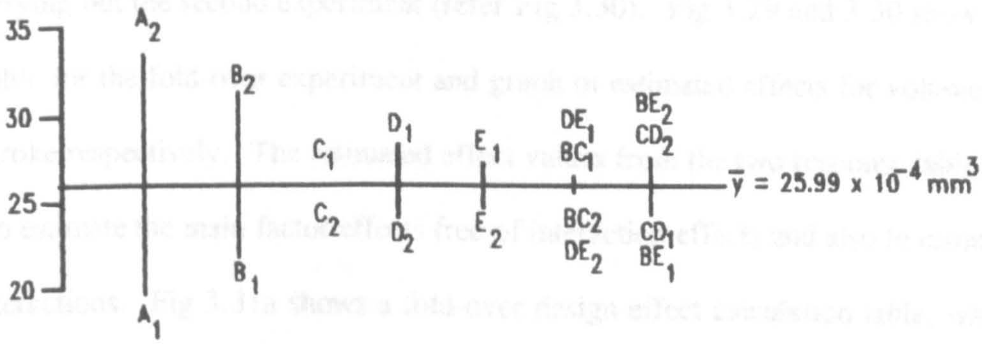


Fig 3.30(b) Graph of estimated effects for volumetric wear per stroke

appear that factors A (load), B (sand content) and D (speed) have significant effects. However, any attempt at analysing five factors with eight trials tend to produce confounding of the main effects and 2-factor interaction, ie being unable to determine which of two or more factors may be affecting the response variable. This results in the following structure [3.6] :

A=BD=CE; B=AD; C=AE; D=AB; E=AC; BD=DE; BE=CD

Due to this structure, it may be possible that the effect on factor A could actually be due to interaction BD (given that factor C has too small an effect to influence interaction CE). The effect on factor B may be due to interaction AD while effect on D may be due to interaction AB. At this stage, it would appear difficult to conclude whether factors A, B or D or interactions BD, AD and AB may have significant effects. A 16-run fold-over experiment would clear any doubts. A fold-over experimental design is one which combines two 2-level fractional factorial and is used as one. Folding over an 8-trial experiment entails constructing an experiment which would fit with the first experiment, so that data from both experiment could be used in the estimates [3.6]. This was achieved by performing a second 8-run experiment where all the factor levels are the opposite of what they were in the first experiment. That was interchanging the 1s and 2s in Fig 3.28 prior to carrying out the second experiment (refer Fig 3.30). Fig 3.29 and 3.30 show the response table for the fold-over experiment and graph of estimated effects for volumetric wear per stroke respectively. The estimated effect values from the two response table are now used to estimate the main factor effects free of interaction effects and also to estimate 2-factor interactions. Fig 3.31a shows a fold-over design effect calculation table, which combines the experiment effect estimates from Fig 3.29 and 3.30. The effects estimated in the last two columns of Fig 3.31a for five factors are summarised in Fig 3.31b.

Considering the column for main effects of Fig 3.31a, it can be seen that factors A (load) and B (sand content) have significant effects on the response values and hence the wear process. These seem to agree with the conclusions reached by Zhang [1.48]. The column for interaction clearly indicates that it is the interaction AB and not factor D which have significant effect. Compared with factors A and B, it appears that factor D does have some effect, though to a lesser extent. This can be explained by the fact that in offshore mud pump operation, the pressure (as related to the normal load) is controlled by varying the speed of reciprocation. The average responses at the four different combinations of levels for factors A and B are given in Fig 3.32a and plotted in Fig 3.32b. Since the aim is to identify factors and settings which result in the most wear (larger-the-better), setting factors A (load) and B (sand content) at their high levels would produce the largest average response. The estimated volumetric wear of the two block fold-over experiment set at the levels of the main effect factors for maximum wear are obtained and averaged out as follows (refer Fig 3.29 and 3.30) :

Block 1 :

$y_{(1)}$ = (grand mean) + (A contribution) + (B contribution) + (D contribution)

$= 28.4 + (11.02) + (3/2) + (8.6/2)$

$= 39.71 \times 10^{-4}\ mm^3$

Block 2 :

$y_{(2)}$ = (grand mean) + (A contribution) + (B contribution) + (D contribution)

$= 25.99 + (13.2/2) + (10.8/2) + (3.9/2)$

$= 39.94 \times 10^{-4}\ mm^3$

Fig 3.31(a) Fold-over design effects calculation table

Combined Experiment Estimates for Volumetric wear/stroke

Column	Experiment 1 Effect E1	Experiment 2 Effect E2	Main Effect $\frac{E1 + E2}{2}$	Interaction $\frac{E1 - E2}{2}$
y average	28.4	25.99	27.2	1.21
1 A	11.02	13.2	12.11	-1.1
2 B	3	10.8	6.9	-3.9
3 C	-2.35	-0.05	-1.2	-1.15
4 D	8.6	-3.9	2.35	6.25
5 E	-3.6	-2.6	-3.1	-0.5
6 BC = DE	3	-1.4		2.2
7 BE = CD	1.4	3.6		-1.1

Effects estimated using Fig 3.31(a)

	5 Factors Effect Column Main	Interaction
y	average	block
1	A	BD = CE
2	B	AD
3	C	AE
4	D	AB
5	E	AC
6	*	BC = DE
7	*	BE = CD

Fig 3.35(a) Fold-over design effects calculation table

Combined experiment estimates for Energy Expended/stroke

Column	Experiment 1 Effect Ef1	Experiment 2 Effect Ef2	Main Effect $\frac{Ef1 + Ef2}{2}$	Interaction $\frac{Ef1 - Ef2}{2}$
y average	58.7	55.95	57.33	1.375
1 A	40	34.4	37.2	2.8
2 B	16.8	11	13.9	2.9
3 C	1.9	2.3	2.1	-0.2
4 D	13.3	0.1	6.7	6.6
5 E	1.16	1.3	1.23	-0.14
6 BC = DE	-2.1	1.3		-1.7
7 BE = CD	-7.2	-0.5		-3.35

Effects estimated using Fig 3.35(a)

	5 Factors Effect Column Main	Interaction
y	average	block
1	A	BD = CE
2	B	AD
3	C	AE
4	D	AB
5	E	AC
6	*	BC = DE
7	*	BE = CD

Fig. 3.32(a) Average response table for different levels for factors A and B

		A : LOAD	
		70N	140N
B : SAND (WT.)	1 %	25.8×10^{-4} 25.5×10^{-4} 15.6×10^{-4} 16.2×10^{-4} 83.1×10^{-4} $A_1B_1 = 83.1/4 = 20.8 \times 10^{-4}$	33.3×10^{-4} 22.9×10^{-4} 24×10^{-4} 26.4×10^{-4} 106.6×10^{-4} $A_2B_1 = 106.6/4 = 26.7 \times 10^{-4}$
	3 %	18.6×10^{-4} 21.6×10^{-4} 20.4×10^{-4} 25.2×10^{-4} 85.8×10^{-4} $A_1B_2 = 85.8/4 = 21\ 5 \times 10^{-4}$	40.5×10^{-4} 39×10^{-4} 43.8×10^{-4} 36.3×10^{-4} 159.6×10^{-4} $A_2B_2 = 159.6/4 = 39.9 \times 10^{-4}$

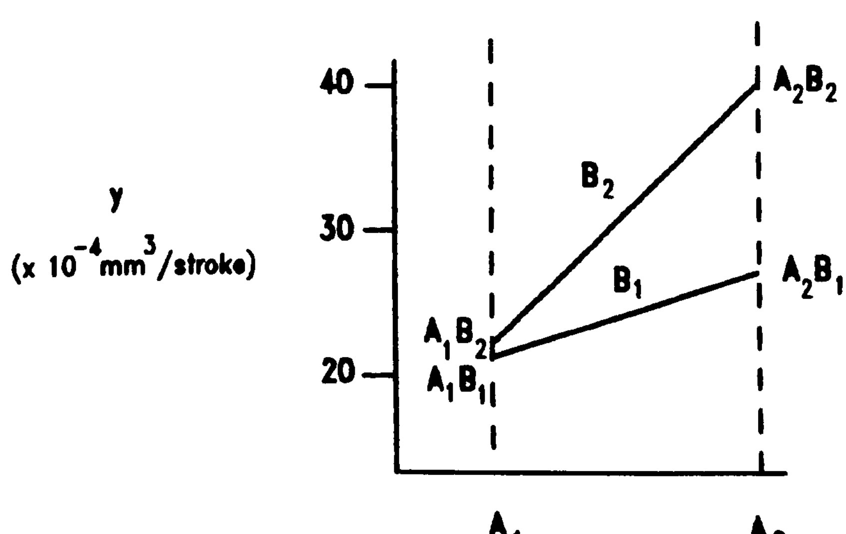


Fig. 3.32(b) Plot of average response for different levels of factors A and B

Average of Blocks 1 and 2 = (39.71 + 39.94)/2 = 39.83 x 10^{-4} mm^3.

The resulting estimate of 39.83 x 10^{-4} mm^3 closely compares with that obtained from the average response table (A^2B^2) in Fig 3.32.

(Note : $y_{(1)}$ and $y_{(2)}$ is used to denote the estimated average responses (volumetric wear per stroke) for a given combination of factor levels).

3.2.4.2 Energy expended per stroke as the response value

Considering the energy expended per stroke (J/stroke) as the response value for the same experiment, Fig 3.33 and 3.34 shows the response tables for the two block of the fold-over experiment respectively. From Fig 3.33b, showing the graph of estimated effects for energy expended per stroke, factor D (speed) appears to have significant effect on the wear process. However, Fig 3.34b, showing the graph of estimated effects for the fold-over experiment, appear to indicate otherwise. As can be seen from Fig 3.35, from the Experiment 1 column, factor D appears to have significant effect. However, it is also possible that the alias of factor D ie interaction AB (refer Fig 3.35b) may have caused the effect. After running a fold-over experiment and combining the effects on the Main Effect and Interaction column, it can be concluded that interaction AB appears to have significant effect while the effect of factor D does not appear to be as significant. This agrees with the conclusion reached in Section 3.2.4.1 for volumetric wear per stroke. The average responses at the four different combinations of levels for factors A and B are given in Fig 3.36a and plotted in Fig 3.36b. Setting the factors A (load) and B (sand content) at their high levels would produce the largest average response (85.93 J/stroke).

Fig 3.33(a) Response table for 5–factor experiment (Block 1)

Random Order No.	Standard Order No.	Response Values y (J/stroke)	A Load (N)		B Sand (% wt.)		C Temp. (C)		D Speed (spm)		E Mud Weight (ppg)		BC / DE		CD / BE	
			70	140	1	3	20	50	60	120	12.8	14				
			1	2	1	2	1	2	1	2	1	2	1	2	1	2
4	1	39.07	39.07		39.07		39.07			39.07		39.07		39.07	39.07	
6	2	34.76	34.76		34.76			34.76		34.76	34.76		34.76			34.76
2	3	37.57	37.57			37.57	37.57		37.57			37.57	37.57			37.57
8	4	43.4	43.4			43.4		43.4	43.4		43.4			43.4	43.4	
3	5	57.42		57.42	57.42		57.42		57.42		57.42			57.42		57.42
7	6	69.83		69.83	69.83			69.83	69.83			69.83	69.83		69.83	
1	7	96.82		96.82		96.82	96.82			96.82	96.82		96.82		96.82	
5	8	90.56		90.56		90.56		90.56		90.56		90.56		90.56		90.56
Total		469.43	154.8	314.6	201	268.4	230.8	238.6	208.2	261.2	232.4	237	238.9	230.5	249.1	220.3
No. of Values		8	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Average		58.7	38.7	78.7	50.3	67.1	57.7	59.6	52	65.3	58.1	59.3	59.7	57.6	62.3	55.1
Effect			40		16.8		1.9		13.3		1.16		−2.1		−7.2	

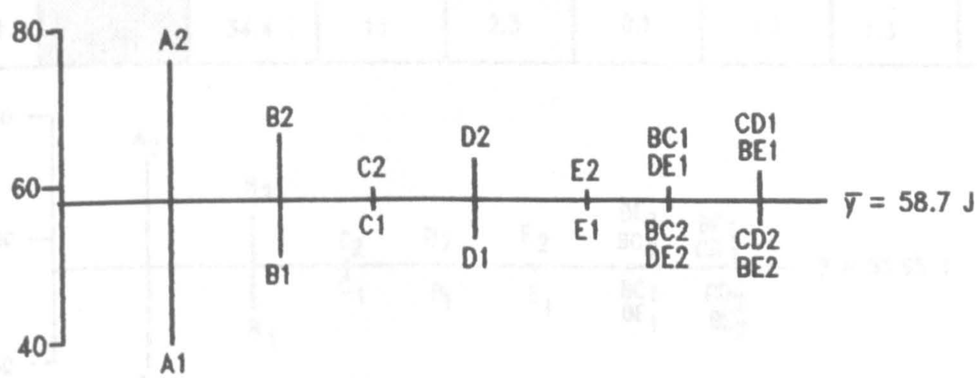


Fig. 3.33(b) Graph of estimated effects of energy expended per stroke

Fig 3.34(a) Response table for 5-factor fold-over experiment (Block 2)

Random Order No.	Standard Order No.	Response Values y (J/stroke)	A Load (N)		B Sand (% wt)		C Temp. (C)		D Speed (spm)		E Mud Weight (ppg)		BC / DE		CD / BE	
			70	140	1	3	20	50	60	120	12.8	14				
			1	2	1	2	1	2	1	2	1	2	1	2	1	2
4	1	77.8		77.8		77.8		77.8	77.8		77.8		77.8			77.8
6	2	78.5		78.5		78.5	78.5		78.5			77.8		77.8	77.8	
2	3	69.6		69.6	69.6			69.6		69.6	69.6			69.6	69.6	
8	4	66.7		66.7	66.7		66.7			66.7		66.7	66.7			66.7
3	5	45.2	45.2			45.2		45.2		45.2		45.2	45.2		45.2	
7	6	42.4	42.4			42.4	42.4			42.4	42.4			42.4		42.4
1	7	35.9	35.9		35.9			35.9	35.9			35.9		35.9		35.9
5	8	31.5	31.5		31.5		31.5		31.5		31.5		31.5		31.5	
Total		447.62	155	292.6	203.7	243.9	219.1	228.5	223.7	223.9	221.3	226.3	221.2	226.4	224.8	222.8
No. of Values		8	4	4	4	4	4	4	4	4	4	4	4	4	4	4
Average		55.95	38.8	73.2	50.9	61	54.8	57.1	55.9	56	55.3	56.6	55.3	56.6	56.2	55.7
Effect			34.4		11		2.3		0.1		1.3		1.3		-0.5	

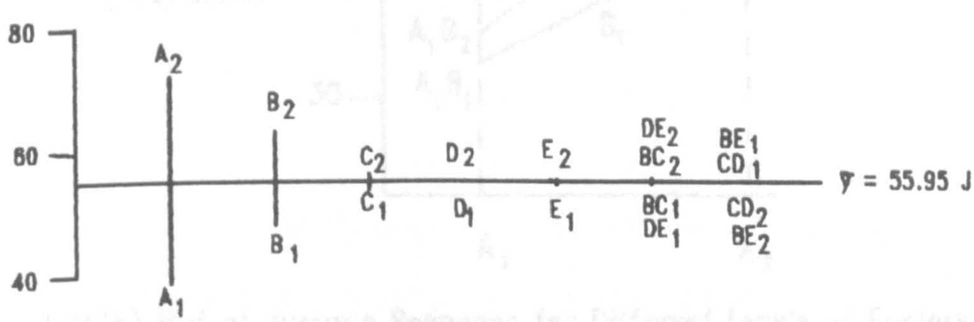


Fig. 3.34(b) Graph of estimated effects for energy expended per stroke

Fig. 3.36(a) Average response table for different levels for factors A and B

		A : LOAD, 70N	A : LOAD, 140N
B : SAND (WT.)	1 %	39.07 34.76 35.9 31.5 141.23 A_1B_1 = 141.23/4 = 35.31 J	57.4 69.8 69.6 66.7 263.5 A_2B_1 = 263.5/4 = 65.88 J
	3 %	37.57 43.4 45.2 42.4 168.57 A_1B_2 = 168.57/4 = 42.14 J	96.8 90.6 77.8 78.5 343.7 A_2B_2 = 343.7/4 = 85.93 J

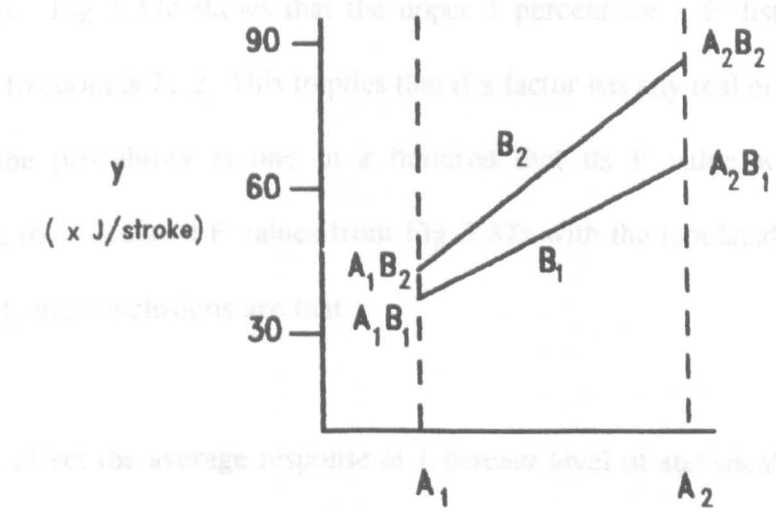


Fig. 3.36(b) Plot of Average Response for Different Levels of Factors A and B

Fig 3.31 and 3.35 showed that factors A and B have significant main effects and interaction AB have significant effects while its alias, factor D, (see Fig 3.35b) do not appear to have significant effect. Curiously, Fig 3.35 showed that both factor D and interaction AB have some lesser effect on the response values. An analysis of variance (ANOVA) is performed for the main effects and interactions of the response values of both volumetric wear and energy expended (Fig 3.37). In the ANOVA for volumetric wear per stroke (Fig 3.37a), factors A (load) and B (sand) are considered as is interaction AB given that factor D has a lesser effect. The $(Ef)^2$ value in the 'no effect' row is obtained from the average of the four $(Ef)^2$ values for interaction effects and factor C which has insignificant main effect. Hence the 'no effect' $(Ef)^2 = ((2.2)^2 + (-1.1)^2 + (-1.2)^2 + (-3.1)^2/4 = 4.275$. The degree of freedom 'df' is the number of factors used in the row, which for the 'no effects' would assigned df = 4. The column 'F' refers to the F statistics and is obtained by dividing the $(Ef)^2$ values in a row by the no effect $(Ef)^2$. These calculated F values are then compared to the theoretical values for the F distribution. Fig 3.37c shows that the upper 1 percent for a F distribution with (1,4) degrees of freedom is 21.2. This implies that if a factor has any real effect on the response variable, the probability is one in a hundred that its F value would exceed 21.2. Comparing the calculated F values from Fig 3.37a with the tabulated F values from Fig 3.37c [3.6], the conclusions are that :

- Factors A affect the average response at 1 percent level of statistical significance.
- Factor B affect the average response at 5 percent level of statistical significance.
- Factor D did not have significant effect on the average response value.
- Interaction AB affect the average response at the 5 percent level of statistical

Fig.3.37 Analysis of variance for the 2 response values

(a) ANOVA for Volumetric wear/stroke

Column	Factor	E	E^2	df	F	Significance Level
1	A (load)	12.1	146.4	1	34.2	1 %
2	B (sand)	6.9	47.6	1	11.13	5 %
4	D (speed)	2.35	5.5	1	1.3	
	AB (interaction)	6.25	39.1		9.14	5 %
3,5,6,7	No Effect		4.275	4		

(b) ANOVA for Energy expended/stroke

Column	Factor	E	E^2	df	F	Significance Level
1	A (load)	37.2	1383.84	1	276.8	1 %
2	B (sand)	13.9	193.21	1	38.6	1 %
4	D (speed)	6.7	44.89	1	8.96	5 %
	AB(interaction)	6.6	43.56		8.69	5 %
3,5,6,7	No Effect		5.01	4		

(c) F Distribution critical values for (1,J) degrees of freedom (Ref. 3.6)

	Significance Level	
J	5%	1%
1	161.4	4052
2	18.51	98.5
3	10.13	34.12
4	7.77	21.2
5	6.61	16.26
10	4.96	10.04
∞	3.84	6.63

significance

In the second ANOVA for energy expended per stroke (Fig 3.37b), the 'no effect' $(Ef)^2$ value is obtained through the average square of the interaction effects and factor C effect, ie $(Ef)^2 = ((1.23)^2 + (1.7)^2 + (3.35)^2 + (2.1)^2/4 = 5.01$. The degree of freedom for the 'no effect' is also four. The conclusions of the second ANOVA for the response values of the energy expended are that :

- factors A and B affect the average response at the 1 percent level of statistical significance
- factor D (speed) and interaction AB affect the average response at the 5 percent level of statistical significance.

In comparing the conclusions of the two ANOVA, those for the volumetric wear is considered in greater significance given that the failure criteria is the volume loss of the piston rubber. In this light, factors A (normal load) and B (sand) and its interactive effects are categorised as having significant effect on the wear process.

To verify the factors deemed as having significant effect on the wear process, a confirmatory test is run. The parameters selected are; load 140N (A2), sand 3% (B2), Temperature 50°C (C2), speed 120 spm (D2), mud weight 1.678 kg/lit (E2). The trial is run for 40,000 strokes and the wear loss monitored every 5000 strokes. The resulting plot of volume loss against total strokes slided (Fig 3.38) showed the volume loss is slightly more than the predicted value, a 4.5 percent difference. One possible explanation

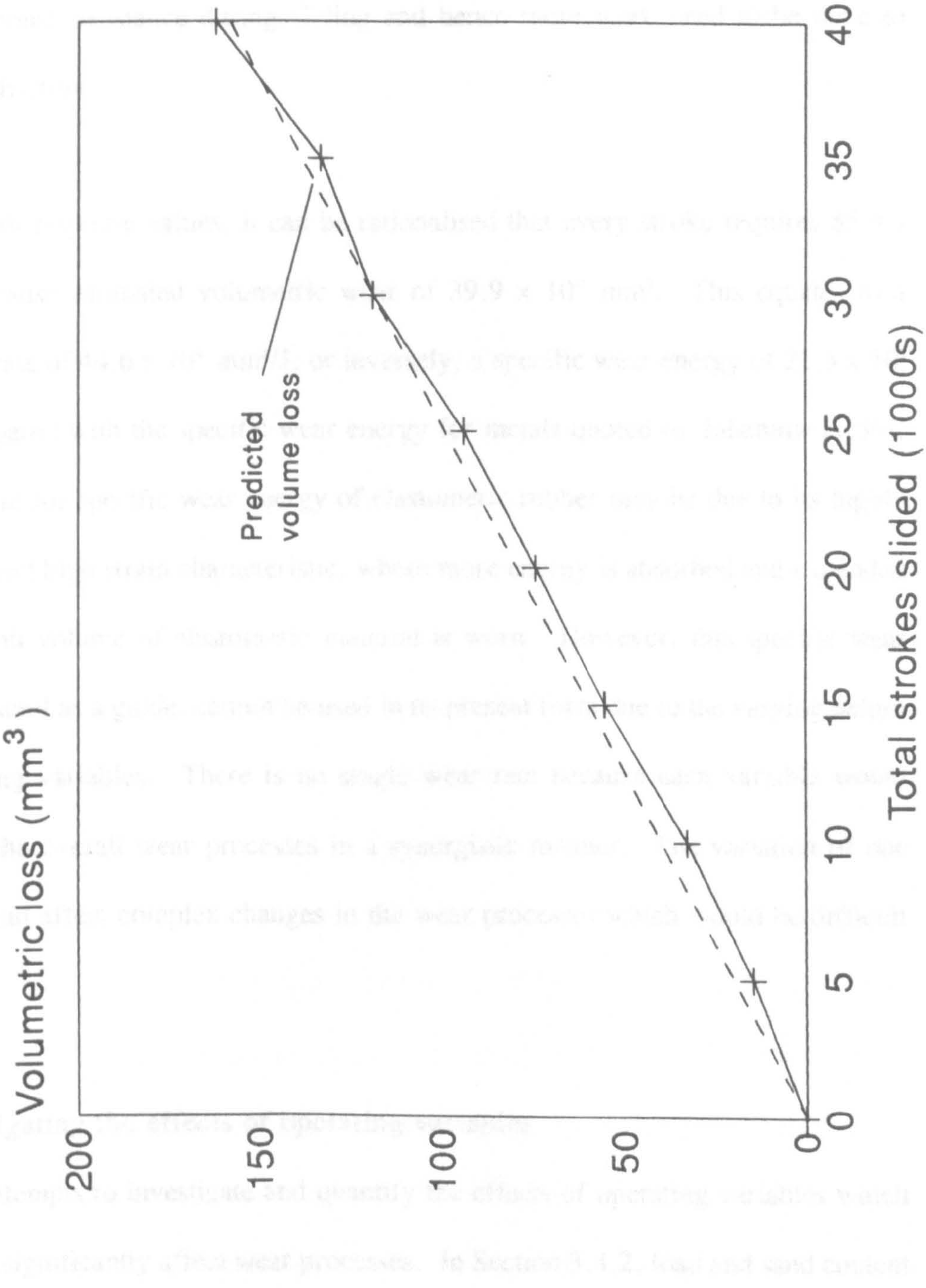


Fig 3.38 Confirmatory test run

is that the volume loss do not necessarily follow a straight line. The higher energy expended per stroke, 87 J, could have been due to the heavier mud weight which result in higher frictional resistance during sliding and hence more work need to be done to overcome the friction.

Combining both response values, it can be rationalised that every stroke requires 85.9 J of energy to cause estimated volumetric wear of 39.9 x 10^{-4} mm^3. This equates to a specific wear rate of 44.6 x 10^{-6} mm^3/J, or inversely, a specific wear energy of 21.5 x 10^6 MJ/m^3. Compared with the specific wear energy for metals quoted by Jahanmir [1.50], the higher value for specific wear energy of elastomeric rubber may be due to its highly elastic nature and high strain characteristic, where more energy is absorbed and expended before each unit volume of elastomeric material is worn. However, this specific wear rate, though useful as a guide, cannot be used in its present form due to the varying nature of the operating variables. There is no single wear rate because each variable would contribute to the overall wear processes in a synergistic manner. The variation of one parameter would affect complex changes in the wear processes which would be difficult to quantify.

3.4.3 Investigating the effects of operating variables

This section attempts to investigate and quantify the effects of operating variables which are deemed to significantly affect wear processes. In Section 3.4.2, load and sand content are founded to have significant effect on the wear process when rubber is slided on chrome steel in the presence of drilling mud, and these two parameters would be studied further. Although temperature is not found to be significant in the Taguchi experiments, this may possibly be due to the inadequate temperature range selected which could not cover the effects due to higher temperatures. This parameter would be looked in greater detail to confirm its effect on the wear process. Similarly, mud weight in the form of mud viscosity would also be studied further. The combined effects of load with sand, and with temperature would also be studied.

3.4.3.1 Sand content

To look at the effect of sand on the wear process, an experiment is conducted where the operating parameters are kept constant while the sand content is varied. A normal load of 150 N is selected to run at a constant 120 strokes per minute (spm) at 30°C (room temperature). The drilling fluid selected is water-based mud with a mud weight of 1.378 kg/lit (11.5 ppg). Five experimental runs are executed with sand content ranging from 1% to 5% by weight. Feedback from offshore installations have indicated that sand content at 2% and above is considered very detrimental to the mud pump fluid end service life, while some drilling contractors have strived for the optimum 0% sand threshold with 1% as the absolute maximum limit permissible. For every experimental run, 1000 strokes are allowed for the test specimen to 'run-in' before the stroke count is commenced and measurements taken for the volumetric loss of the rubber specimen. The results of the experiment indicated (Fig 3.39) that the sand content appeared to have a direct effect on the wear processes. Analysis of the experimental results (Fig 3.40) showed a correlation between the wear loss, $V_{w(s)}$ and the frictional work, E (J) which can be described by the form :

$$V_{w(s)} = a_{(s)} E^{b_{(s)}} \tag{3.1}$$

Fig 3.39 Effects of sand on sliding wear

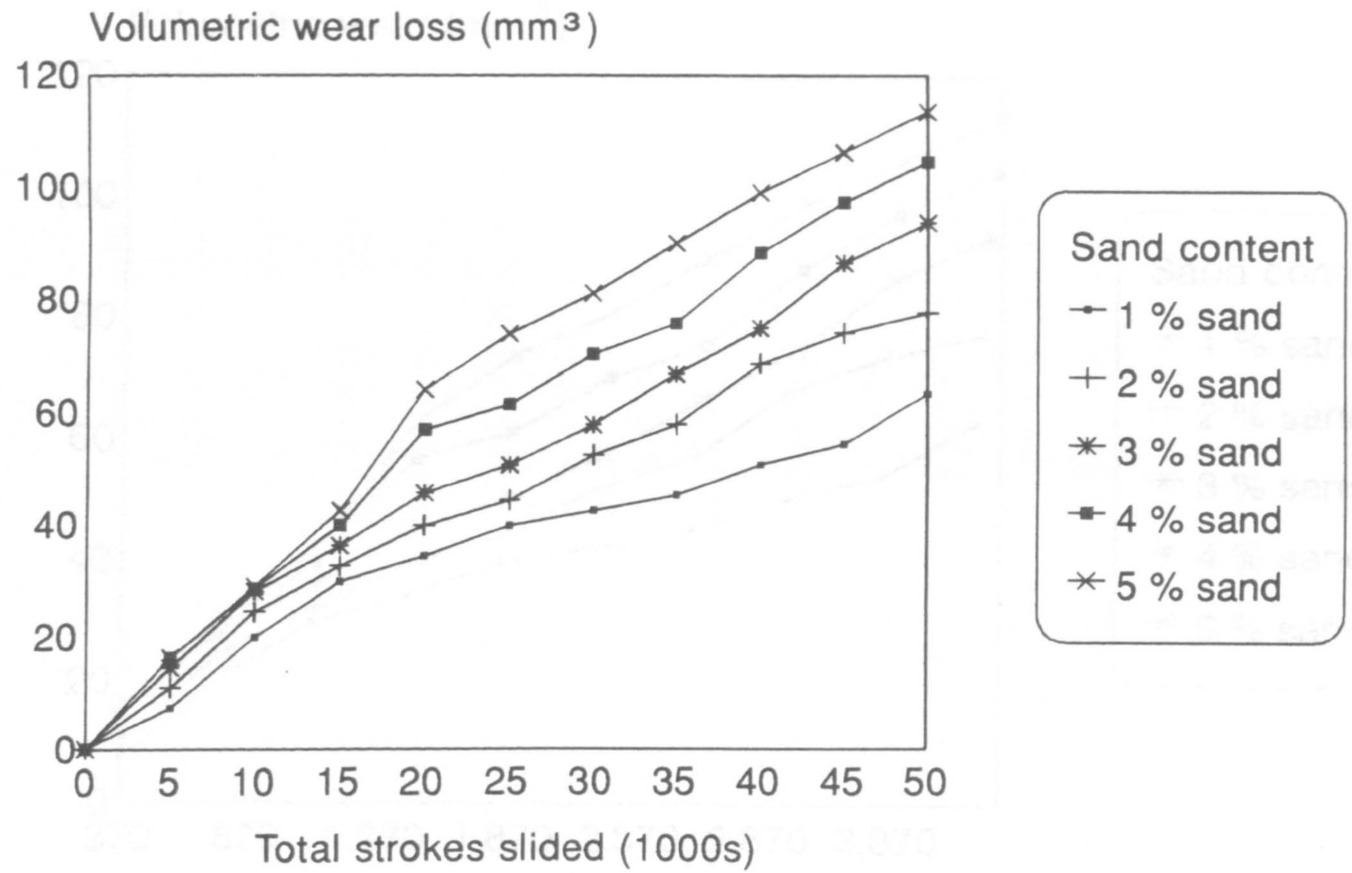
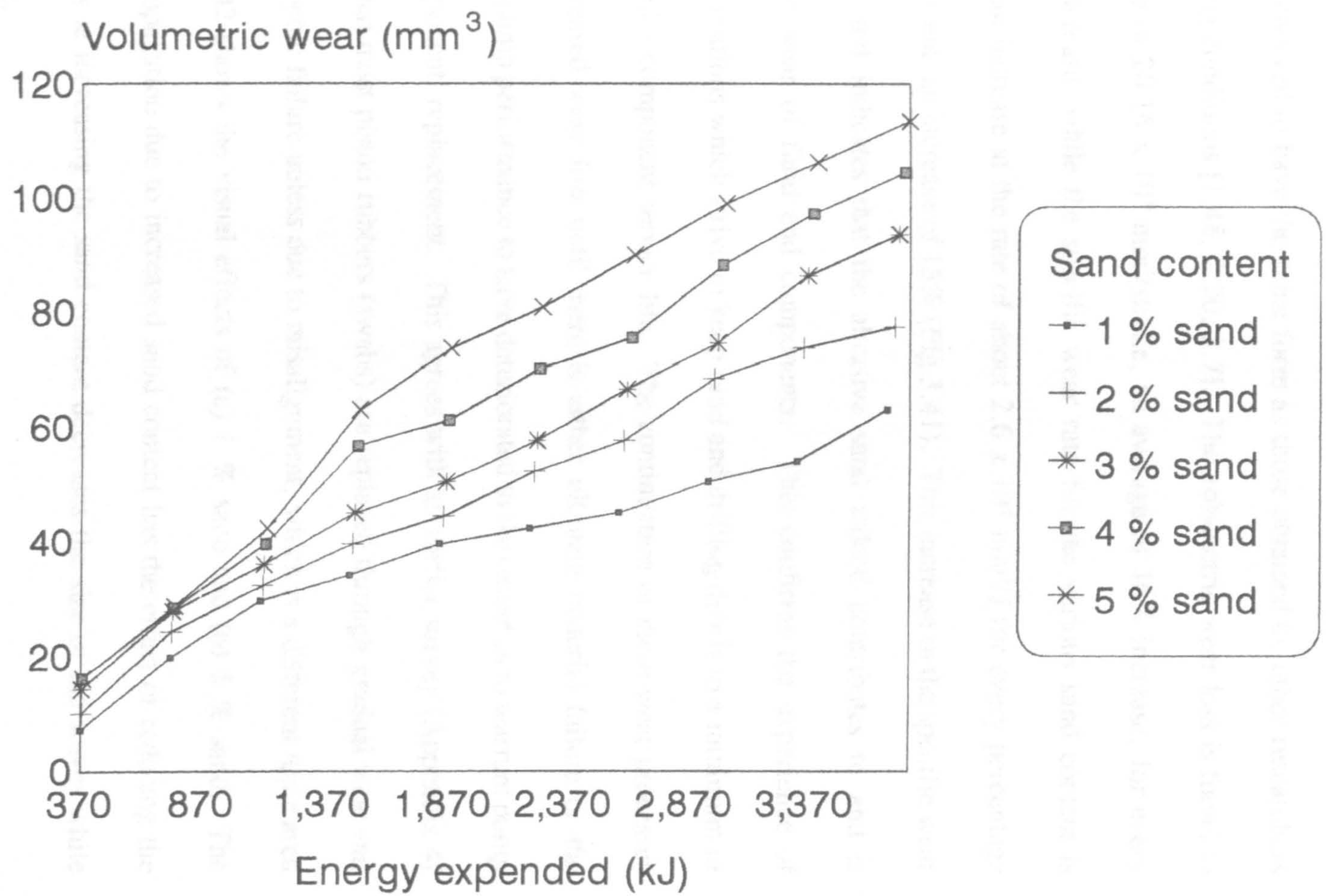


Fig 3.40 Volume loss and energy expended due to sand effects



where the coefficient $a_{(s)}$ is found to have a constant value of 0.00265 while exponent $b_{(s)}$ ranges from 0.665 to 0.705 for sand contents of 1% to 5% respectively (Table 3.1). Equation (3.1) is observed to have the same form as those obtained by other researchers for different running conditions [1.48, 2.20, 3.7]. The volumetric wear loss is found to increase at the rate of 20.16 x 10^{-6} mm^3/stroke, an average of 16% increase, for every percentage sand increase, while the specific wear rate for the various sand content is observed to show an increase at the rate of about 2.6 x 10^{-6} mm^3/J for every percentage increase in sand content, an increase of 15% (Fig 3.41). This increase in the specific wear rate is significant and indicates that the abrasive sand indeed contributes to and is responsible for the wear of fluid end components. This confirms the experience of offshore drilling operations which strive to keep sand and drilling debris to a minimum in the effort to maximise component service life. The continuation of these wear processes would lead to increased wear loss until there is either ultimate material failure or the operators consider pump performance to have deteriorated to the extent as to warrant pump downtime for component replacement. This agrees with an earlier survey [Appendix 4] which concluded that most piston rubbers (swabs) are replaced through gradual wear-out rather than catastrophic failure unless due to misalignment, which is a different topic area altogether. Fig 3.42 shows the visual effects of (a) 1 % sand and (b) 5 % sand. The resultant waves propagation due to increased sand content has the effect of reducing the peaks of the waves ie increasing the sand content decreases the size of the waves while flattening its peaks.

Fig 3.41 Sand effects on specific wear rate and volume loss

Sand % by weight	Total volume loss (mm^3)	Energy expended per stroke (J)	Specific wear rate (mm^3/J)
1	63	74	17×10^{-6}
2	77.4	74.6	20.8×10^{-6}
3	93.6	75	25×10^{-6}
4	104.4	75.5	27.7×10^{-6}
5	118.8	75.9	31.3×10^{-6}

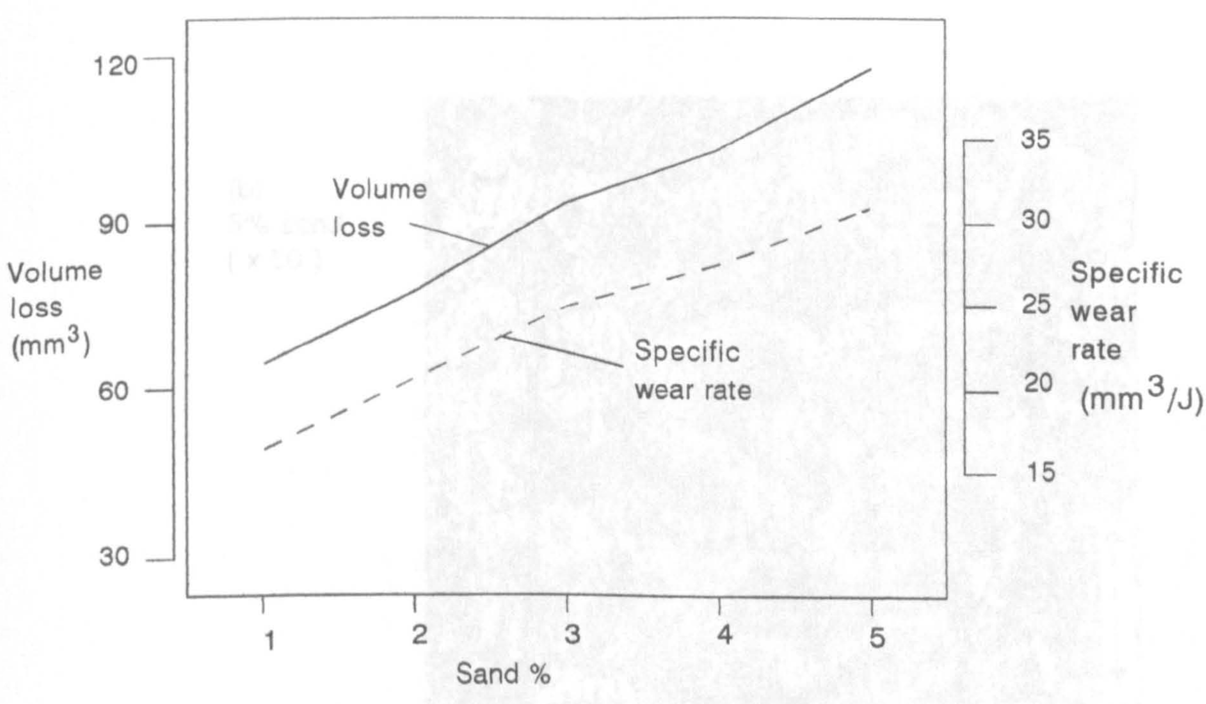
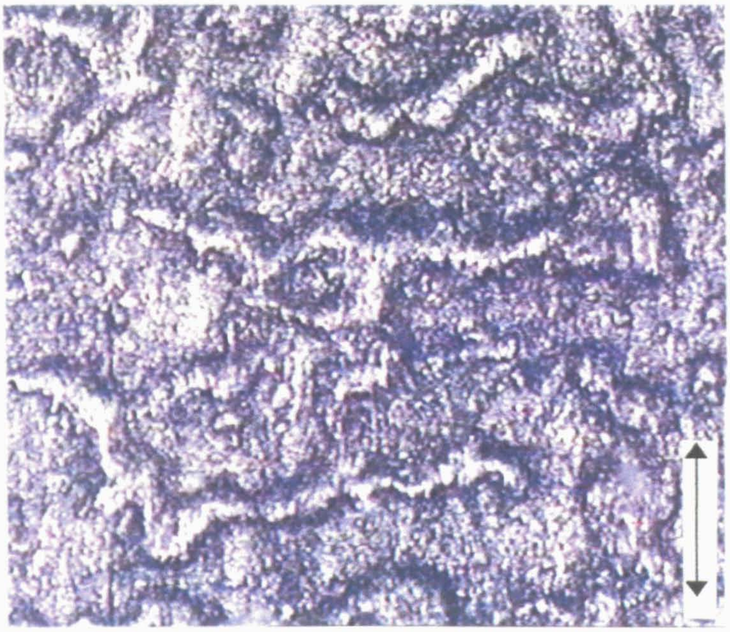
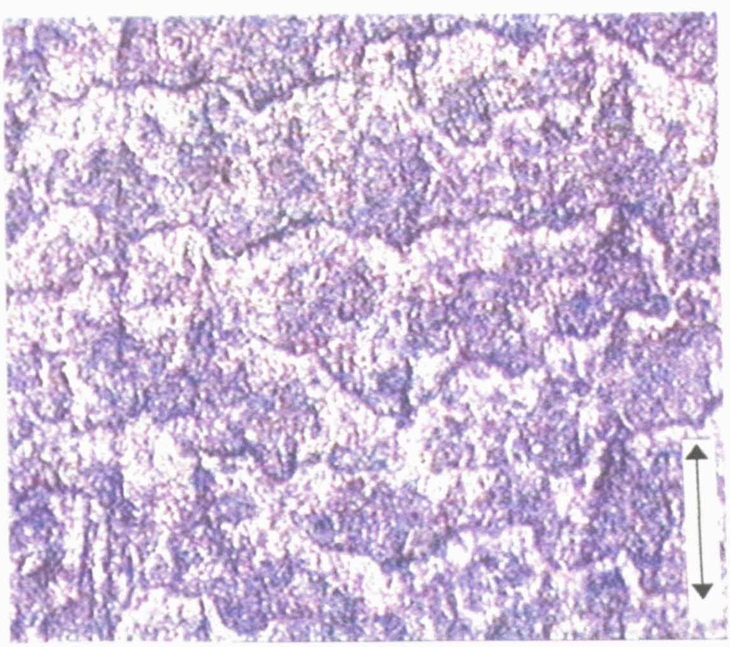


Fig 3.42 Effects of sand on wear surface



(a)
1% sand
(x 50)



(b)
5% sand
(x 50)

Table 3.1 Coefficient $a_{(s)}$ and exponent $b_{(s)}$ for frictional work due to various sand content.

Sand %	Coefficient $a_{(s)}$	Exponent $b_{(s)}$
1	0.00265	0.665
2	0.00265	0.678
3	0.00265	0.69
4	0.00265	0.696
5	0.00265	0.705

3.4.3.2 Load

To investigate the effects of load on wear processes, a series of experimental runs are performed where the load is varied while the other parameters are kept constant. The wear tester is set to run at a speed of 120 strokes per minute (spm) with the temperature set and maintained at 30°C. The drilling fluid is water-based mud with a mud weight of 1.378 kg/lit (11.5 ppg). One percent sand by volume is prepared and mixed with the mud before set to drip from the mud decanter. Five loads are selected ranging from 70N at increments of 20N up to 150N, the latter being the maximum load the rubber specimen can reasonably sustain without obvious deformation or distortion. In field operation, pump operators tend to vary the pump speed to achieve different pressures and flow rates within the liner size. As in earlier experimental runs, the test specimens are allowed to 'run-in' prior to commencing stroke count and measurements for volume loss. The results (Fig 3.43) showed the effect of load on the wear processes. Analysis of the experimental results (Fig 3.44) showed that the frictional work E is related to the wear of the rubber

Fig 3.43 Effects of load on sliding wear

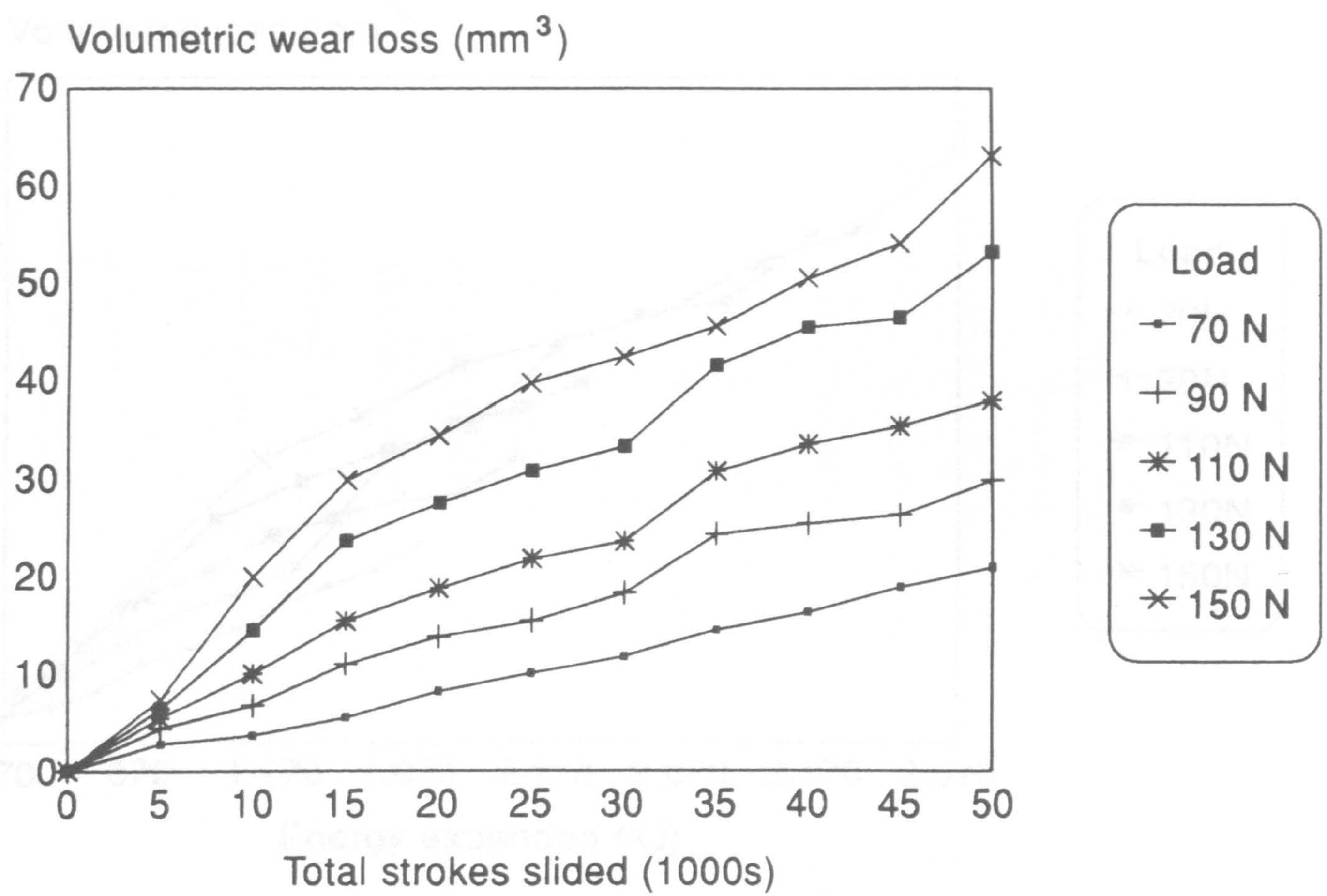
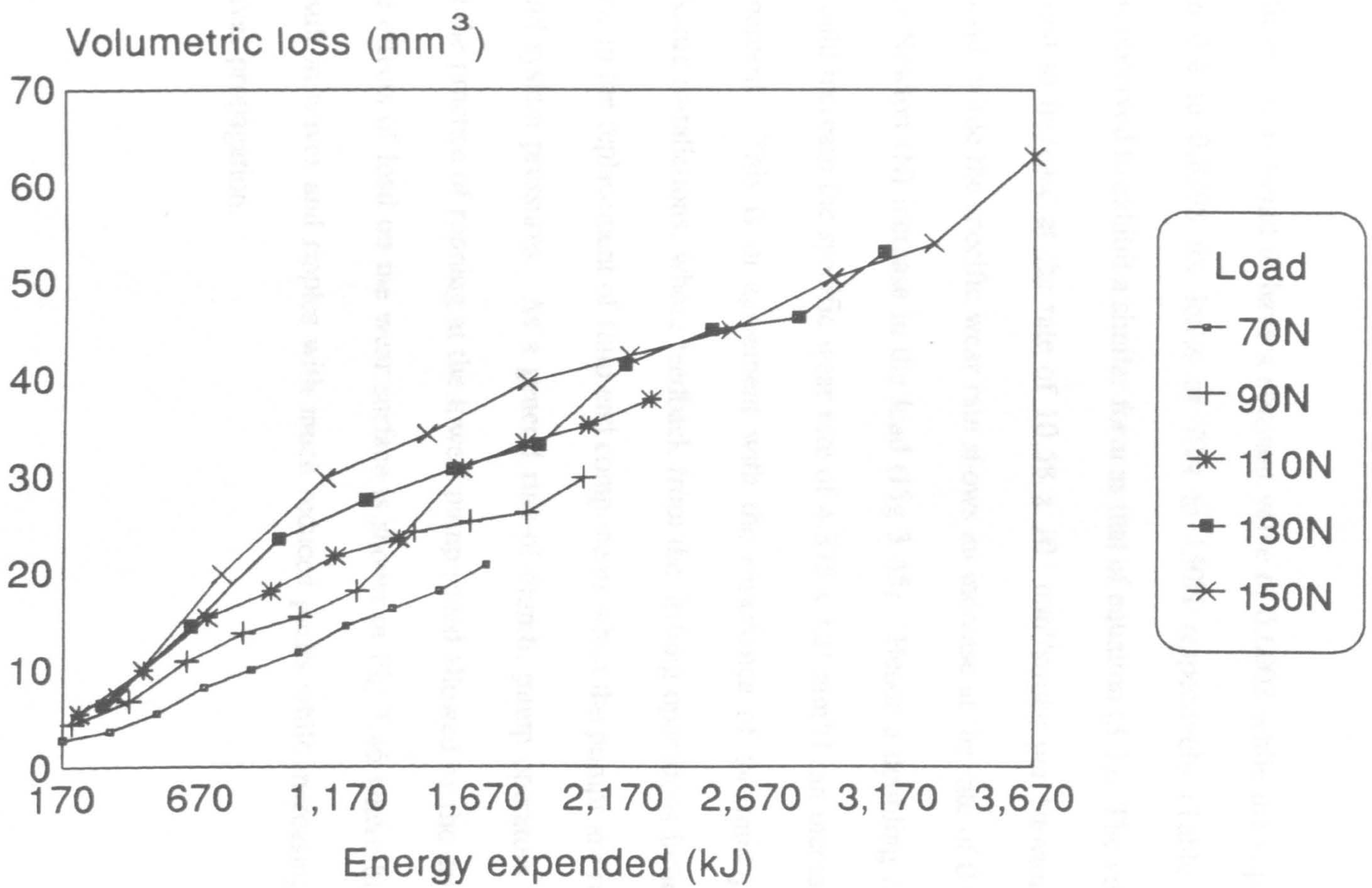


Fig 3.44 Volume loss and energy expended due to load effects



in the form described by :

$$V_{w(l)} = a_{(l)}E^{b_{(l)}} \tag{3.2}$$

where the coefficient $a_{(l)}$ is found to have a constant value of 0.0002 while the exponent $b_{(l)}$ ranges from 0.8 to 0.8398 for loads of 70N to 150N respectively (Table 3.2). Equation (3.2) is observed to exhibit a similar form as that of equation (3.1). The volume wear loss is found to increase at the rate of 10.58 x 10^{-6} mm^3/stroke per Newton (N) increase in the load, while the specific wear rate shows an increase at the rate of 0.0625 x 10^{-6} mm^3/J per Newton (N) increase in the load (Fig 3.45). Hence a doubling of the load to 140N would increase the specific wear rate of 4.375 x 10^{-6} mm^3/J, an increase of more than 35 percent. This is in agreement with the experience of the mud pump operators in offshore installations, whose feedback from the drilling operations indicated a marked increase in the replacement of fluid end components when the pumps are run at higher speeds and system pressures. As a general rule-of-thumb, pump operators have always followed the practice of running at the lowest pump speed allowed by the drilling conditions. The effects of load on the wear surface is shown in Fig 3.46 where higher loads tend to result in waves and ripples with much reduced peaks while increasing the frequency of waves propagation.

Fig 3.45 Load effects on specific wear rate and volume loss

Load	Total volume loss (mm^3)	Energy expended per stroke (J)	Specific wear rate (mm^3/J)
70 N	20.7	34	12×10^{-6}
90 N	27	41	13.2×10^{-6}
110 N	37.8	46	16.1×10^{-6}
130 N	52.2	63	16.6×10^{-6}
150 N	63	74	17×10^{-6}

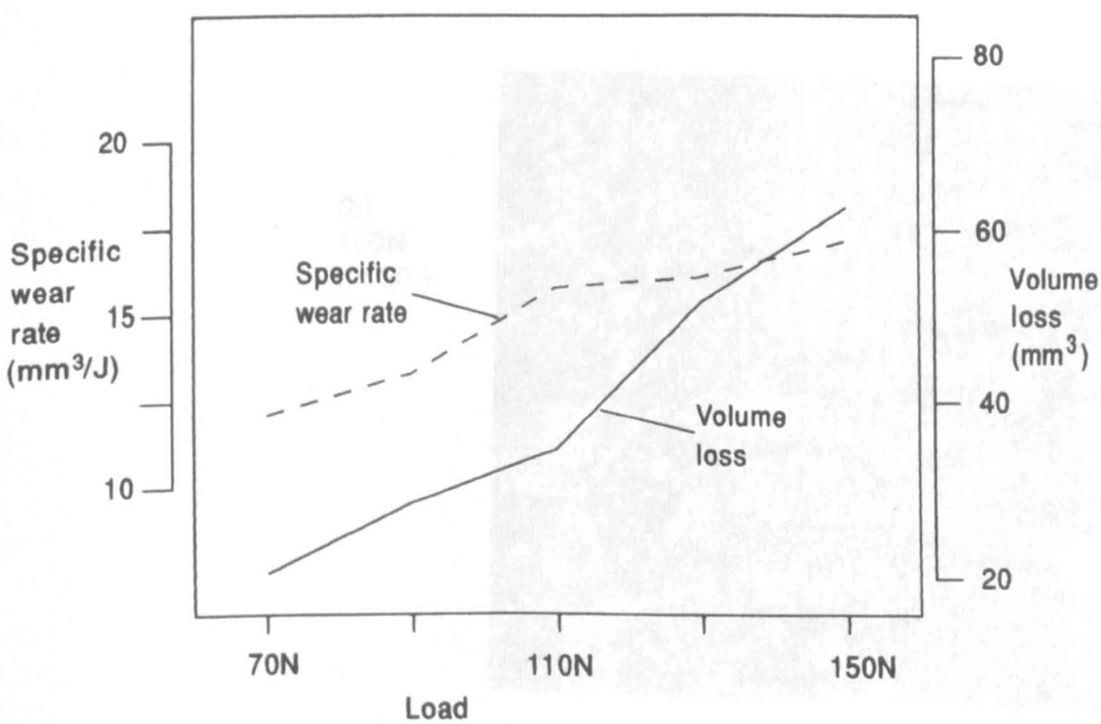
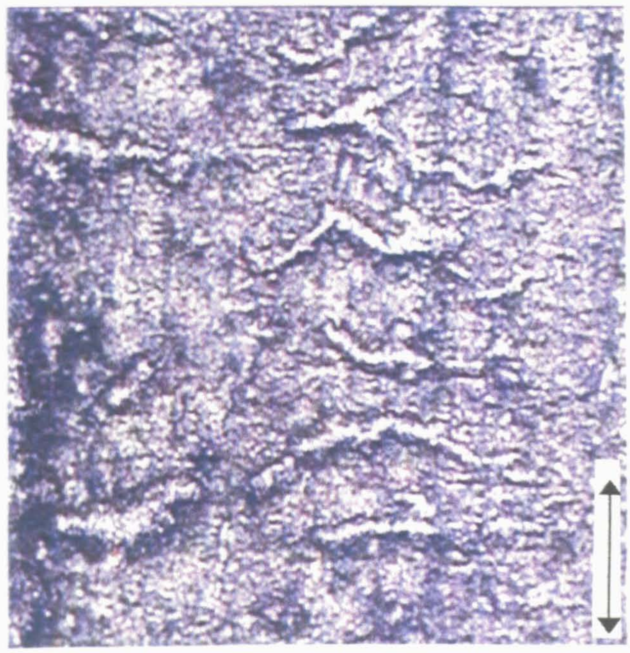


Fig 3.46 Load effects on wear surface

(a)
70N
(x 50)



(b)
150N
(x 50)

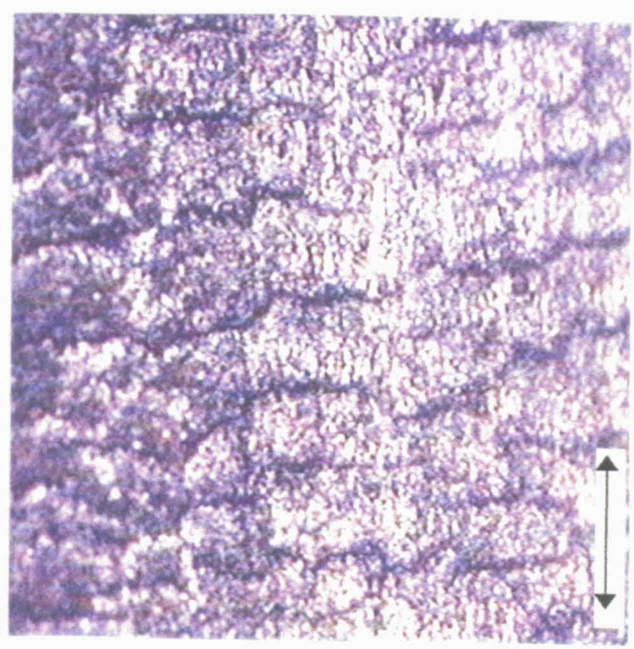


Table 3.2 Coefficient $a_{(l)}$ and exponent $b_{(l)}$ for frictional work due to various loads.

Load	Coefficient $a_{(l)}$	Exponent $b_{(l)}$
70N	0.0002	0.8
90N	0.0002	0.8195
110N	0.0002	0.83
130N	0.0002	0.8375
150N	0.0002	0.8398

3.4.3.3 Temperature

To better understand the effects of temperature on the wear of rubber sliding on steel in the presence of drilling mud, a series of experiments are run with the temperature varied while the other parameters are kept constant. In the context of the experimental set-up, the temperature effect on mud viscosity was considered to be minimal since the mud was frequently replenished and the mud which was exposed to heat was not static. Similarly, in the case of mud pumps in field operation, each stroke of compression and discharge would encounter different a volume of mud. The effect of heat on the mud properties would then not be significant enough to cause any changes in the viscosity. This in no way dispute the effects of temperature on mud viscosity which would apply if the same amount of mud had been static and constantly exposed to heat. In mud pumps, this scenario would not occur. The parameters are a normal load of 150N, a sliding speed of 120 strokes per minute, water-based drilling mud with a mud weight of 1.378 kg/lit, and mixed with the appropriate amount of sand to take the sand content to one percent. The selected temperature ranges from 30°C to 90°C at increments of 15°C. These limits represents the working range of temperatures which most mud pump operators feel are the acceptable norm. The upper limit represents the temperature beyond which the piston rubber starts to show signs of softening perhaps due to material deterioration as a result of the heat. Most pumps have efficient water cooling systems to keep the mud temperatures below a certain limit, of which 70°C is frequently mentioned. For each selected temperature setting, the temperature controller is adjusted to allow the current flow to heat up the silicone mat heater until the desired temperature is obtained. Once the temperature has been verified by a digital thermometer, the capillary thermostat is brought into firm contact with the chrome liner sleeve in order to ensure the temperature is maintained. The procedure is similar to previous sections, with the wear tester allowed to run for 1000 strokes to allow the test specimen to 'run-in' at the set temperature and speed before wear loss measurements are taken at every 5000 strokes. The procedure for obtaining the frictional work per stroke is detailed in Appendix 8.

Analysis of the results (Fig 3.47 & 3.48a) showed that the relation of the frictional work, E, with the volumetric wear loss, $V_{w(t)}$ could be described by :

$$V_{w(t)} = a_{(t)} E^{b_{(t)}} \tag{3.3}$$

where the coefficient $a_{(t)}$ had a constant value of 0.00275 while the exponent $b_{(t)}$ ranged from 0.6645 to 0.69 for temperatures of 30°C to 90°C respectively (refer Table 3.3). Equation (3.3) was observed to be similar in form to equations (3.1) and (3.2). A comparison of the actual volumetric wear loss with the theoretical wear loss obtained using

Fig 3.47 Temperature effects on sliding wear loss

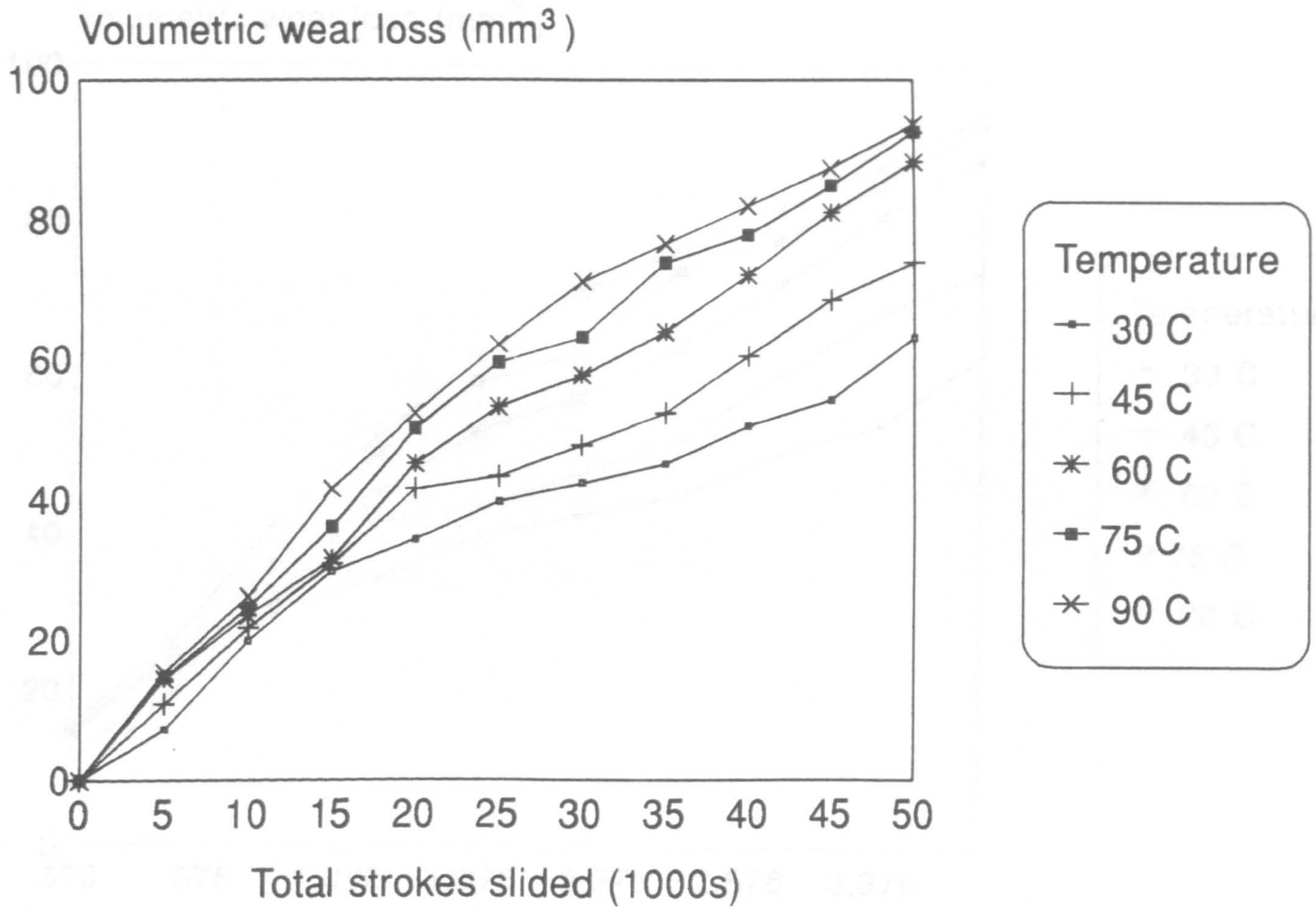
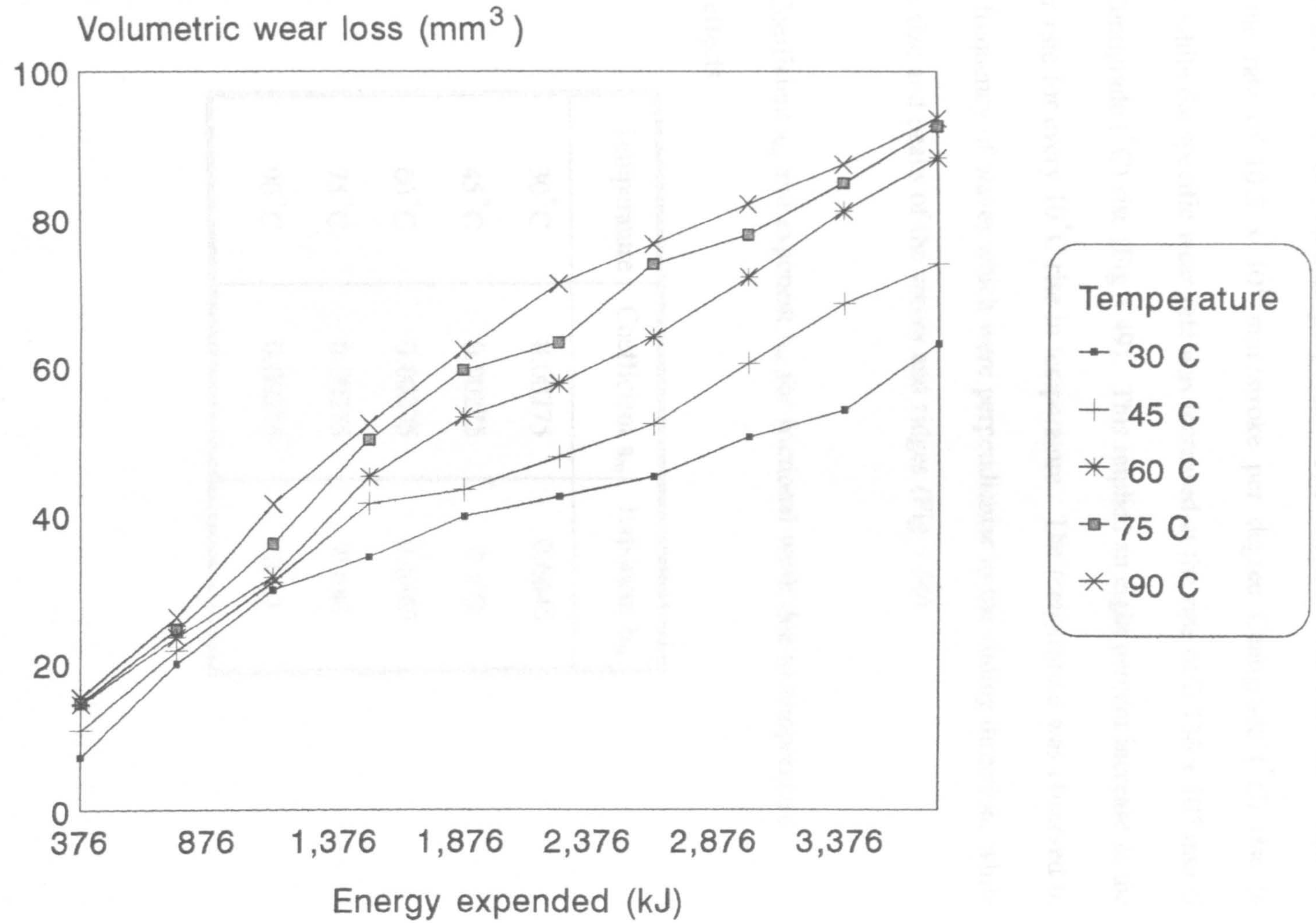


Fig 3.48(a) Energy expended and volume loss due to temperature effects



equation (3.3) as shown in Fig. 3.48b indicated the curve, although not a perfect fit, was deemed sufficient for most practical purposes. The volume wear loss was found to increase at the rate of 10.2 x 10^{-6} mm^3/stroke per degree Centigrade ($^\circ$C) rise in temperature, while the specific wear rate was increased at the rate of 0.136 x 10^{-6} mm^3/J per degree Centigrade ($^\circ$C) rise (Fig 3.49). This implied an eight percent increase in the specific wear rate for every 10°C rise in temperature. The temperature was observed to increase the frequency of waves which were perpendicular to the sliding direction, while reducing the size and peaks of the waves and ridges (Fig 3.50).

Table 3.3 Coefficient $a_{(t)}$ and exponent $b_{(t)}$ for frictional work due to temperature effects

Temperature	Coefficient $a_{(t)}$	Exponent $b_{(t)}$
30°C	0.00275	0.6645
45°C	0.00275	0.672
60°C	0.00275	0.6845
75°C	0.00275	0.689
90°C	0.00275	0.69

Fig 3.48(b) Comparison of actual volume loss due to temperature effects with theoretical model

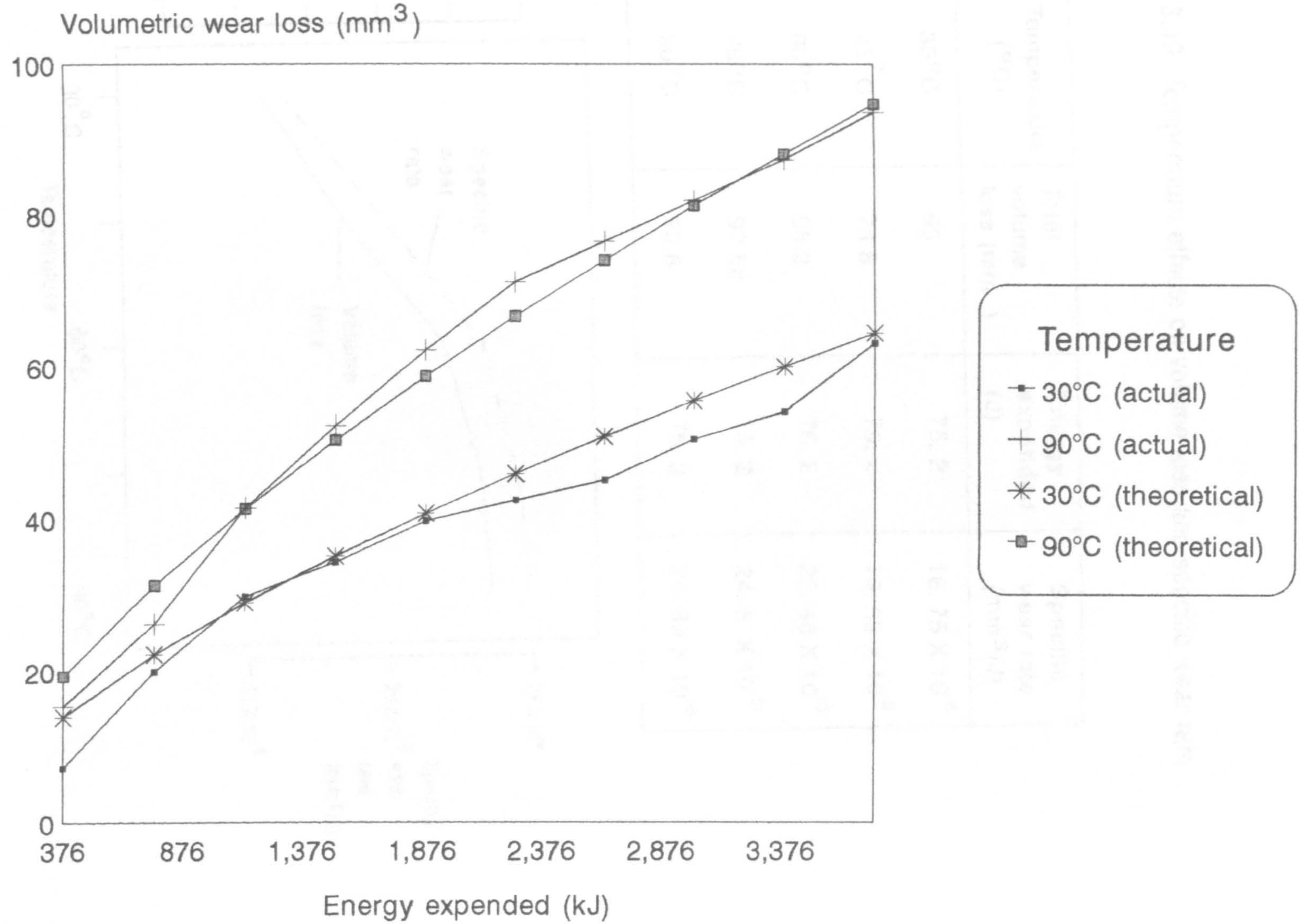


Fig 3.49 Temperature effects on volume loss and specific wear rate

Temperature (°C)	Total volume loss (mm)	Energy expended (J)	Specific wear rate (mm^3/J)
30°C	63	75. 2	16.75×10^{-6}
45°C	73.8	75. 2	19.63×10^{-6}
60°C	88.2	75. 2	23.46×10^{-6}
75°C	92.52	75. 2	24.6×10^{-6}
90°C	93.6	75. 2	24.89×10^{-6}

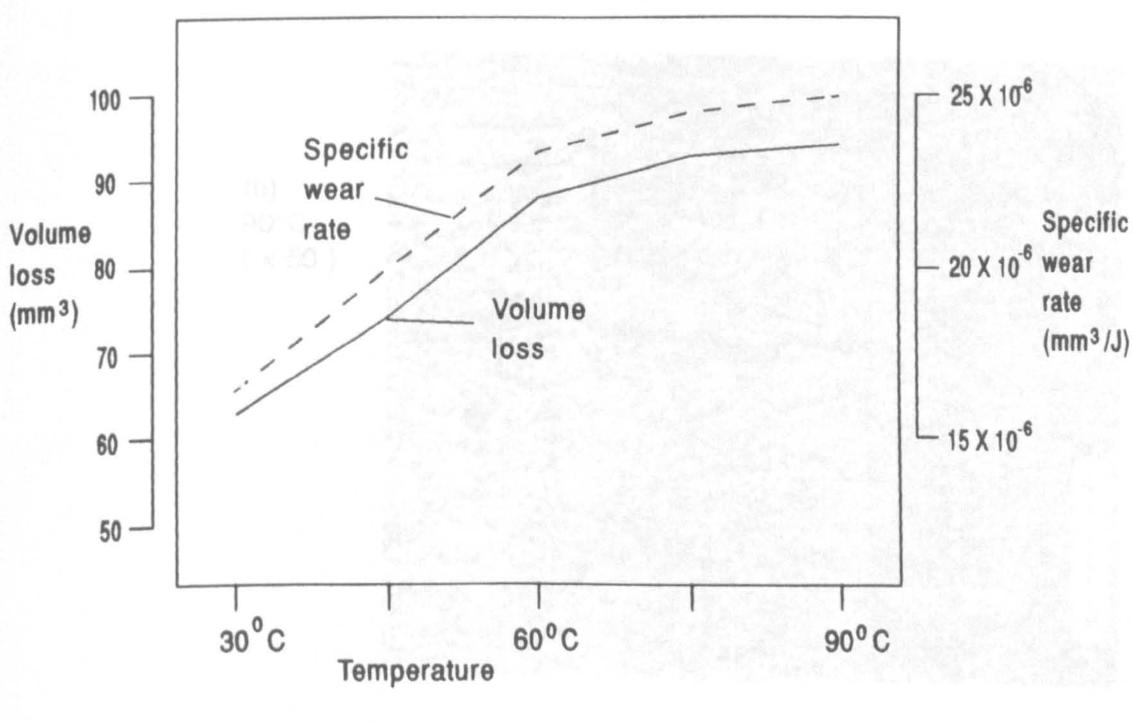
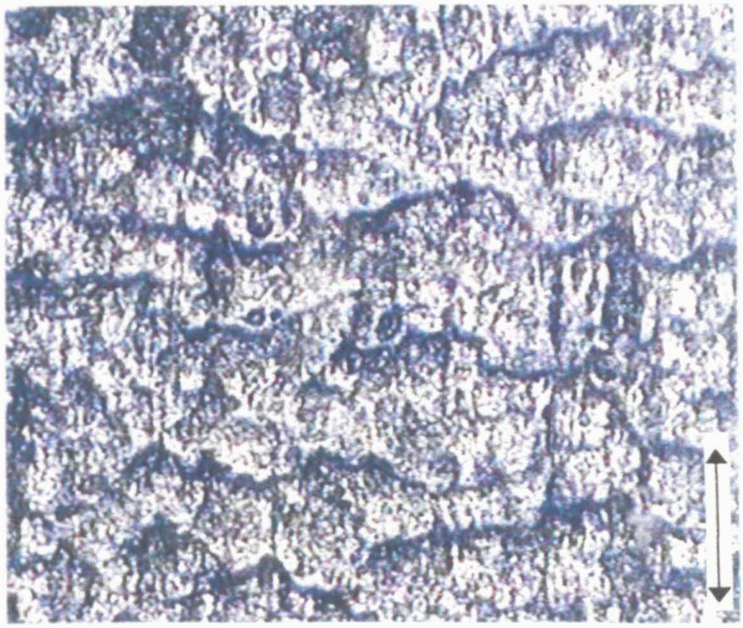
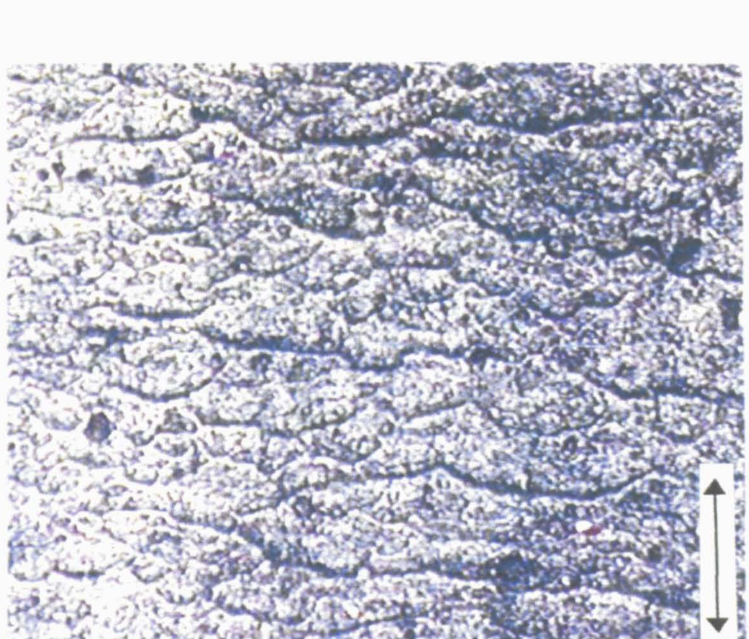


Fig 3.50 Thermal effects on wear surface

(a)
30°C
(x 50)



(b)
90°C
(x 50)



3.4.3.4 Mud viscosity

The experiments in section 3.4.2 did not indicate mud weights as one of the variables which affected wear while sand was identified as one of the variables which did. There was a possibility given that only two mud weights have been considered in the Taguchi trials, the viscosity span had not been sufficient to determine decisively the effect of mud viscosity on wear. Since sand was known to increase the viscosity of mud, it was deemed relevant to look at the effect of mud viscosity on wear processes. Also because the frictional work per stroke was affected, this effect in turn was translated to the specific wear rate. To investigate the effects of mud viscosity on wear processes, a series of experiments were run with both oil-based mud and water-based muds, without the presence of sand. The parameters are a normal load of 150N, a sliding speed of 120 strokes per minute, the temperature maintained at 30°C. The selected mud viscosities ranged from 1.0 to 2.0 with increments of 0.25. These viscosities were in the normal range of mud weights normally encountered in field drilling operations. As with earlier procedures, the wear tester was run for 1000 strokes to 'run-in' the test specimen prior to monitoring the volume loss at every 5,000 strokes. The procedure for obtaining the frictional work per stroke was explained in Appendix 8.

Analysis of the results (Fig 3.51 & 3.52) showed that the relation of the frictional work, E, with the volumetric wear loss, $V_{w(wm)}$, could be descibed by :

$$V_{w(wm)} = a_{(wm)} E^{b_{(wm)}} \tag{3.4}$$

Fig 3.51 Mud viscosity effects on sliding wear

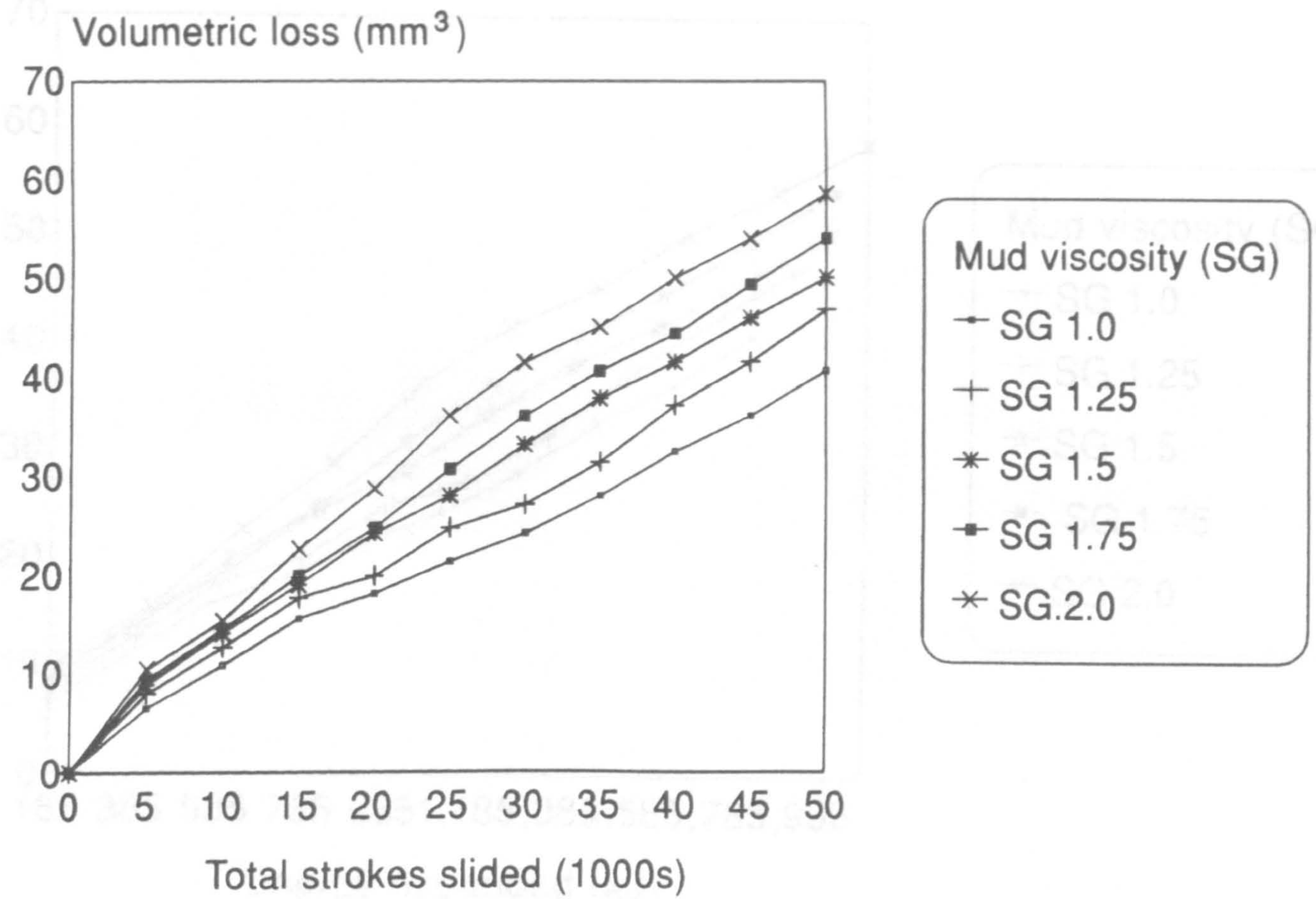
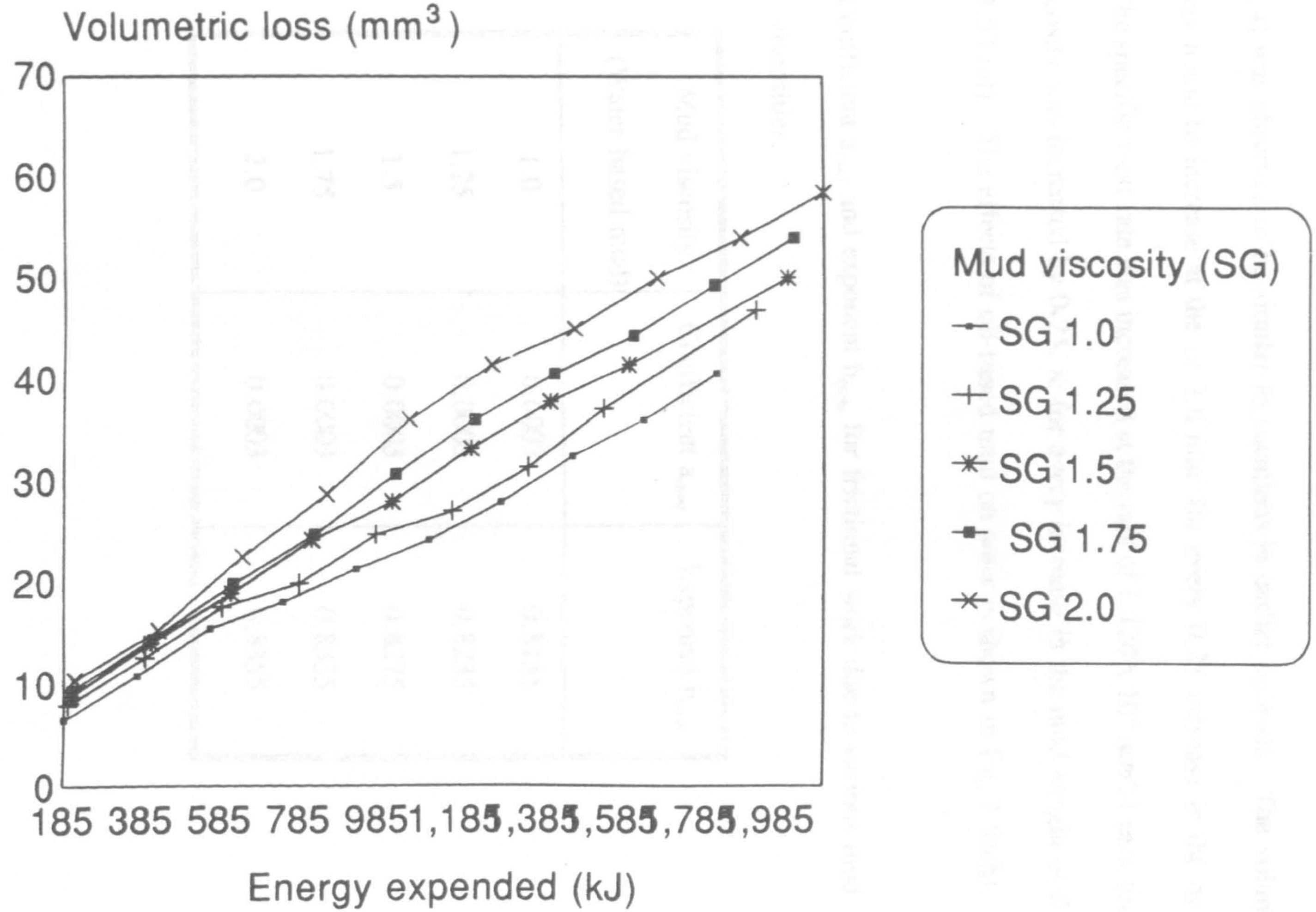


Fig 3.52 Energy expended and volume loss due to mud viscosity effects



where the coefficient $a_{(wm)}$ had a constant value of 0.0003 while the exponent $b_{(wm)}$ ranged from 0.8155 to 0.8355 for mud viscosities of 1.0 to 2.0 respectively (refer Table 3.4). Equation (3.4) was observed to be similar to equations in earlier sections. The volume wear loss was found to increase at the of 3.6 mm^3 for every 0.25 increase in the mud viscosity. The specific wear rate was increased at the rate of 1.128 x 10^{-6} mm^3/J each time the mud viscosity was increased by 0.25, ie for every increase in the mud weight of 0.2 kg/lit (Fig 3.53 (a)). The effects of oil-based mud on wear is shown in Fig 3.53(b).

Table 3.4 Coefficient $a_{(wm)}$ and exponent $b_{(wm)}$ for frictional work due to various mud viscosities.

Mud viscosity (Water-based mud)	Coefficient $a_{(wm)}$	Exponent $b_{(wm)}$
1.0	0.0003	0.8155
1.25	0.0003	0.8235
1.5	0.0003	0.8275
1.75	0.0003	0.8335
2.0	0.0003	0.8355

Fig 3.53(a) Mud viscosity effects on volume loss and specific wear rate

(Water-based mud)

Mud viscosity (SG)	Total volume loss (mm^3)	Energy expended per stroke (J)	Specific wear rate (mm^3/J)
1.0	40.5	37.04	21.87×10^{-6}
1.25	46.8	39.07	23.96×10^{-6}
1.5	50.04	40.71	24.58×10^{-6}
1.75	54	41.03	26.32×10^{-6}
2.0	58.5	42.53	27.51×10^{-6}

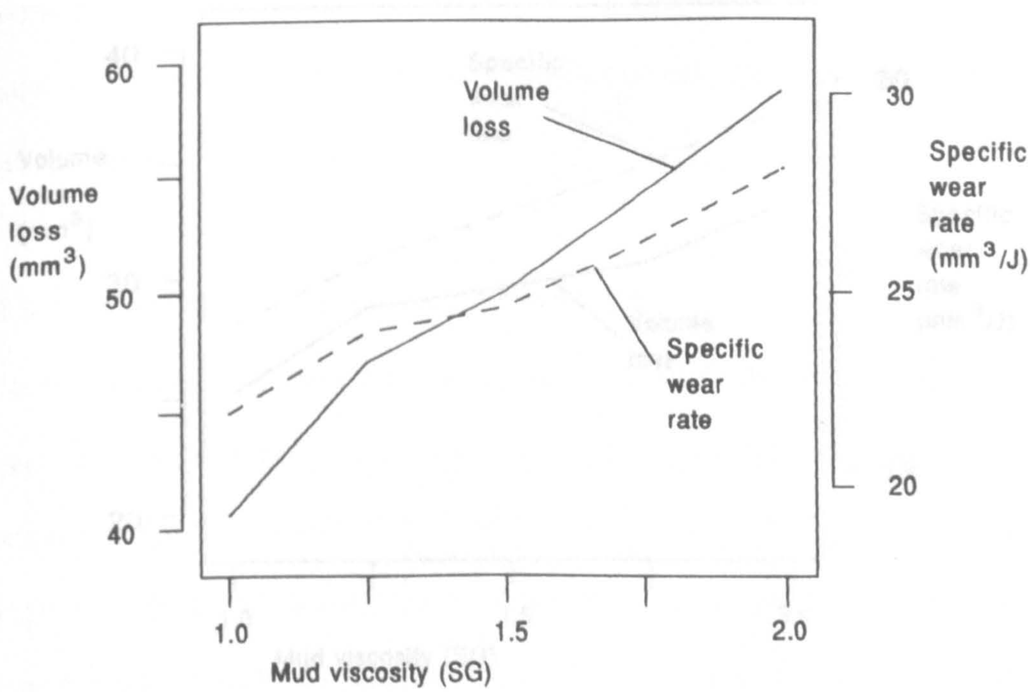
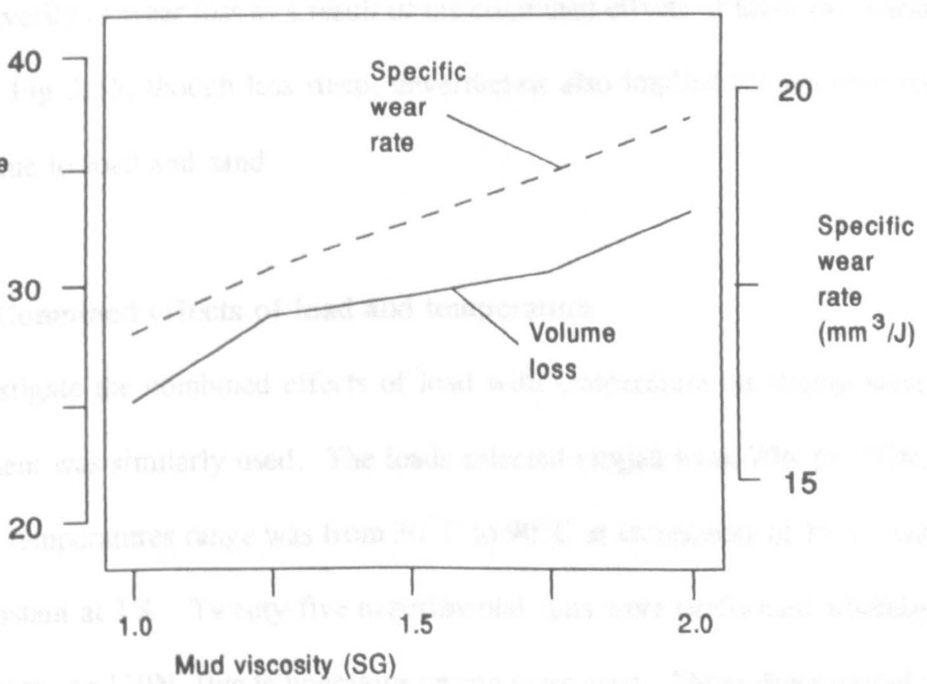


Fig 3.53(b) Mud viscosity effects on volume loss and specific wear rate

(Oil-based mud)

Mud viscosity (SG)	Total volume loss (mm^3)	Energy expended per stroke (J)	Specific wear rate (mm^3/J)
1.0	25.2	30.4	16.8×10^{-6}
1.25	28.44	31.9	17.8×10^{-6}
1.5	29.16	32.3	18.06×10^{-6}
1.75	30.6	33	18.55×10^{-6}
2.0	33.3	34.01	19.58×10^{-6}



3.4.4 Combined effects of load and sand

To investigate the combined effects of load and sand on the wear of rubber sliding on steel, a matrix set of experimental runs were conducted. Loads ranging from 70N to 150N at increments of 20N were used in conjunction with sand contents of 1% to 5%. With a temperature kept constant at 30°C and a sliding speed of 120 strokes per minute, twenty-five experiments were run whereby for every load setting, eg 70N, five runs were performed at different sand contents, and so on. Each experiment was run for 50,000 strokes and the cumulative volume loss and energy expended recorded, to obtain the specific wear rate. Three-dimensional plots of the volume loss against load and sand (Fig 3.54) and that for specific wear rate (Fig 3.55) were prepared. It was observed that the maximum volume loss occurred at the maximum load of 150N and 5% sand content. The steep slope (Fig 3.54) resulting from conditions of 110N and 3% sand onwards pointed to the severity of wear loss as a result of the combined effects of these two variables. The slope in Fig 3.55, though less steep, nevertheless also implied the possible combinatory effects due to load and sand.

3.4.5 Combined effects of load and temperature

To investigate the combined effects of load with temperature on sliding wear, a matrix experiment was similarly used. The loads selected ranged from 70N to 150N, while the selected temperatures range was from 30°C to 90°C at increments of 15°C, with the sand kept constant at 1%. Twenty-five experimental runs were performed whereby for every load setting, eg 110N, five temperature setting were used. Three-dimensional plots of the load and temperature against volume loss (Fig 3.56) and specific wear rate were prepared (Fig 3.57). The combined effects of load and temperature were evident in a three-

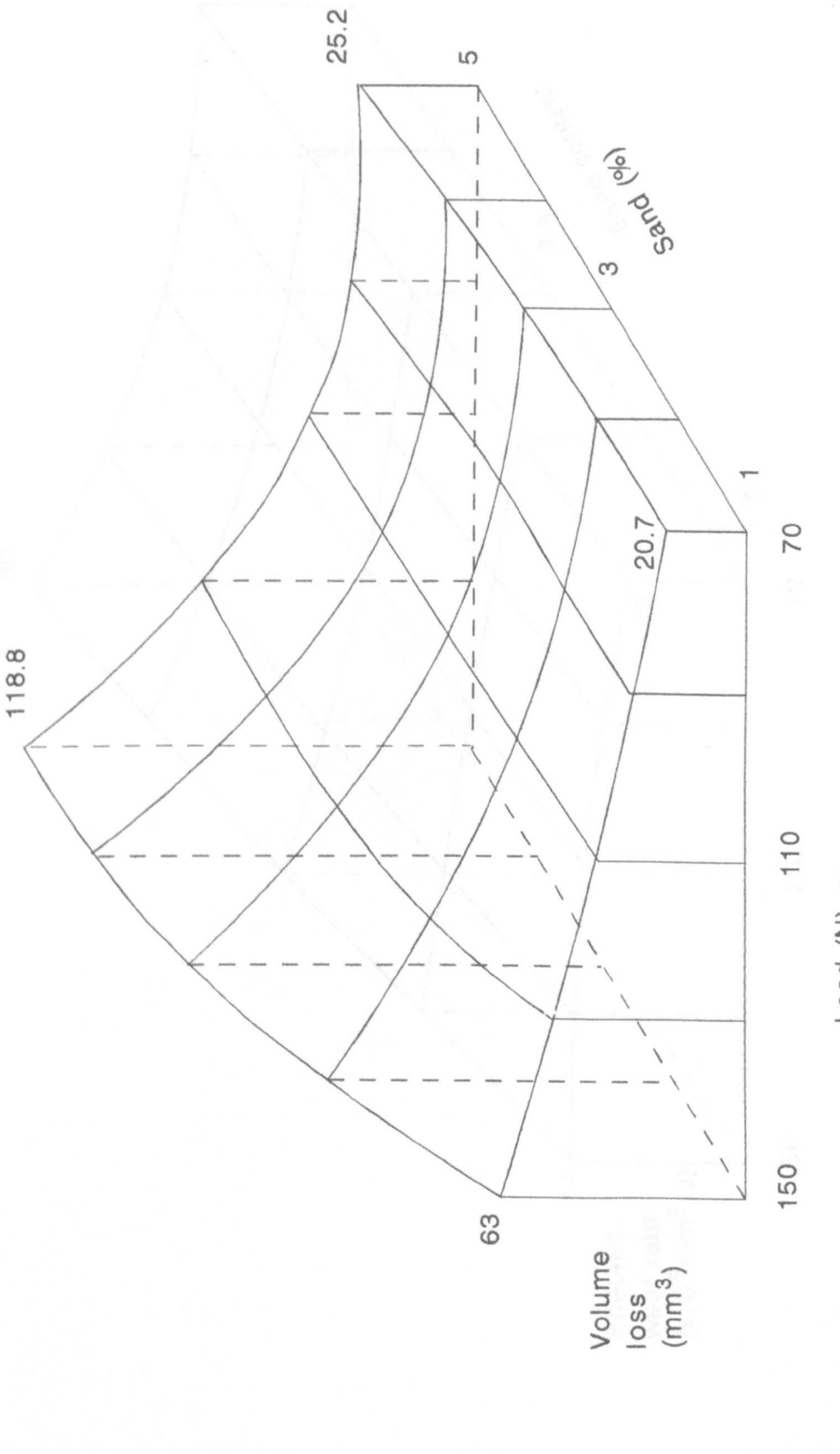


Fig 3.54 Combined effects of load and sand on volume wear loss

Fig 3.55 Combined effects of load and sand on the specific wear rate

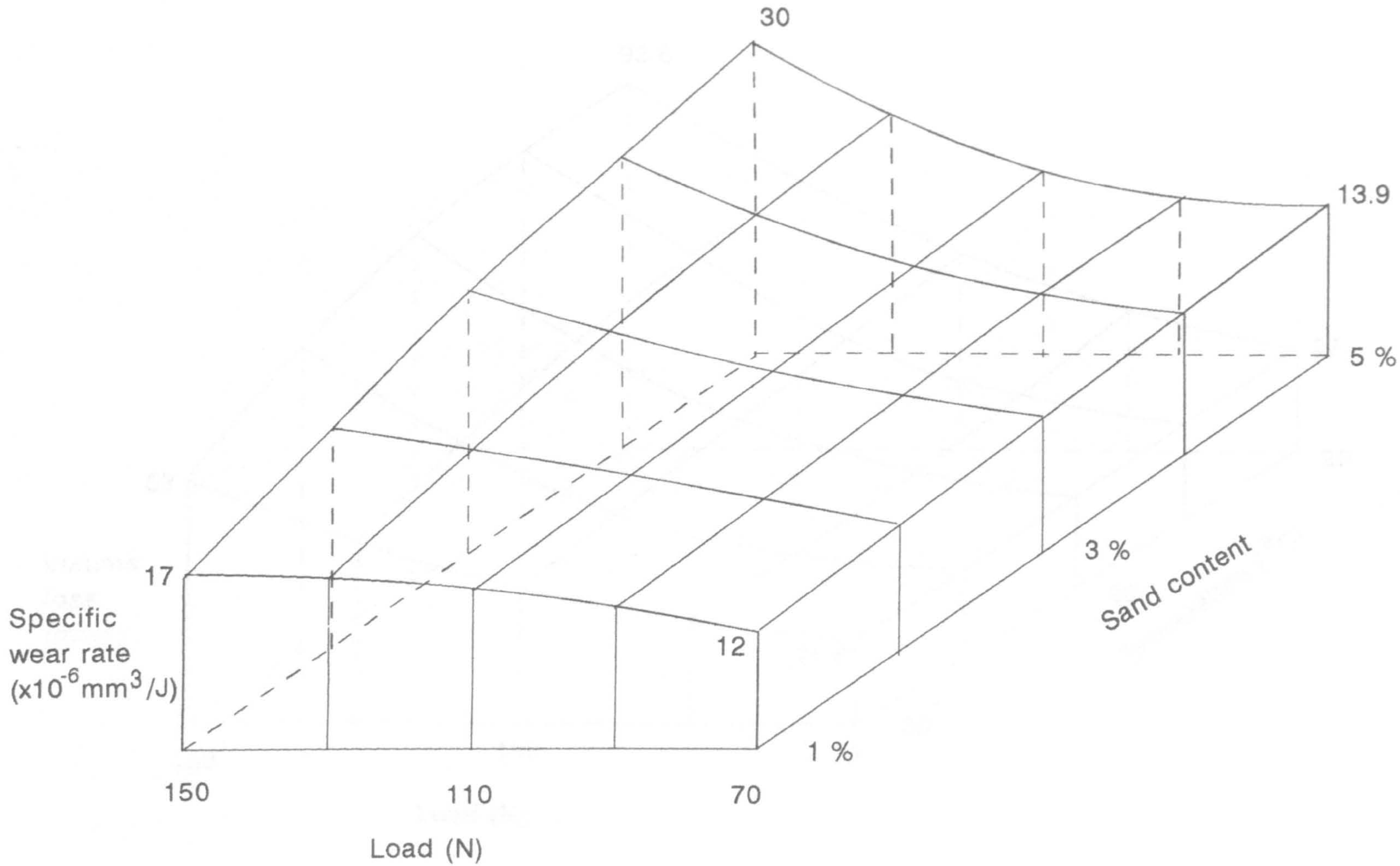


Fig 3.56 Combined effects of load and temperature on volume wear loss

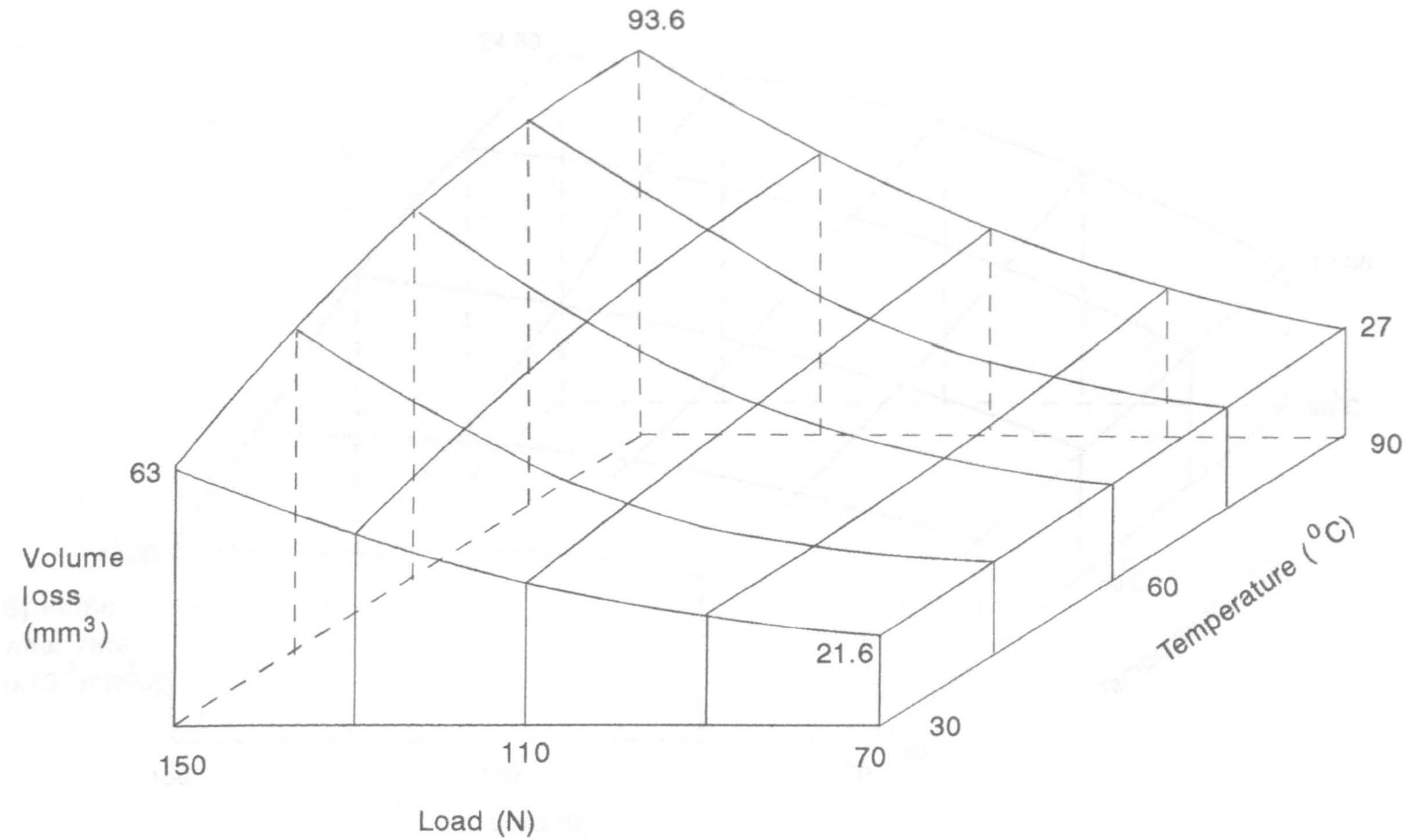
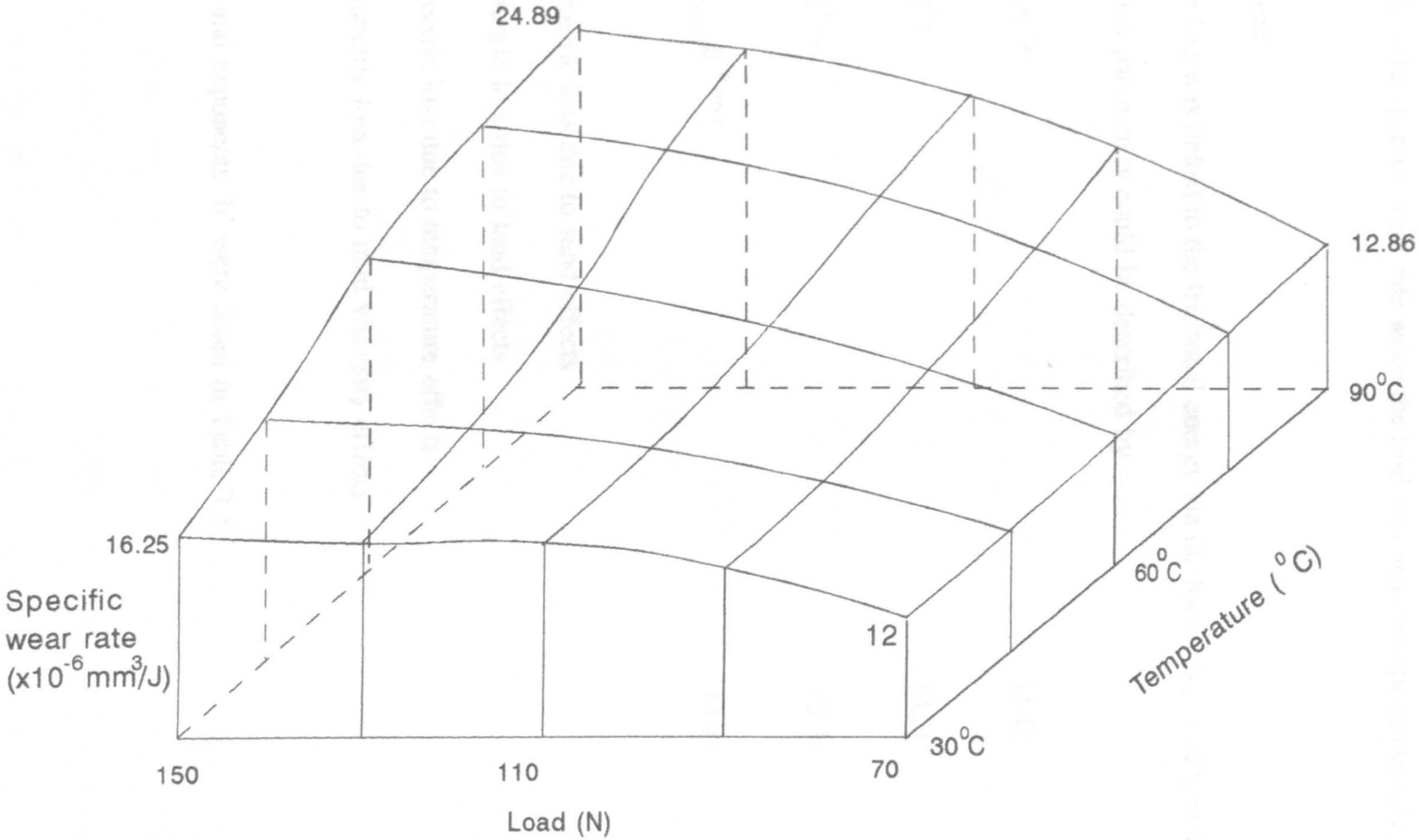


Fig 3.57 Combined effects of load and temperature on the specific wear rate



dimensional plot (Fig 3.56) where increasing the temperature progressively with load resulted in significantly higher volume loss. The slope in Fig 3.57 indicated the effects that temperature had on the specific wear rate when the load was increased progressively.

3.5 Wear coefficient

The volumetric wear loss was linked to the frictional energy via the form : $V_w = aE^b$, and the effect of individual parameters could be described by :

$$V_{w(s)} = a_{(s)}E^{b(s)} \tag{3.1}$$

$$V_{w(l)} = a_{(l)}E^{b(l)} \tag{3.2}$$

$$V_{w(t)} = a_{(t)}E^{b(t)} \tag{3.3}$$

$$V_{w(wm)} = a_{(wm)}E^{b(wm)} \tag{3.4}$$

where $V_{w(s)}$ = volumetric loss due to sand effects

$V_{w(l)}$ = volumetric loss due to load effects

$V_{w(t)}$ = volumetric loss due to temperature effects

$V_{w(wm)}$ = volumetric loss due to mud viscosity effects.

The coefficients 'a' and exponents 'b' were listed in Table 3.5.

Table 3.5 Combined table of coefficients and exponents of different operating variables

Sand % Load(N) Temp(°) Mud SG	Coeff. $a_{(s)}$ (sand)	Expnt. $b_{(s)}$ (sand)	Coeff. $a_{(l)}$ (load)	Expnt. $b_{(l)}$ (load)	Coeff. $a_{(t)}$ (temp.)	Expnt. $b_{(t)}$ (temp.)	Coeff. $a_{(wm)}$ (mud weight)	Expnt. $b_{(wm)}$ (mud weight)
1/70/ 30/1	0.00265	0.665	0.0002	0.8	0.00275	0.6645	0.0003	0.8155
2/90/ 45/1.25	0.00265	0.678	0.0002	0.8195	0.00275	0.672	0.0003	0.8235
3/110/ 60/1.5	0.00265	0.690	0.0002	0.83	0.00275	0.6845	0.0003	0.8275
4/130/ 75/1.75	0.00265	0.698	0.0002	0.8375	0.00275	0.689	0.0003	0.8335
5/150/ 90/2.0	0.00265	0.705	0.0002	0.8398	0.00275	0.69	0.0003	0.8355

3.6 Discussion

The preceding sections had looked at the design and construction of the wear tester and the experiments done, which included comparing the appearance of wear surfaces of test specimens with field samples collected from offshore installations, and identifying the parameters which significantly affected wear and the effects by these variables. The limitation of the wear tester was that it could not simulate pressure in a closed system. This was due to the physical and cost constraints imposed by an actual mud pump. Even a small reciprocating triplex piston pump would require a capital outlay of £25000-30000 for the pump alone, which was beyond the scope of funding for this project. However, initial groundwork had been done on the design of a flow loop which involved the modifications on a small triplex ram pump, capable of pumping up to 13.8 MPa (2000 psi), to accommodate a miniature piston scaled down proportionally to that of an actual sized mud pump piston. This design was not continued to fruition due to constraints imposed by laboratory premises and other factors. The flow loop system would be able to validate the accuracy of the wear coefficients obtained via the wear tester, as well as facilitating the study of the effects of parameters on elastomeric wear under pressure. Further information on this flow loop design can be found in Chapter 5 : Mud flow loop. In field service, the pump pressure and flow could be controlled by varying the speed for the range of pressures which a particular liner size would facilitate. This correlation between pressure and speed could not be quantified with the existing wear test system. However, the wear tester allowed periodic and progressive wear measurement of test specimens, whilst a closed system, with its robust machinery to withstand high pressures, would not be able to facilitate fast and easy access to the test pistons (in a reciprocating pump) for periodic wear measurements. The wear test system represented the fifth complexity level of tribo-testing approaches which investigated the fundamental tribological processes of wet abrasion (see section 2.4.1).

The majority of mud pump operators appeared to be using nitrile budadiene rubber (NBR) or its commercial equivalent, Buna-N, as the material for piston swab, though the alternative material which is currently being marketed and slowly accepted in the North Sea is polyurethane. According to one mud pump manufacturer, the polyurethane piston marketed by some specialised fluid end companies had the compound bonded to the piston, and were primarily designed for liner tail-out service ie for the third piston run in a liner before the liner was replaced. While polyurethane was recognised as having good abrasion resistance, it generally could not be used in situations where mud temperatures were high. The upper limit was about 71°C (160°F) above which it begun to soften. Users of polyurethane pistons must ensure that the cooling system was absolutely efficient and reliable. Because NBR was still the most common piston material, it was considered preferable to investigate the wear characteristics of this material as opposed to polyurethane. One manufacturer of polyurethane pistons saw a need to protect what it consistently considered as proprietary information, rendering it difficult to gather further information on the mechanical properties and what special grades of polyurethane were used.

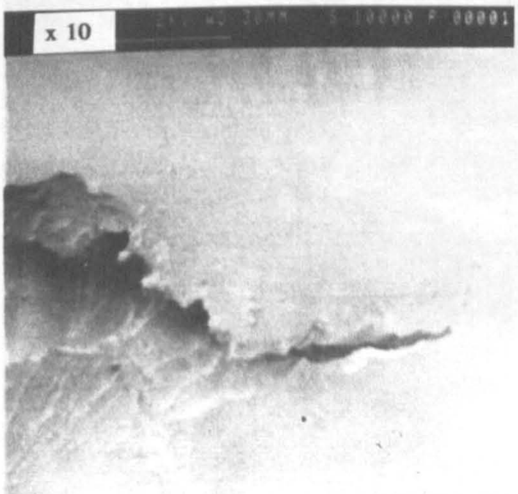
Failure of piston swabs in the context of mud pump operation was categorised as material wear (usually radial) loss to the extent considered sufficient to cause a drop in output pressure and pump performance. This type of failure was generally gradual but the varied nature of reciprocating pump operations meant failure prediction could not be time based but rather condition based and would only be possible if there was some way of quantifying or monitoring the deterioration due to the operational conditions. Catastrophic failure may be due to other factors such as misalignment, which could be attributed to wear in the lower cross-slide, resulting in the lower piston surface rubbing against the liner surface and causing severe metal-to-metal wear. The end result of this severe condition were aberrations and roughened liner surface, causing accelerated wear in the piston swab. The rubber could also cause ploughing and longitudinal grooves which disrupted the sealing action of the swab, resulting in pressure loss during the forward ('pressure') stroke. Although the wear tester looked at the fundamental tribo-processes of wet abrasion, there was insufficient load/pressure to produce the severe aberrations in the 28% chrome steel sleeve to quantify wear due to surface conditions. Since the only surface changes in the chrome sleeve were some very faint lines after more than 1.3 million strokes (see Appendix 9), this was another inherent limitation of the wear test system.

In comparing the appearances of wear surfaces between field and test specimens (see section 3.4.1), the fair closeness of some samples (Fig 3.18 - 3.21) implied the wear test system had achieved the simulation criteria which was appearance of worn surfaces. Some abrasion patterns found on field specimens such as ripples (Fig 3.23) were slightly different from those on test samples (Fig 3.22), possibly because the higher pump pressures had resulted in 'eroding' the peaks of the ridges, with the subsequent 'rounded' peaks which looked like flattened 'domes'. Another interesting feature found on field samples were the 'criss-cross' lines (Fig 3.24) on the radial surface of piston, much like the 'spirals' on cylinder bores after a honing operation. These may be due to the abrasion of wear particles when they became embedded in the softer rubber material and were 'dragged' across the liner surface during the pumping action. Fig 3.25 showed the path of one such wear particle as it ploughed the material surface and changed direction.

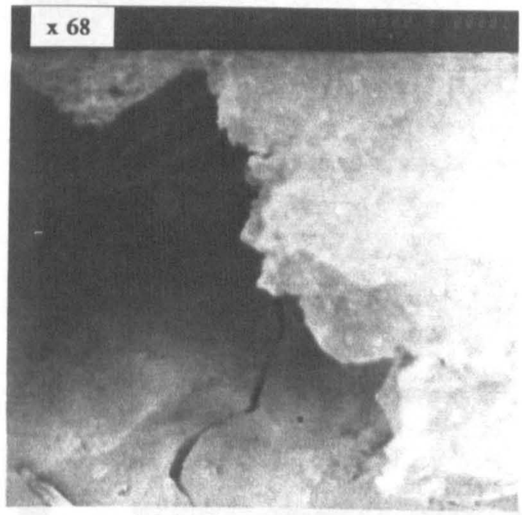
Preliminary work on dry sliding with similar loads and speeds had showed the presence of the three phases of wear [2.10] which may occur simultaneously all over the contact surfaces. In the pre-detachment phase, the dry sliding had resulted in the deterioration of the material, where the surface may be deformed and altered but not necessarily resulting in loss of material, giving rise to crack formation & growth, as exemplified by Fig 3.58 (a) - (d). In the second phase, the detachment of surface material occurred by fracture or smearing by thermal or mechanical degradation of the polymer structure [2.10]. The surface material may detach by delamination (Fig 3.59(a)) or abrasion (b), and in severe wear conditions exhibit abrasion patterns described as waves or ridges (Fig 3.22). In the post-detachment phase, the detached material may be transported in various modes (i) as loose debris; (ii) smearing and transfer through deposition; and (iii) back transfer to the rubber itself. Fig 3.60 exemplified the back transfer and smearing as they occur in dry sliding.

The Taguchi methodology (section 3.4.2), which looked at identifying the operating variables which significantly affect wear, used orthogonal arrays which were termed fractional factorial designs because they were a particular combination of trials from a full factorial. This approach made it possible to perform a few well chosen representative experiments to give a high level of confidence that the experiments were representative of all the conditions encountered. Such an orthogonal array was on average more than 90% efficient as compared to running a full factorial experiment, and this loss in efficiency was more than justified in terms of resources and time-savings. Two sets of response values were selected ie volumetric loss per stroke and energy expended per stroke. The analysis of variance (ANOVA) concluded that the normal load and sand and its interaction significantly affected the average response. The Taguchi trials did not include sliding distance since it was considered axiomatic that the tangential frictional force was

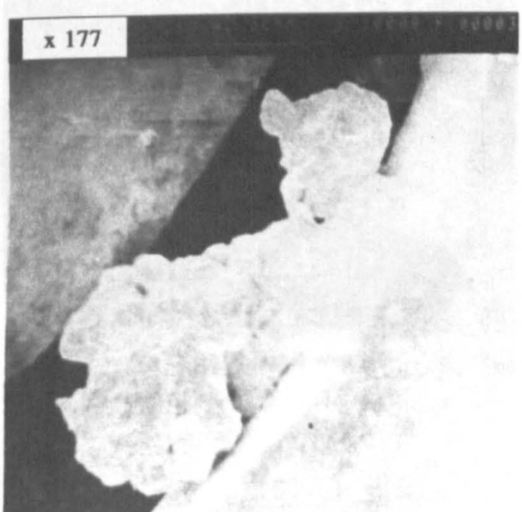
Fig 3.58 Phase 1 - Predetachment phase



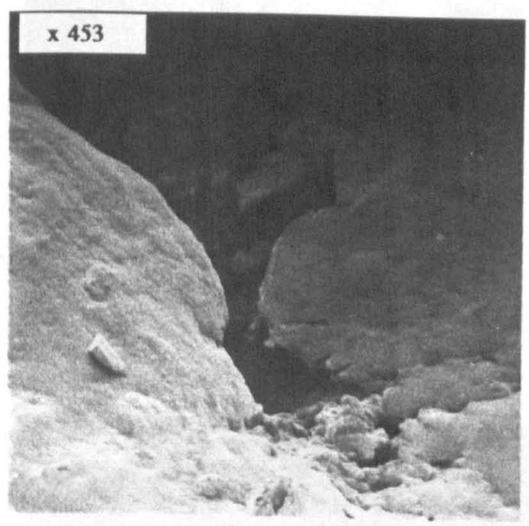
(a) Tearing



(c) Fracture/Cracks

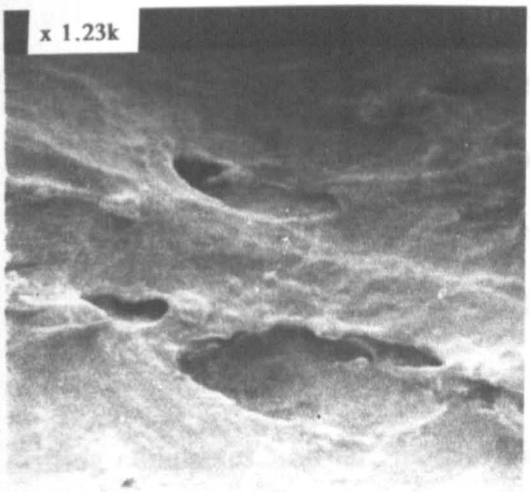


(b) Micro-tearing



(d) Crack nucleation

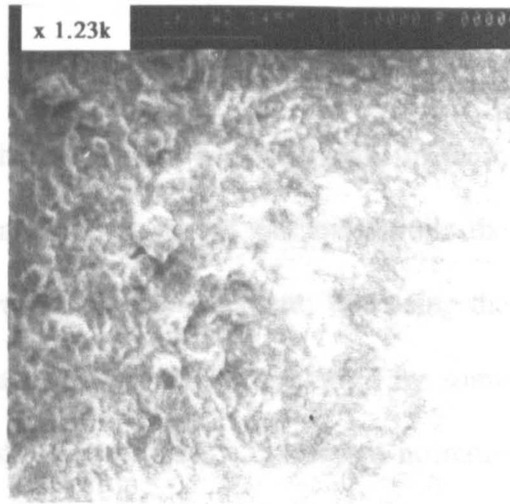
Fig 3.59 Phase 2 : Detachment of surface material



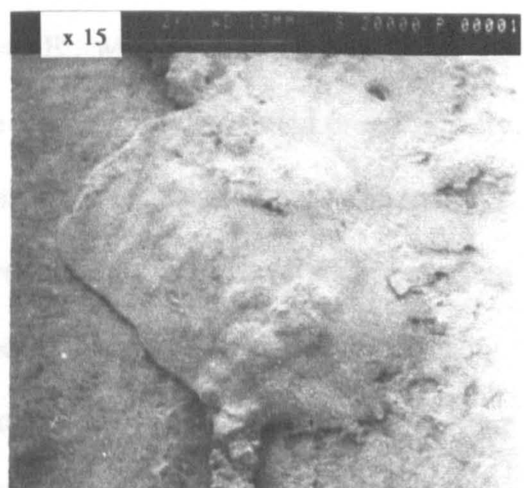
(a) Delamination wear

(b) Abrasive wear

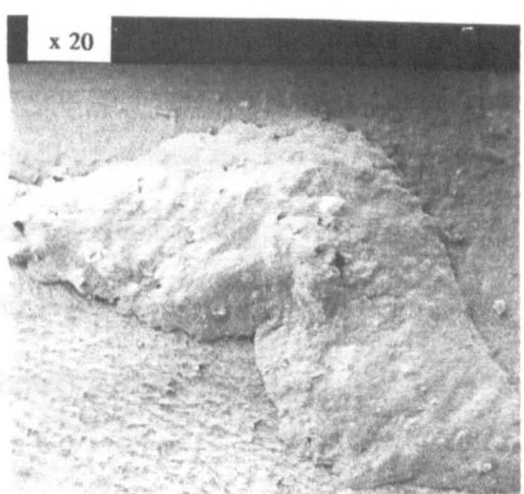
Fig 3.60 Phase 3 : Postdetachment phase
(Transportation of wear debris)



(a) Roll formation by wear fragments



(b) Back transfer/smearing



(c) Back transfer/smearing

proportional to the distance.

On the premise that the factor levels selected at the Taguchi trials were not adequate enough, individual trials were conducted where other variables were fixed at the medium setting while the parameter being investigated was varied. As an example, while the normal load was investigated, the sand content was fixed at one percent, this being the mid-setting between zero percent and the optimum two percent as preferred by some drilling companies. So far, most drilling operation had been known to accept no more than two percent sand, with some drilling companies adopting zero percent sand as the standard. There appeared to be a common correlation between the volumetric loss, V_m, and the energy, E, expended via the form : $V_m = aE^b$, where the coefficient, a, of the parameters remained approximately constant while the exponent, b, varied proportionally as the parameter (see Table 3.5). Hence the exponent 'b' increased from 0.665 to 0.705 as the sand content increased from 1% to 5% respectively. However, each of these parameters were considered to affect the wear process independently and synergistically. This rendered it difficult to establish a single coefficient 'a' to represent all the four parameters, just as the exponent 'b' had been known to vary differently for each of the parameters.

Two matrix experimental sets were performed, which combined the effect of load with sand, and with temperature, and the results showed accelerated volume loss and greater specific wear rate when the load and sand were at the highest setting (Fig 3.54-3.55). The results from the effects of load and temperature showed a similar trend (Fig 3.56-3.57).

During a drilling operation, the pressure and temperature were linked whereby the mud pump speed was varied to control the mud flowrate, changing the frictional load and also the heat generated due to the pumping action. The generated heat was removed by the cooling system and hence was difficult to quantify its effect on the wear process. The mud weight was dependant on the well condition and lithology, while the mud viscosity was dependant on other factors such as the efficiency of the mud conditioning system. The only parameters which could realistically be monitored irrespective of the drilling operations were the pump pressure and sand content, and the pump speed and/or cumulative strokes. One approach of tracking failure is to monitor the cumulative strokes and pump pressure, and identify the trends which indicate piston failures occurring due to what pressures and cumulative strokes, and in the presence of what sand content. A proposed novel approach, detailed in Chapter 4, analyses the pump data from the field and explores the possibility of identifying failure trends which are tagged to a proposed new working unit, based on the field data analysis, which is applicable for offshore drilling operations.

Chapter 4

FIELD DATA ANALYSIS

4.1 Introduction

The focus of experimentation discussed in Chapter 3 : Experimental had been on the investigation of fundamental tribological processes and represented the fifth level of complexity of tribo-testing approaches, as outlined in section 2.4.1. It had been shown that limitations were encountered at this level of complexity as regards the understanding of the tribological processes. One of the constraints was the inability of the wear test system to simulate flow of pressurised fluid to facilitate the study of the wear characteristics of fluid end components. To address this issue, efforts had been initiated to analyze the information contained in the records of mud pumps operating on offshore installations. This initiative represented the first complexity level of tribo-testing approaches mentioned in section 2.4.1, whereby the 'field tests' were carried out on actual mud pump systems operating under actual drilling conditions. These drilling activities were carried out on different reservoirs and would have encountered different lithology and rock formations. Irrespective of the effect of rock formations on component wear or failure, the main objective in this procedure had been to identify the trends and failure patterns of typical fluid end components, especially the liners and piston rubbers (swabs), through the use of a novel measurement unit, called Pressure-strokes (PS), which is similar to the way PV (pressure-velocity) factors were derived from pressure and velocity to form a relative unit. PV factors had been used in various forms as illustrated by the following examples : Fuchsluger and Vandusen [4.1] reported using PV correlation to predict wear of compression ring life in compressors. In the PV test method used to establish the effectiveness of seal face materials, Johnson and Schoenherr [4.2] defined PV as the friction power per unit area. They suggested, in the context of considering material wear as a fundamental process, that basic measurements such as volumetric wear, load and sliding distance, used in the wear coefficient, k, can be modified algebraically to allow the use of PV data from seal tests. Hence the wear coefficient, K, expressed as :

$$K = \frac{WH}{Ld} \tag{4.1}$$

where K = wear coefficient

W = volumetric wear

H = hardness

L = imposed load

d = sliding distance,

could be changed to the form :

$$K = \frac{hH}{t\,(PV)} = \frac{linear\ wear}{(time)} \times \frac{hardness}{PV} \tag{4.2}$$

where h = linear wear = W (volumetric wear)/A (contact area)

t = time

PV = pressure x velocity.

Glaeser [4.3] described how the PV factor had been used as a rough estimation of heat input to a bushing due to friction. Suh [4.4] stated that though the PV limit had been used to designate the commencement of the rapid deterioration of polymeric bearings, the PV limit was not constant and depended on the specific load and specific velocity. Hence the PV limit would have to be 'specified in terms of a limiting load at a given sliding speed or in terms of a limiting speed at a given load.'

'Pressure-strokes' (PS) would be specific to mud pump operation and had been adapted as an alternative to energy units, in view of the existing format of pump records which were almost universally similar on most, if not all, drilling installations. PS values could be used as a rough estimate of total material deterioration. Just as PV factors could be used to predict the onset of material failure, PS values could also be used to pinpoint the 'failure point' of fluid end components, ie the point at which the components exhibited such rapid deterioration as to warrant replacement. One possible difference as compared with PV factors was that the resultant PS values had been derived from the analysis of the pump records for actual fluid end components operating within the genuine conditions. The 'failure point', once it had been established, in terms of PS values, would facilitate the next logical step which would be to predict the component failures by monitoring the pressure and using a simple mathematical means to obtain the value of the cumulative strokes which would lead to component failure. The pump pressure and running speed may vary greatly during a drilling phase, but for the duration of each of the two daily 12-hour drilling shifts, these two operating conditions had been known to be fairly constant. These operational characteristics of the mud pump would benefit the PS method of monitoring the pressure-strokes since the cumulative material wear would be directly related to the work done on the component by the mud, and not dependent on the cumulative time.

This approach of using PS, instead of computing energy units eg Joules, would negate the need to install additional instrumentation to the mud pumps, an activity which was discouraged due to operational safety considerations and the need to impose minimum interruptions to the drilling operations. This had been dictated by the competitive and critical nature of the drilling business which allowed little room for manoeuvre as regards field monitoring of mud pump performance and component wear and failure. One limitation of this approach had been the difficulty of accurately monitoring the installation and replacement of each and every fluid end component. This had been complicated by the fact that the nature of drilling operations required the occasion use of different liner sizes during certain drilling phases. Some drilling contractors had been known to change-out the fluid end components and store them for future use if deemed insufficiently worn to warrant classifying them as component failures.

Pressure-strokes (PS) was to be derived from the discharge pressure and cumulative strokes of the mud pump. Both these data could be obtained from the pump records which were periodically updated along with the daily drilling records. Because of the need to maintain the confidentiality of the respective oil fields on which drilling operations had been carried out, there would be no mention of any oil fields nor offshore installations from which the mud pump records were obtained.

Within the limitations of this approach, the resultant PS values for swab failures with and without sand would signal the preliminary conclusion of the first phase of this method of predicting fluid end component failure. Further development work would be required to fine-tune this strategy and it was hoped that oil operators and/or drilling companies could be persuaded to be involved in the next phase of this method.

4.2 Analysis and correlation of field data

Most mud pump records include running hours, pump speed in strokes per minute (spm), pump discharge pressure, usually taken from the standpipe and the sand content. These records were entered twice daily. Although the pump speed may had been varied through the duration of the 12-hour shift, feedback from offshore mud pump operators have concluded that the pump speed do not generally deviate much from the set speed selected for that drilling phase. This was particularly true for continuous and stable drilling conditions where the mud flow rate must be maintained. It should be noted that the pump speeds were changed in order to obtain varying flow rate and mud pressure. Much data on offshore installations seemed to be still in Imperial units, and mud pump pressures were no exception, as exemplified in Fig 4.1. Hence for the purpose of analysis, the units on the pump records for pressure had to be changed from 'PSI' to 'MPa' prior to the analysis

Fig 4.1 Sample of mud pump record

MUD PUMP RECORD

OP 148

RIG ________ PUMP ________

POWER END

#1 #2 #3

GENERAL INFORMATION									#1 CYLINDER				#2 CYLINDER				#3 CYLINDER						
DATE	TOUR	CONTINUOUS HRS.	HRS. BETWEEN CHNG.	PUMP PSI	PUMP SPM	SAND CONTENT	WEIGHT	TEMP.	LINER	PISTON	SUCTION VALVE	DISCHARGE VALVE	LINER	PISTON	SUCTION VALVE	DISCHARGE VALVE	LINER	PISTON	SUCTION VALVE	DISCHARGE VALVE	PARTS USED	COST	REMARKS
15-9-92	JC DAY								Pump NOT Run														
	RM NIGHT	15538	4	800	75		S/W		26	26	328	328	26	26	328	328	26	26	328	328			
16-9-92	JC DAY								Pump NOT Run														
	RM NIGHT	1559	1	500	50		S/W		27	27	329	329	27	27	329	329	27	27	329	329			WASHING DOWN PRIOR TO TYING BACK 13⅜
17-9-92	RM DAY								PUMP				NOT				RUN						
	NIGHT								Pump				NOT				RUN						
18-9-92	RM DAY								PUMP				NOT				RUN						
	PD NIGHT								Pump				NOT				RUN						
19-9-92	RM DAY	15548	9	2400	96	.75	9.6	28	36	36	338	338	36	36	338	338	36	36	338	338			drill out 16" hole water base polymer mud
	PD NIGHT	15557	11	2500	90	.75	10-1		47	47	349	349	47	47	349	349	47	47	349	349			DRILLING. WATERBASE MUD.
20-9-92	RM DAY	15567	8	2450	90	.50	10.0	122	55	55	357	357	55	55	357	357	55	55	357	857			" " "
	PD NIGHT	15579	12	2500	90	.5	9.9	52°C	67	67	369	369	67	67	369	369	67	67	369	369			" " "
21-9-92	RM DAY	15591	12	2400	90	.5	9.7	125	79	79	381	381	79	79	381	381	79	79	381	381			" " "
	PD NIGHT	15597	6	2500	90	.5	9.7	125	85	85	387	387	85	85	387	3.67	85	85	3.61	387			" " "
TOTAL																							

process. The strokes remained unchanged, being dimensionless unless needed to calculate the sliding distance. The cumulative strokes could be obtained from manipulation of the data in the records for running hours and speed, expressed in strokes per minute (spm). Hence the PS units expressed in 'MPa.strokes' were obtained via the equation :

PS = (MPa) x (strokes/minute x hours x 60) = Pressure.strokes

By monitoring the cumulative PS units for change of liners and/or pistons, it would be possible to determine the PS value for the service life of the fluid end components. If the pump records included sand content, the PS values were obtained and correlated for the amount of sand, usually expressed in percentages.

The gist of the pump records had been extracted and analyzed to identify the failure trends and pinpoint the average of the most common PS values of component replacement ie the 'failure point'. From the average PS values obtained due to the analysis of the pump records which covered a time period of four years from three installations and involving six pumps, a plot of the pressure and strokes would permit the definition of a 'PS' band which allowed a known pressure to predict the cumulative strokes deemed to result in component replacement. By monitoring the pressure, this facility should be able to allow 'failure points' to be estimated via the cumulative strokes, and at what sand content.

4.3.1 'Pressure-strokes' (PS)

This method of using pump records to determine failure points of fluid end components would bound to have some assumptions and limitations, and they included :

- some liners/swabs were changed when required to provide a different pressure range or flow rate - ie the changed component may still have some run life remaining.

- a new drilling phase may require continuous pump service; some drilling contractors have been know to opt for the 'safer option' of changing to new parts prior to start of the drilling, so it may not be able to pinpoint accurately the 'failure point'.
- most, if not all, drilling contractors used running hours as means of monitoring component run life. However this method did not account for the different pressures operating within the component run life. This implied that the varying degrees of material deterioration due to wear were not considered. Hence a period of 100 hours with the pump running at low pressure would be taken to have the same wear as that of another period of 100 hours of the pump running at high pressures. The advantage of using the PS methodology would be its ability to account for effect of pressure on wear since the PS computation had to include pressure as well as strokes.

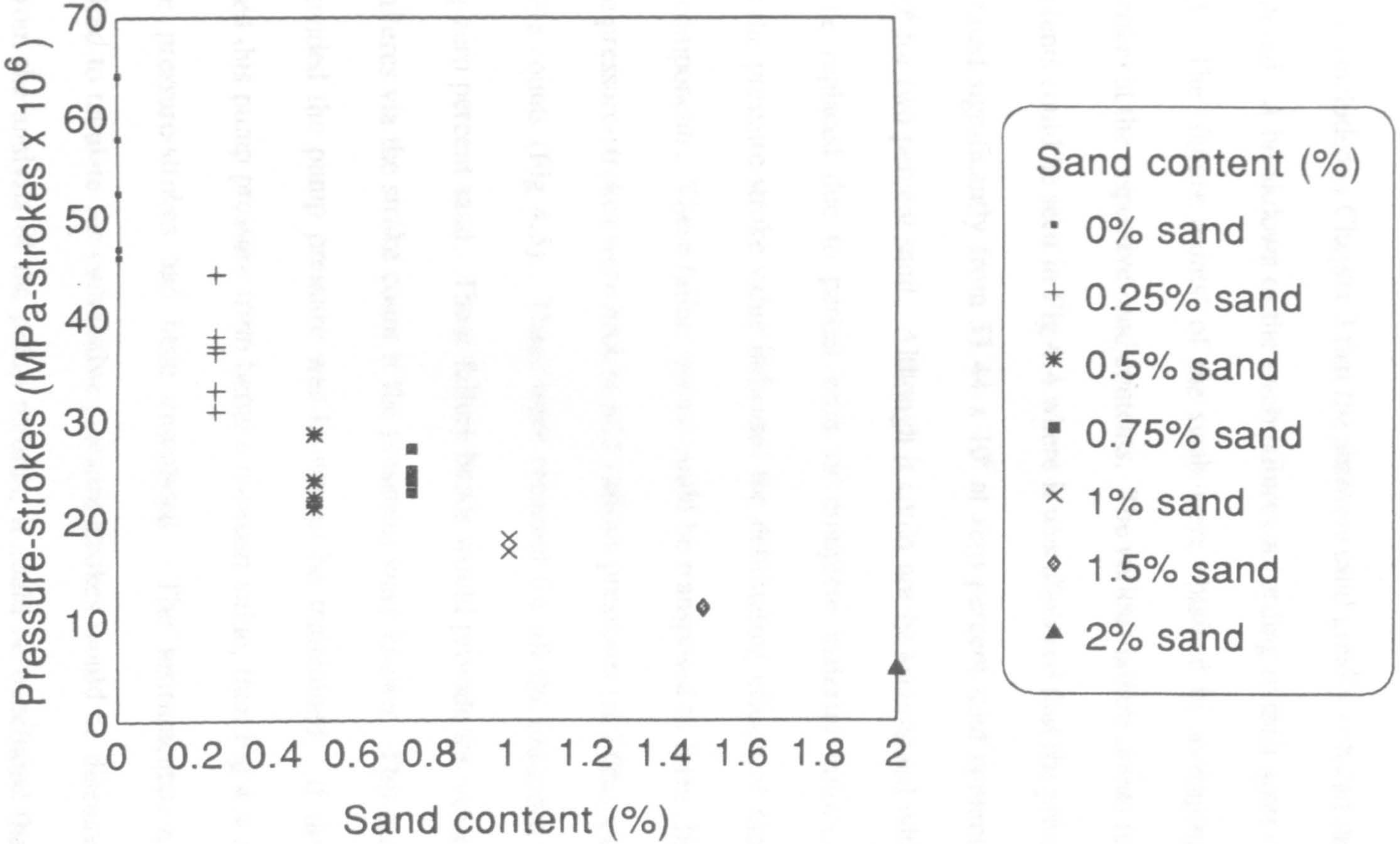
4.3.2 'Failure points'

The analysis of the PS values were concentrated mainly on the piston, and the piston rubber or 'swab' in particular. This was because there were almost three times more swab changes for each liner replacement. The small amount of PS data for liners would not facilitate a reasonable 'failure point' to be derived conclusively. The same logic applied for the other fluid end components including valves and valve seats.

A summary of the failure frequency of swabs according to the amount of sand could be seen in Fig 4.2. The two most common swab sizes occurring in the pump records ie 152mm and 165mm diameter, had been considered together. It should be pointed out that

Fig 4.2 Failure frequency of swabs vs sand content

(152 mm and 165 mm swab size)



the maximum pressure-strokes occurred when the sand content was zero percent and gradually decreased to a minimum when the sand content was two percent. This was as expected as had been concluded in Chapter 3 that the abrasive sand greatly reduced the run life of the swab material. A breakdown of the swab failures according to their sizes could be seen in Fig 4.3. The 'failure points' of the swab were obtained by averaging the various pressure-strokes at the respective sand contents. The various failure points for the respective sand contents could be seen in Fig 4.4 where it was observed that the pressure-stroke values decreased significantly from 53.44 x 10^6 at zero percent sand content to a low of 5.2575 x 10^6 for two percent sand. Although it could not be ascertained whether the swab had to be replaced due to partial wear or complete material failure, this significant drop in the pressure-stroke value indicated the debilitating effects of sand on the run life of the components. These failure points could be transposed to form 'failure bands' whereby the pressure-strokes were broken into various pressures (in MPa) and the corresponding stroke counts (Fig 4.5). These were repeated for all the indicated sand contents, including zero percent sand. These failure bands would provide the vehicle to predict the swab failures via the stroke count if the pressures were known. This method was applicable provided the pump pressure was known to be maintained. If drilling operations prevented this pump pressure from being a constant value, then Fig 4.4 could be used when the pressure-strokes had been cumulated. The instrumentation and methodology required to tabulate the cumulative pressure-strokes would be discussed in the Chapter 5. From the analysis of the pump records, it could be concluded that the curve in Fig 4.4 was applicable for swab sizes of 152mm and 165mm diameters. Further work would be needed to compile a complete list of 'failure curves' for all the swab sizes

Fig 4.3 Swab failures in pressure-strokes
(152 mm and 165 mm swab size)

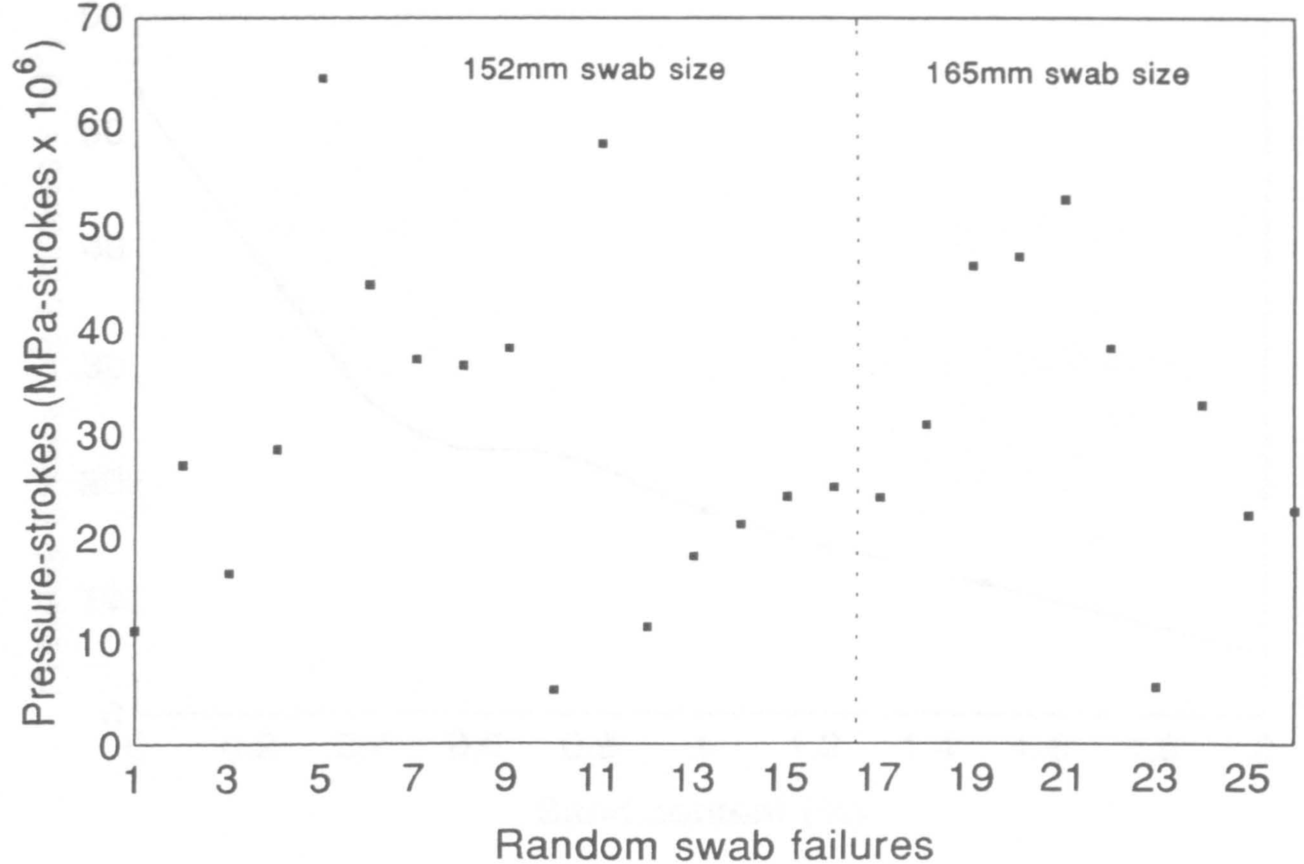


Fig 4.4 Failure points of swabs
Pressure-strokes (MPa-strokes) vs sand content

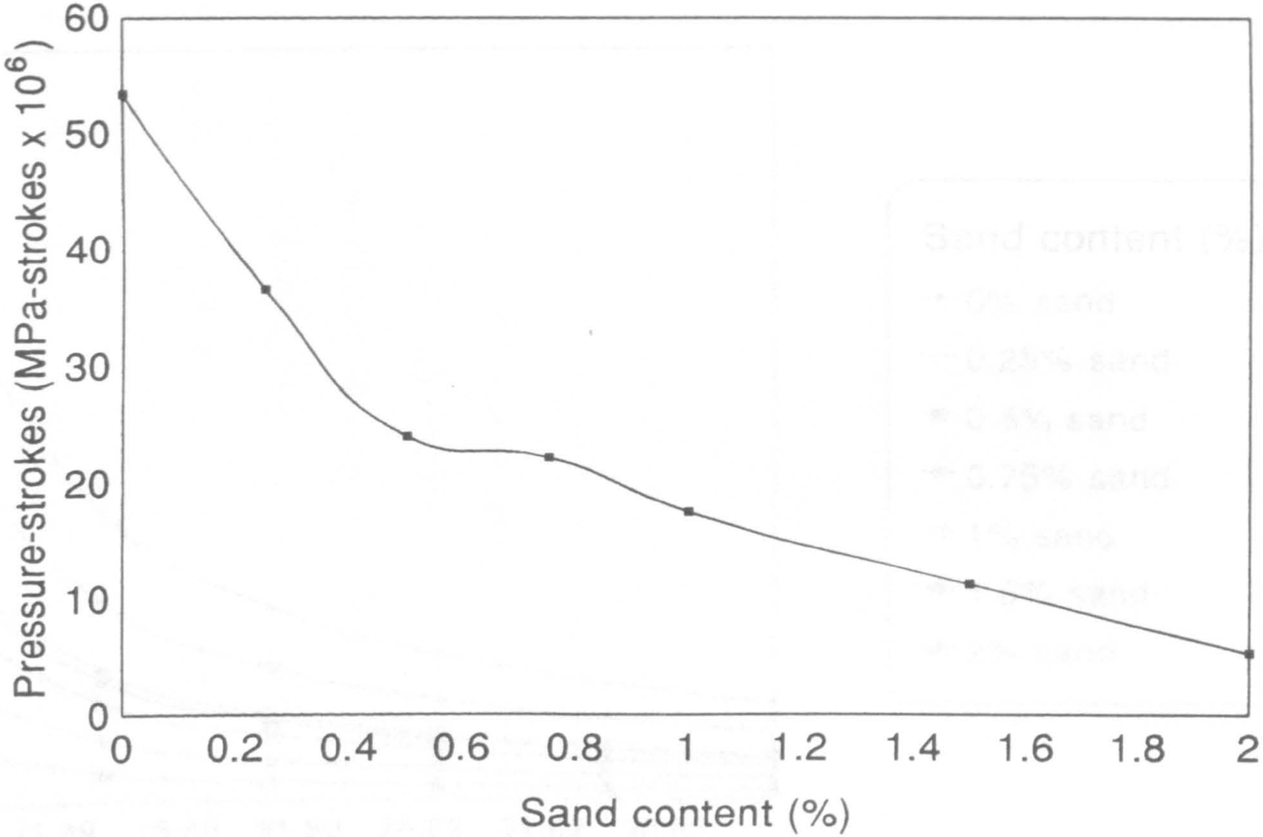
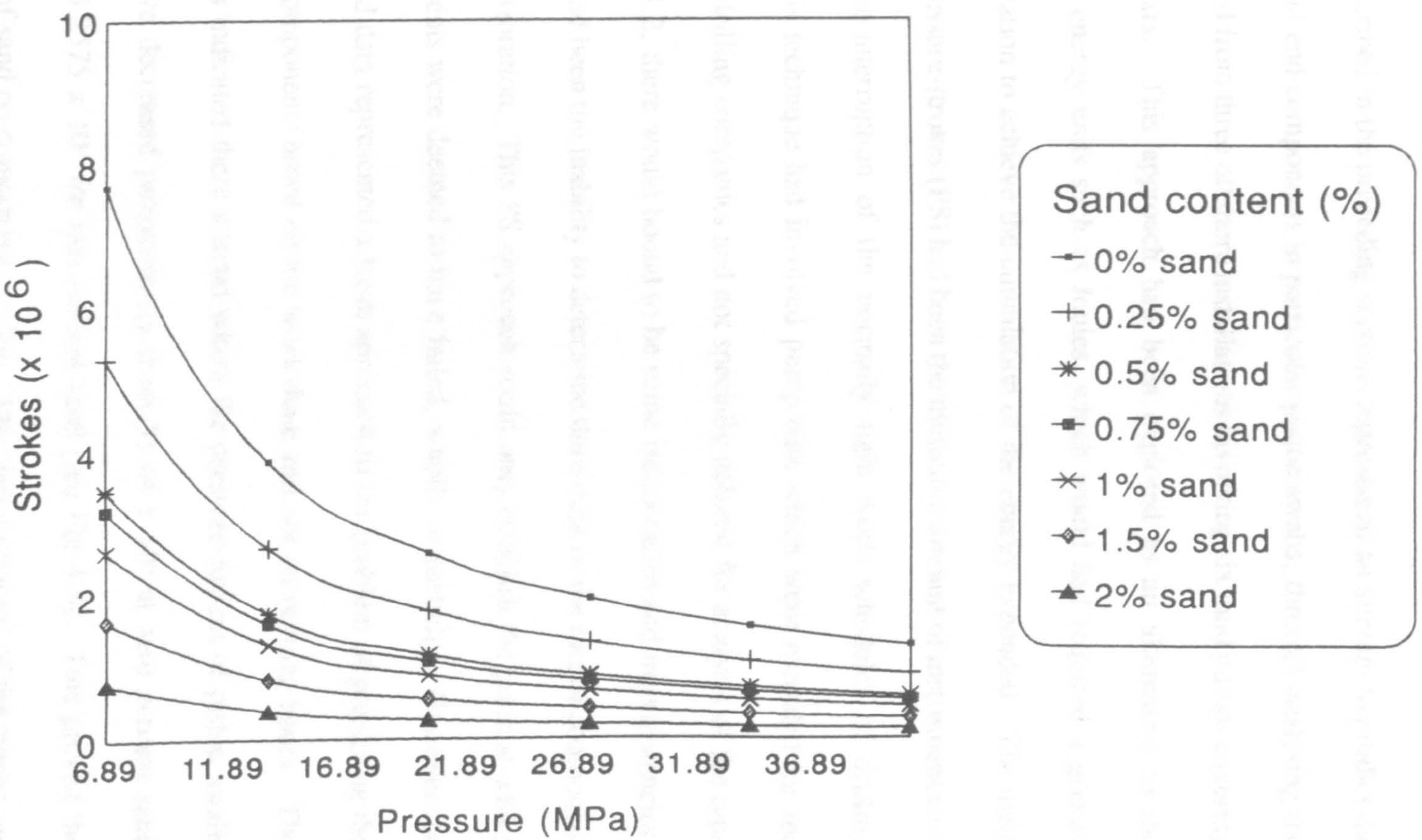


Fig 4.5 Failure bands of swabs

(Pressure vs strokes)



currently used.

4.4 Discussion : Practicality & applicability

The methodology described in the preceding sections represented an attempt to predict the 'failure points' of fluid end components in particular piston swabs, through analysing the pump records collected from three offshore installations involving six mud pumps covering a period of four years. This approach had been explored as an alternative to the monitoring of actual energy units such as Joules, which would had required a greater amount of instrumentation to achieve the cumulation of the energy expended. The main advantage of using pressure-strokes (PS) had been the minimum amount of instrumentation involved and the non-interruption of the normally tight work schedules of drilling operations. Since this technique had involved pump data which were recorded for the operational needs of drilling companies and not specially tailored for analysis of the type described in section 4.2, there would bound to be some inaccuracies and inconsistencies. The main drawback had been the inability to determine the extent or the rate of component wear or material deterioration. This PS approach could only establish the point at which the fluid end components were deemed to have failed, wholly or partially. However the analysis of actual field data represented a fresh approach to the problem of predicting the failure of fluid end components based on the work done and not on running hours. The results of the analysis indicated there a trend where the pressure-strokes of piston swabs were observed to have decreased progressively from 53.44×10^{6} at zero percent sand content to a low of 5.2575×10^{6} for two percent sand (see Fig 4.4). This proved the deteriorating effects of sand on component run life. The 'failure points' of the swabs can be transposed to a form of 'failure band' which allowed the strokes to be predicted for a known pressure. Failure bands had been plotted for sand contents ranging from 0.25% to two percent. These bands could be used as charts for the pump operators to determine the failure points of swabs provided the sand content was known. Alternatively the failure points on Fig 4.4 could be incorporated into an instrumentation device which could total up the pressure-strokes on-line when the pumps were operated and provide an alarm when the set PS had been reached. However further collaboration with drilling companies and oil operators would be needed to compile a complete list of failure points for all the fluid end components which require regular maintenance. Because of the minimum instrumentation involved, this method could be very relevant for mud pump operators who require to know the extent of run life remaining in the component prior to the start of another drilling phase. Like the PV technique, this method of using pressure-strokes (PS) involved a relative unit of measurement which could provide the facility for monitoring material wear and/or component failure. For most practical purposes, the ability to predict component failure or run life would be just as relevant as the capacity to measure the exact amount of material wear. The latter case would be particularly difficult for equipment which had to be employed in situations which required variable operating conditions with no standard operating parameters. This could be best characterised by mud pumps which were used for drilling operations.

Chapter 5

MUD FLOW LOOP

5.1 Introduction

This chapter looks at the next stage of the research, which was the design and development of a mud flow loop and associated equipment. This flow loop contained several features, explained in greater detail in the following pages, which were mainly focused on new methodologies and devices envisaged to aid in the overall maintainability and reliability of mud pumps. The main objective of developing a scaled down mud system was the duplication of offshore mud pump operational characteristics which proved very difficult to incorporate into experimentation without interrupting real-time drilling activities and increasing the already high operational risks. The flow loop could perform several functions, including the following :

(1) facilitating the gathering of pump performance data under controlled conditions, including accelerated wear of piston swabs and the effect of temperature and sand on component material deterioration.

(2) by using a small triplex reciprocating piston pump, the wear characteristics of liners and pistons could be studied in greater depth; the results of experimenting with different piston swab designs and/or different materials could be translatedti full sized piston designs for mud pumps.

(3) expedite the design, installation and commissioning of an energy processing unit (EPU) which could monitor the energy expended by the piston/liner and correlate with component wear or failure. When the concept of using energy to monitor piston wear has been verified, the EPU could then be adapted for industrial application.

(4) the applicability of different technology such as ultrasonics and thermography for fault detection could be explored via the flow loop.

(5) new technologies such as radiation tagging, could be investigated for their suitability in predicting pump wear. The suitability of a sand probe as a means of providing a sand correction factor could also be explored.

Further details of the above listed features could be found in the following respective sections. Due to factors beyond the control of the researcher, the material presented here represented the groundwork which had been completed and which required further industrial collaboration to bring the suggested designs and development work to fruition.

5.2 Mud flow loop

The flow loop system had progressed from concept up to the design stage where the appropriate equipment had been identified and where the system component layout had been completed. The additional work which required to be done was the actual procurement and assembly of the materials and components. The heart of the flow loop (Fig 5.1) would be the reciprocating triplex pump. The suggested requirements of the flow loop (see Section 5.2.1) were based on a typical triplex ram pump system found on an offshore production installations in the UK North Sea, which could not be named for reasons of confidentiality. Because of the cost implications of procuring a new reciprocating triplex piston pump, the suggested option would be a used reciprocating ram pump. Development work which had been done on the suggested modifications on the pump were further detailed in Section 5.2.2.

5.2.1 Suggested flow loop requirements

The suggested flow loop would be expected to operate at conditions similar to those of mud pump systems though at a lower discharge pressure. The selected pump, a single acting horizontal three throw ram pump with 41mm diameter plungers and 102mm stroke, and with a pump speed of up to 150 revolutions per minute (rpm) could supply a discharge pressure of up to 13.8 MPa. This could be powered by a 26 kW DC electric motor coupled to a reduction gearbox to bring the motor speed of 888 rpm down to 150 rpm. A DC controller could provide variable speed control required to provide different pump flow rates round the flow loop. To reduce the expected vibration from the reciprocating pump, the motor, reduction gearbox and ram pump could be installed on a skid equipped with rubber dampeners. The skid, which needed to be robust, could be easily fabricated out of steel sections and welded together. Technical specifications of the pump and prime mover were provided in Appendix 10.

A variety of pipe fittings and components were required for a flow loop, as indicated by Fig 5.1. A suction stabiliser would required to be located on the suction line as close to the pump as possible. This stabiliser could improve pump performance by reducing flow-induced pulsations and preventing the accumulation of gases in the suction line, effectively diminishing pump knock and noise. A flexible hose or expansion joint in the suction and discharge lines would provide for thermal expansion and help to minimize vibrations [1.7]. A discharge pulsation dampener would also required to be located close to the pump in order to reduce pump flow variations and flow-induced discharge pulsations.

Besides the discharge pulsation dampener, the flow loop also required the following fittings :

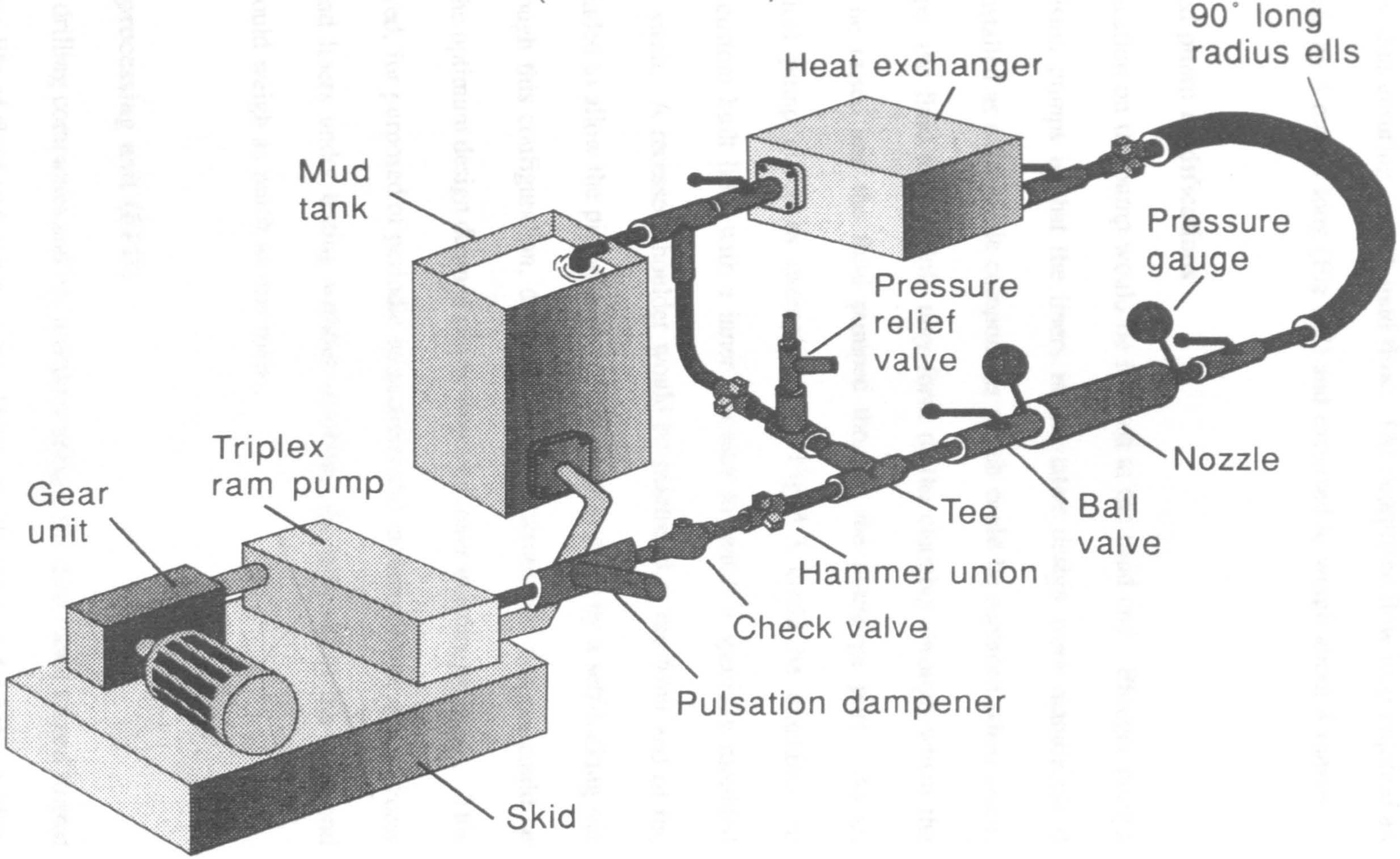
- a check valve would need to be installed immediately adjacent to the discharge pulsation dampener, to prevent backflow of the mud;
- pressure gauges to monitor the mud pressure;
- pressure relief valve, to be set no higher than 1.25 times the maximum plunger working pressure;
- nozzle, to provide pressure drop and allow the pump to do work;
- heat exchanger, to allow heat generated by the pump to be dissipated under controlled conditions. This facility could expedite the study of the effects of

heat on fluid end components, notably polymer swabs;

- thermometers before and after the heat exchanger, to facilitate temperature readings;
- the mud supply tank, with a slope of 1 in 48 from tank to pump, as recommended by Miller (1.7);
- for a similar pump as described above, the selected discharge line size was 50mm with the suction line size suggested as 76mm diameter. This preference for the suction pipe size was to facilitate improved suction with reduced cavitation. Similarly, long-sweep 45° elbows in the discharge system were preferred to 90° elbows for much the same reasons. Where possible, high points in the suction line were avoided to reduce the accumulation of air or gas pockets.
- safety aspects of the system included safety shutdown controls for all malfunctions of the system, pump and prime mover such as :
    - low mud tank level or low suction pressure
    - low system discharge pressure
    - high system discharge pressure
    - high system mud temperature
    - excessive vibration, overloading of prime mover etc.
- other safety aspects of the system included providing adequate work area around the pump unit for servicing and operation, safety guards on all exposed moving machinery, periodic inspections, routine and preventive maintenance.

A schematic layout of the suggested mud flow loop (Fig 5.1) gives an indication of the

Fig 5.1 Schematic of flow loop
(not to scale)



size and shape of the whole system. The layout had been chosen to minimize the need for bends and elbows, thus ensuring smooth mud flow. The completed flow loop required an area 2 meters wide by 5 meters long (Fig 5.2) and expected to weigh about 4 tonnes.

5.2.2 Suggested pump modifications

The main modification on the pump would be focused at the fluid end. Plunger pumps differed from piston pumps in that the liners in the latter design were manufactured separately and installed as expendable components which could be replaced when worn. In plunger pumps, the fluid end module may form of the chamber through which the pressure would be raised and the fluid pumped through the discharge port. As an example, a typical pump cover as exemplified in Fig 5.3 could be machined to accommodate a custom built liner with a inner diameter to match a specially moulded miniature piston swab. A recessed shoulder would be machined at the front end of the plunger and threaded to allow the piston swab to be locked in place by a self-locking nut (Fig 5.4). Through this configuration, different piston materials and designs could be tested to verify the optimum design deemed an improvement over existing products. This layout also allowed, for purposed of periodic inspections and measurements, easier access to the pistons and liners under testing without involving the removal of the fluid end module which could weigh as much as one tonne.

5.3 Energy-processing unit (EPU)

Most, if not all, drilling companies and oil operators utilised the time-based record format to monitor the run life of fluid end components. However, this format did not consider

Fig 5.2 Flow loop layout
(not to scale)

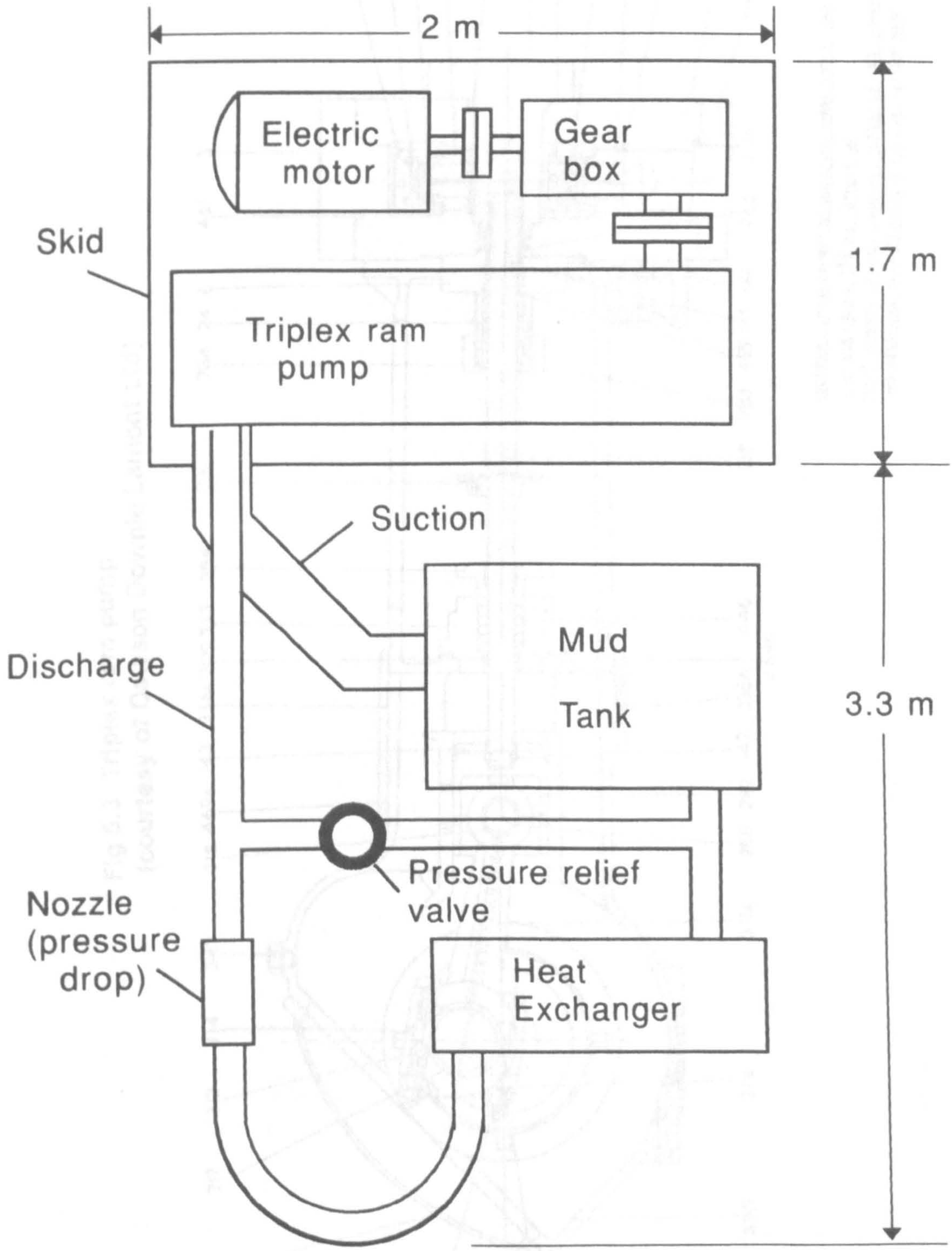


Fig 5.3 Triplex ram pump
(courtesy of Dawson Downie Lamont Ltd)

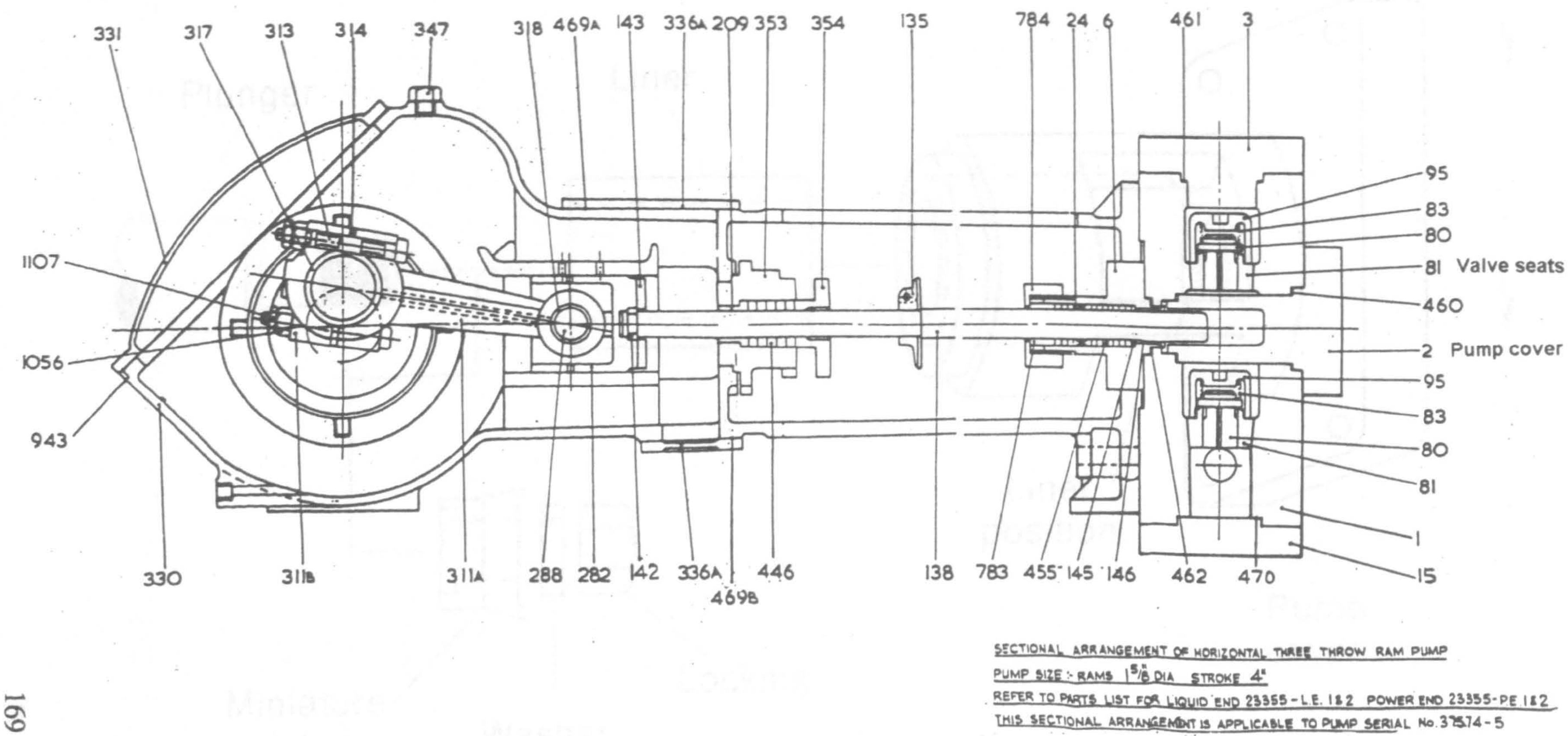
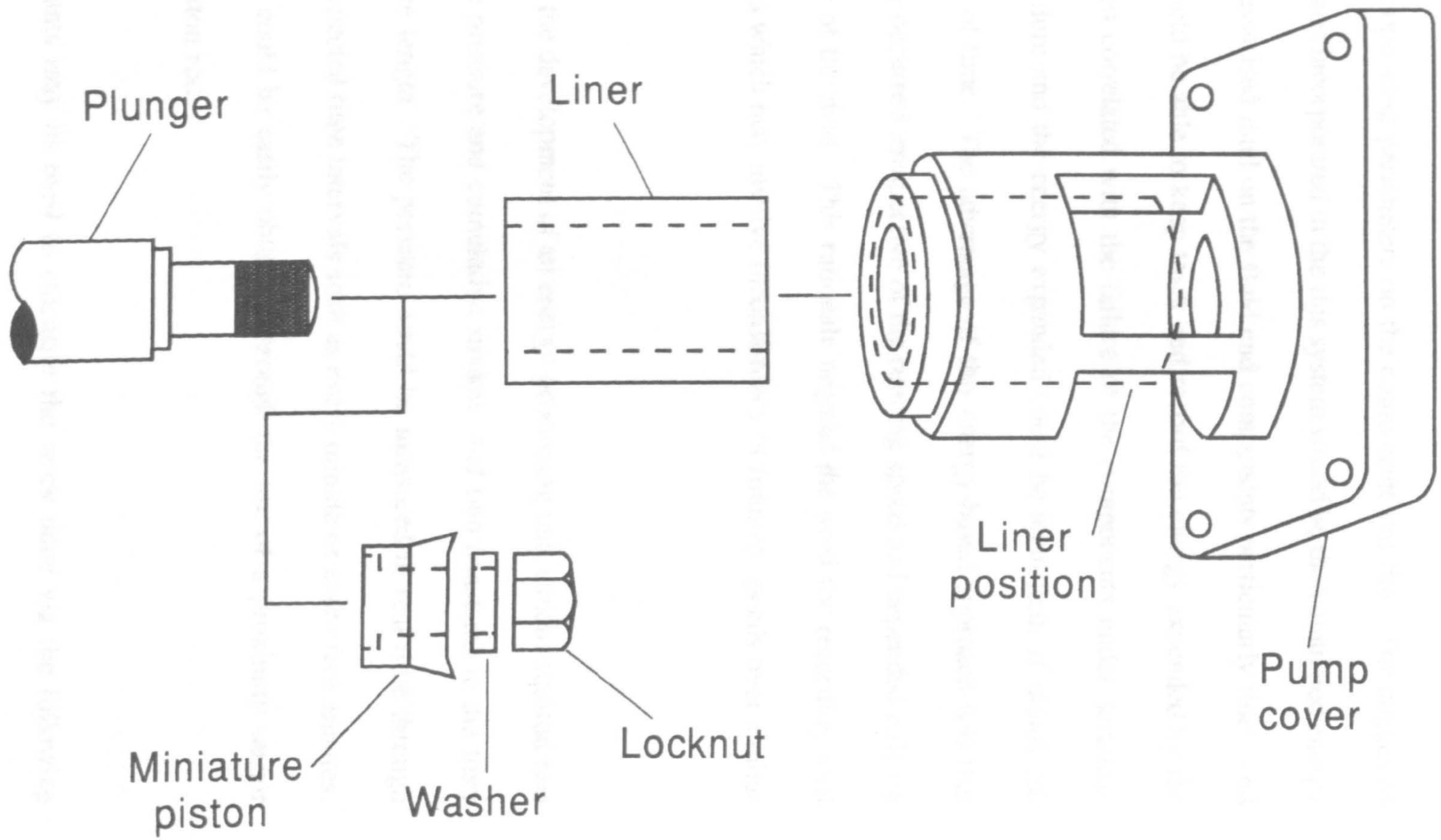


Fig 5.4 Modifications on a typical ram pump fluid end module



the effect of operating variables on the rate of material deterioration of the components. One suggested alternative would be the use of a system which could account for the combined impact of the operating parameters on the component run life. The suggested medium to be explored and incorporated in the this system would be the cumulated energy or work done by the pressurised mud on the fluid end components particularly liners and swabs. This system would be able to keep track and record the energy expended by the liners and swabs. When correlated with the failure of the components under scrutiny, trends of component failure and the energy expended could be identified, if monitored over a sufficient period of time. The advantage of this energy-based approach was that the rate of material wear occurred irrespective of the running speed and depended only on the load or the pressure of the mud. This rationale negated the need for recording total equipment running hours which may involve inconsistency of running speeds over similar time frames.

The system necessitated the development of an energy-processing unit which required two inputs ie mud discharge pressure and cumulative strokes, and two constants ie the liner bore diameter and stroke length. The pressure could be measured in real-time through pressure transducers at selected time intervals such as every minute or every five minutes. The cumulative strokes could be easily obtained through the use of a proximity sensor fitted adjacent to the piston rod.

These inputs and constants may be used to calculate the work done via the following formula :

$$\mathbf{WD = (V_{disp})(P_p)(N)} \qquad (5.1)$$

where WD = work done (Nm)

P_p = pressure (N/m^2)

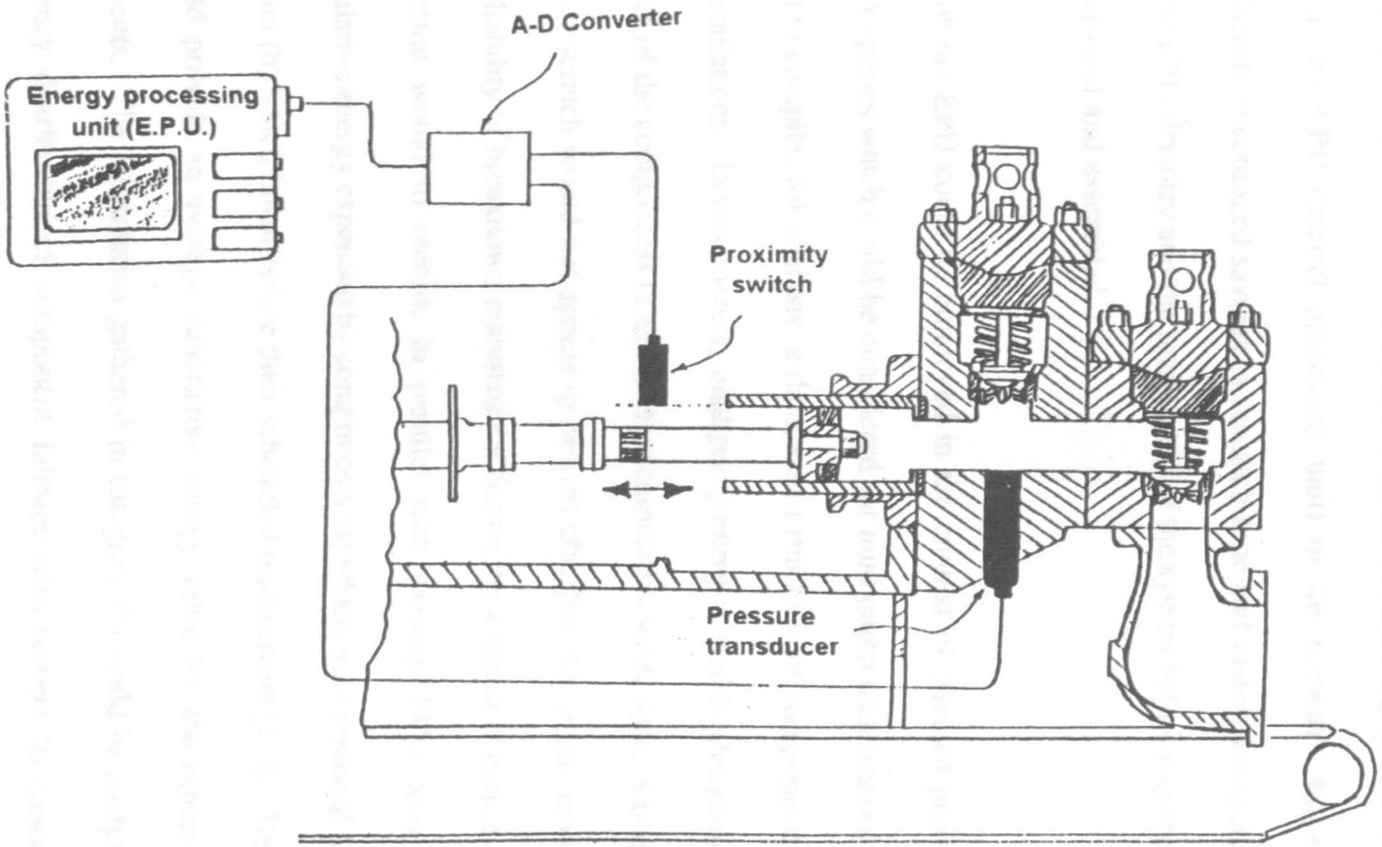
V_{disp} = volumetric displacement (m^3)

N = cumulative strokes

Equation (5.1) indicated that energy would be expended during every stroke of reciprocation. The required cumulative energy could be obtained by totalling up the energy expended per stroke for any number of strokes. This method denoted that running speed do not feature in the computations which by implication leaves out the time factor in the calculation of energy.

To achieve this computation of energy in practice would require the development of a compact instrumentation unit which incorporated a microprocessor, an AD (analog-digital) conversion unit, an LCD (liquid crystal display), proximity sensor (stroke counter) and pressure transducer, and reset buttons. The inputs from the pressure and proximity transducers could be entered onto a microprocessor together with the constants (ie liner bore and stroke length) for conversion to energy and displayed on the LCD in units of in Joules (J). The programming may be done in a suitable language such as BASIC or Machine code. The system may use 4-20 mA current load, as recommended by practising instrumentation engineers. Fig 5.5 showed a schematic of an EPU and the location of the pressure transducer on a typical triplex single acting mud pump.

Fig 5.5 Energy processing unit (EPU)
(not to scale)



A suggested alterative to actually building the actual EPU would be to use a graphical instrumentation package such as LABVIEW. This software package allowed the EPU to be built on the computer screen and directed signals from the pressure transducer through an AD converter to the CPU (central processing unit) of the computer for energy calculation. This facility encouraged savings in research time and material resources by actually verifying the EPU circuitry and the practicality of the system before requiring the hardware to be hardwired and assembled.

The full potential of the EPU could be employed in the context of planned preventive maintenance (PPM) options which could be considered for mud pump maintenance. The EPU could be used to compile, for analysis, a database of run lives of components from offshore drilling installations. In this context, mud pump energy could be designated for recording the run life of the components in terms of its cumulative work done. Compiling such a database from scratch would not appear to be cost effective due to the envisaged poor operational reliability of breakdown maintenance during the evaluation period [1.1]. One alternative option would to record, in parallel with existing PPM time-based strategies, the cumulative energy expended by components (a) which were replaced at their scheduled times, and (b) which failed before their scheduled replacement [1.1]. The data from case (a) could provide an average cumulative energy value for the replacement interval of components. The information gathered in category (b) could be analyzed to provide the frequency distribution of component failures with respect to cumulative energy. This could help to identify the 'failure point' of components and provide the impetus to stretch the replacement interval in terms of work done.

The flow loop would allow the practicality of the EPU to be verified besides facilitating the monitoring of pump component wear in terms of the energy expended in the fluid chamber. Once verified in terms of its functionality and reliability, the EPU could be developed further and adapted for operation on full-sized mud pumps on offshore installations. The expected harsh working environment in offshore drilling would allow any weaknesses of the EPU system to be highlighted and resolved. Further industrial collaboration would be required to consider any commercial potential of the EPU system.

5.4 Fault-detection systems

The failures of fluid end components may be catastrophic or imperceptible. The mud system may experience discharge pressure loss when the clearance between liners and pistons are gradually increased by wear. Catastrophic failures tend to occur when the system experiences extreme operating changes such as rapid temperature rises due to a faulty cooling system. Both types of failures could be addressed through the installation of fault-detection systems which could monitor and detect faults before they become serious enough to cause equipment failure and drilling downtime. Two possible areas which could benefit from further development work included liner wall thickness and temperature regulation.

5.4.1 Ultrasonics

The use of ultrasonic techniques had been well established for applications such as corrosion condition monitoring [5.1] , non-destructive testing [5.2] and detection of cracks [5.3]. Ultrasonic inspection had also been an integral part of preventive maintenance program for some organisations [5.4]. In this respect, ultrasonic techniques could also be adapted for monitoring the thickness of liner walls, as a result of the pumping action, in real-time during the operation of the mud pumps. Compact ultrasonic inspection systems, which are commercially available, could be installed onto mud pump systems and modified to provide audio visual indications and possibly even emergency shutdown facilities once a pre-set thickness value has been breached. This facility may help to reduce the occurrence of damaged liner due to piston misalignment

5.4.2 Thermography

The temperature rise due to the reciprocating pumping action in mud pumps had been known to be so great that cooling systems had to be used to keep it within the limits of material endurance, especially for the elastomeric piston swabs. If and when the cooling system become faulty, the continued pumping action could cause the swab to overheat and fail within 10 minutes only [1.42]. By monitoring the temperature during operation, early detection of rapid temperature rise becomes possible and could be used to prevent equipment downtime by emergency shutdown measures. Equipment which operate on the principle of infrared thermometry are commercially available which could be installed permanently on the mud pump to provide easy and fast readout of the temperature of the liner wall. The location of the liners of most mud pumps which required the liners to be easily accessed for maintenance had favoured the possibility of infrared thermometry as a fault-detection system.

5.5 Other new development areas

One new technology area which could have great benefit to the forecasting of fluid end failures was the development of thin layer activation (TLA) [5.5-5.6]. This technique involved exposing a section of the selected component to mild radio-activity. For cylindrical components the patch area could be a round band up to 100mm wide. The treated material would emit gamma radiation which could be monitored via remote sensors. The material which were lost from the treated patch due to wear would cause a drop in the radioactivity level which could then be used to calculate the wear rate. The main advantage of this methodology had been the potential savings as a result of wear tests which could be completed in a fraction of the normal time required.

The sensitivity of this technique, averaging about one percent of the active layer depth, could be used in conjunction with the EPU to derive a wear rate based on the energy expended by the fluid on the component.

Another area which merits further development involved the adaptation of sand probes to detect the percentage of sand content in the mud and transposing it to a sand correction factor before incorporating into the EPU. Sand probes available commercially in industry were generally used in the analysis of sand produced in oil and gas production. Since sand probes were originally used in less severe environments involving less pressures, further development work would be required to adapt them for detecting sand content instead of calculating amount of sand as in oil production.

5.6 Discussion

The availability of a flow loop would offer the various advantages of simulating operating conditions of mud systems similar to those on offshore installations. Thus the risks generally associated with offshore drilling could be avoided while providing a platform to investigate various innovative technologies, and commission custom-built equipment which would enhance the reliability and maintainability of mud pumps. The improved performance of mud pumps resulting from such investigations, especially the possibility of stretching the run lives of fluid end components, would enable drilling companies and oil operators to realize the full potential of mud pump systems with reduced frequency of equipment downtime.

Based on the material presented here, further industrial collaboration would be required to bring the suggested designs and development work to fruition. Fault detection systems involving ultrasonics and thermography could be developed and verified for their functionality, via the flow loop, and adapted for industrial application.

The flow loop could also facilitate the development of new piston materials and designs, as well as exploring the adaptation of sand probes for translating sand content to sand correction factors which could incorporated into the EPU.

The flow loop and EPU would permit the measurement of cumulative energy and the parallel comparison of 'pressure-strokes'. The 'failure point' of fluid end components could then be identified and compared in both units of 'pressure-strokes' and energy

(Joules) for further analysis.

Chapter 6

DISCUSSION and CONCLUSIONS

6.1 Mud pumps and maintenance strategies

Reciprocating pumps constituted the heart of mud systems used in drilling operations on offshore installations. The fluid end modules of these mud pumps, due to the variable and severe operating conditions associated with offshore drilling, formed the main problem areas when compared to the power end [1.1]. The design of the fluid end modules, particularly the dynamic sealing nature of the piston swab, lend itself to increased maintenance frequencies which were considered not desirable nor tolerable especially when the occurrence of equipment downtime could not be forecast. The presence of particles in the mud such as drill cuttings added to the severity of wear problems encountered in mud systems, as do other slurry pumping systems.

Existing maintenance strategies adopted by most, if not all, drilling companies and oil operators included breakdown maintenance, time-based maintenance and condition monitoring. Breakdown maintenance were applicable to machinery which had broken down and required corrective measured, and were generally not favourably considered in the modern competitive industrial environment. Certain techniques of condition monitoring, in their present form, do not seemed suitable to mud pump operation. One of the techniques, oil debris analysis, could not be applied since the pumping medium ie mud do not circulate back to the system in the same manner as lubricating oil in engines. The mud which flowed back to the pump would generally be subjected to mud conditioning processes where most of the particles would be screened and removed. The very fine particles which remained in the mud tended to be the most destructive. Another technique, vibration analysis, seemed more suitable for radially rotating machinery such as gas turbines although new developments could not be discounted in their applicability for reciprocating equipment such as mud pumps. Time-based maintenance would have only considered time-related factors such as speed and running hours, and not the fluid pressures which were load related. Hence identical operating time periods with different pressures would be considered to experience similar amounts of wear when the different loading conditions clearly dictated otherwise.

Customised computer-based systems had been developed by drilling companies which offered the facilities of real-time performance monitoring, event detection and alarms, enabling operators to spot early trend changes and prevent further equipment damage. However, these systems lack the ability to forecast possible component failure. A maintenance strategy based on the energy expended appeared to have practical implications for reciprocating pumps which did not operate on standard conditions. The varying oil well hydrostatic pressures encountered during drilling required different mud pressures which were met by changing the pump speeds. This would have an effect on condition monitoring, which were best suited for constant operating conditions, but would not have influenced an energy-based maintenance strategy since the energy consumption depended on load-related factors such as pump pressure and the cumulative strokes of reciprocation, irrespective of pump speed.

A two pronged approach was adopted to investigate this axiom further. In the first approach, the wear characteristics and variables which were considered to affect the failure of fluid end components were studied. The other method involved analysing pump records collected from offshore installations and exploring the link between component failure and operating parameters such as pressure and cumulative strokes.

6.2 Wear characteristics and mud pumps

The majority of work done by tribological researchers had been focused on two- and three-body wear, particularly dry wear. By comparison, the research on wet abrasion or hydroabrasive wear, particularly for elastomeric wear in steel, had been on a lesser scale. Another area of research with industrial practicality was reciprocating wet abrasion with abrasive mud as the interfacial medium. This was indicative of a survey of mud pump operators which concluded that the sand content in the mud system was one of the factors which had a direct effect on the wear and failure of liners and piston swabs. Research had been done on wear of piston rings for engines and gas compressors, where the lubricating medium was not the pumping medium, and the sealing elements were metallic rings. This differed significantly from the dynamic piston seals of reciprocating pumps which were made of elastomers and had to seal and pump highly abrasive slurries (or mud). The mud had the additional function of being a lubricant for the reciprocating sealing element (or swabs). The presence of sand particles would have rendered the lubricating mud highly abrasive. Wet three-body abrasion in reciprocating pumping would be normally centred on the mud, as the third body, where sand particles provided the catalyst for accelerated abrasion which led to component failure.

The nature of drilling operations required variable speed variable pressure (load) operating conditions. This made condition monitoring techniques difficult to be applied since standard operating conditions would be needed from which to reference performance trends. The variable operating conditions also implied that there would be varying rates of wear within any identical time frame. This made time-based maintenance scheduling not very reliable in terms of forecasting actual material wear and hence the service life of the components. Maintenance which were scheduled on the basis of energy consumption would have taken into account the effects of varying operating factors on wear, and need not be burdened by the differing pump speeds or operating durations.

The postulation that component failure were associated with wear and work done had been established. Wear and friction were linked entities. Friction was essentially an energy dissipation process where the frictional work generated frictional heat which had an effect on tribological interactions which in turn resulted in wear processes leading to generation of loose wear particles. The frictional work was considered to be dissipated in two separate regions in the interfacial region. The interfacial zone though considered narrow corresponded to high rates of energy dissipation while the other process entailed deformation within a large volume of material and lower rates of energy dissipation.

These processes were characterised by microcutting; fatigue cracking and tearing with sub-surface heating and damage; interfacial sliding and zone shear with material rupture and transfer and finally the propagation of abrasion patterns termed as Shallamach waves. These processes may occur and overlap concurrently. An energy consideration of friction would divide the 'loss' process of mechanical energy into different phases where the mechanical energy, after first introduced into the contact zone, would be transformed by ploughing, adhesion and plastic deformation, before dissipated via the effects of thermal dissipation, storage and emission processes.

To appreciate the link between material wear and loading conditions, it would be advantageous to understand that the design and operational nature of reciprocating pumps presented problems and characteristics unique to this category of machinery. They included the possibility of sudden failure due to misalignment caused by significant wear in the lower crosshead guides as a result of the 'overrunning' crankshaft rotation; the Jacoby leakage phenomena where the slight leakage of mud due to the relaxation of the deformed swab immediately after the end of the stroke gave rise to accelerated three-body abrasion on the liner and swab; the effect of stroke reversals on wear caused by the sudden change of the swab back to its original shape which trapped abrasive sands between the swab and liner walls; the effect that the extrusion of the elastomeric swab material, due to piston-liner clearance, had on the life of the piston swab. The rationalisation that liner wear was caused by stroke reversals indicated the effect of discharge pressure on liner and piston wear.

6.3 Application of systems concept to mud pumps

The main characteristics of a systems approach include :

- the structure of a system, comprising the elements (A), properties of the elements (P) and the relations between the elements (R), represented as $S = \{A,P,R\}$.
- inputs and outputs, provided the system could be hypothetically separated from its environment by a systems envelope.
- function of a system, which was to transform the inputs into outputs.

A systems function involved identifying a system envelope separate from the environment, compiling the inputs and outputs and describing the functional input-output relations. A system structure, which undergo changes relative to the action of tribological processes of friction and wear, would identify the elements of the system and the interrelations and interactions between them as well as specify their relevant properties.

There were different complexity levels of tribological processes where a systems envelope may be located. In the case of mud pumps, the systems envelope in the first level was located broadly around the mud pump system operating in field conditions. Dynamic interactions of the pump (driven) with other system components, electric motor or engine (driver), well lithology, the drilling team (human factor) and the drilling conditions would influence the behaviour of the system. At the next level, the system envelope centres around the mud pump under bench tests. The fluid end module of mud pumps may be considered as a sub-system at the third complexity level. The next level would involve the behaviour of basic components such as pistons and liners. At the most fundamental level of complexity, the fundamental tribological processes between two relatively moving surfaces may be investigated on suitable wear tester under controlled conditions in a laboratory environment.

The main function of a tribo-mechanical system whose system envelope was centred on the fluid end of a mud pump would be to transform inputs of motion and energy into work for hydraulic transmission.

The four elements which participated in the wear process include (i) piston swab (moving tribo-element); (ii) liner (stationary tribo-element); (iii) drilling mud and sand (interfacial medium); and (iv) environment (water/coolant, air, chemical activity). The interactions of the systems' elements comprised (a) friction and wear processes; (b) three-body abrasion between (i), (ii) and (iii), and tribo-effect of (iv) on (i), (ii) and (iii). The tribological behaviour of the system would be affected by the properties of the elements such as the mechanical properties of the piston swab, the mud weight of the mud and the sand content.

The concept of quantity flows had been used to explain the different processes occurring within a system. An adaptation of Czichos' three-dimensional conceptual planes [1.32] had been used to explain the transfer processes happening in the piston-liner component couple of mud pumps. Work, if considered as a quantity flow in the same context as a quantity mass flow, would be the result of the influence of one element upon another similar to the effect of motion and speed on increasing the mud pressure during the pressure stroke of the piston. This work would had occurred following the transmission of mud (mass) through the system resulting in the generation of wear particles (mass transformation) from the wear processes. The frictional work would most likely had been transformed into heat (energy losses), giving rise to the assumption that friction caused energy losses while wear processes resulted in material losses. These losses may be categorised into three phases, where (i) work was initiated into the contact zone, forming contact areas, (ii) the transformation of work within the contact zone, resulting in plastic and elastic deformation of the swab, three-body abrasive wear processes, and (iii) the dissipation of energy as heat, friction, vibration or acoustics. An energy balance may be considered, where the work input of a tribo-system (ΣE^w_x) must equal the useful work (ΣE^w_y) + loss-output energy (ΣE_z) + energy transformed to other forms (ΣE^w_t), represented through the equation :

$$\Sigma E^w_x = \Sigma E^w_y + \Sigma E_z + \Sigma E^w_t \qquad (6.1)$$

The responses resulting from the effect of surface forces on steel and rubber tended to be different. The elastic-plastic nature of steel would result in energy being expended and dissipated as heat on the surface, leading to the general view that friction was a surface phenomenon in steel. The visco-elastic characteristic in rubber allowed the strain to propagate throughout the material until the fracture limit resulted in energy dissipation through a bulk effect. The wear process in a tribo-system would be affected by many variables under the sub-divisions of material, operational, geometric and environmental. Since the steel liner in mud pump fluid end was the stationary element while the piston rubber was the sliding element, the capacity of the rubber to trap or transmit solids or sand particles during the pumping cycle would determine the mode of wear and debris transportation. There are three phases in the wear process of a tribo-system involving a material couple with a rubber element, namely (i) the pre-detachment phase, where the material surface layer became deformed without any resultant material loss; (ii) detachment of surface material, where material deterioration and subsequent detachment result in abrasion patterns, called Schallamach waves, which were perpendicular to the direction of motion; and (iii) post-detachment phase, where the detached material may be transported via three modes, (1) as loose debris; (2) transferred to the steel countersurface through particle deposition and smearing; and (3) back transfer to the rubber itself, reinforcing the phenomenon of stroke reversal and Jacoby leakage effects. The second and third modes would result in the wear debris ploughing and gouging into the softer rubber surface as well as the chrome steel giving rise to 'streaking' in the liners as commonly found in field service.

A system function would not be affected and continue to perform as expected if the structure of a system remained unchanged. Since the systems structure include elements and the interrelations between them, any changes in the elements or its interrelations would affect the system outputs via its functions. Changes in the tribo-system may include alterations of system elements such as wear particle generation; changes in the relations between elements such as wear mechanism transition from abrasive to micro-tearing; and changes in the properties of the system elements such as streaking-induced grooves and surface damage which would affect the performance of the piston rubber.

Functional failures of a system may occur when operating conditions exceed certain set tolerances. In mud pumps, a sudden pressure drop would indicate failure in the fluid end while excessive clearances signify failed pistons and/or liners. The demands of high pressure pumps dictated that the higher the mud pressure, the smaller the piston-liner gap allowed. Hence tribo-processes such as friction and wear could lead to structural changes which in turn cause functional failures.

Drilling operations required mud pumps to be operated in a variable speed, variable load mode. This placed uneven stresses on the machine components particularly the fluid end, resulting in wear which were proportional to the extent of work output of the pumps. Abrasive sand in the mud induces aggressive three-body wet abrasion generating substantial damage to the liner and piston due to gouging, ploughing and 'nibbling' of the rubber. This was the result of the sand becoming embedded in the piston rubber partly due to stroke reversals and partly as a result of the fluid nature of rubber under high pressure. Changes to the system due to wear would affect the operational reliability of the pump. The postulation that piston and liner wear were directly dependant on the amount of work transmitted from the tribo-mechanical system coupled with the wear characteristics of the fluid end components lead to the difficulty of predicting the reliability of mud pumps. Forecasting the wear life of the fluid end relative to the work transmitted from the pump system would then be more practical than correlating the inputs and outputs of the system.

An investigation of the wear processes of a tribo-system may involve either the actual practical system or a test system under laboratory conditions. Both approaches presented pros and cons, and particular attention would be required in the case of an open system structure such as in drilling under field conditions. As an illustration, the mud circulating in the pump system and out to the oil well would encounter many variables such as rock formations and well pressures which imply that operating input to tribo-systems in the fluid end are not constant and would add to the complexity of maintaining pump operating conditions. Given these complications, tribo-test investigations would be more realistic if the type of motion and materials of the system elements were similar to those of practical systems. In the case of fluid end of mud pumps, this would involve a test system which utilised the actual liner and piston as used on mud pumps for offshore drilling, and entail reciprocating sliding in the presence of drilling mud with properties suitable for field applications. Suitable sands could be added to the mud to simulate drill cuttings and solids concentration. A suitable simulation criteria which indicated a similarity of tribological processes in both practical and test systems could be the appearances of worn surfaces. This test procedure formed the main experimental work of the research.

Another approach which was explored involved analysing field data of mud pump records. These pump records reflected the true working conditions in offshore drilling environment. Analysis of these data would enable the researcher to identify the 'failure points' of fluid end components, the refinement of which could permit mud pump operators to forecast the failure of components such as piston swabs and liners, and plan drilling operations around such downtime events.

6.4 Effects of variables on wear processes

The experimental work were performed on a specially built wear tester with reciprocating motion and not on an actual mud pump system, the reasons being the size and weight of the average mud pumps used on offshore installations, weighing as much as 36000 kilograms and measuring 6 m by 2.9 m and 2 m in height, prohibited its feasibility under laboratory conditions. Also, given there were usually only two mud pumps on most drilling installations, the competitive nature of the drilling business and safety requirements would not permit any experiments to be performed nor any instrumentation to be installed which would affect the routine operations of the equipment. Besides the cost factor, where a mud pump system would require a capital outlay of more than £1.2 million to install, a closed pressurised system would not allow the periodic and progressive measurements of the targeted piston rubber under experimentation.

Rubber samples from initial experiments were examined and compared with field samples of failed pistons in order to validate the design of the experimental wear tester as having achieved the selected simulation criteria (see Section 4.3.1), which was the appearance of worn surfaces. Having established the simulation criteria, the next stage of experimentation was focused on identifying the operating variables which significantly affect wear processes using Taguchi experimental design (see Section 3.4.2) and investigating the effects of these operating variables on wear processes (see Section 3.4.3).

The variables which were found to have significant effect on the wear processes included sand content, load, temperature and mud viscosity. A matrix set of experimental runs (see

Section 3.4.4) concluded that the combined effects of load and sand were significantly greater than the single effects of load and sand considered separately, and resulted in accelerated volume loss and greater specific wear rate. A similar matrix experiment (see Section 3.4.5) also reached the conclusion that the combined effects of load and temperature exerted significantly greater influence on wear than the separate and single effects of load and temperature, and showed a similar trend in terms of volumetric wear loss and specific wear rate. It was found that the volumetric wear loss was linked to the frictional energy via the form $V_w = aE^b$, and the effect of individual parameters could be described by the following equations :

$$V_{w(s)} = a_{(s)}E^{b(s)}$$

$$V_{w(l)} = a_{(l)}E^{b(l)}$$

$$V_{w(t)} = a_{(t)}E^{b(t)}$$

$$V_{w(wm)} = a_{(wm)}E^{b(wm)}$$

where $V_{w(s)}$ = volumetric loss due to sand effects

$V_{w(l)}$ = volumetric loss due to load effects

$V_{w(t)}$ = volumetric loss due to temperature effects

$V_{w(wm)}$ = volumetric loss due to mud viscosity effects.

Each of these parameters were considered to affect the wear processes independently and synergistically. This rendered it difficult to establish a single coefficient 'a' to represent all the four parameters, just as the exponent 'b' had been known to vary differently for each of the parameters. The wear test system represented the fifth complexity level of tribo-testing approaches and therefore could only facilitate the study of wear characteristics and determine the variables affecting wet abrasion of elastomers, but cannot simulate the high pressures in a closed system. This constraint has been addressed by the groundwork done on the design and development of a flow loop (see Section 5).

6.5 'Pressure-stroke' (PS)

During the operation of mud pumps, the pressure and temperature were linked whereby the pump speed was varied to control the mud flowrate, changing the frictional load and also the heat generated due to the pumping action. The effect of generated heat on the wear processes was difficult to quantify because the heat would have been dissipated and removed. The mud weight depended on the well condition and lithology, while the mud viscosity was reliant on other factors such as the efficiency of the mud conditioning system. The only parameters which could realistically be monitored irrespective of the drilling conditions were the pump pressure and sand content, and the pump speed and/or cumulative strokes. One approach of tracking failure would be to monitor the cumulative strokes and pump pressure, and identify the trends which indicated piston failures caused by pressure, cumulative strokes and sand content.

These pump records were analyzed using a different approach through the combination of pressure and strokes to form a new measurement unit called 'Pressure-stroke' (PS) (see Section 4.3.1), similar to the manner in which PV (pressure-velocity) was derived from pressure and velocity to form a relative unit of measurement. PS were specific to mud pumps and could be expressed in 'MPa.strokes' via the equation :

$$\mathbf{PS = (MPa) \times (strokes/minute \times hours \times 60) = Pressure.strokes}$$

This method of using pump records to determine 'failure points' of fluid end components require certain assumptions including the fact that most, if not all, drilling contractors used running hours to monitor component run life. However, time-based methods do not account for the different pressures operating within the component service life, hence implying that varying degrees of material loss due to wear were not considered. The advantage of using PS methodology would be its ability to account for the effect of pressures on wear since the PS computation had to include pressure as well as strokes.

The essence of the pump records were extracted and analyzed to identify the failure trends and pinpoint the average PS values of component replacement ie the 'failure point'. The resultant plot of the pressure and strokes permitted the definition of a 'PS' band which facilitated a known pressure to forecast the cumulative strokes deemed to result in component replacement. Hence, by monitoring the pressure, this facility should allow 'failure points' to be estimated via the cumulative strokes, and at what sand content. The significant drop in the pressure-stroke values shown in Fig 4.4 where it was observed to decrease from 53.44×10^6 at zero percent sand content to a low of 5.2575×10^6 for two percent sand, indicated the debilitating effects of sand on the service life of the fluid end components. This agreed with the conclusions of earlier experimental work about the effects of sand on wear processes. These failure points, when transposed to form 'failure bands' whereby the pressure-strokes were broken into various pressures and the corresponding stroke count (see Fig 4.5), provided the vehicle to predict the swab failures through the stroke count if the pressure were known. The plotted curve in Fig 4.4 was applicable for swab sizes of 152mm and 165mm diameters. Further work would be needed to compile a complete list of 'failure curves' for all the swab sizes currently used on offshore drilling operations.

6.6 Mud flow loop and other further work

Considerable groundwork had been completed on the design and development of a mud flow loop and associated equipment. This flow loop offer the various advantages of simulating operating conditions of mud systems similar to those on offshore installations, thus avoiding the risks generally associated with offshore drilling while providing a platform on investigate innovative technologies and custom-built equipment which could enhance the reliability and maintainability of mud pumps. The functions of the mud loop included facilitating the design, installation and commissioning of an energy processing unit (EPU) which could monitor the energy expended by the pistons/liners and correlate with component wear or failure. Once the EPU has been verified on the flow loop, it could be adapted for industrial applications. The flow loop could also help to explore the applicability of different technologies such as ultrasonics and thermography for fault detection. New technologies such as thin layer activation (TLA) could be investigated on the flow loop for their suitability in predicting pump wear, a facility which could not be conveniently provided by offshore installations. In addition, new piston materials and design could be developed and verified through the availability of the flow loop system, as well as exploring the adaptation of sand probes for translating sand content to correction factors which could be incorporated into the EPU. The preliminary work presented here would require further industrial collaboration to bring the suggested designs and development work to fruition.

6.7 Conclusions

The relationship between material wear loss and the energy expended was investigated. A dedicated reciprocating wear tester was developed for experiments which represented the fifth complexity of tribo-testing approaches. The simulation criteria, which was the appearance of worn surfaces, was achieved through comparing experimental worn with field samples from offshore installations. Abrasion patterns such as Schallamach waves, which were normally reported for dry friction tests, were identified and found on experimental samples during wet abrasion experiments. Taguchi experimental design and analysis of variance (ANOVA) were used to identify variables which affect wear. Within the confines of level settings, sand content and load and their interaction were found to have significant effect on the wear processes.

The effects of variables on wear were investigated. The volume loss were found to be related to the energy expended through the form : $V_m = aE^b$. Linear regression analysis was used to obtain the values of the coefficient 'a' and exponent 'b'. The coefficients for the variables found to be constant for the range of values investigated were :

Factors	Coefficient 'a'
Sand content (1% - 5%)	0.00265
Load (70N - 150N)	0.0002
Temperature (30°C - 90°C)	0.00275

Mud SG (1.0 - 2.0)	0.0003

The values of the exponent 'b' were not constant and found to increase proportionally as the value of the variables, thus :

Factors	Exponent 'b'
Sand content (1% - 5%)	0.665 - 0.705
Load (70N - 150N)	0.8 - 0.8398
Temperature (30°C - 90°C)	0.6645 - 0.69
Mud SG (1.0 - 2.0)	0.8155 - 8305

Since each of these variable were considered to affect the wear process independently and synergistically, this rendered very difficult to establish a single coefficient 'a' and exponent 'b' which were representative of all the four parameters, especially given that the latter were found to vary for different values of each of the parameters.

Another approach was explored which analyzed pump records collected from three offshore installations for six mud pumps over a period of four years. The analysis used 'pressure-strokes' (PS) in the similar analogy as the relative unit 'pressure-velocity' (PV) adapted by other researchers. The results of the analysis agreed with earlier experimental work about the debilitating effects of sand on component wear, and permitted the definition of a PS 'failure band' which facilitated, through a known pressure, to forecast the cumulative strokes at the 'failure point' which require component replacement. The essence of the results was applicable for piston sizes of 152mm and 165mm. Further work would be required to compile a complete list of 'failure bands' for all the swab sizes currently used on offshore drilling operations.

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Appendix 1

PRINCIPAL FUNCTIONS of the DRILLING FLUID (MUD) are :

1. To remove the cuttings from the bottom of the hole and carry them to the surface.

2. To transmit hydraulic horsepower to the drill bit.

3. To cool and lubricate the drill string and bit.

4. To exert sufficient hydrostatic pressure to control fluids encountered in formations penetrated.

5. To minimize settling of cuttings and weight material insuspension when circulation is temporarily stopped. The mud, however, should have properties which allow the cuttings to settle in the surface system.

6. To support and protect the walls of the hole.

7. To reduce to a minimum any harm to the formations penetrated.

8. To insure maximum information about the formations penetrated.

Appendix 2

MAINTENANCE STRATEGY OF MUD PUMPS OF OIL OPERATORS AND DRILLING COMPANIES

The maintenance strategies adopted by some oil operators and drilling companies are similar and as varied at the same time. Most oil operators contract out the drilling operation to independent drilling companies and may impose certain requirements as far as maintenance of drilling equipment are concerned. Others may include specific details of penalties incurred for 'unacceptable pump downtime'. Marathon, for example, has its own drilling rigs and provide the parts and equipment and decides on the timing of maintenance and replacement of pump fluid end comsumables. Big drilling concerns like Sedco Forex follow a flexible maintenance strategy according to the client's drilling policy or the drilling program. Smaller companies like Atlantic Drilling, on the other hand, have been chroming the liners for re-application, for the past 15 years. The most recent addition on the market are ceramic liners which can run up to 5000 hours as compared to the average-800 hours conventional liners, though at 3 times the cost. At the end of the day, each company will follow a maintenance strategy either dictated by the either dictated by the client oil company or according to its own operations capability and economic pragmatism. Below is a summary of the various maintenance strategies of the oil companies and drilling companies which have responded to the request for information :

1) **Atlantic Drilling Co. Ltd.** follows a policy of planned periodic maintenance with regular checks on the drive-end of mud pumps eg. crankshafts and bearings, while other checks are done after every well eg. oil contamination. These are subjected to the annual certificate of fitness checks by bodies like Lloyds. Components on the fluid-end section follow a time-based maintenance schedule carried out based on running hours and whenever conditions permit eg. cementing. The pistons are rotated 120° every 300 hours and the valves changed every 700 hours. The liners average 500 - 1000 hours and are checked for wear and tolerance. Those liners which exceed the set limits are sent for chroming and re-used. An additional 500 hours of service life have been normally achieved, bringing the average chromed-liners service life to 1500 hours. Atlantic Drilling

have been doing this for 15 years and claim to have been among the pioneers in this respect. Other machining process eg. metal-spraying have been used, with such jobs handled by such firms as V[illegible] Der Horst. It seems at one stage, the liner manufacturers were seriously considering buying out the chroming industry because of the threat to their products. But now the cost of liners and chroming have narrowed so much as not to warrant such commercial considerations. One European company have produced ceramic liners which Atlantic Drilling cosidered as ideal under laboratory conditions but not very suitable in the harsh environment of real-life drilling.

2) **Stena Drilling Ltd**. follows a maitenenance schedule which is dictated by the type of drilling section. For example, prior to drilling with 12 1/4 or 17 1/2 sections, the fluid-end of mud pumps are stripped and all consumables replaced. This may be done during cementing. The Manager of Stena Drilling felt that heat is also an important consideration and found the closed-loop cooling system rather "medieval" as compared to the current method of using sea-water line in a open system. The current working conditions which average 180°-220° at pressures of 4000 psi with oil-based mud warrants the use of open-line system. Furthermore, there is the feeling that pump design have not advanced with technology changes. In the 60s, the fluid-end components are changed when required, ie run-to-breakdown type of maintenance. In the 70s, the trend was towards time-based maintenance, following time-cycles. In the 90s, the demand for increased hydraulic power have seen the change from duplex to triplex pumps. Modern pump design are taking hydraulic to the extreme, as exemplified by the working pressure of 4200 psi of oil-based mud, which is twice the pressure about 3 decades ago.

3) The type of contract arrangements between **Marathon Oil Uk Ltd** and the drilling contractor dictates the type of maintenance strategy.

a) Marathon has its own drilling rig and provide the parts and equipment while the drilling contracator provide all the labor and expertise. In this situation, Marathon will decide when to change the fluid-end consumables and will provide the components.

b) The other arrangement will have the drilling contractor responsible for all the drilling and completion of the well as well as all the equipment involved, including the mud pumps. There may be the situation where a clause in the contract stipulates penalties if there is any downtime in the drilling operation. It is recognised that the fluid-end components are the main source of most mud pump problems. Drilling contractors will try to maximise the service of the fluid-end consumables and if possible will run-to-failure. From Marathon's point of view, run-to-failure is neither recommended nor acceptable because of the high cost of downtime. Hence when situation (a) applies, Marathon tend to incur more fluid-end cost than may actually be needed, just to be on the 'safe side'. This is especially so if downhole problems are anticipated when drilling deep into the reservoir with higher working pressure. In this context, the decision is made to change all the fluid-end parts before proceeding into the hole section.

The fluid-end components categorised as consumables are :

1) liners, usually 1000 hours service life;
2) pistons and piston rings, where there may be 3 piston ring changes to each piston change. Pistons are changed every time a liner is changed;
3) Valves and valve seats - usually 1000 hours service life, there may be 2 or 3 valve disk changes for every change of valves.

Marathon concluded, from experience, that the fluid-end consumables are directly affected by the sand content in the mud. The ideal is 0 % sand while 2 % of sand will signal a critical situation.

Although the normal maximum pressure of mud pump is 5000 psi, the usual working pressure ranges between 2800-4000 psi. There have been cases when pressures of 4600 psi have been used when drilling through particularly soft rock formations. The surface pressure, not the downhole pressure, is deemed to have a direct effect on the fluid-end service life. The surface pressure refers to the pump pressure which consist of the friction losses inside and outside the drillpipe and the pressure drop at drill-bit during jetting and/or cleaning. The hydrostatic mud weight must be always more than the formation pressure.

Higher pump pressures may translate into greater fluid-end component wear and hence higher maintenance cost. However the higher pressure also mean greater rate of penetration (ROP) and hence favourable drilling rates and faster well completion. This translates into earlier return on investment from faster oil production. In this context, the maintenance cost of mud pump of fluid-end consumables is considered negligible when the overall situation encompass minimum drilling downtime and earlier oil and gas production. On the other hand, one mud pump fluid-end failure may mean a mere 1 hour or up to 5 days of drilling downtime, depending on the criticality of the hole section. The situation may be complicated when performing directional drilling. This may be so because when mud circulation stops during directional drilling, the drill cuttings suspended in the mud tend to start settling. The inclination of the hole means the cuttings have a shorter distance to settle, subsequently resulting in a buildup of cuttings around the drill pipe, especially round the bends (Fig. 1). When this happens, the drill pipe may become stuck and expensive downtime incurred to free the drill pipe before drilling can resume.

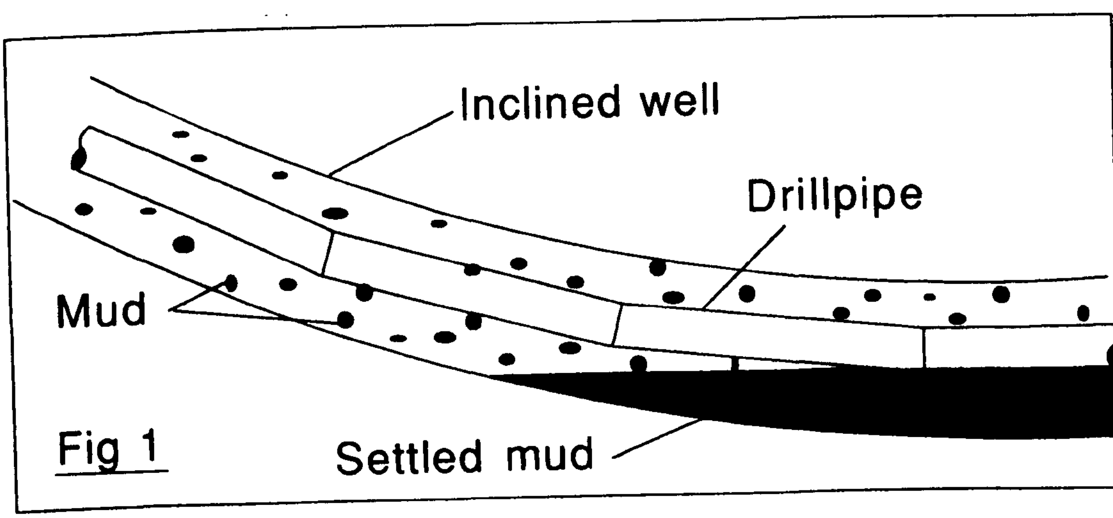


Fig 1

This situation may be minimised by proper hole cleaning. Marathon's experience has been to use the hydraulic pressure of the mud more for hole cleaning (considered more critical) and less for jetting. When drilling to depths of 4300 meters, or even up to 7000 meters, the pressure drop is considerable and may not be effective for hydraulic jetting. Hence the emphasis for more efficient hole cleaning through better mud circulation. The mud pump is not operated during tripping. When tripping-out, the volume of the drill-pipe is replaced by mud supplied from a trip tank connected to the hole. This ensure the hydraulic mud pressure is always greater than the formation pressure, effectively killing the well.

4) **Hamilton Brothers Oil & Gas Ltd.** do not own any mud pumps. These are standard equipment fitted to and supplied with the rigs contracted by them.There is a legal requirement placed on the vessel owner to have its own planned maintenance system. Hamilton Brothers felt it would not be reasonable to expect the rig owner to change its system to suit each oil company it worked for, the mainstay of any planned maintenance system is that it is consistent. They audit the rig owner's system to ensure compliance with the law.

5) **Deminex UK Oil & Gas Ltd.** do not currently own any mud pumps and therefore for drilling operations the mud pumps are supplied with the third-party chartered drilling rigs. The planned maintenance of mud pumps is considered important in order to minimise downtime on mud pump repair. Deminex therefore review the Drilling Contractor's preventive maintenance program for all rig equipment (including mud pumps) prior to contracting the rig. An example of the type of testing a mud pump cound be subjected in order to check its fitness for purpose is given in Section 5.1

On an operating rig the mud pump records would be inspected and discussions would take place with the current opertor on the mud pump performance. Deminex would also review the mud pump planned maintenance procedure and request proof that this was being conducted. The onus for mud pump maintenance remains with the drilling contractor. Deminex would insist, however, that during any maintenance work, a permit-to-work system is in operation and that all power to the pump is completely locked-out under the electricians' supervision prior to anyone working on the pumps.

5.1 HP Mud Pumps Testing

- Start up with rods disconnected
- Check proper function of lubricating system of chain cases and liner rods
- Graduclly vary spm from 0 to 120 in 20 spm steps. Run for 10 minutes at each step

- Run on no load for 2 hours at 100 spm
- Connect pony rods
- Load test by pumping through the choke
- Run separately each pump at 60 spm, choke open
- Record pressure, amps and volts at each step
- Increase discharge pressure to 2000 psi and run for 30 minutes
- Increase spm from 60 to 90 to 120
- Maintain 90 spm for 30 minutes and 120 spm for 60 minutes adjusting choke to maintain 2000 psi.

6) Amerada Hess Ltd (AHL) do not operate any platform drilling rigs until the Scott platform comes on stream in 1993. The pumps to be installed on the Scott platform, which are Emsco FB-1500 Triplex Pumps (Appendix 1), are to be maintained by the Scott drilling contractors (Appendix 2 outlines the maintenance requirements). With regards to strategy based concern, the mud pumps are considered a vitally important and integral part of the 'prime' drilling equipment, hence their maintenance is viewed in the same manner. Downtime in this area is very costly as it usually relates directly to rig down-time. As an operator, AHL depends on their drilling contractors to maintain their own pumps and hence responsibility lies with them. They do penalise contractors for unacceptable pump 'downtime' and this information is contained in the contract including specific details of penalties incurred. This open approach is deemed typical of many operators in the North Sea. AHL therefore stress the importance of pro-active maintenance and back it up with cost penalties. Historically these 'prime' movers are reliable and stand up well to the punishment they receive as part of their normal operating cycle.

6.1 Maintenance requirements of mud pumps :

The following is a guide to some of the key points which should be observed during routine inspection;

Daily :

- Observe for any unusual conditions such as loose bolt, fluid leaks, vibration or noise

- With pump shut down, check oil levels in power
- End gear case and chain guards
- Check chain lube spray nozzles for proper syray pattern
- Check oil pressure of power end lube system
- Check oil pressure of chain lube system
- Check suction/discharge damper pre-charge pressures
- Check that liner spray system is operating properly
- Add oils as required
- Check relief valve setting, set at 10% above maximum pressure for liner size being used
- Routinely observe the discharge pressure gauge

Monthly :

- Visual inspection of pump auxiliaries, associated valves pipework and hoses for leakage, damage, corrosion or abnormal running conditions
- Drain sludge traps on crankcase
- Liner spray reservoir - clean and refill as required
- Check condition of suction filter
- Remove inspection door on the rear of gear case, inspect conditon of main gear teeth and pinion gear teeth
- Remove crosshead inspection doors, inspect condition of crossheads and slides, also drain off any accumulation of water or contaminants from these areas
- Inspect drive chains for broken lines or wear
- Record pump operating parameters for condition monitoring
- Piston rod clamp for tightness
- Check condition of piston and intermediate rods

Three Monthly

- All monthly inspections
- Take oil samples of the gearcase and chaincase oils

Six Monthly

- All monthly inspections

- Change gearcase and chaincase oils
- During the flushing procedure of the gearcase thoroughly clean the oil throughs and the compartment on top of the crossheads guides. Remove covers from settling chamber and thoroughly clean. Also clean or replace the filter element in the air breather cap and clean suction screen.
- Check crosshead to guide clearance
- Check shaft end clearance and gear backlash

Condition Maintenance
- Oil changes will also be done as condition of oil as will the inspeciton of bearings
- Change out of fluid end parts, such as liners pistons, valve seats and valves will be on condition or as dictated by operations.

7) **Sedco Forex** operates a flexible maintenance system which can accomodate the dictates of the client oil company and its own maintenance schedule. In conjunction with the requirements of the oil company, the depth of the hole drilled and the type of geological formation encountered are also taken into consideration.

The maintenance system fundamentally follows a 2-fold basis :

1) - On-going checks are conducted regularly on systems such as oil lubrication system, drive ends etc.
   - Monthly checks are carried out by the drilling department on the fluid-end components.
   - Quarterly checks are made by the mechanical and electrical departments, more on the drive-ends and 'prime movers' eg. electrical motor, crankcase etc.

2) The drilling program may encounter specific situations where downhole problems are anticipated, in which case fluid-end consumables like liners and pistons are replaced with new. The changed out parts especially the liners are marked and the number of running hours noted. These liners are stored to be re-used for less critical situations.

The latest development include a recommendation from their French research department to consider the use ceramic liners and self-aligning mud pump rods to improve the service life and efficiency.

Appendix 3

SURVEY OF SWAB FAILURE CHARACTERISTICS

(50 responses from five North Sea offshore drilling installations)

1. Causes of Swab changes

a) due to material failure / rubber extruded 10

b) misaligment 7

c) due to old age 19

d) material deterioration due to corrosive mud . . . 40

e) gradual wear-out 38

2. Percentage of swab changes-out due to material failure vs gradual wear-out......average range : 5-30% ; mainly 10-15% (for material failure)

3. Swab failure due to the following :

a) abrasive sand content. Percentage (14 : 2-3 % / 12 : 1-4 %) . 41

b) fatigue due to cyclic loading 10

c) excessive gap bet. liner ID and piston OD (rubber extrusion) . . 14

d) heat, due to pumping, not adequately removed 26

e) liner streaking, premature wear 39

f) excessive speed/pressure (2 : pressure) 19

g) insufficient suction pressure 1

4. No. of swabs change-out for every liner change . . . 1 : 2

. . . 2 : 25

. . . 3 : 29

. . . 4 : 2

. . . 5 : 0

Comments & suggestions from respondents :

- No real problem with swab failure since converting to South-West Components
- Since using South-West pistons swab, material failure has been cut dramatically. Suspect it is because South-West rubbers are bonded to the piston head whereas National & Mission are not
- Since changing from Mission swabs to Dual Duro, the percentage of failures due to material failures has dropped
- Misalignment cause severe damage to pony rod extension rods, piston rods & clamps
- Since changing from Mission swabs to Dual Duro, the percentage of failures due to material failure has dropped
- Excessive hours (for swab change-out)
- Poor coolang facility (influence on swab failure)
- High pump pressures/pump rate (cause of swab change-out)
- systematic change-out of pistons; 1st after 400 hrs, 2nd after 250 hrs, 3rd 150 hrs
- Ceramic liners would allow 5 piston swabs per liner and approximately 2000 hr on liner
- Over 1% sand - minor effect; 4% and above - noticeable effect
- Keep sand content as low as possible
- Using water-based mud does affect swabbing more, compared to OBM
- A better cooling system using soluble oil & heat exchanger could extend liner & piston life but cost of such a system would not be economical to run
- Swab/liner change-out depend on running hours of liner, high pressure & high spm. It would be interested to see how many hours would get if South-West piston heads are used in ceramic liners
- South-West swabs & liners are felt to be a good combination, less swabs used. The combination of ceramic liners & South-West swabs would be interesting to see, but would the results justify the extra expense?
- South-west piston swabs & liners are more economical, less swabs used
- Heat failures seem to occur only when using closed system without adequate heat

exchangers

- Most failures experienced are due to gradual wear-out

Appendix 4

A SUMMARY OF BMW'S COMPUTERISED PM SYSTEM FOR CARS

BMW's computerised system for its cars have a variable PM schedule, dependent on the driver's driving habits, distance covered and the time elapsed. The system clock will still run, irrespective of car being idle or running. Eg. if the car is not used for 18 months, the system clock would flash amber light to indicate servicing requirements after the elapsed time period. The distance is tabulated via speed pulses picked up from instrumentation in the drive axle in the ratio of 6 pulses to each revolution of the wheel. In this way, the distance covered are cumulated.

The engine speed is picked/monitored via frequencies emitting from the ignition system. Although there is a threshold value for engine speed, the frequency of usage and operation at high RPMs would mean more wear and tear, and hence lead to an increased servicing schedule per period of time (eg year). Hence the 3 factors which affect the servicing/maintenance schedule are :

1. engine speed (via frequency)
2. distance (via speed pulses)
3. time elapsed (via system clock).

Distance and time may be pre-determined, but the engine speed varies and depend on driving conditions and habits, load, road conditions etc. The CPU of BMW's system will correlate these 3 factors and determine the PM schedules. The implication is that the system is programmed to operate on a series of set targets of usage/performance, perhaps similar to expert systems which uses rules- or knowledge-based principles. It seems many experiments have been conducted before reaching the conclusive targets of performance or limits of failure. The experiments helped determine the predictive maintenance limits and hence the PM schedules. BMW considers these knowledge as their trade secrets and would not elaborate.

In this context, the fixed criteria eg. time and distance may be likened to the mud pump's valve seats or piston seals which are of polyurethane material and would wear out any way; while the ngine spee likened to the variable load conditions of the mud pumps which cause irregular wear patterns in the fluid end expendables, notably the urethane valve seats and piston seals, as well as the liners and pistons. The varying factors may be many, eg sand in the mud, depth etc. but the main indicator is the discharge pressure and/or volume delivered. In terms of condition monitoring, it is felt that pressure should be the main factor.

Appendix 5

FAILURE MODE OF MUD PUMPS

(DUPLEX & TRIPLEX)

MUD PUMP DOWNTIME

- LINERS and/or PISTON BODY
- VALVES and SEATS
- PISTONS
- PUMP KNOCK
- DISCHARGE
- SUCTION

LINER and/or PISTON BODY

- Excessive wear
- Streaking of liner bore and piston rubbers
- Pitted liner
- Localised wear on one side of piston or liner
- Swollen and torn piston rubbers
- Swelling of face and chunks of rubber
- 'Burned' piston rubbers in dry liners
- Torn or broken lips on rubber or end of piston
- Short part life
- Flashing

VALVES and SEATS

- Fluid cut sealing members of parts
- Fluid cut taper of seat and pump deck
- Damaged pump deck
- Abnormal wear or breakage
- Too much pressure from supercharge pump

PISTON

- Piston failure
  - Excessive gap clearance between piston flange and liner bore
  - Jetting fluid
  - Rapid wear
  - Piston rubber burn — inadequate cooling
  - Worn piston in new liners or new piston in worn-out liners

SUCTION

- Low volumetric efficiency
  - high mud temperature
  - entrained air/gas
  - excessive pump speed

PUMP KNOCK

- Mechanical knock
  - incorrect size and part number
  - valve lift incorrect
  - loose liners or piston
  - loose pony rods, lock-nut
  - breakage, loosening or wearing of parts
  - excessive wear of crosshead slides
- Hydraulic knock
  - too high mud temperature
  - poor priming
  - air hammer
  - low mud level
  - leak occurs in suction line
  - whirlpool forms in tank/pit
  - suction valves sticking
- At high speeds only
  - clogged suction strainer
  - large deposits of sand and drilled solids

DISCHARGE

- Severe discharge vibration
  - resonate situation
  - mud composition
  - discharge pressure
  - pump speed

Appendix 6

BASIC PROGRAM FOR CALCULATING THEORETICAL MECHANISM FORCES OF RECIPROCATING MACHINES

```
10   REM THEORETICAL MODEL FOR RECIPROCATING MACHINES
20   PRINT "PROVIDE INPUT FOR SPEED (RPM),CRANK RADIUS
     (R,INCHES),CONROD LENGTH(L,INCHES),CROSSHEAD MASS (M,LBS)"
30   INPUT RPM,R,L,M
40   PRINT "PROVIDE INPUT VALUE FOR DISCHARGE PRESSURE "
50   INPUT DISCHP
70   PRINT "THEORETICAL MODEL FOR MECHANISM FORCES OF
     RECIPROCATING PUMPS "
80   PRINT "---------------------------**********--------------------------- "
90   PRINT "SPEED IS ";RPM;"RPM        ";"DISCHARGE PRESSURE IS ";
     DISCHP;"N/M2 "
100  PRINT "CRANK RADIUS IS ";R;"INCHES ";"CONROD LENGTH IS ";
     L;"INCHES "
110  PRINT "CROSSHEAD MASS IS ";M;"LBS "
120  PRINT "CRANK ANGLE ";"ACCELERATION ";"LINEAR VELOCITY ";
     "CONROD LOAD ";"REACTION FORCES "
130  LET PI=3.141592
140  LET OMEGA=RPM*PI/30
150  LET R=R/12
160  LET L=L/12
170  LET C=R/L
180  FOR J=10 TO 360 STEP 10
190  LET A1=1
200  LET A2=C+(1/4)*C^3+(15/128)*C^5
210  LET A4=-(1/4)*C^3-(3/16)*C^5
220  LET A6=(9/128)*C^5
230  LET X=R*(A1*COS(ANG1)+A2*COS(2*ANG1)+A4*COS(4*ANG1)+
     A6*COS(6*ANG1))
240  LET A=-OMEGA*OMEGA*X
250  LET ANG1=J*PI/180
260  LET W=C*SIN(ANG1)
270  LET ANG2=ATN(W/SQR(-W*W+1))
280  LET V=-OMEGA*R*(A1*SIN(ANG1)+A2*SIN(2*ANG1)+A4*SIN(4*ANG1)+
     A6*SIN(6*ANG1))
290  IF ANG1<PI THEN FP=0
300  IF ANG1>PI THEN FP=DISCHP*SIN(ANG1-PI)
310  LET F=(M*A-FP)/COS(ANG2)
320  LET NR=M*9.81+(M*A-FP)*TAN(ANG2)
330  PRINT J,INT(A*100+0/5)/100,INT(V*100+0.5)/100,INT(F*100+0.5)/100,
     INT(NR*10+0.5)/10
340  NEXT J
350  END
```

Appendix 7

Procedure to import data from oscilloscope to PC for energy calculations

1) After connecting oscilloscope to personal computer (PC), put wear test system on line and enter force profile on screen.
2) Run program 7A on PC which triggers the scope to record and store force profile on screen after 1000 cycles. An ASCII file called CHN1.PRN is opened on disk which transfers the data of the force trace on the screen to the file. The force trace is made up of 1022 lines of numerical data.
3) Run program 7B which would identify the two positive slopes to indicate zero crossings which form one period (one cycle).
4) Having selected the two reference points, program 7B would next delete and segregate out other data outside these two reference points. All data between these two reference points would be selected in suitable increments to yield 36 readings which correspond to 10° per reading, and saved into program 7C as file with suitable extension 'PRN' which can be loaded onto any spreadsheet for calculation of energy consumption.

Program 7A : this is a BASIC program which captures and stores all the numerical data on the oscilloscope, of which there are 1022 lines of data. The program reads:

```
10   CLOSE
20   KEY OFF: CLS
30   TIMER ON
40   T1=TIMER
50   FOR X= 1 TO 1000
60   GOSUB 500
70   IF CHVAL > 2500 THEN GOTO 80
80   IF CHVAL < 1000 THEN GOTO 90
90   LOCATE 22,29: PRINT "NUMBER OF REVOLUTIONS =";X
100  IF X=1000 THEN GOTO 120
110  NEXT X
120  CLOSE
130  OPEN "COM2:9600,N,8,1,DSO,LF" AS #1
140  GOSUB 600
150  PRINT #1,"HSA,ST1VS;ST2VS"
160  A$="HSA="
170  B$="ST1VS="
180  C$="ST2VS="
```

```
190    WHILE NOT EOF(1)
200    X$=INPUT(1,#1)
210    IF INSTR(J$,A$) THEN J$=""
220    IF INSTR(K$,B$) THEN K$=""
230    IF INSTR(L$,C$) THEN L$=""
240    J$=J$+X$
250    K$=K$+X$
260    L$=L$+X$
270    H=VAL(J$)
280    V=VAL(K$)
290    P=VAL(L$)
300    WEND
310    Q$="ST1="
320    OPEN"CHN1.PRN" FOR OUTPUT AS #3
330    GOSUB 600
340    PRINT #1,"ST1;ST2"
350    Y=.035*V
360    WHILE NOT EOF(1)
370    X$=INPUT$(1,#1)
380    IF INSTR( A$Q$) THEN A$=""
390    A$=A$+X$
400    IF X$="," THEN A=VAL(A$):COUNT=COUNT+1
410    IF X$="," THEN C=A*Y:PRINT #3,C
420    IF X$=2," THEN A$=""
430    CLOSE #3: CLOSE #1
440    T2=TIMER
450    RUN=T2-T1
500    OUT &H702, 2
510    OUT &H702, 1 * &H10 + 3
520    CHVAL = ( INP(&H701) AND &HF) * 256 + INP(&H700)
530    RETURN
600    PRINT #1,"ARM;SRQV"
610    T3=TIMER
620    T4=TIMER
630    TIME=T4-T3
640    IF TIME < 5 THEN GOTO 620
650    RETURN
```

Program 7B : this is a BASIC program which determines the two reference points on the force trace which form a cycle, assuming the cycle starts at the first zero force reading :

```
10     OPEN "CHN1.PRN" FOR INPUT AS #1
20     I=1
30     OPTION BASE 0:DIM Y(1023)
40     WHILE NOT EOF(1)
50     INPUT #1,Y(1)
60     IF(Y(I-1)<0AND Y(I)>=0)THEN G=(I-1)+(-Y(I-1))/(Y((I)-Y(I-1)):K=K+1
70     IF Z<Y(I) THEN P=Y(I)
```

```
80    IF P>Y(I) THEN P=Y(I)
90    IF K=1 THEN A=G
100   PRINT " FIRSI ORDER = ";:PRINT A
110   PRINT " SECOND ORDER = ";:PRINT G
120   WEND
130   CLOSE #1
```

Program 7C : having defined the two reference points on the force trace which form a cycle, all data between these two reference points will be read, squared and square-rooted to remove the negative signs, and stored on disk for import to any spreadsheet for energy calculations :

```
10    LET A=A
20    LET G=G
30    OPEN "CHN1.PRN" FOR INPUT AS #1
40    I=1
50    WHILE NOT EOF(1)
60    INPUT #1,Y(I)
70    IF I<A THEN GOTO 130
80    IF I>G THEN GOTO 150
90    LET BETA=(Y(I))^2
100   LET ALPHA=SQR(BETA)
110   LET X=ALPHA
120   PRINT USING "###.##";Y(X);
130   I=I+1
140   WEND
150   PRINT USING "##### numbers found";(I-1)
160   CLOSE #1
170   OPEN "ENRGY.PRN" FOR OUTPUT AS #1
180   LET PHI=(G-A)/36
190   FOR I=A TO G STEP PHI
200   H=Y(X)
210   PRINT #1,H
220   NEXT I
230   CLOSE
```

Appendix 8

Properties of materials :

- Mechanical properties of water-based mud

• Mud weight	1.378 kg/lit
• Plastic viscosity (PV)	32 centipoise
• Yield point (Y_p)	89.77 Pa
• Gel strength	14.6/50.1 Pa
• Particle size	19-50.9 μm
• pH value	10

- Chemical composition of water-based mud

• Water	80%
• Reactive fractions of mud (eg bentonite/attapulgite)	3%
• Low gravity solids eg sand/chert/drilled solids	5%
• High gravity solids (eg barite/lead suphide)	6%

  • Chemical additives to control viscosity, yield point, gel and fluid loss mud properties: mud thinners eg phosphates/ surfactants; mud thickners eg lime/cement, polymer extenders eg sodium polyacrylate.

- Mechanical properties of abrasive sand

• Finess grade (calcium silicate)	
• Grain size	0.01-0.5 mm
• Typical conductivity	10 millisiemens/m
• Chloride content (ppm)	< 7
• Hardness (mohs)	5 - 6
• Bulk density (kg/dm^3)	1.35
• Specific gravity (kg/dm^3)	2.8

- Chemical composition of abrasive sand (% range)

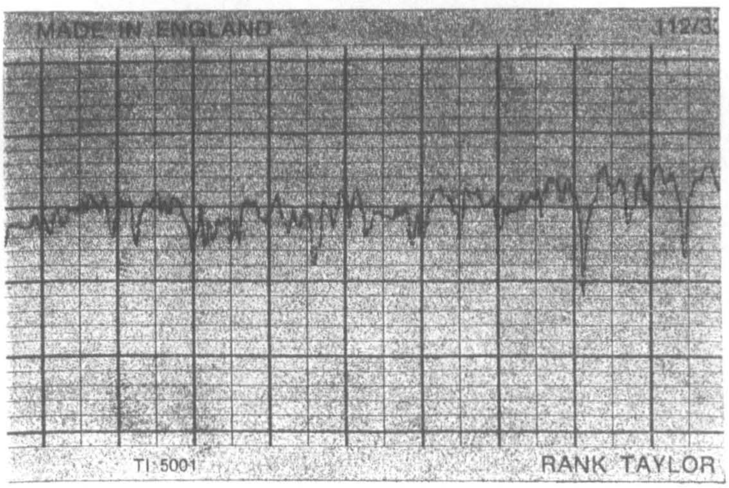
• Iron oxide	0 - 10
• Silica	35 - 45
• Aluminium oxide	10 - 15
• Calcium oxide	35 - 45
• Magnesium oxide	6 - 10
• Manganese oxide	0 - 10
• Free silica	< 1%

- Properties of liner sleeve
  - 28% chrome
  - Hardness : Rockwell C - 58
  - (further details of metallurgical composition unobtainable from mud pump manufacturers which cited proprietary information confidentiality)

Appendix 9

COMPARISON OF SURFACE ROUGHNESS OF 28% CHROME STEEL LINER

Surface roughness after 0 strokes : CLA 0.075 μm



Surface roughness after 1.5 x 10^6 strokes : CLA 0.035 μm