THE EFFECTS OF SOLID CONTAMINATION IN HYDRAULIC FLUIDS ON AXIAL PISTON PUMPS.

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SUMMARY

The fluids used in hydraulic systems inevitably contain large numbers of small, solid particles, a phenomenon known as 'fluid contamination'. Particles enter a hydraulic system from the environment, and are generated within it by processes of wear. At the same time, particles are removed from the system fluid by sedimentation and in hydraulic filters.

This thesis considers the problems caused by fluid contamination, as they affect a manufacturer of axial piston pumps. The specific project aim was to investigate methods of predicting or determining the effects of fluid contamination on this type of pump.

The thesis starts with a theoretical analysis of the contaminated lubrication of a slipper-pad bearing. Statistical methods are used to develop a model of the blocking, by particles, of the control capillaries used in such bearings. The results obtained are compared to published, experimental data. Poor correlation between theory and practice suggests that more research is required in this area before such theoretical analysis can be used in industry.

Accelerated wear tests have been developed in the U.S.A. in an attempt to predict pump life when operating on contaminated fluids. An analysis of such tests shows that reliability data can only be obtained from extensive test programmes. The value of contamination testing is suggested to be in determining failure modes, and in identifying those pump components which are susceptible to the effects of contamination. A suitable test is described, and the results of a series of tests on axial piston pumps are presented and discussed.

The thesis concludes that pump reliability data can only be obtained from field experience. The level of confidence which can be placed in results from normal laboratory testing is shown to be too low for the data to be of real value. Recommendations are therefore given for the ways in which service data should be collected and analysed.

Contamination
Hydraulics
Pumps
Reliability
Testing

To Heather Louise

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GLOSSARY

Hydraulics

Hydraulic System	:	A system of power transmission in which an incompressible fluid is used as a power transmitting medium.
Hydraulic Fluid	:	Any fluid used as the power transmitting medium in a hydraulic system.
Hydraulic Pump	:	A device which converts mechanical power into hydraulic fluid power.
Hydraulic Motor	:	A device which converts hydraulic fluid power into mechanical power.
Fluid Power	:	The theoretical rate at which a fluid could do work by virtue of its flow and pressure.
Hydrostatic Transmission	:	A type of hydraulic machine which provides a continuously variable ratio between input and output speeds. Usually consists of a coupled hydraulic pump and motor.

Fluid Contamination

Fluid contamination	:	The presence of solid particles in a hydraulic fluid. (Note that some authors use the term to refer to the presence of water, other liquids, gas or organic matter in a hydraulic fluid.)
Particle	:	A small, solid body, usually defined to be of a "diameter" between 1 and 100 microns.
Contamination Level	:	A quantitative description of the mass, volume or numbers of particles present in a hydraulic fluid.
Gravimetric Analysis	:	Any technique for determining the mass of particles in a fluid.
Particle Counter	:	A device for detecting the numbers of particles (of a specified size or sizes) in a sample of fluid.

Electro-optical counter (Hiac, Royco) Coulter Counter	Different types of particle counter.
Contamination Sensitivity	A quantitative or semi-quantitative measure of the ability of an item of hydraulic equipment to operate on contaminated fluids.
Particle Ingression	The addition of contaminating particles to a hydraulic fluid, whether by generation within the system or by entry from the environment.
Air cleaner fine test dust	A dust, composed predominantly of quartz and feldspar, widely used in contamination testing.
Silting	The progressive obstruction by particles of a fine clearance.
Hydraulic Filter	A device for removing particles from a hydraulic fluid.
$Beta_{10}$ ratio	A measure of the effectiveness of a hydraulic filter. The ratio represents the number of particles larger than 10 microns upstream of a filter, compared to the number of such particles downstream of the filter.

<u>Tribology</u>

Tribology	The study of wear, friction and lubrication.
Wear	The progressive loss of substance from the operating surface of a body, occurring as a result of relative motion at the surface.
Abrasive (or cutting) Wear	Wear by the displacement of material caused by hard particles or protruberances.
Adhesive wear (Rubbing wear)	Wear by the transferrence of material from one surface to another during relative motion, due to a process of solid-phase welding.

Erosive Wear (erosion)	: The loss of material from a solid surface due to relative motion in contact with a fluid which contains solid particles.
Running-in	: The process by which machine parts improve in conformity, surface topography and frictional compatability during the initial stages of use.
Lubrication	: The reduction of frictional resistance and wear between two load-bearing surfaces through the application of a lubricant.
Hydrostatic lubrication	: Lubrication maintained by an externally pressurised oil supply system.
Hydrodynamic lubrication	: Lubrication self-generated by favourable surface geometry and a relative sliding velocity between components.
Elasto-hydrodynamic lubrication	: Hydrodynamic lubrication where favourable surface geometry is produced by elastic deformation of the surfaces due to induced fluid pressures.
Ferrography	: A method for preparing a microscope slide of wear particles from a fluid sample. (See also appendix A.)
Talysurf/ Talylin	: Methods of producing a trace of a surface profile.

NOTATION

Symbol	Description	Defined on page
a	Radius of circle of contact between a particle and a surface.	61
A	Constant in Meyer's relation (section 4.5).	64
A_B	Bearing area exposed to contamination.	72
A_o	Initial system fluid contamination level.	
A, A(t)	System fluid contamination level at time t.	31
$A_a(t)$	Effective, abrasive contamination level of system fluid at time t.	126
b	Bearing depth.	191
B	Constant in log-log^2 particle size model.	82
B(t)	System (boost circuit) fluid contamination level at time t.	209
c	Constant relating pump leakage flow to pump clearances.	78
d	'Diameter' of an impression left by a particle.	64
d_c	Capillary diameter.	93
d_p	Piston diameter.	54
E_1; E_2	Young's modulus.	-
f(t)	Probability density function at time t.	104
F(t)	Cumulative probability density function at time t.	110
g_1, g_2	Time constants for exponential changes in contamination levels.	210
h	Clearance between pump components.	54
h_o	Mean bearing equilibrium clearance.	69
h_o	Maximum clearance between a flat slipper and a 'dished' slipper plate (section 6.4.9).	184

Symbol	Description	Defined on page
h'	Clearance at radius r between a flat slipper and a 'dished' slipper plate (Section	184
i,j	Individual reference for an array of variables.	-
k	Index in a wear relation.	76
K_c	Constant relating flow through a capillary to fluid viscosity and capillary pressure differential.	53
l	Length of wear track produced by a particle.	71
L_c	Capillary length	93
m	Index in Meyer's relation (section 4.5).	64
M_h	Log-mean particle size.	82
n	Number of pumps tested in a reliability prediction (section 5.5).	110
n,n_i	Particle concentration per unit fluid volume.	71, 77
n_c	Critical particle concentration per unit fluid volume.	94
$n(h)$	Particle size probability density function.	82
$n_a(t)$	Concentration of abrasive particles per unit fluid volume at time t.	126
N_w	Number of particles entering a bearing per second.	71
$N(h)$	Number of particles of diameter greater than h per unit fluid volume.	82
$\bar{p}$	Representative material hardness.	71
P_i	Fluid pressure at a hydrostatic bearing pocket.	54
P	Unbalanced load on a slipper.	69

Symbol	Description	Defined on page
P_m	Mean pressure between a particle and a surface.	64
P_r	Fluid pressure at radius r	183
P_s	Pump delivery pressure.	54
dP/dr	Fluid pressure drop per unit distance.	-
q	Slipper leakage flow.	53
q_p	Leakage between parallel plates.	191
q_{tot}	Total pump leakage flow.	77
q_w	Leakage flow/unit width between plates having sinusoidal wear tracks.	178
Q	Pump delivered flow.	77
Q_B	Flow through boost circuit filter.	209
Q_f	Filter flow.	32
Q_f	Final pump flow in a contamination test (section 6.4.8).	167
Q_o	Initial pump flow in a contamination test.	167
Q_v	Theoretical maximum pump delivered flow.	77
r	Particle radius (section 4.5).	63
r	Radius.	-
R, R'	Contaminant ingression rate.	32, 198
Ri	Inner radius for slipper land.	54
Ro	Outer radius of slipper land.	54
$R(t)$	Pump reliability at time t.	104
s	Index in a wear relation.	78
S_A	Number of abrasive wear particles entering a bearing per second.	82
t	Time.	-
t_a, t_b	Ratios for contaminant transfer between main and boost circuits in a hydrostatic transmission.	209

Symbol	Description	Defined on page
T_i	Failure time for pump i.	110
V, V_A, V_B	Hydraulic system fluid volumes.	32
V_c	Critical fluid volume.	94
V_p	Particle volume.	71
V_w	Volume of wear metal removed by a particle.	71
$\frac{dV_{TOT}}{dt}$	Total volumetric wear rate.	72
w	Bearing unblocked width.	90
W	Load.	61
x	Particle 'diameter'.	15, 17
X	Mesh size of rigid filter model.	58
y	Actual particle count in a fluid volume.	94
y	Height of sinusoidal wear track (section 6.4.9).	179
Y	Material yield stress.	-
z(t)	Failure rate.	94
dz/dt	Surface wear rate (linear).	72
α	Parameter of a Weibull distribution.	113
α	Parameter describing 'dishing' of a worn surface (section 6.4.9).	183
α_v	Particle shape factor.	71
β	Parameter of a Weibull distribution.	113
β_{10}	Filter performance as measured in an ISO multipass filter test (see glossary).	-
γ, γ_i	Pump wear coefficient.	77
η	Filter efficiency.	32

Symbol	Description	Defined on page
θ	Parameter of an exponential distribution (mean life)	94
$\hat{\theta}$	Estimate of θ.	110
μ	Fluid kinematic viscosity.	-
μ	Mean of a Gaussian distribution.	-
ρ	Constant relating contaminant generation rate to existing fluid contamination level.	198
σ	Standard deviation.	-
σ_1, σ_2	Poisson's ratio	-
τ, τ', τ''	Time constants for changes in contaminant concentrations	126, 127
ϕ	Angle defining the aspect ratio of a particle.	70

ABBREVIATIONS

ACFTD	:	Air cleaner fine test dust.
ANSI	:	American National Standards Institution.
ASLE	:	American Society of Lubrication Engineers
ASTM	:	American Society for Testing and Materials.
BFPR	:	Basic Fluid Power Research programme of OSU
BHRA	:	British Hydromechanics Research Association
BSI	:	British Standards Institution
CHL	:	Commercial Hydraulics Ltd. (Gloucester Division)
CSI	:	Commercial Shearing Inc.
I.Mech.E.	:	Institution of Mechanical Engineers
ISO	:	International Standards Organisation
MOD	:	Ministry of Defence
NAS	:	National Aerospace Standard
NFPA	:	(US) National Fluid Power Association
OSU	:	Oklahoma State University, Fluid Power Research Center.
SAE	:	Society of Automotive Engineers
SP	:	Special Publication
μ	:	Micron

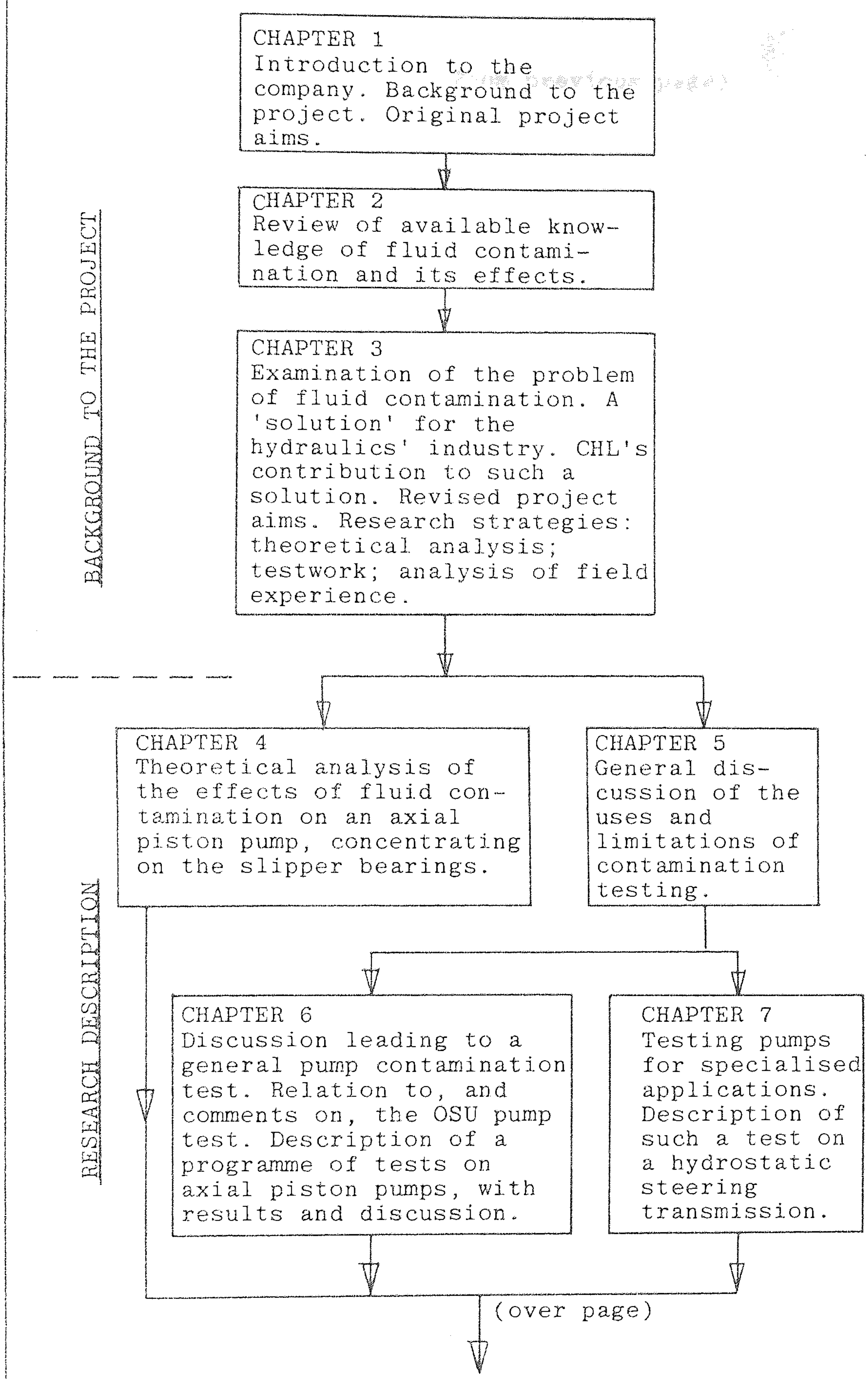

Figure 0.1. THESIS PLAN (continued overleaf)

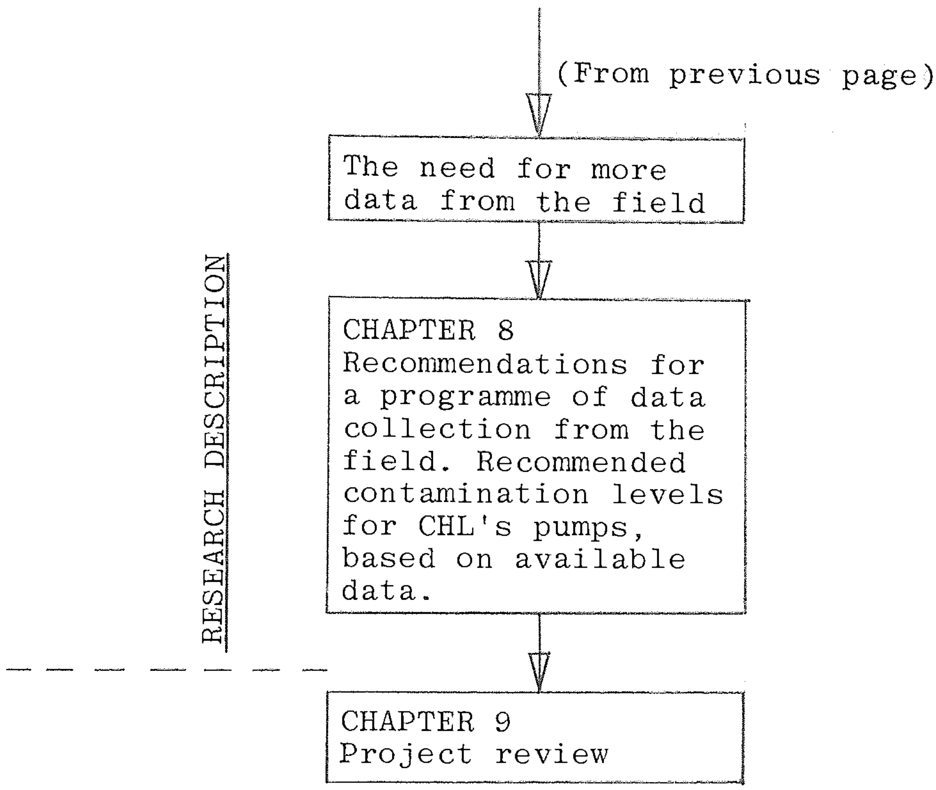

Figure 0.1. THESIS PLAN (continued)

1.0 INTRODUCTION

1.1 Preface

This thesis describes work conducted through the Total Technology higher degrees scheme of the University of Aston in Birmingham. The work was sponsored by the Science Research Council, and by the Gloucester Division of Commercial Hydraulics Ltd. (CHL), and was conducted at the Company's works, and in the departments of mechanical engineering and physics at Aston University. In the second year of the project, a period of three months was spent working and studying at the Fluid Power Research Centre at Oklahoma State University, U.S.A.

1.2 Commercial Hydraulics Ltd.

Commercial Hydraulics is a wholly-owned U.K. subsidiary of a large American group called Commercial Shearing Inc. (CSI). CSI manufactures a range of engineering products, and has eighteen plants throughout the World. In 1978-79, the group employed 3500 people, and had sales of $ 200 millions.

Commercial Hydraulics is entirely concerned with the design, development and manufacture of hydraulic equipment, and has two British plants. A factory at Bedford manufactures valves, gear pumps and gear motors, whilst the Gloucester Division is primarily concerned with equipment based on axial piston pumps and motors. (The operation of these units is described in chapter 2.) Throughout the rest of the thesis, the name "Commercial Hydraulics" will refer to the Gloucester division of the company.

For the duration of this research, (1976 to 1979) CHL's main project was to develop a high-power hydrostatic steer transmission for a military vehicle. However, CSI intended that the Gloucester facilities should also act as an engineering research and development centre for the group's European operations in hydraulics, and the Gloucester division therefore had an interest in all aspects of hydraulic equipment and systems.

1.3 Company Aims

There are many theories as to the economic aims of business organisations. Without entering into a lengthy debate, it may be argued that CHL's main objective is to survive as an organisation by maintaining, or preferably by expanding its sales of hydraulic equipment. (It must also do so at a profit and provide an acceptable return on employed capital.) CHL can achieve its main objective either by expanding its share of the existing market, or by increasing the overall market for hydraulic equipment in competition with alternative technologies.

A recent NEDO report [1] examined how users of hydraulic equipment select their suppliers. Users gave as commercial factors (in order of importance) delivery reliability, good communications, after-sales-service, and delivery length. But 42 out of 50 companies interviewed stated that product "quality" is more important than price. The precise nature of "quality" was not defined in the report, but it is possibly a combination of good technical performance and high reliability, achieved through sound engineering design and careful manufacture. The importance of equipment reliability may be appreciated by considering the nature of hydraulic systems and the applications in which they are used.

1.4 Hydraulic Systems

In an hydraulic system, a pressurised fluid is used to transmit power between various points in a hydraulic circuit. Fluid pressure and flow ('fluid power') are usually generated by a hydraulic pump driven by a prime mover. Fluid power may be reconverted to mechanical power in a hydraulic motor, if rotary motion is required, or in a hydraulic cylinder, if linear motion is needed. This variety of motion makes hydraulic systems very versatile. Linear and rotary output speeds can also be varied fairly easily, in contrast with electric drives. Further advantages of hydraulic equipment are high power densities, rapid response, and the facts that the working fluid can also be used as a lubricant, coolant, corrosion inhibitor, and as a medium for transmitting control signals. These factors allow hydraulic systems to be very compact compared to systems based on competing technologies.

The above features of hydraulic systems have led to their widespread use in construction and earthmoving equipment, machine tools, aircraft and missile flying controls, ships, agricultural equipment, and in process plant in the mining, mechanical handling, chemicals and metal-working industries. In all these applications, the costs incurred through any failure of the installed hydraulic systems are likely to be considerable. The main expense is not that of replacing failed equipment, which will have probably cost only a small proportion of the cost of the original plant. The bulk of financial losses stem from lost production while plant is 'down' for repair.

1.5 Hydraulic Equipment Reliability

It can be appreciated therefore that users of hydraulic equipment are likely to be more concerned about equipment reliability than they are about the initial cost of a system (although this latter aspect will still be a major consideration). This being the case, CHL may be able to improve its market share if it can produce and sell highly reliable pumps. Conversely, the company risks losing sales if its products are less reliable than is acceptable to its customers. Similar remarks apply to the hydraulics industry as a whole. It has long been suspected that poor equipment reliability has been a major factor limiting the market penetration of hydraulic systems.

At the start of this project, CHL was aware of the importance of product quality, and was interested in any aspect of the design, manufacture or operation of hydraulic equipment which might affect the reliability of its products. The company had known for some time that the overall condition of the fluid used in a hydraulic system (usually known as the "working fluid") can have an effect of the reliability of system components in general and of axial piston pumps in particular.

One aspect of fluid condition is the presence in a fluid of large numbers of microscopic, solid particles, a phenomenon which will be referred to as "fluid contamination". (It should be noted that some authors use the same term to refer to the presence of water, air, bacteria, etc.) CHL wanted to learn more about the nature of fluid contamination, of its effects, and of the means to control it, and the company sponsored this research in pursuit of these aims.

Although determined primarily by factors described above, the research was given further impetus by two other developments.

1.6 American Research in Fluid Contamination

A considerable amount of research into fluid contamination has been conducted by organisations in the U.S.A. The majority of this work has been undertaken by staff at the Fluid Power Research Center at Oklahoma State University (OSU), where the subject has been studied since 1959.

Some of the work at OSU has led to the development of a test method for hydraulic pumps. This test has been designed to give results which will allow the determination of the life of a pump when operating on fluid contaminated to a specified level. The test is under consideration as an ISO standard, and is already being employed by users of hydraulic pumps, particularly in the U.S.A., to compare the performance of competing pump designs.

The OSU test undoubtedly represents a potential threat to CHL's sales, and the company wanted to know more about it. Specifically, the company needed to know if the test is valid and, if it will be applied to CHL pumps anyway, how the company's products might be improved so that they perform better in the test.

1.7 Vehicle Steering Transmissions

It has been mentioned already that, at the start of this research,CHL's main project was a vehicle steering transmission. There are substantial advantages to be gained from operating such transmissions on the same fluid as is used in the vehicle's main gearbox and transmission. However, this does mean that the fluid is likely to be heavily contaminated, especially if 'wet' brake and clutch plates are employed. This was the case with CHL's hydrostatic transmission, and both CHL and the vehicle manufacturers were anxious to know the likely effects of fluid contamination on the life and operation of the steering transmission.

1.8 Initial Project Objective

Although CHL wanted to gain information on all aspects of fluid contamination, the main problem, as initially perceived by the company, was to determine whether or not it is possible to rate the ability of a hydraulic pump to operate on contaminated fluids. It was on this basis that the project proceeded.

The first task was obviously to find out about fluid contamination, its size, nature, concentration and effects. These will be described in the next chapter.

2.0 FLUID CONTAMINATION

2.1 Introduction

Previous researchers into fluid contamination have tended to concentrate on one of four separate, though related areas. These are: the measurement and description of fluid contamination levels; the determination of filter performance; the sensitivity of hydraulic equipment to operation on contaminated fluids; the precise nature of the effects produced by contaminants.

Until the mid 1960's, research was largely concentrated on the problems of accurately determining fluid contamination levels. Until this could be done, it would have been difficult to have conducted work in other areas. It will be shown, later in the thesis, that the severity of the effects of fluid contamination are related to both the numbers of particles present, and to their 'size'. (The definition of particle size is considered in section 2.3). To determine the contamination level of a fluid sample, a known proportion of the particles in the sample must therefore be counted and allocated to different size ranges. (Particles in hydraulic fluids always possess a broad range of sizes.)

It has generally been assumed that the particles in hydraulic fluids which are likely to cause damage are those of a size larger than or equal to the clearances between components in hydraulic equipment. The hypothesis put forward was that smaller particles should pass through the equipment without causing damage, although Wusthof and Hezemann [2] suggest that, to avoid silting, the maximum permitted particle size should not exceed one third of the smallest clearance.

Farris [3] gives values of the clearances between components in a variety of hydraulic equipment. These are shown in table 2.1.1. The clearances in CHL's products can only be considered in relation to the operation of an axial piston pump.

2.2 Axial Piston Pump Operation

Figure 2.2.1. shows a CHL pump design. The shaft (1) is driven by a prime mover (not shown), and drives, in turn, the cylinder block (2). As the block rotates, the pistons (3) are forced to reciprocate in the piston bores (4). This motion draws in fluid through the low pressure port (5) and discharges it, half a revolution later, through the high pressure port (6) in the port-plate (7). The pistons are supported by slipper-pad bearings (8) which 'float' on the slipper-plate (9). The figure also shows the pump's associated boost pump (10), relief valve (11) and servo controls (12) for the swash-plate (13). Figure 2.2.2 shows a simplified view of the pump, emphasising bearing surfaces and leakage paths. Axial piston motors are basically piston pumps operating in reverse.

Part of the 'sales-appeal' of this type of pump is that it can operate at high speed (3500 rpm) and high pressure (5000+ psi), whilst achieving high volumetric efficiency. However, under these conditions, the clearances between moving components are very low indeed. Kakoullis [4] has measured a separation of 1 to 10 microns[1]

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1 One micron equals 10^{-6} metres.

COMPONENT	CLEARANCE IN MICRONS[1]
GEAR PUMP (pressure loaded)	
Gear to side plate	0.5 - 5.0
Gear tip to case	0.5 - 5.0
VANE PUMP	
Tip of vane to case	0.5 - 1.0
Sides of vane to case	5.0 - 13.0
PISTON PUMP	
Piston to piston bore	5.0 - 40.0 (radial)
Portplate to cylinder block	1.0 - 25.0
Slipper to slipper plate	1.0 - 10.0
SERVOVALVE	
Orifice	130.0 - 450.0
Flapper wall	18.0 - 63.0
Spool to sleeve	1.0 - 4.0 (radial)
HYDROSTATIC BEARING	1.0 - 25.0

NOTE 1. One micron = 10^{-6} metres

Table 2.1.1

Clearances in hydraulic equipment.

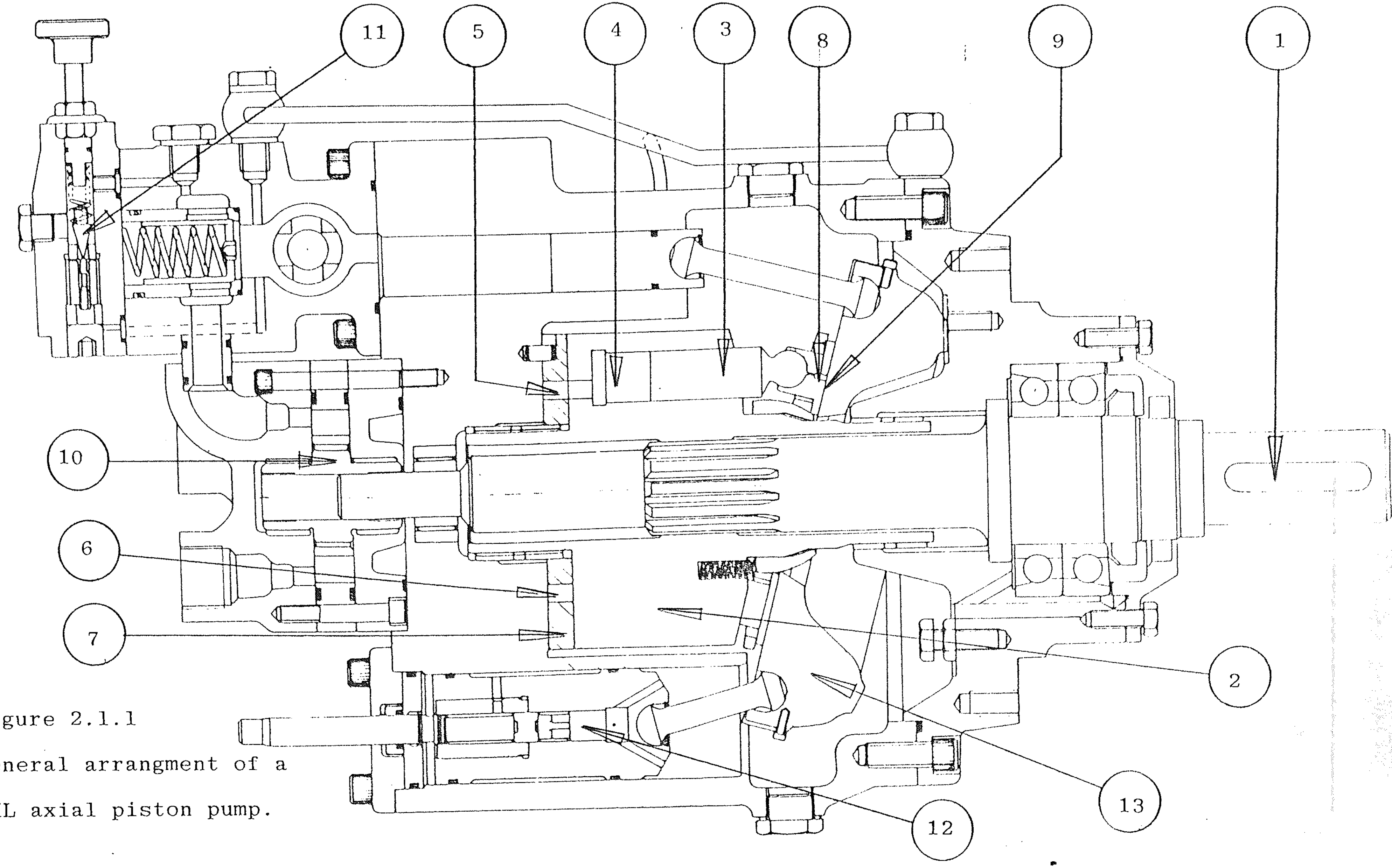


Figure 2.1.1

General arrangment of a

CHL axial piston pump.

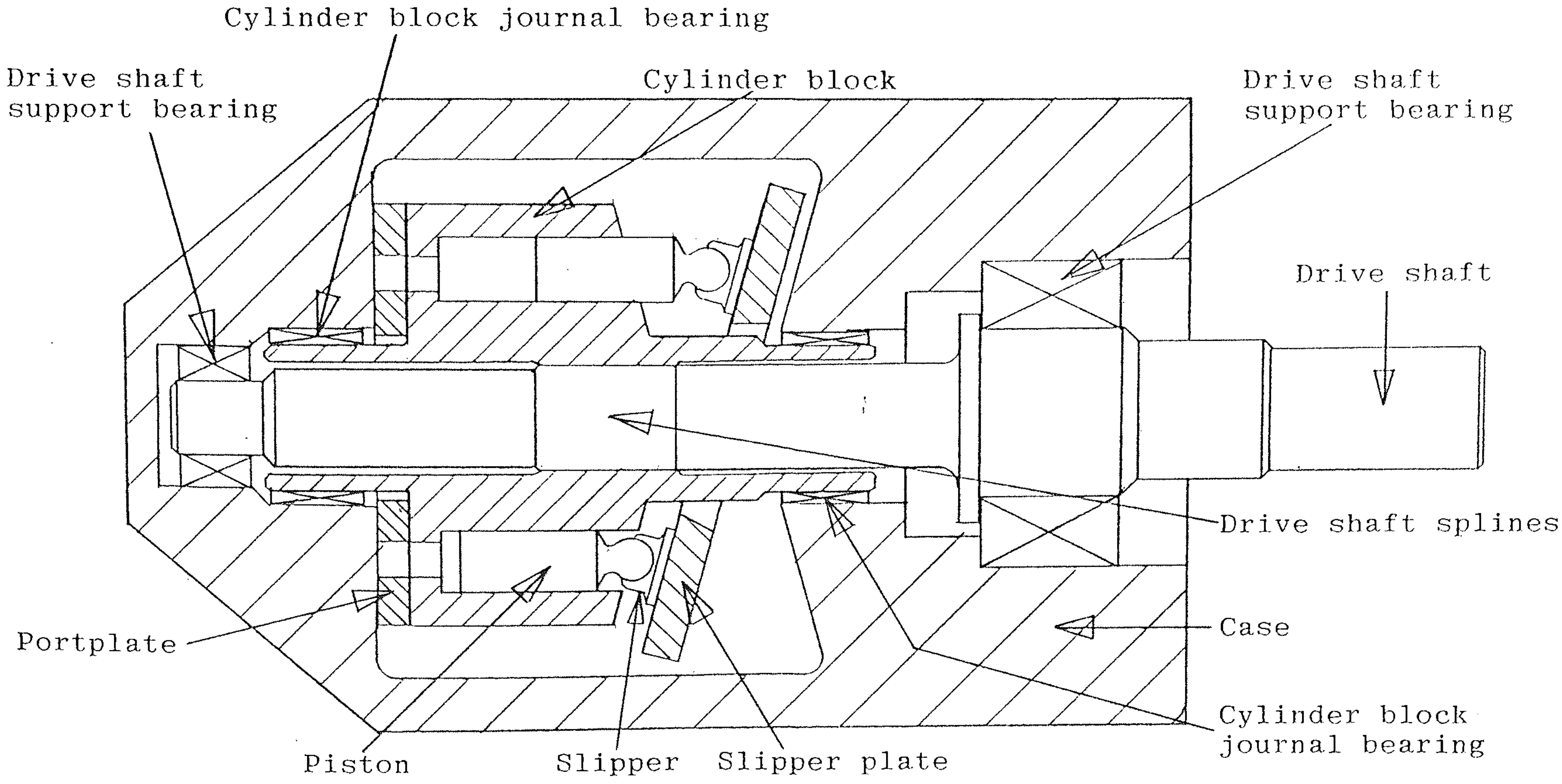


Figure 2.2.2

Simplified diagram of a CHL piston pump.

between slipper-pad and slipper-plate of an operating pump. Madera [5] calculated an oil-film thickness of 5 to 20 microns between portplate and cylinder block, and Shute and Turnbull [6] suggest an optimum radial separation of 10 to 25 microns between piston and piston-bore.

2.3 Determining Particle Size and Concentration

It was decided, on the basis of the data given above, that particles in a size range of 1 to 100 microns were those likely to be of concern to CHL. The lower size limit of 1 micron represents the smallest clearance found in hydraulic equipment, but it also represents the limit of resolution of most techniques for counting and sizing contaminants, and it would therefore have been impractical to have studied smaller particles. The upper size limit of 100 microns was selected for two reasons. Firstly, even the coarsest filters installed in hydraulic systems should remove the vast majority of particles above this size. Secondly, it was calculated that particles of this size and above would have settling velocities, in commonly-used fluids, and at normal system operating temperatures ($40^{o}C$ - $60^{o}C$), of the order of a mm/sec. Such particles should therefore settle out in well-designed system reservoirs, and will cease to be of concern.

(It should be mentioned that, although particles above 100 microns diameter were not considered in this thesis, such particles would probably cause severe damage if they entered a system and reached vital components.)

The simplest method of determining particle concentration is to remove the contaminant from a small sample of fluid (typically 10 to 200 mls), either by centrifuging or by filtration through a fine membrane. The weight of contaminant is then determined, and the contamination level is expressed as a gravimetric level in mg. of contaminant per litre of fluid. The analysis is usually conducted in accordance with a recognised standard method [7], Such procedures are generally referred to as "gravimetric analysis".

Although gravimetric analysis is relatively simple, it gives no information on the size or numbers of particles present in the fluid. These sizes (1 to 100 microns) may be placed in perspective by considering that a human red blood cell is about 8 microns across, and the resolution of the unaided human eye is no better than 40 microns. This means that the appearance of a fluid sample to the naked eye is no guide whatsoever to the fluid's contamination content.

The small size of fluid contamination suggests the use of optical microscopes for counting and sizing particles, and this is a widely used technique. In making a microscope count, a small volume of fluid (usually 10 to 100 mls) is passed through a fine membrane. Particles are retained on the membrane surface, and may then be examined, sized and counted under an optical microscope. National and international standards are available to give guidance on suitable techniques [8, 9, 10, 11].

Allen [12] and Dwyer [13] both discuss the use of optical microscopes for fluid analysis, and highlight some of the problems encountered. Among these are the length of time taken for an analysis, operator fatigue, and the relatively low repeatability of the technique. Day and Lee [14] found that the standard deviation of a series of microscope counts on the same sample was between 25 to 60% of the average count.

The accuracy of microscope counts may be improved by automating the counting process - certainly much of the tedium is relieved for the operator. Both Millipore and Cambridge Instruments (Quantimet) produce suitable equipment. A television image of the microscope slide is scanned, and particles can be detected as changes in light intensity. Associated electronic equipment converts these changes to equivalent particle sizes, and registers counts into specified ranges.

The Coulter company produces an instrument which measures and counts particles after they have been suspended in an electrolyte. The electrolyte is passed between a pair of charged plates. Any particle present produces a change in the electrical resistance of the fluid between the plates, and this can be detected and registered as a count at the appropriate size. The instrument is widely used in the fields of pharmacy and chemical engineering, and can be used to analyse hydraulic fluids.

In the mid-1960s, both HiaC and Royco developed electro-optical particle counters. These work by passing a fluid sample through a narrow passage, across which shines a light beam. Particles interrupt the beam, and the resulting reductions in light intensity are detected by a photo-diode. Electronics convert the electrical output of the photo-diode to particle counts. These instruments are easy to use and provide repeatable (although not necessarily accurate) counts. The author, using a HiaC PC.320 at OSU, was able to count particles so that the standard deviation of a series of counts on the same fluid sample was only 1.5% of the mean count.

Although the development of electro-optical particle counters allowed researchers in fluid contamination to direct their attention to other aspects of the problem, difficulties still remain in measuring contamination levels. For example, Day and Saunders [15] reported recently that correlation between particle counts obtained by different techniques is poor. The differences involved were not small, being up to two orders of magnitude in some counts. One reason for this is that different techniques rely on different features (length, projected area, volume) of a particle to define its 'size'. Particles from hydraulic fluids are rarely regular in shape. Figure 2.3.1 shows sketches of particles found in samples of hydraulic fluids taken during the tests described in Chapter 6. It will be appreciated that any definition of the size of such irregularly shaped particles must be essentially arbitrary, but it must also be a clear definition if data on fluid contamination levels are to have any meaning. Allen [op cit] lists several definitions of particle size, and some of those more commonly-used are illustrated in figure 2.3.2. Unless stated otherwise, particle size will be taken, in this thesis, to refer to the longest

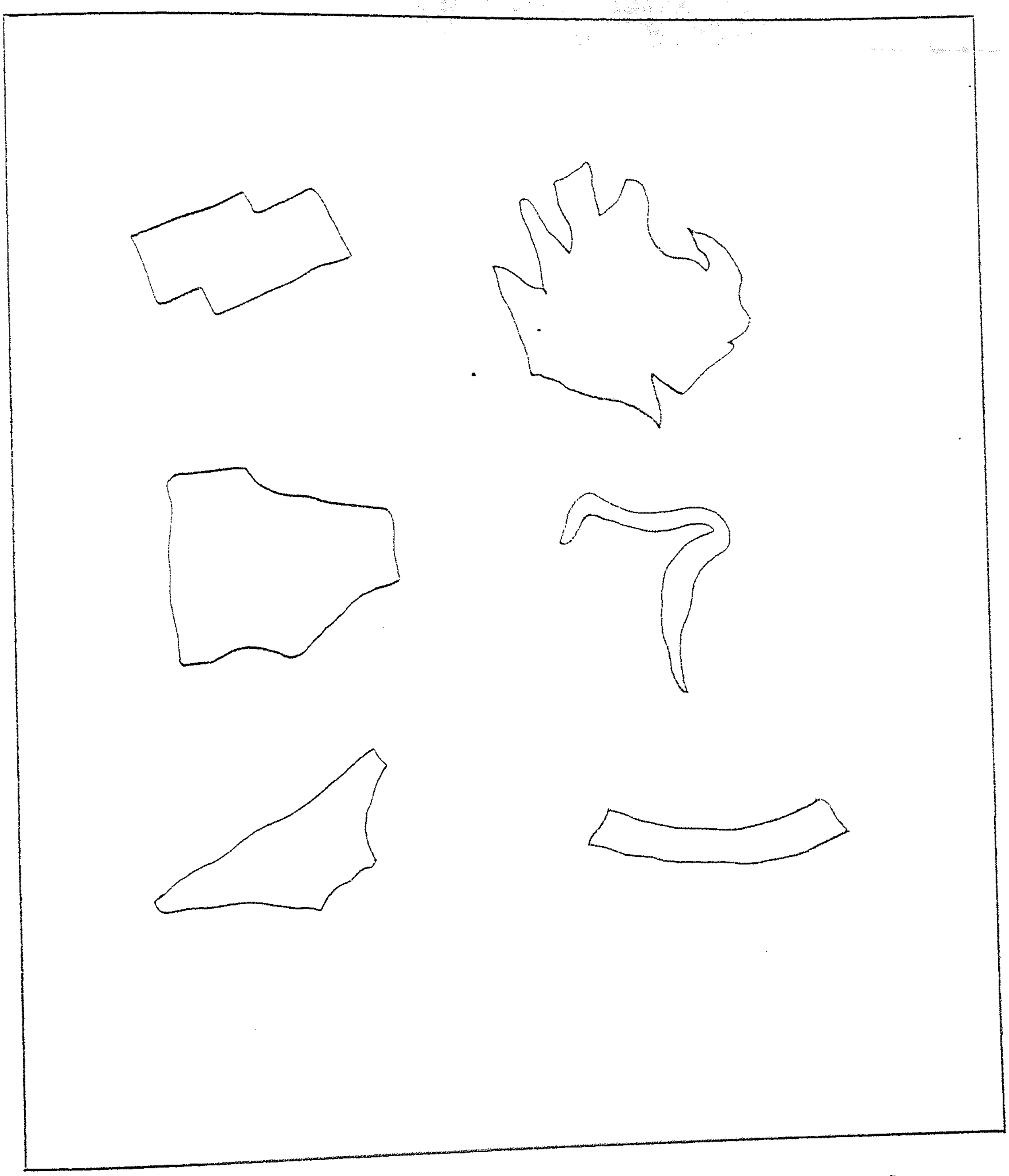
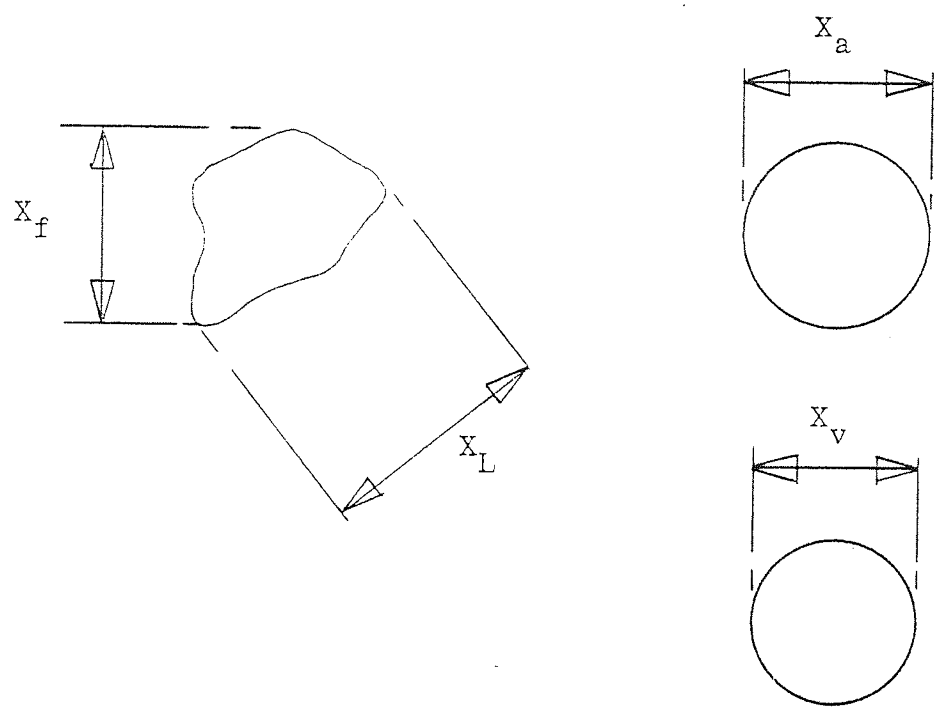


Figure 2.3.1 Sketches of the profiles of particles present in samples of hydraulic fluid, as viewed through an optical microscope.



X_L - longest dimension in any direction of projected image of particle

X_f - longest chord in specified direction of projected image of the particle

X_a - diameter of a circle having the same area as the projected image of the particle

X_v - diameter of a sphere having the same volume as the particle

X_s - stoke's diameter - the diameter of a sphere having the same sedimentation velocity as the particle.

Figure 2.3.2 Definitions of particle diameter.

dimension of a particle when viewed in random orientation.

With the possible exception of electro-optical counters operating at fairly low contamination levels, none of the above analytical techniques can be used to measure contamination levels 'on-line' in an operating system. Most fluid analysis therefore requires that a small sample of fluid (typically 100 to 200 mls) be drawn into a 'clean' bottle, with the actual particle counts being performed in a remote laboratory. Two main problems arise here. The first is that of ensuring that the sample contamination is representative of the overall system fluid. Akers and Stenhouse [16] conducted experimental and theoretical work into the design and location of fluid sampling points. Day and Lee [14] also considered this problem, and standards are available to assist in sampling. [17, 18, 19].

A second problem with bottle sampling is to prevent contamination of the fluid, both when the sample is taken and during subsequent analysis. Day and Lee [op cit] describe suitable methods of cleaning apparatus, and OSU have developed a procedure to check the cleanliness of sample bottles processed in batches. This procedure is now an ISO standard [20].

2.4 Reporting Fluid Contamination Levels

There is still no industry-wide consensus as to the best methods of reporting fluid contamination levels. Allied to the poor correlation between results obtained by different counting techniques, and to a general lack of precision in defining particle size, this makes it almost impossible to compare data from different sources.

Early researchers often described the amount of contamination in a fluid sample as a gravimetric level. While this method gives a simple and unambiguous result, it provides no information as to the size of particles present, or of their numbers. Complete particle size distributions provide more information. They may be given as interval data, as in table 2.4.1, and in this form the data may also be plotted as a histogram. It is, however, more common to express data as "cumulative oversize counts", which give the numbers of particles larger than specified sizes per unit volume of fluid. The data of table 2.4.1 is given in this form in table 2.4.2. Cole [21] developed a log-log^2 model for particle size distributions, and researchers at OSU [22] developed a log-log^2 graph-paper to give more nearly linear plots of particle counts. The data of table 2.4.2 are shown in this form in figure 2.4.1.

When particle counts from hydraulic fluids from operating systems are plotted as cumulative oversize counts, it becomes apparent that they tend to possess a similar 'shape'. This allows comparison of a given count to a set of standard contamination levels, the result being expressed as a class number. This method has obvious advantages in industry, where information on contamination levels must often be communicated rapidly and without confusion. Four systems are in common use. These are NAS 1638 [23], SAE 749 [24], a Royal Navy standard [25], and a commercial system developed by the Thermal Control Company [26]. The different systems are discussed in Chapter 8, and in an internal report given to CHL [27].

Size range (microns)	2-5	5-10	10-15	15-25	25-50	50+
Number of particles in size range per ml. of fluid	1943	33.44	6.60	3.84	1.03	0.23

Table 2.4.1 'Typical' particle count data (obtained on a sample of hydraulic fluid analysed on a Hiac PC 320 counter calibrated on air cleaner fine test dust).

Particle size (microns)	2	5	10	15	25	50+
Number of particles larger than this size per ml. of fluid.	1988	45.14	11.70	5.10	1.26	0.23

Table 2.4.2 Particle count data of table 2.4.1 converted to "cumulative oversize counts".

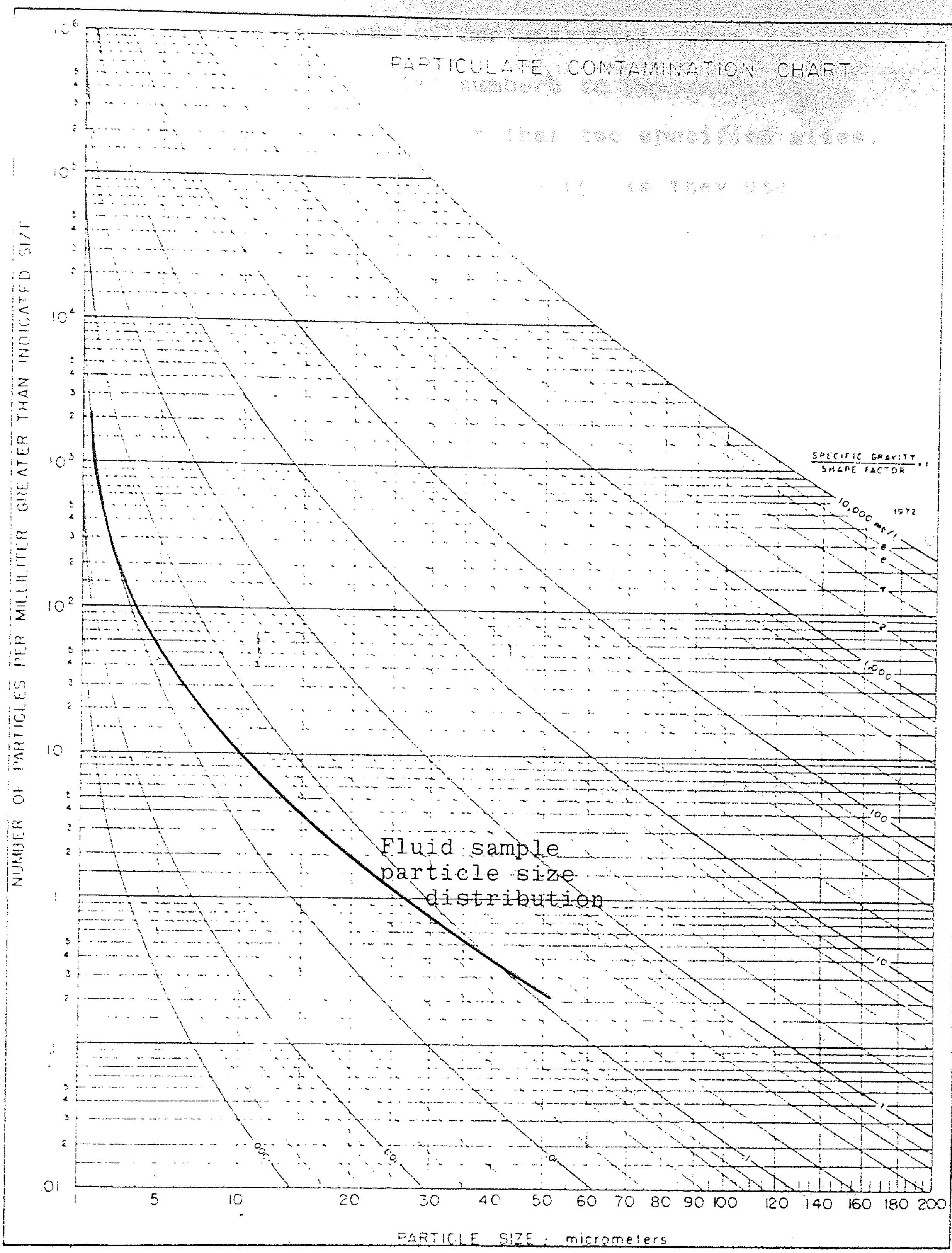


Figure 2.4.1 Particle count data plotted as a cumulative oversize distribution.

Recently, systems of contamination codes have been developed. These use two numbers to represent the numbers of particles greater than two specified sizes. The different systems are confusing, as they use different numbers for the same counts. They too are described and discussed in reference [27].

2.5 Hydraulic Fluid Contamination Levels

Despite the importance apparently attached to fluid contamination by the hydraulics industry, Wusthof [2] stated, in 1969, that "there are practically no data at all on contamination levels in hydraulic systems", although Barta [29] reported that the average industrial machine circuit has a contamination level just under 20 mg/l. The lack of data was alleviated considerably in 1973, when Lee [30] and Bensch and Bonner [31] published the results of major studies into fluid contamination levels.

It is difficult to compare the results of the two studies. Bensch and Bonner used gravimetric analysis and a HiaC counter to obtain their data, whereas Lee determined contamination levels by optical microscope counts. Bensch and Bonner measured levels as high as 855 mg/l in fluid from an agricultural tractor, but samples from cranes, machine tools, hydro-electric plants and aircraft were all below 40 mg/l. The two researchers also gave particle counts for the samples studied. When these counts are related to the Thermal Control system of classes, shown in figure 2.5.1, the results indicate that 'typical' fluids have contamination levels between classes 1 to 8. This lends support to the Thermal Control system, whose classes range from 1 to 9. (Comparing Bensch and Bonner's results to the Thermal Control classes was not entirely straight-forward. The

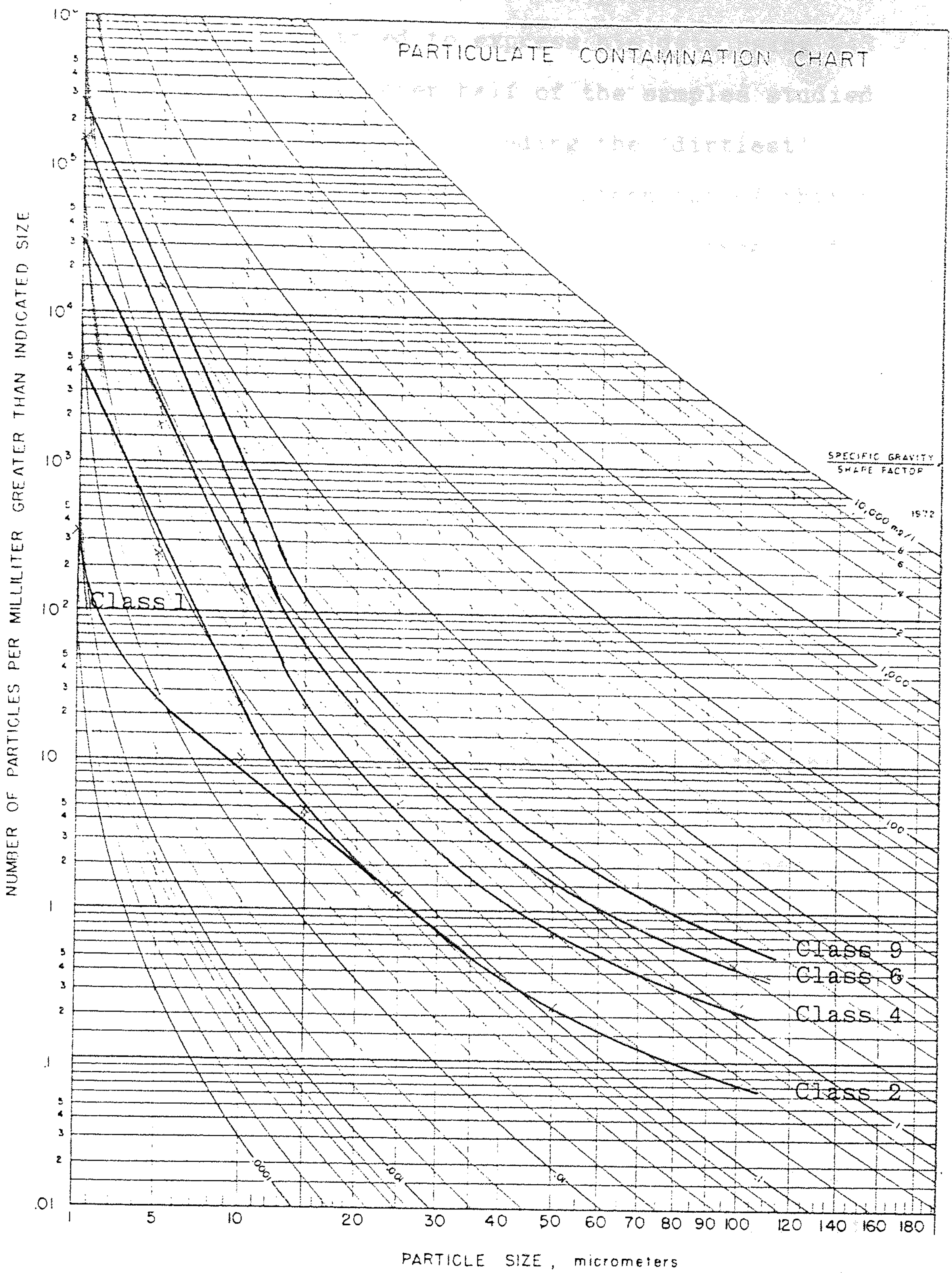


Figure 2.5.1. Thermal Control Company Contamination Classes. (some classes omitted for clarity).

exercise is described in reference [27].)

Lee [op cit] tried to express his data using NAS 1638, but found that over half of the samples studied had contamination levels exceeding the 'dirtiest' level covered by this system. Fourteen out of thirty-seven samples had contamination levels in excess of class 9 on the Thermal Control system, the remainder being spread among classes 1 to 8.

2.6 The Nature and Sources of Contamination

Fluid contamination in a hydraulic system can only come from three general sources: it can be built into a system during manufacture and assembly; it can be generated within a system during operation by processes of wear and corrosion; or it can be introduced into a system from the environment and during maintenance. Wray [32] suggests that contaminants are often the residues of machining operations such as honing and grinding. Seielstad and Sherlock [33] list other contaminants such as weld-scale, sand from castings, blast grit and waste and scrap from system reservoirs. CHL's own service engineer has often found contamination from hydraulic tube and from poor fitting practices during system construction.

Contamination may be a sympton of wear, as much as a cause of it. Klemgard and Coty [34] found that 50-85% by weight of contaminants in aircraft hydraulic fluids had been worn from rubber seals. Tessmann [35] lists possible sources of wear particles in a variety of equipment, and makes quantitative estimates of associated wear rates. The new technique of Ferrography relies on examining wear particles as a diagnostic tool. Collacott [36] considers this and other methods of fluid analysis.

Particles are always present in the atmosphere, and may enter hydraulic systems from this source. Dwyer [13] reports that, in a 'clean' room, 21000 particles larger than 5 micron settled onto each square foot of area in a 24 hour period. Ruskell and Westcott [37] and Denny [38] showed that hydraulic seals are separated from shafts and cylinders by a thin film of lubricant and contaminant particles might therefore enter a system through these devices. King, Tessmann and Bensch [39] estimated that up to 10^7 particles (larger than 10 microns) could enter a system per minute in this way.

Poor maintenance practices may introduce contamination. Brown [40] and Wray [op cit] warn that new fluid may be heavily contaminated. This was confirmed in an experimental study by Moore [41], who found contamination levels in new fluid up to 218 mg/litre. Riehl and Hawkins [42] found that new filter elements were themselves contaminated, and that this contamination could be released into a system when elements were changed.

Actually identifying contaminant materials is not easy, especially if quantitative data are required. Particles can sometimes be identified by microscopic examination (optical or SEM), and the "Particle Atlas", [43] by McCrone et al is invaluable in this work. Other approaches are to use micro-chemical analysis (Bensch and Bonner [op cit]), electron probe micro-analysis, X-ray diffraction and emission and absorption spectrography. Collacott [op cit] gives an excellent resumé of the available techniques. Lee [op cit] used spectrographic analysis in his study, but Westcott and Seifert [44] report that the technique is 'blind' to particles above 8 microns diameter, which must limit its usefulness.

The main conclusion which can be drawn from published research is that contaminant materials are extremely varied. The contaminants found in a given system will depend on the system's operating environment, on the materials of which it is made, and on its manufacturing and maintenance histories.

2.7 Factors Affecting System Contamination Levels

Very little published work was found on the factors which determine the contamination level in a given system. Intuitively, one would expect a specific fluid contamination level to be determined by a balance between the rate at which particles enter the system and are generated within it, and the rate at which they are removed by the system filters. However, Lee [op cit] found no relation between the nominal rating of a system filter and the contamination level of the system fluid. This finding is explained in sections 3.2 and 7.4. Fitch and Tessmann [45] considered the build-up of contamination in a simple filter test, but no studies were found into the factors which determine contamination levels in more complex hydraulic systems.

2.8 The Effects of Fluid Contamination

The effects of fluid contamination can be considered from three viewpoints: effects perceived by the user of hydraulic systems; consequences for manufacturers of hydraulic equipment; effects on the equipment itself.

Many users of hydraulic systems believe that fluid contamination can either cause equipment to break down, or that breakdowns can be prevented only by expensive, planned preventive maintenance. The costs of failures in hydraulic systems are often immense. Busby [46] and Collacott [36] quote the cost of downtime in steelworks as being (in 1972) £1200/hour. Collacott also gives £8000/day as being the cost of the enforced laying-up of a ship for repairs. A 600 MW electrical generating set can cost upwards of £25000/day in replacement generation if it lies idle, and it is impossible to calculate the value of human lives which might be lost if an aircraft's hydraulic system should fail.

There is a considerable body of evidence, both published and unpublished, to suggest that fluid contamination can cause hydraulic systems to fail. Barta [29] reports that "service records indicate that 70% of all hydraulics' problems can be traced directly to the system fluid". O'Connor [47] gives details of the improvement which was obtained in the reliability of hydraulic systems used on agricultural tractors, largely through attention to fluid contamination. The OSU annual reports [48] describe the results of many contamination tests on hydraulic equipment, in which abrasive contaminants produced considerable damage. It should also be mentioned however, that Lee [op cit] found that only one out of the thirty-seven machines which he studied was apparently showing any serious problems which could be directly attributed to fluid contamination.

Whether or not fluid contamination is a major problem affecting reliability, users obviously perceive it to be so. The implications for equipment manufacturers who fail to tackle the problem are that they may lose orders to competing companies; the implications for the industry as a whole are that it may lose orders to competing technologies. This is not speculation or prediction, but fact. O'Connor [op cit] reports that his company changed to a new pump on an agricultural tractor, largely because the original choice was affected by contamination. Lancer-Boss [49] gave poor tolerance to dirty fluids as a reason for not installing a hydrostatic transmission in a new forklift truck. Vardy [50], in a paper critical of all concerned, stated that the poor reliability of hydraulic systems (felt to be largely due to contamination) almost led to their complete abandonment in BSC steelworks. He concluded that his analysis clearly indicated "the need for a concerted effort by manufacturers, users and researchers to improve knowledge and understanding of the effects of contamination and of the means to control it".

Unfortunately, this understanding seems to be some way off.

There are several published papers describing the effects of contamination in broad terms. It seems fairly clear that fluid contamination can cause wear by"abrasion" or"erosion". Fine clearances in servo valves can be silted up, or particles can jam valves in one position. Orifices in valves can be blocked, and lubricant feed channels can be obstructed. Collacott [op cit] suggests that metal particles can act as catalytic agents to cause breakdown of oils.

However, what is lacking is detailed understanding of how these effects occur, and how they may be reduced. Tessmann [51] admits that the OSU pump test has been introduced, relying on "component degradation as an indirect measure of contaminant wear ..... without a thorough understanding of the wear phenomenon involved." Wusthof [52] states that "almost no documentation about built-in low contaminant sensitivity is available". Broeder and Heijnekamp [53] found, from tests on journal bearings, that abrasives in the oil could initiate surface pitting. Fitzsimmons and Clevenger [54] found that the wear of rolling-element bearings depended on the size, concentration and hardness of contaminants. Barwell [55] has postulated that particles could initiate adhesive wear[1], and Scott [56] believes that contaminants may induce a complete breakdown of a lubricant film. The processes of wear are considered by Bowden and Tabor [57], Rabinovicz [58] and Kragelskii [59], but the conditions under which they conducted their experiments were very different from those existing in an axial piston pump and the results may be difficult to apply. Finally, Laurenson [60] and Rimmer [61] have conducted studies into the blocking of fine orifices and capillaries by particles.

2.9 Areas of Insufficient Knowledge

It became clear, from a wide survey of relevant sources, that useful work could be pursued on almost any aspect of fluid contamination. It also became clear

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1 See glossary.

that fluid contamination affects, in some way, all branches of the hydraulics industry. For CHL, the problem became one of deciding which aspect or aspects of the subject would be most suitable for study. It was mentioned, in 2.7, that very little work has been published into the factors which determine the contamination level of a fluid. A consideration of these factors helped to define the fundamental problem posed by contamination, and to delineate the areas of research which should be pursued. This forms the subject of the next chapter.

3.0 PROBLEM DEFINITION AND RESEARCH STRATEGIES

3.1 Introduction

It might be thought that a viable solution to the problem of fluid contamination would be to 'simply' remove all potentially damaging particles from hydraulic fluids. However, it is shown in this chapter that hydraulic fluids are inevitably contaminated, and that it is expensive to obtain even 'relatively clean' fluids. The problem of fluid contamination is therefore to identify and maintain an economic fluid contamination level for a given system. Consideration of an ideal solution to this problem shows that it can only be tackled effectively through a concerted effort by all sections of the hydraulics industry. It is argued that CHL's contributions to an 'industrial solution' should be: to improve general understanding of fluid contamination; to determine 'acceptable' fluid contamination levels for its products; to improve the ability of its pumps to operate on contaminated fluids. The design and development history of a typical pump is outlined, and this suggests three possible research strategies.

3.2 Factors affecting fluid contamination levels

To examine the factors which determine the contamination level of a hydraulic fluid, the contamination behaviour of a hydraulic circuit was modelled as in figure 3.2.1. A(t) represents the fluid contamination level at time t. (This can only refer strictly to particles in a narrow size range, and a complete analysis would have to consider many such size ranges.) Contaminant will enter the system from the environment, and will be generated within the system by processes of wear and corrosion. These processes were assumed to take place at a combined 'ingression rate' R, whose units are consistent with those of A(t). R was assumed constant, although it will probably vary with time in an actual system, and may also be a function of A(t) itself. (The results of tests described in chapter 6 confirm that this latter suggestion is correct.)

Particle removal will occur through filtration, sedimentation and silting. Filtration was assumed to be the dominant process, partly because it is the one over which the system designer has most control. Filter efficiency was given the symbol η, such that the filter will remove a proportion η of the contaminant passing through it. (Again, filter efficiency can be defined only for a narrow range of particle diameter.) If flow through the filter is Q_f, then the filter will remove particles from the system at a rate $A(t)Q_f\eta$. For a system of overall volume V, where all changes in contaminant content are distributed immediately throughout the system, it was calculated that:

$$A(t) = \frac{R}{Q_f\eta}\left[1-e^{-Q_f\eta t/v}\right] + A(o)e^{-Q_f\eta t/v} \qquad 3.2.1$$

Many of the assumptions made in setting up this model are obvious oversimplifications. For instance it is known that filter efficiencies vary over the "life" of a filter element. However, the model is of the correct form, certainly for short periods of time, and, as such, it illustrates several important points.

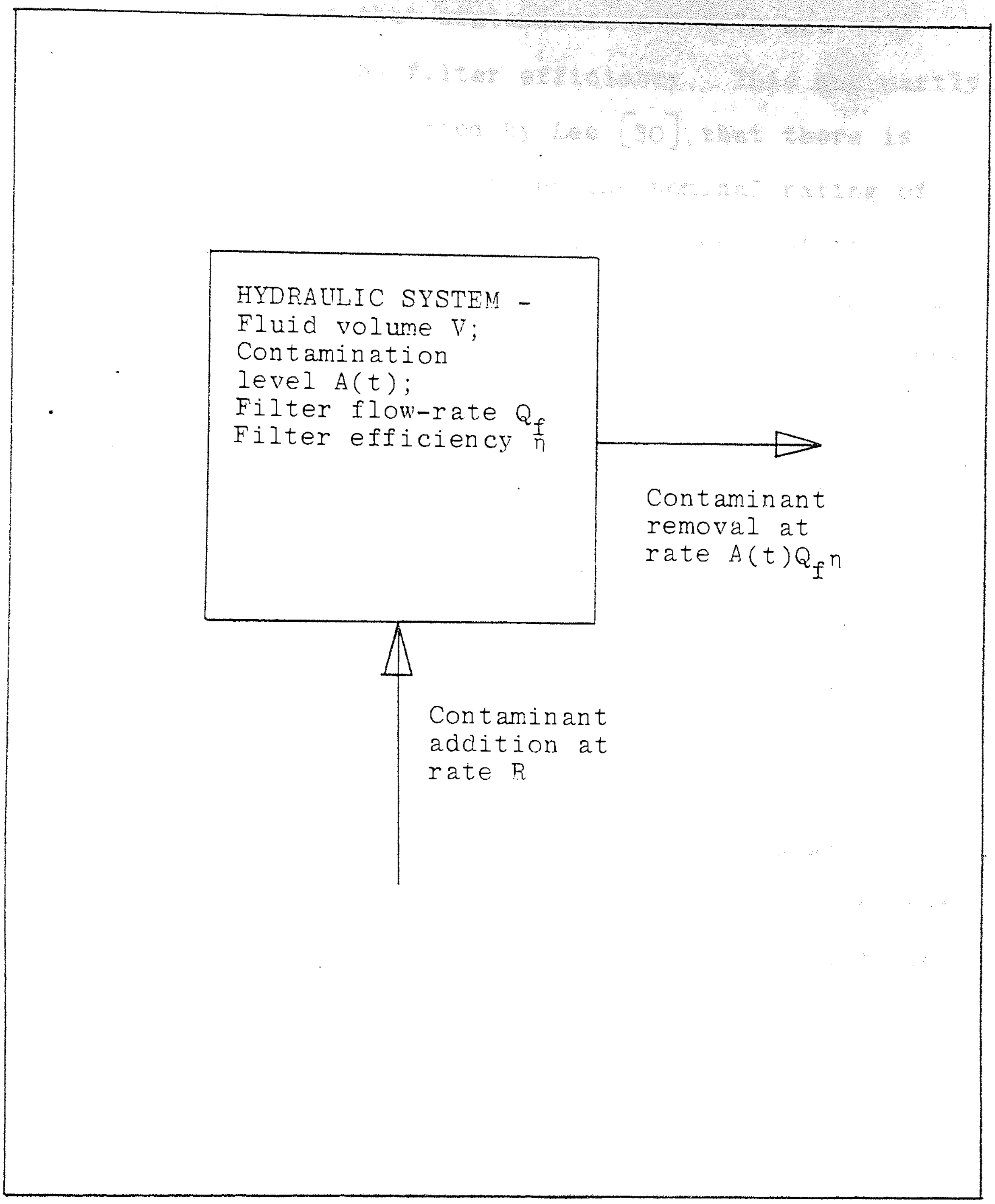


Figure 3.2.1 Simple model of the contamination behaviour of a hydraulic circuit.

The first is that contamination levels are only partly determined by filter efficiency. This may partly explain a finding reported by Lee [30], that there is very little correlation between the nominal rating of a system filter and the contamination level of the system fluid. (There are also other factors connected with the existing methods of testing and rating filters. These are discussed in section 7.4.)

The second prediction of the model is that the contamination level of a hydraulic system fluid will tend, in an exponential manner, towards a final, steady value. A typical variation of a contamination level with time might be as in figure 3.2.2. This prediction, if correct, has important consequences related to the timing of fluid sampling. Contamination levels must be allowed to stabilise before samples are taken.

The most important prediction of the model is that the final, steady-state contamination level of a fluid can be reduced to zero in only two ways. The first is to eliminate R. It is well established by now that this is not practical. A completely sealed system operating normally will still generate particles. The only way to completely eliminate particles would therefore be to install a 100% efficient filter immediately upstream of all hydraulic system components likely to be affected by contamination. It is probably possible to produce filters which are 100% efficient for particles larger than 10 microns, although it is impossible to verify such performance with complete confidence. However, the cost of providing filters finer than this rises steeply with decreasing pore size, and can often

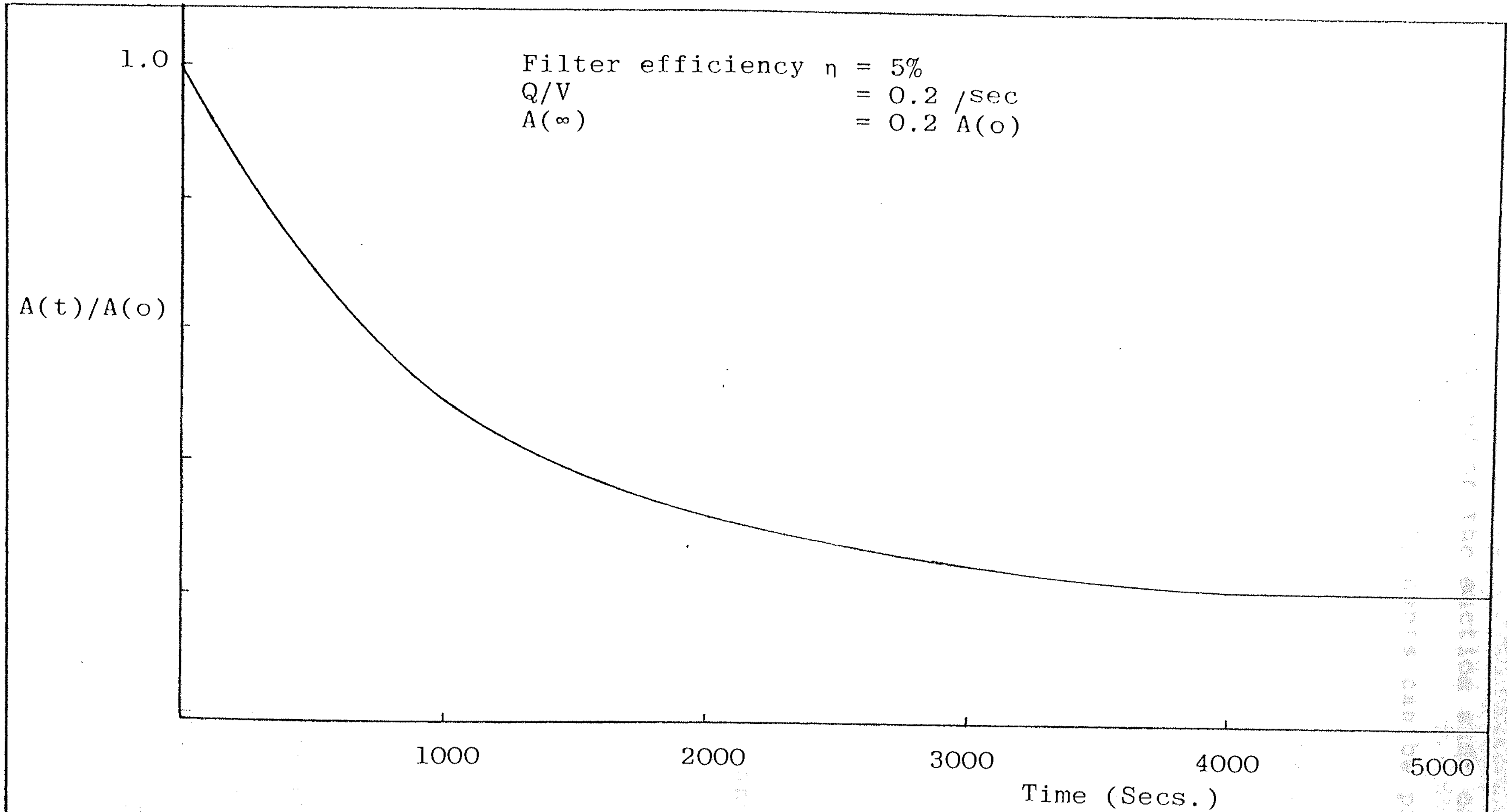


Figure 3.2.2 Predicted variation of fluid contamination level with time.

exceed the cost of the component to be protected. Also, the pressure drop across such filters is too great for them to be installed on the suction side of an unboosted pump, so not all components can be protected in this manner.

The conclusion to be drawn is that hydraulic fluids are inevitably contaminated to some extent. The model of equation 3.2.1 indicates the ways in which fluid contamination levels may be reduced, and these will now be briefly discussed.

3.3 Reducing Fluid Contamination Levels

Fluid contamination levels can be reduced, but not totally eliminated, in five ways.

(a) Filter efficiency ($\bar{\eta}$, in the model of section 3.2) can be increased by using a finer filter medium. However, finer elements are expensive, and they may need more frequent replacement than less efficient units. (The author knows of one filter manufacturer who believes that this need not be the case because R, the particle ingression rate, is itself a function of the contamination level. Some supporting evidence for this belief is provided in test results given in chapter six.) A finer element will also produce a greater pressure drop across the filter, and a larger, and hence more expensive unit may have to be provided.

(b) Filter flow-rate, Q_f, may be increased. This may require a larger filter at an increased cost.

(c) The contaminant ingression rate, R, may be reduced. This can be achieved through improved maintenance techniques, better seals and the use of wear and corrosion-resistant materials. All these approaches cost money.

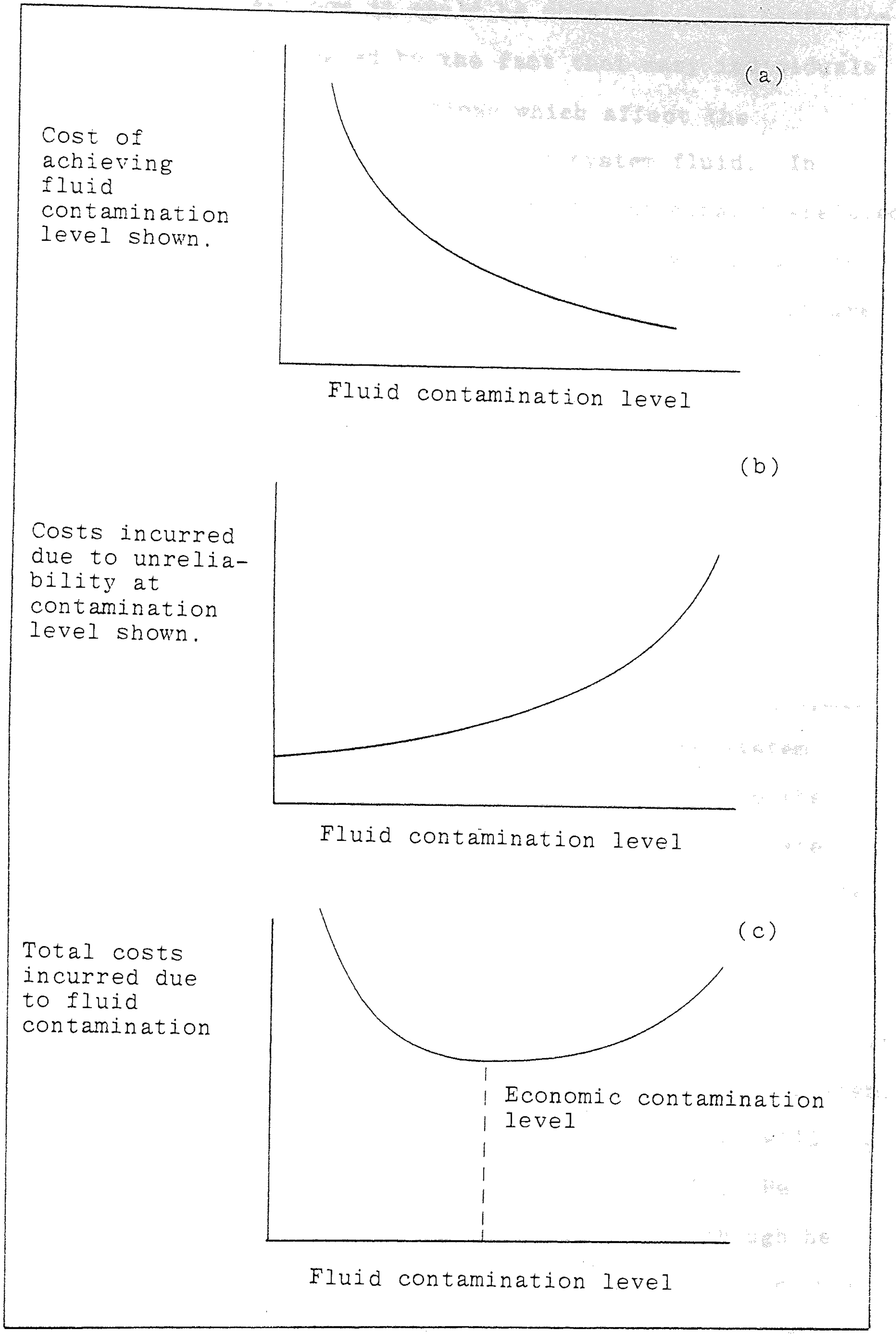
(d) The system's "built-in" contamination level, $A(o)$, may be reduced through better cleaning of components during manufacture and assembly. This is expensive to accomplish.

(e) The ratio of filter flow to system volume (Q_f/v) can be increased. This requires either an increase in filter flow (with the problem described in (b) above) or a decrease in system volume. This latter approach may lead to aeration of the fluid, which often creates operational problems. Also, greater cooling capacity may have to be installed.

3.4 The Problem of Contamination

The above analysis may seem to be over-emphasised, but it does illustrate the basic problem of fluid contamination. It is intuitively obvious that the damage caused by fluid contamination increases with increasing fluid contamination levels. Work described later in this thesis will confirm this argument. However, it is expensive to reduce contamination levels. The basic problem of fluid contamination is therefore that a balance must be made, in the design and operation of hydraulic systems, between the cost and inconvenience involved in reducing contamination levels, and the expense which will be incurred if levels are left too high and equipment fails as a result.

An ideal situation would be to have data of the form shown in figure 3.4.1, so that an 'economic' contamination level could be selected for any system. At the start of this research, data of this form was not available,



Ideal (hypothetical) approach to the economics of fluid contamination.

Figure 3.4.1

nor was it clear how it could be obtained. The situation is further complicated by the fact that many individuals and organisations take actions which affect the contamination level of a hydraulic system fluid. In addition, those who bear the cost of contamination-related equipment failure are often not those who would have to bear the cost of reducing the likelihood of such failure. These last points can be appreciated by considering the organisation of a typical project involving a hydraulic system.

3.5 The Organisation of Hydraulics Projects

Figure 3.5.1 illustrates a fairly typical project involving hydraulics, of the kind in which CHL has recently been involved. The central personality is the system designer, who will specify and order the equipment to be installed in the hydraulic system. The system manufacturer will build the system and pass it to the overall plant manufacturer. Eventually, the complete plant will pass to the customer, who will be responsible for operation and maintenance.

Each individual or organisation involved in a project of the form shown in figure 3.5.1 will be partly responsible for the final contamination level of the hydraulic system. Referring to equation 3.2.1, the system designer will specify filter flow (Q_f) and system volume (V). He will also determine filter efficiency (η), although he will specify filter performance as a nominal or absolute rating. (These ratings are discussed in section 7.4). The system manufacturer, overall plant manufacturer and the original equipment manufacturers will together determine the initial fluid contamination level (A(o)),

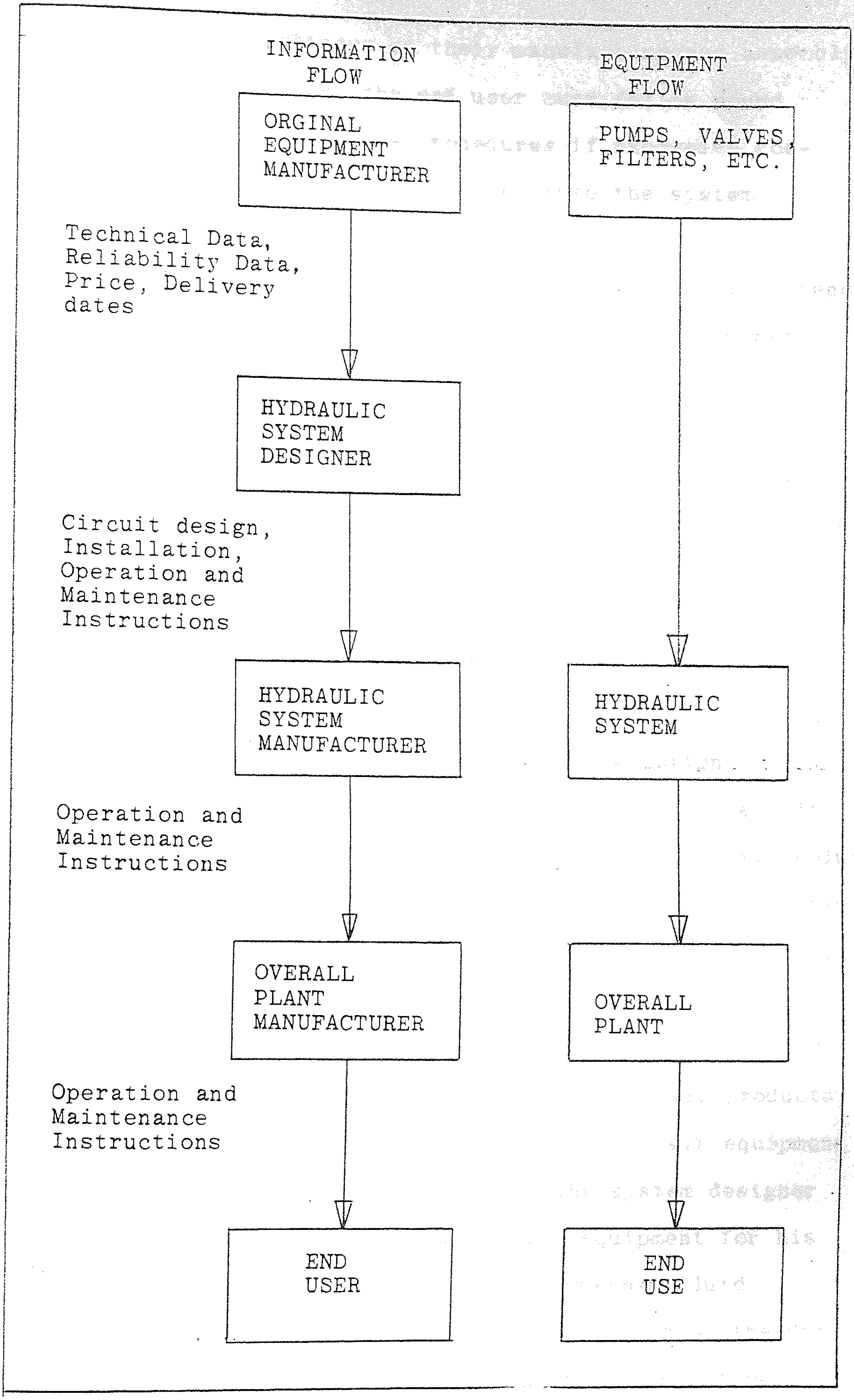


Figure 3.5.1 A typical hydraulics project.

through the cleanliness of their manufacture and assembly operations. Finally, the end user must follow sound operation and maintenance procedures if excessive contamination is not to be introduced into the system while it is in his care.

The costs of any contamination-related failures tend to fall on the system user. Even if pumps fail under warranty, it is not unknown for the pump manufacturer to use "fluid contamination" as an excuse for not meeting the warranty claim. It can be appreciated, from figure 3.5.1., that there is considerable scope for shifting the responsibility for equipment failure!

3.6 An Ideal Solution

The central point to emerge from the above analysis is that fluid contamination represents a problem for the entire hydraulics industry, and that a 'solution' cannot be obtained by any one section action alone. It is instructive to consider the form which an 'ideal' solution might take. Figure 3.6.1 illustrates a possible industry-wide approach to the problem, on the lines of that being adopted in the U.S.A.

Under this approach, equipment manufacturers would give recommended contamination levels for their products, whilst striving to improve the ability of their equipment to operate on contaminated fluids. The system designer could then select the least sensitive equipment for his system and pass on to the user a recommended fluid contamination level for the system. It would be the duty of the user to monitor and maintain this level, albeit with assistance from other parties involved.

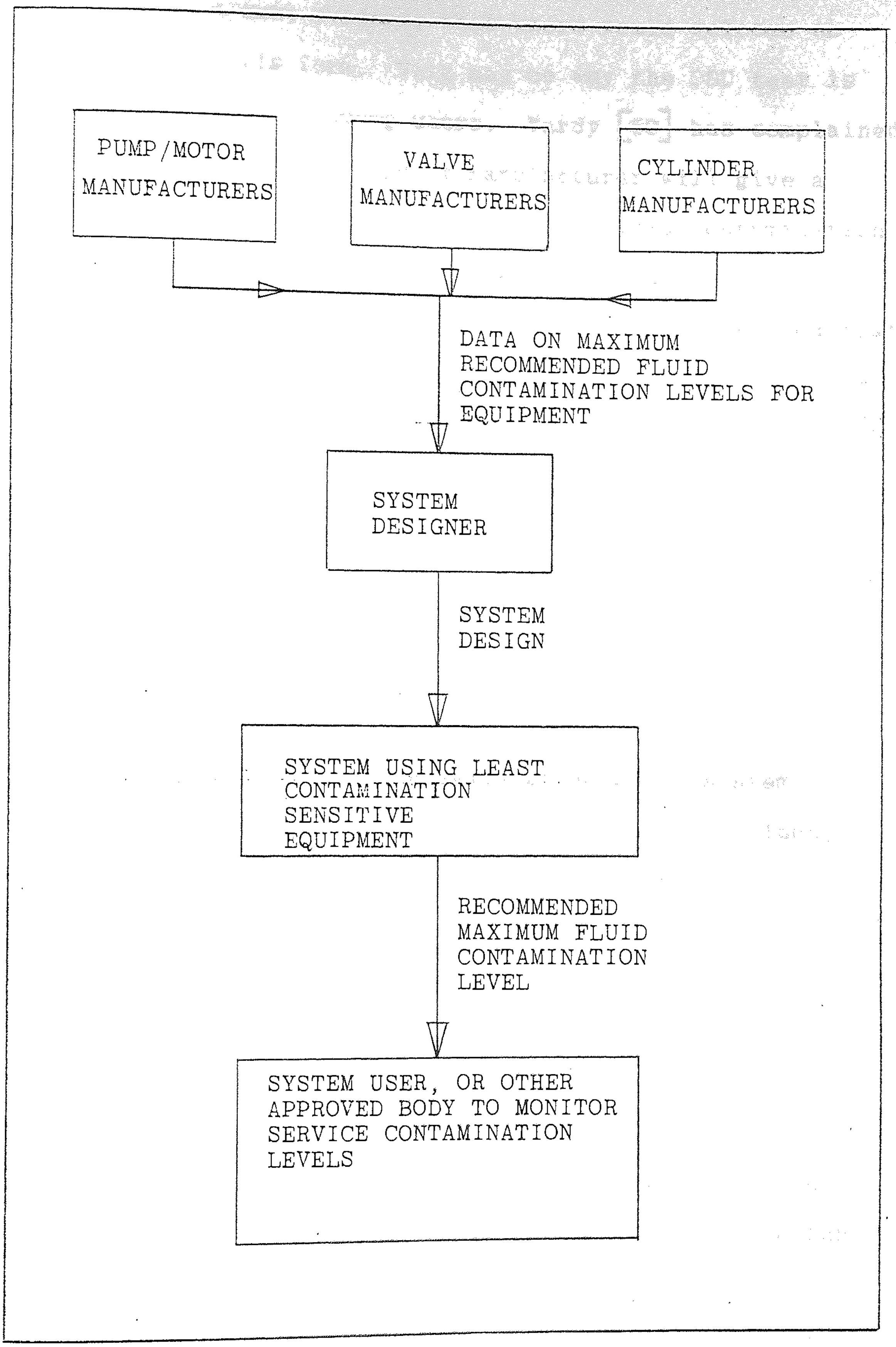


Figure 3.6.1 An 'ideal solution' to the problem of fluid contamination.

There is some evidence that users would welcome an approach of this form. This may be why the OSU test is gaining popularity among users. Vardy [50] has complained that "no hydraulic equipment manufacturer will give a practical classification to which the fluid contamination should be kept". Briggs [62] makes the same point.

A survey of pump manufacturers' catalogues showed that Vardy's complaint is justified. Most pump manufacturers specify only a nominal filtration level (often 10 microns) for their products. This is completely unsatisfactory.

(a) As shown in section 3.2, filter performance is only one of five factors which together determine a fluid contamination level.

(b) Bensch [63], Day and Lee [14] and Tsai and Way [64] have found that the performance of a filter in a system can be substantially affected by system conditions, and especially by flow surges. Thus, it cannot be assumed that a filter's test performance will be reproduced in service.

(c) As will be shown in section 7.4, nominal filter ratings are an imprecise way to measure filter performance, and also mean different things to different filter manufacturers.

The author is firmly convinced that the approach illustrated in figure 3.6.1 represents the way in which the hydraulics industry should tackle the problem of fluid contamination. The aims of this project therefore depend on the most effective contributions which CHL can make to such an approach. These will now be summarised.

3.7 Revised Project Aims

It was decided that CHL should pursue work in three areas.

(a) CHL should develop its own knowledge of fluid contamination and then pass on its knowledge to customers. Such an educational process need not be undertaken through altruistic motives. It would qualify as "good communications" and "after-sales-service". It will be remembered, from chapter one, that users of hydraulic equipment rated these factors highly in their choice of suppliers.

(b) The company should try to determine practical fluid contamination levels for its products.

(c) The company should try to improve the ability of its pumps to operate on contaminated fluids.

It can be seen that the original project aim, that of "rating the ability of a pump to operate on contaminated fluids", could well form part of work in areas (b) and (c).

3.8 Possible Research Strategies

Possible research strategies became clear through a consideration of the design and development history of an axial piston pump. Figure 3.8.1 illustrates a typical design/development process. A new pump starts with a specification of pump performance, reliability, weight, cost etc. (CHL's specifications do not, at present, contain any reference to fluid contamination tolerance, although this may change with the increasing adoption of the OSU test.) To meet the specification, a design is prepared, based on past experience, mathematical analysis, computer-aided design studies, published data and "engine-

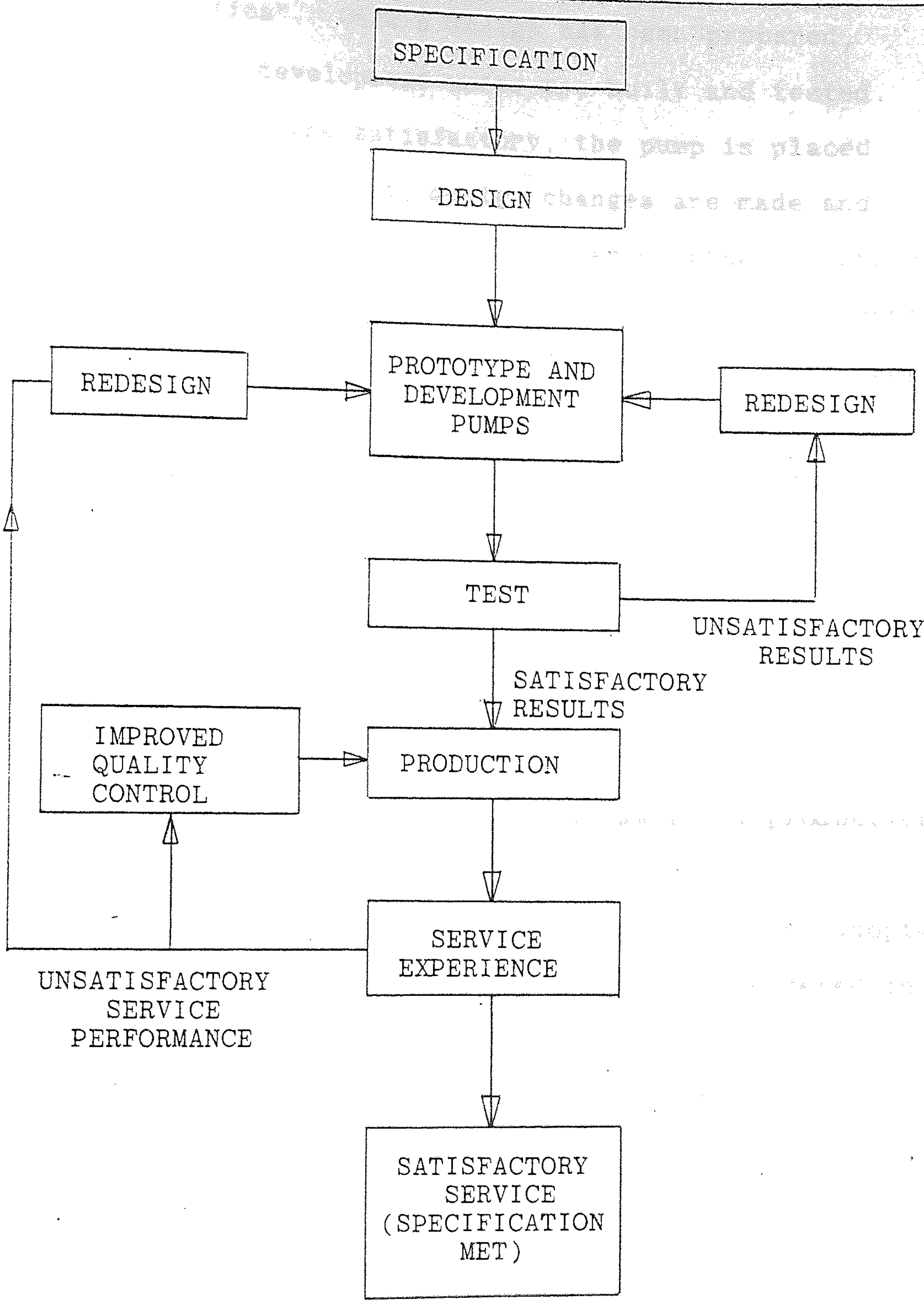


Figure 3.8.1 Design/development of a hydraulic pump.

ering intuition". Once a design has been prepared, prototype and development units are built and tested. If test results are satisfactory, the pump is placed into production; if not, design changes are made and units are tested until acceptable performance is obtained. Test results do not guarantee good performance in service, so field experience is usually evaluated to rectify any problems which may arise.

There are three stages in this design and development process at which CHL may obtain data on the effects of fluid contamination. These are:

(a) by theoretical analysis at the design stage;

(b) by conducting tests on prototype and development pumps;

(c) by monitoring the service performance of production units.

None of these approaches had previously been adopted by the company. All three were therefore considered in pursuance of the project aims outlined in section 3.7. Priorities for the work were determined by an economic consideration. The earlier in the design and development process that information can be obtained, the cheaper that information tends to be, and the cheaper it tends to be to take any necessary action. Chapter four therefore examines theoretical analysis of fluid contamination-related problems. Chapters 5, 6 and 7 deal with the contamination testing of hydraulic pumps. Chapter 8 considers analysing service experience of fluid contamination and related failures.

4.0 THEORETICAL ANALYSIS OF THE EFFECTS OF FLUID CONTAMINATION

4.1 Introduction

The main attraction of a theoretical analysis of the effects of fluid contamination is that the work involved will be conducted during pump design. This has two advantages. Firstly, design work is inherently less expensive than are the alternatives of testing and the analysis of service data. Secondly, it is less costly to rectify a design error or to introduce a modification when the design exists only as a set of drawings than when the pump is in production. Data from theoretical work is therefore relatively cheap to act upon.

Design analysis thus represents the cheapest way to improve CHL's knowledge of fluid contamination and of its effects. It is also obviously in the company's own interest to minimise, through design changes, the effects of contamination on its pumps. CHL may be forced to adopt this approach anyway, in the near future, to improve the performance of its products in the OSU pump contamination test.

This chapter therefore contains a general description of the possible effects of particles on a simple axial piston pump. Because of the number of effects, and the number of pump components which may be affected, attention is subsequently concentrated on the slipper/slipper-plate bearing area. A brief description is given of the mechanism of slipper lubrication on "clean" fluids. The calculations involved are related to the problem of filter specification and it is shown that potentially damaging particles will inevitably enter axial piston pumps. Three possible effects of these particles are then considered; abrasive wear, silting of fine clearances, and blocking of capillaries and orifices. In each case, the analysis covers three questions.

(i) Is the effect likely to occur?

(ii) If it is, is it possible to relate the severity of the effect to the concentration, size and properties of the fluid contamination present?

(iii) Can the analysis be used to suggest ways of minimising or eliminating the form of damage considered?

The chapter closes with a discussion of the present, and the possible future value of this type of pump design analysis.

4.2 The Effects of Fluid Contamination on Axial Piston Pumps

Figure 4.2.1 shows a general arrangement of a simple, fixed delivery axial piston pump. The example shown is the Reyrolle A70 unit used in the tests described in Chapter 6. Although this pump looks very different to the CHL design shown in figure 2.2.1 the two pumps are of the same basic type. The main differences are firstly that the A70 has the drive shaft and cylinder block machined as a single unit and supported on common bearings, whereas in the CHL pump the shaft and block are separate. Secondly, the A70 pump has a thrust-plate between the slipper and swashplate. The thrust-plate spins at pump rotational speed so that the slippers are almost stationary with respect to the plate. In the CHL design the slippers slide around the slipper-plate.

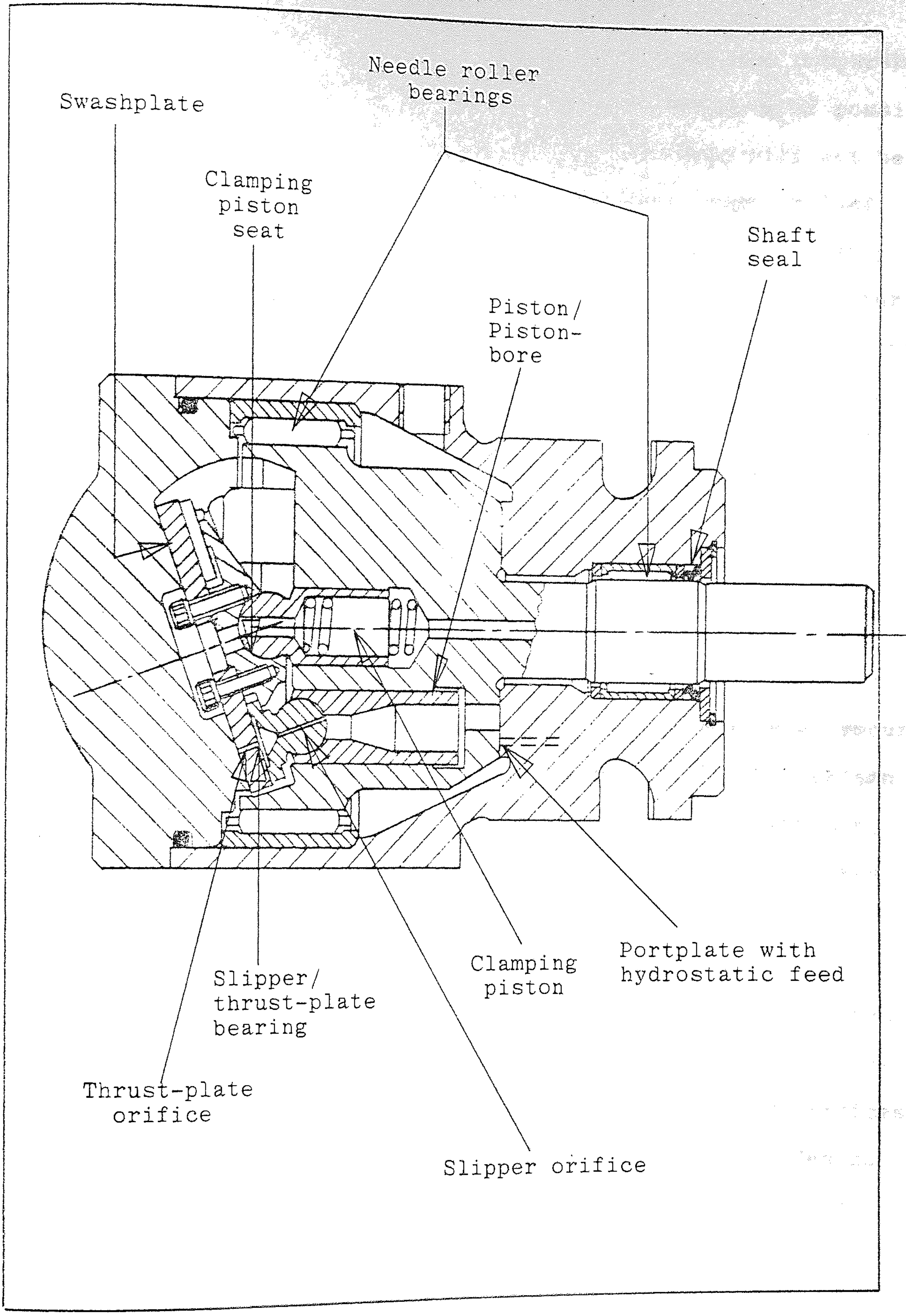


Figure 4.2.1 Areas of possible contamination damage in the A70 axial piston pump.

Apart from these features, the two designs are comparable, and the A70 will be used in a general estimate of possible contamination effects, partly because these will not be complicated by the presence of the boost pump, relief valve and servo-controls of the CHL design, and partly to allow comparison with the tests described in Chapter 6.

Any theoretical analysis must start with an assumption of the effect to be analysed. This is not a problem with scientific research, and experiments can be designed to verify any theory developed. But in a pump design analysis, the experiment already exists as the commercial use of the pump. The approach here must be to assume what might happen, and then work backwards to deduce an analytical theory. This involves very real dangers that an effect will be overlooked, or that scarce resources will not be concentrated on the most important problems. These are major limitations of a theoretical approach, and ways to circumvent them are considered in chapters 5 and 6.

Recognising these limitations, it was assumed, in this work, that the most severe effects of fluid contamination would be to cause abrasive wear, to silt up fine clearances, and to block capillary passages and orifices. These possible effects were related to the pump design shown in figure 4.2.1, the conclusions reached being summarised in table 4.2.1.

Given the time and resources available during this research, it would clearly have been impossible to have investigated all possible failure modes for all pump

PUMP AREA (see fig. 4.2.1)	PRIMARY EFFECT	SECONDARY EFFECTS
Shaft seal	Wear	Leakage, entry of dirt, bearing damage.
Bearings	Wear	Seizure, noise, inefficient operation, contamination of rest of pump.
Portplate: surface : orifice	Wear Erosion Blockage	Increased leakage, complete failure of pump through seizure.
Cylinder block	Wear	Damage to portplate, piston seizure, increased leakage, complete failure.
Slipper orifice	Erosion Blockage	Increased leakage. slipper wear, failure.
Slipper/thrust plate	Wear Silting of slipper pocket	Inefficiency, failure, debris.
Thrust-plate orifice	Blockage	Wear of thrust plate/ swashplate
Swashplate	Wear	Seizure, inefficiency, debris.

Table 4.2.1 Possible effects of fluid contamination on the A70 pump.

areas likely to be affected by fluid contamination. It was decided that the work should concentrate on the slipper/slipper-plate bearing. This area was selected for three reasons.

(i) CHL's own past experience suggested that the slipper bearing of an axial piston pump is especially sensitive to the effects of fluid contamination (although the results of the tests described in chapters 6 and 7 do not entirely support this view).

(ii) CHL had already sponsored research into the lubrication of slippers, so relevant data were available.

(iii) Research data has been published into the blocking of fine capillaries by small particles. Such capillaries are an important part of slipper bearings.

The work which follows concentrates on the slipper bearing of the CHL 5.6 in^3/rev. pump, but similar calculations and results would apply to any piston pump. It was assumed, in this section of the work, that the pressure distribution through a contaminated fluid, and the viscosity of that fluid, would be unaffected by the presence of particles. Very little work was found on these aspects of fluid contamination. Einstein [65] predicted a slight reduction in the viscosity of a suspension of particles as compared to the viscosity of the clean fluid, but his results indicate that the effect should be negligible at the contamination levels typical of hydraulic fluids. However, Turvey [66] measured a significant reduction in the viscosity of Shell Tellus 37 contaminated to 300 mg/l with coal particles. His results are considered in more detail in chapter 6.

In the absence of more published data, the analysis will proceed with a brief description of slipper lubrication on clean fluids. It will then be assumed that the same analysis will apply to slipper lubrication on contaminated fluids.

4.3 Slipper Lubrication on Clean Fluids

Figure 4.3.1 shows a typical piston and slipper unit. CHL have always designed slippers as hydrostatic thrust bearings, although Kakoullis [4] has recently suggested that the dominant mode of lubrication may be hydrodynamic or elasto-hydrodynamic.

In a hydrostatic slipper design, small quantities of fluid are bled from the piston bore, through a controlling capillary and into a pocket beneath the slipper. The fluid then flows across the slipper land. To achieve a balanced design, slipper geometry is chosen so that, at the required operating clearance, loads imposed on the piston, and hence on the slipper, by supply pressure in the piston bore, match those generated in the fluid beneath the slipper. Friction forces on piston and slipper, and inertial loads, are usually ignored. When the slipper is balanced, it can be shown (see for example Ernst [67]) that, at zero swash angle.

$$h^3 = \frac{1}{K_c}\left[\frac{12(Ro^2-Ri^2)}{d_p^2} - 6\ \ln\ (Ro/Ri)\right] \qquad 4.3.1$$

where the control capillary has a characteristic such that

$$P_s - P_i = \mu K_c q \qquad 4.3.2$$

μ is the fluid's absolute viscosity and K_c is a constant which depends on capillary dimensions. The leakage flow under the slipper, q, is given by

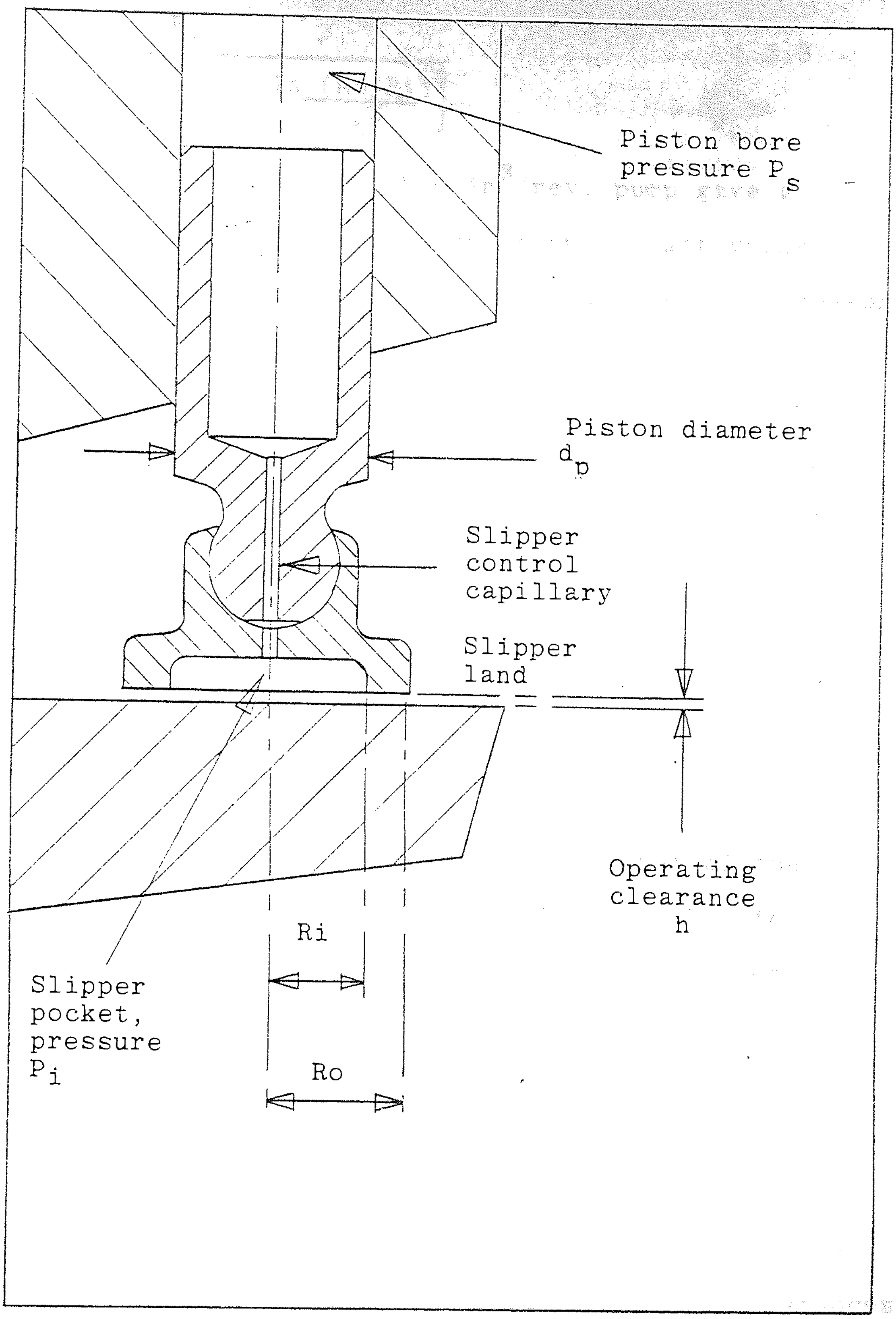


Figure 4.3.1 Slipper bearing lubrication.

$$q = \frac{\pi P_s}{\mu\left[K_c + \frac{6 \ln (Ro/Ri)}{h^3}\right]} \qquad 4.3.3$$

Calculations on a CHL 5.6 in^3/rev. pump gave a slipper clearance of 5 to 7 microns (the exact value depending on production dimensions). At a supply pressure of 200 bar and fluid viscosity of 26 cP, the leakage flow would be about 0.4 ml/sec. Under these conditions 4.3.3 may be approximated by

$$q = \frac{\pi P_s h^3}{6\mu \ln(Ro/Ri)} \qquad 4.3.4$$

Even if such a slipper was operated on fluid contaminated to only 10 mg/l, it would still be receiving about 2000 particles larger than the operating clearance every second. (This assumes that the contamination of the leakage flow would be the same as that of the bulk system fluid.) In a 5000 hour industrial life, the slipper would receive 3.6×10^{10} such particles. Before considering the possible effect of this contamination, attention was given to a suggestion made by Scott [56] (among others).

4.4 Design as a Means of Filter Specification

Scott has suggested that the problem of fluid contamination should be tackled by calculating the clearances between the moving components in the equipment to be protected, and then using these values to "help the designer to specify the level of filtration required for a system". For example, to protect the slipper bearing analysed in section 4.3, Scott would presumably specify a "5 micron" filter, so that all particles in the system fluid would be smaller than the slipper/slipper-plate clearance.

This approach certainly seems plausible and it had to be considered in this research, even though the subject of filtration is somewhat outside the scope of the work. It was discovered that the strategy is not feasible at present. The specific reasons why this should be so are best considered in relation to the three stages which would be involved in an attempt to implement the method. These are as follows.

(i) Calculate the clearances in the equipment to be protected. The author freely admits that the analysis of section 4.3 is an oversimplification. Pump designers and researchers in hydraulics are still not able to predict fluid film thicknesses with complete confidence. Even if this could be done, variations in production components would probably cause actual clearances to be different from those predicted.

(ii) If clearances could be predicted, these would have to be related to the sizes of particles which would be likely to cause damage. This is relatively easy in the case of abrasive wear damage. Particles smaller than fluid film thicknesses will not cause abrasion (although they might still erode surfaces.) But it is much less easy to decide on the size of particles which might cause silting or block a capillary. These problems are considered later in the chapter. All that can be said at the moment is that particles smaller than film thicknesses would have to be filtered out to provide complete protection. Referring to the pump clearances shown in table 2.1.1, this would imply removal of particles down to 1 micron in diameter.

(iii) It might be thought that particles of 1 micron diameter and above could then be removed by the installation of a "1 micron" filter. This is not the case. If a filter medium was a rigid mesh, as shown in figure 4.4.1, then it would be possible to talk of performance in terms of a micron rating. Figure 4.4.1 shows the particle transmission curve of such a filter, and it can be seen that the filter would remove all particles above a certain size.

Unfortunately, this model of a filter, although widely accepted, is inaccurate. A better concept is of a gaussian distribution of filter pore sizes, as shown in figure 4.4.2. An equivalent particle transmission curve is also shown. Day and Lee [14] argue that, due to the inaccuracy of particle counting methods, it is impossible to accurately determine filter efficiencies above 85%. This makes it virtually impossible to determine a complete filter performance curve, and it is not feasible to specify a filter to remove all particles above a given size. The difficulties are increased by the fact, discovered by Tsai and Way [64] among others, that the performance of a filter can be substantially reduced in service, compared to the performance achieved under test.

The above discussion has dealt with specific problems which would be involved in implementing Scott's suggestion. But the overall concept is also an oversimplification of the problem.

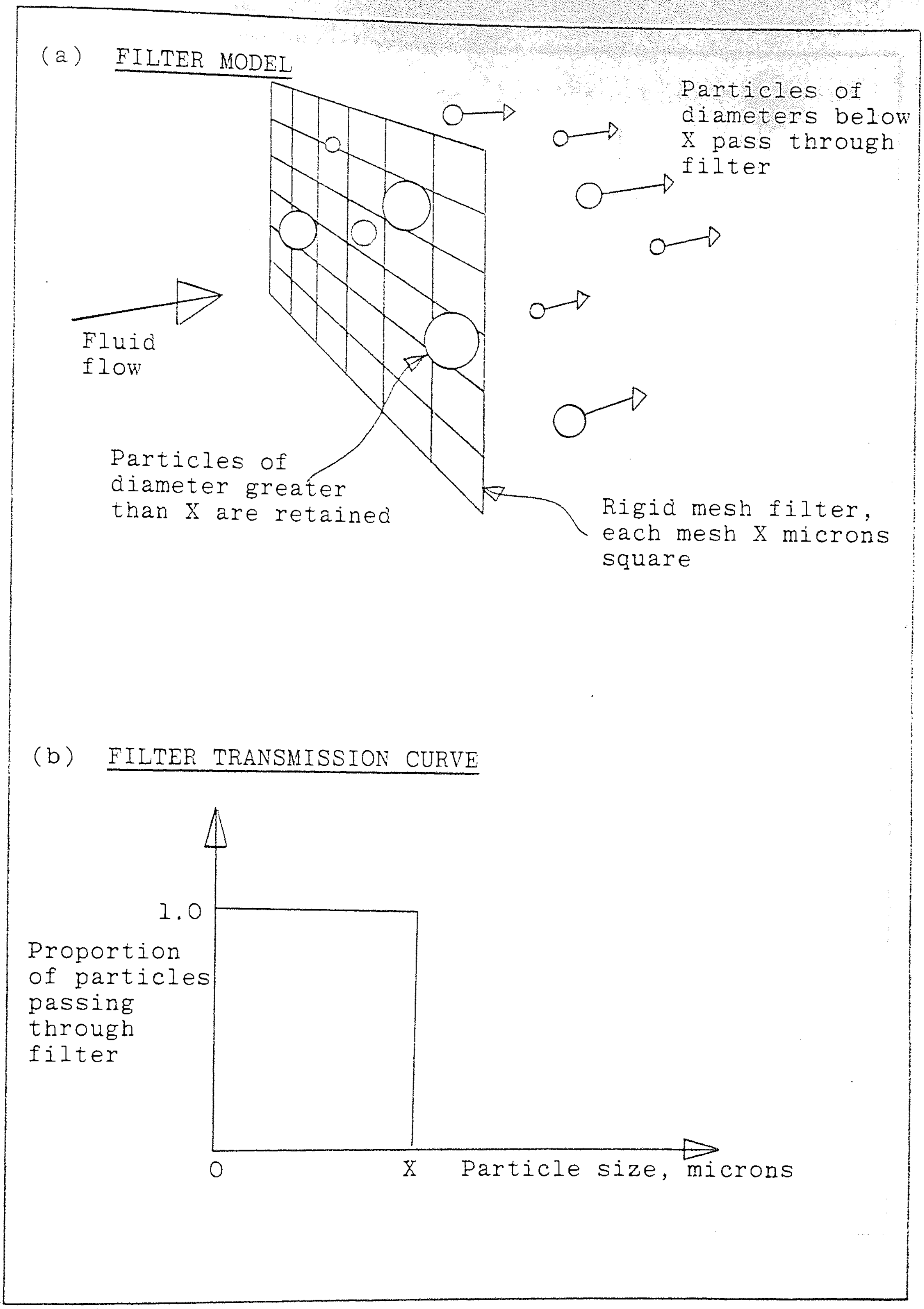
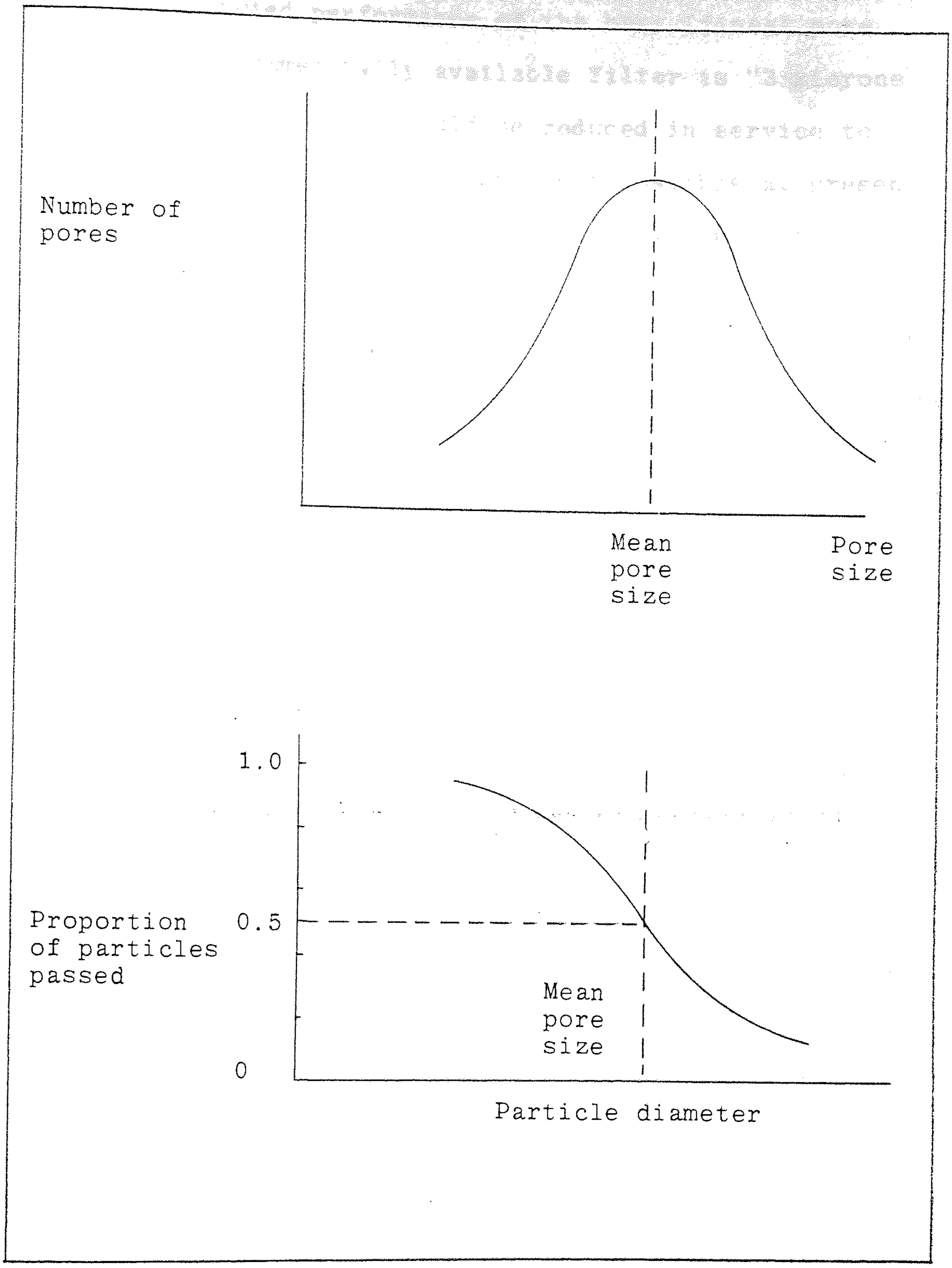


Figure 4.4.1 Rigid mesh filter model.



Model of a filter medium as a Gaussian distribution of pore sizes (upper curve) and equivalent particle transmission curve (lower graph).

Figure 4.4.2

(i) The quoted performance of the best (finest pore size) commercially available filter is "3 microns absolute". This could be reduced in service to 9 microns, so it is simply not possible at present to remove all the potentially damaging particles from a hydraulic fluid. This being the case, attention should be concentrated on the numbers of such particles which a pump can tolerate, and not on their size.

(ii) As shown in chapter three, filter performance is only one of many factors determining the contamination level of a fluid. The system designer should therefore specify an acceptable fluid contamination level, not a filter efficiency or rating.

(iii) Attempting to remove all potentially damaging particles from a fluid is wasteful, because it penalises 'dirt-tolerant' equipment. It also diverts attention away from attempts to reduce, through design modification, the effects of fluid contamination.

The overall conclusions to be reached are that design analysis can show the size of particles which are likely to be of concern, but that this knowledge does not really help in specifying the filtration required for a given system. It is also clear that some potentially damaging particles will be present in any hydraulic pump. The next sections therefore concentrate on the possible effects of these particles.

4.5 The Abrasive Wear of Slipper Bearings

Abrasive wear involves the removal of material from the surface of a body by processes of plastic deformation.

The deformation may be produced by asperities (protuberances) on a second surface (two-body wear), or by particles introduced between two surfaces (three-body wear). The latter situation applies to the possible effects of fluid contamination.

Of course, the mere presence in a pump of abrasive particles does not necessarily imply that abrasive wear will take place. The particles must enter clearances between components and must then be pressed into one or both surfaces. If particles arrived at a clearance as shown in figure 4.5.1 (a) then silting might occur, but abrasive wear would not be a problem. However, any of the situations illustrated in figures 4.5.1 (b) and (c) could allow potentially damaging particles to enter critical pump areas. These situations do exist in pumps and it can be assumed that particles will find their way between components. The question to be answered is whether or not the forces imposed on the particles will be sufficient to induce plastic deformation in pump surfaces.

Hertz [68] considered the elastic deformation of a surface by a spherical indenter (which may be compared to a particle), as in figure 4.5.2. He showed that the radius 'a' of the circle of contact is related to the applied load W by the formula

$$a = \left\{ \tfrac{3}{4}\, Wr \left[\frac{1-\sigma_1^{\,2}}{E_1} + \frac{1-\sigma_2^{\,2}}{E_2} \right] \right\}^{1/3} \qquad 4.5.1$$

where E and σ are Young's modulus and Poisson's ratio respectively for the spherical particle (1) and the surface material (2).

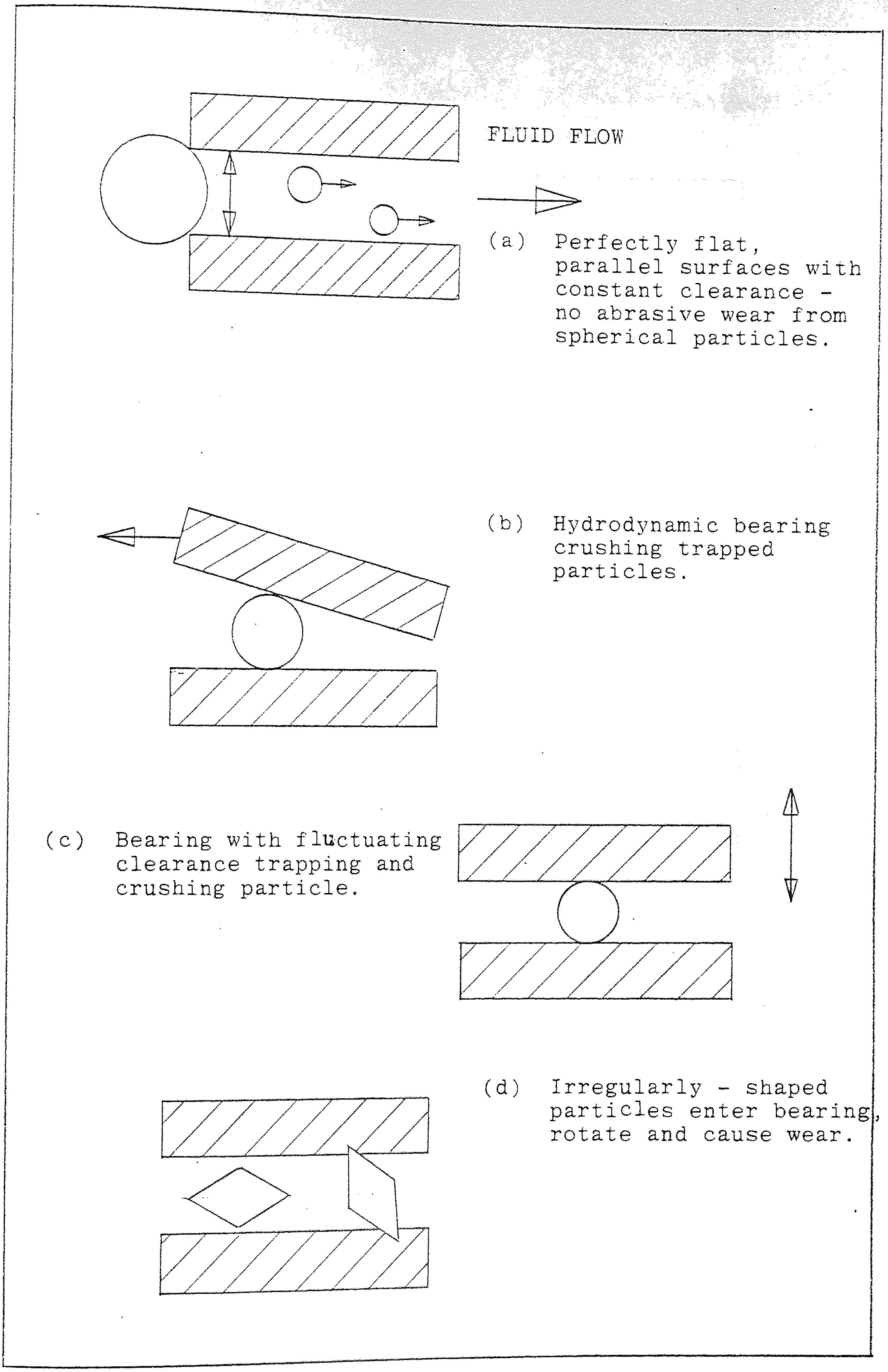


Figure 4.5.1 Abrasive wear situations.

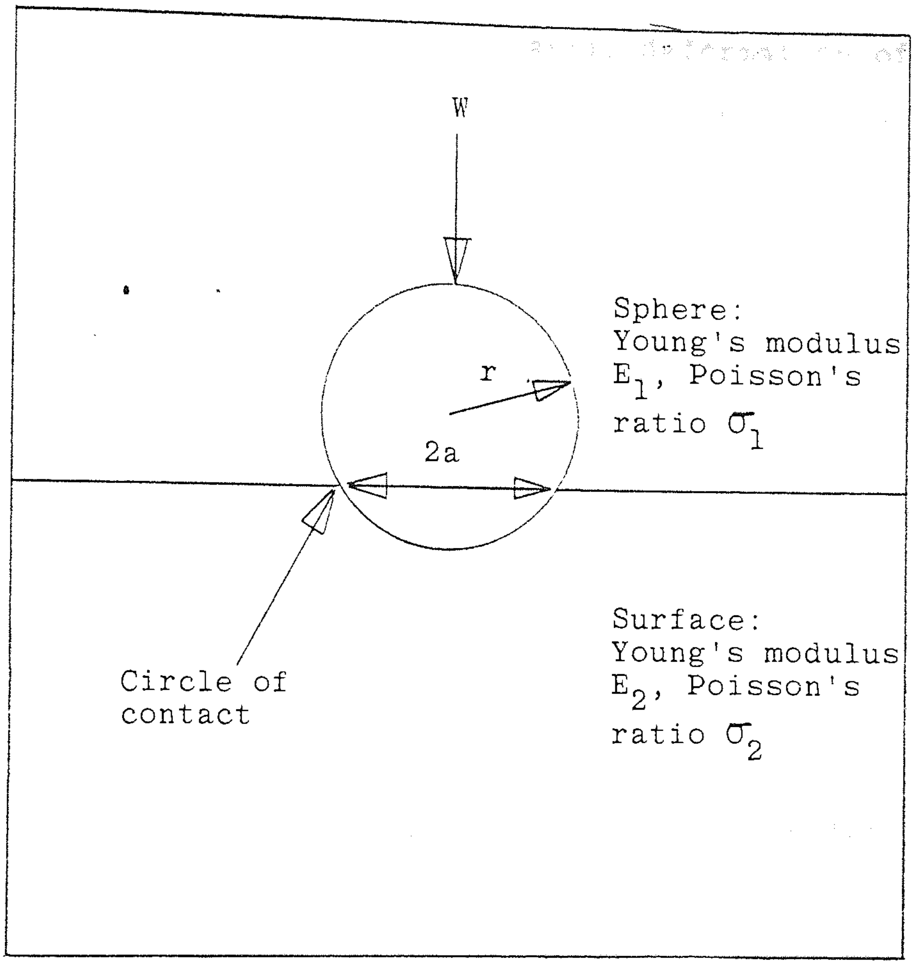


Figure 4.5.2 Rigid sphere impressed into a surface

Timoshenko [69] developed Hertz's analysis and calculated that the maximum shear stress in the surface material is produced approximately 0.5a below the centre of the circle of contact. Plastic deformation of a ductile material will commence when this shear stress exceeds a certain value. Both the Tresca and Hüber-Mises criteria for the onset of plasticity indicate that this value should be 0.5Y, where Y is the yield stress of the surface material. (The analysis assumes that the indenter is not deformed).

Tabor [70] shows that, under certain conditions, full plasticity will be reached when the mean pressure, P_m, $(= W/(\pi a^2))$ is approximately 3Y. He also found, from experiment, that this occurs at a load approximately 150 times that required to initiate plastic deformation.

The author used Tabor's results to calculate the loads under which spherical silica particles will produce plastic deformation of a steel surface with a Vickers' hardness of 300. It was assumed that the particles would not deform themselves, and that Poisson's ratio was 0.3 for both materials. The results were compared to equivalent values derived (also by the author) from experimental data given by Tabor. The data were used in the empirical relation obtained by Meyer [71], such that a sphere impressed, under load W, into a surface will leave an impression of diameter d, where

$$W = Ad^m \qquad 4.5.2$$

A is a constant, and m has a value, depending on the surface material, between 2 and 2.5.

The calculated values are shown in table 4.5.1 and it can be seen that the two different approaches gave similar results. The conclusion to be drawn is that only very light loads will be required to press contaminant particles into pump surfaces. The required loads are almost negligible compared to those which have been calculated to be imposed on pump components during operation, and abrasive wear is therefore a distinct possibility.

The above analysis rested on an assumption that the contaminant particle was harder than pump surfaces, and that the particle itself therefore suffered no deformation. One possible design approach to minimising abrasive wear would be to use pump materials that were harder than the hardest contaminants likely to be encountered. Tabor [op cit] showed that a surface will not be deformed by an indenter (particle) if the surface material is at least 2.7 times as hard as the indenter (both hardnesses being measured on the Vickers' scale).

Table 4.5.2. shows the hardness of various pump materials and possible fluid contaminants. It can be seen that, on Tabor's criterion, complete protection against contaminants such as silica (quartz) could only be provided by carbide materials. Such materials are expensive and difficult to use, and their selection would be a matter of economics. A further consideration is that, if contaminant particles come, in part, from wear of pump components themselves, then building pumps of harder materials will simply produce harder fluid contaminants. As wear particles will also be produced in a fully work-hardened state, the use of harder pump materials may not be a satisfactory approach to the problem.

PARTICLE DIAMETER (MICRONS)	LOAD TO PRODUCE FULL PLASTICITY (N), FROM THEORY/EXPT	DIAMETER OF REMAINING IMPRESSION	
		FROM HERTZ[1] MICRONS	FROM MEYER[1] MICRONS
2	1.18×10^{-4}	0.24	0.29
10	2.95×10^{-3}	1.18	1.43
20	0.012	2.36	2.89
30	0.027	3.54	4.33
40	0.047	4.72	5.72
60	0.106	7.08	8.59
100	0.295	11.80	14.32

[1] See Text

Table 4.5.1

Loads required to produce plastic deformation of steel surface (HV = 300) and diameter of impression formed, for various sizes of silica spheres.

CONTAMINANT	VICKERS HARDNESS	MATERIAL	VICKERS HARDNESS
Carbon	35	Aluminium	45
Calcite	140	Tin-bronze	80
Fluorite	190	Aluminium alloy	170
Glass	500	Brass	190
Feldspar	600-750	Phosphor bronze	200
Flint	950	Mild steel	200
Quartz (Silica)	900-1280	Cast iron	150-240
Emery	1400	Medium C steel	130-250
Corundum	1800	High C steel	140-400
Silicon carbide	2600	Alloy steel	300-600
		Hard chrome	600-1000
		Chromium carbide	1200-1600
		Tungsten carbide	2400
		Titanium carbide	3200

Table 4.5.2 Hardness of fluid contaminants and pump materials. Data from Climax Molybdenum Company Limited .

One popular way of improving 'wear-resistance' is by surface treatment. CHL apply such a technique to many pump components, to produce a thin, but very hard layer in the surface of a low to medium carbon steel. One company undertaking this process claims that it produces a layer up to 8 microns thick, with a surface hardness of 500 to 600 on the Vicker's scale.

It might be expected that such a surface would provide substantial protection against abrasive wear. But the analysis presented above indicates that a particle, pressed into such a surface, could cause plastic flow of the base material, causing the hardened outer surface to break away. CHL have suspected, for some time, that this can occur. Broeder and Heijnekamp [53] also observed pitting, initiated by abrasive particles in the lubricant, when running hardened steel shafts against hard steel bushes. It could be that CHL should move to through-hardened components of more homogeneous properties, if they want to reduce abrasive wear damage in their pumps.

4.6 Abrasive Wear Rates

The conclusion to be reached from the work described in section 4.5 is that some degree of abrasive wear is almost inevitable in an axial piston pump subjected to particles of a hardness above about 100 on the Vickers' scale. These would of course include particles worn from pump surfaces by other processes.

The problem then becomes one of reducing abrasive wear rates. This section presents calculations to determine the wear-rates of a slipper/slipper-plate bearing subjected to a range of particle sizes and concentrations. The analysis was conducted for three reasons:

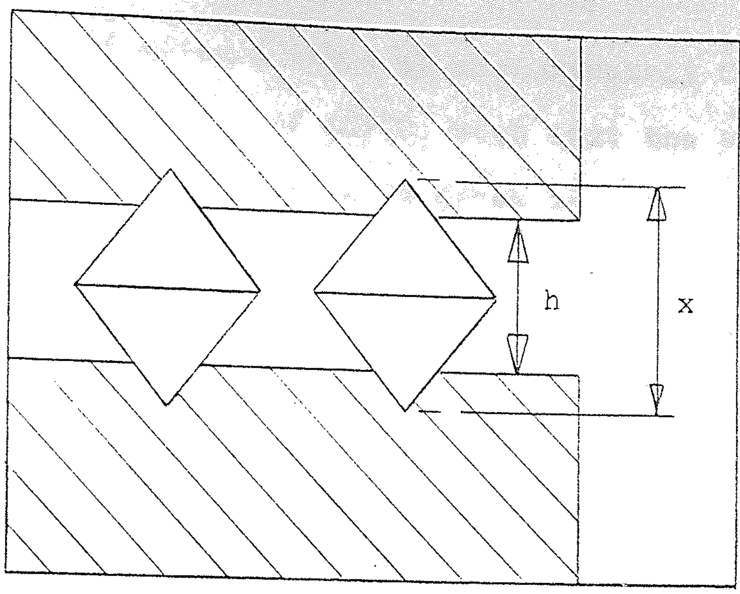
(a) to see if this theoretical approach is feasible, or if it could be feasible given more experimental data;

(b) to see whether theoretical analysis can be used to suggest ways of reducing abrasive wear rates;

(c) to see what reductions in abrasive wear rates may be expected if hydraulic fluid contamination levels are reduced.

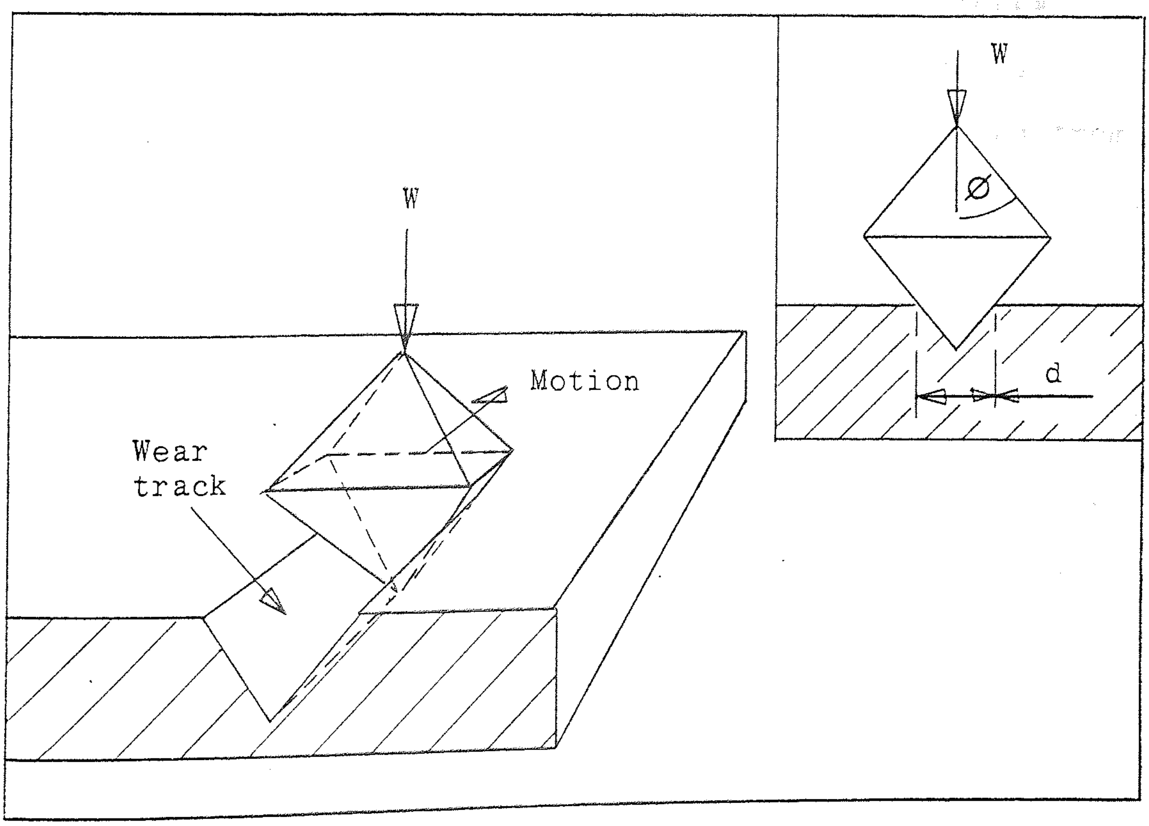
The situation considered was that of figure 4.6.1. Particles have somehow been introduced between two surfaces (in this case a slipper and slipper-plate, although the analysis would be generally applicable). The particles have wedged the surfaces apart to a clearance h, which is greater than the bearing's equilibrium clearance h_o. This has disturbed the force balance on the components and has generated a force P, tending to reduce h. This force has pressed the particles into the surfaces.

The particles were modelled as double pyramids rather than spheres, because optical microscope examination showed that the double pyramid is an appropriate model for silica. Silica is very hard (see table 4.5.2), and is a very common material in the Earth's crust. It therefore represents one of the most abrasive contaminents likely to be found in a hydraulic fluid and as such is a suitable subject for study. Another reason for studying the effects of silica is that Air cleaner fine test dust (ACFTD), the 'artificial' contaminant used in the OSU pump test, is composed predominantly of this material. Any knowledge of its effects, and more especially of ways to reduce them, promised to be very useful.



Abrasive wear of pump components

Figure 4.6.1



Wear by a pyramid-shaped particle

Figure 4.6.2

Cole [21] gives a value of approximately 0.2 for the shape factor (α_v) of ACFTD, such that the volume, V_p, of a particle of diameter x is given by

$$V_p = \alpha_v X^3 \tag{4.6.1}$$

This implies a value of $\emptyset = 38^o$ for the particle model shown in figure 4.6.2.

Bowden and Tabor [57] have considered the wear produced by a pyramid-shaped particle pressed into a surface, as depicted in figure 4.6.2. They suggest that the surface material will deform plastically until loads generated within the surface support those imposed on the pyramid. To account for a certain amount of 'load-spreading', they take the actual load-bearing area to be approximately twice the apparent area, so that:

$$W = 2d^2p \tag{4.6.2}$$

where p is a representative surface material hardness.

If the particle shown in figure 4.6.2 slides a distance 1 before it is either crushed or expelled from the bearing, and if all the material deformed plastically is removed from the surface, then the volume of metal removed by the particle, V_w, will be given by

$$V_w = \frac{d^2 1}{4 \tan \emptyset} \tag{4.6.3}$$

The wear rate of the surface will be a function of the number of potentially damaging particles entering the bearing per second. If the concentration of such particles is n per unit fluid volume, then the rate at which such particles enter the bearing (N_w) will be, from 4.3.4.

$$N_w = \frac{n\pi P_s h^3}{6\mu \ln(Ro/Ri)} \tag{4.6.4}$$

Of course, these particles will have various sizes, and a major difficulty encountered in the analysis was trying to deal with the cumulative effects of particles with a range of diameters. Eventually it was assumed, largely for simplicity, that the particles were of the same size. This is not totally unrealistic, for those particles which actually enter the bearing clearance will be of a similar order of size. However, a more sophisticated analysis would have to tackle the problem of particles with a size distribution.

For monosize particles, 4.6.3 and 4.6.4 together give the total volumetric wear rate of one surface, $\frac{dV_{TOT}}{dt}$, where

$$\frac{dV_{TOT}}{dt} = \frac{\pi P_s h^3 n.d^2 l}{24\mu \tan\emptyset \ln(Ro/Ri)} \qquad 4.6.5$$

For a bearing of surface area A_B, the linear wear rate, that is the rate of reduction in component thickness, is given by $dz/_{dt}$, where

$$\frac{dz}{dt} = \frac{\pi P_s h^3 n d^2 l}{24 A_B \mu \tan\emptyset \ln(Ro/Ri)} \qquad 4.6.6$$

This relation has been derived from simple geometric considerations, given an initial value of d, and hence of h, for a specified particle diameter. The actual value of h will depend on a force balance betwen the load supported by the material beneath the particles, and the load generated due to the bearing imbalance. It was calculated, from 4.6.2, that the former is given by

$$P = nA_B h.2p\tan^2\emptyset(x-h)^2 \qquad 4.6.7$$

The load due to the bearing imbalance was calculated to be

$$P = \frac{\pi P_s K_c (Ro^2 - Ri^2)(h^3 - h_o^{\prime 3})}{12 \left[\ln(Ro/Ri) \right]^2} \qquad 4.6.8$$

4.6.7 and 4.6.6 together give

$$\frac{dz}{dt} = \frac{\pi P_s h^2 Pl}{48 A_B^2 p\mu \tan\phi \ln(Ro/Ri)} \qquad 4.6.9$$

Bowden and Tabor [57] and Rabinovicz [58] found, in studies of abrasive wear, that under dry conditions, only 10% of the particles present were actually producing wear. However, under lubricated conditions, wear rates were 2 to 4 times as high as those obtained under dry conditions. These findings were included in 4.6.9 to give

$$\frac{dz}{dt} = \frac{0.02 P_s h^2 Pl}{A_B^2 p\mu \tan\phi \ \ln(Ro/Ri)} \qquad 4.6.10$$

4.6.7, 4.6.8 and 4.6.10 were used to calculate the linear wear rate of a CHL slipper design operating on fluid contaminated to different levels, and with different sized particles. The results are shown in table 4.6.1 and are plotted in figure 4.6.3. It was assumed, in calculating the results, that both pump surfaces were steel with a Vicker's hardness of 300, and that the average length of a wear track would equal the width of the slipper land.

Before discussing the results of the calculations, one point needs explanation. The particle counts shown are for "notional fluid gravimetric levels". The problems of dealing with a range of particle sizes have already been mentioned. It was calculated that, on the double-pyramid particle model shown in figure 4.6.2, particles of a maximum dimension up to 1.5 times the bearing clearance

PARTICLE DIAMETER (MICRONS)	PARTICLES PER ML. FLUID	h (MICRONS)	P (NEWTONS)	LOAD/ PARTICLE (NEWTONS)	WEAR-RATE dz/dt (m/sec)	$h^2 dz/dt$ m^3/sec	INDEX IN RELATION $dz/dt \; \alpha \; n^k$
8.0	8640	5.08	0.14	0.030	2.4(-8)	6.2(-19)	0.97
	28800	5.23	0.43	0.027	7.9(-8)	2.2(-18)	
	86400	5.55	1.08	0.021	2.2(-7)	6.8(-18)	
	2.88(5)	6.09	2.39	0.013	5.9(-7)	2.2(-17)	
12.0	8640	5.40	0.76	0.154	1.5(-7)	4.3(-18)	1.15
	28800	6.06	2.30	0.124	5.6(-7)	2.1(-17)	
	86400	7.10	5.50	0.085	1.8(-6)	9.3(-17)	
	2.88(5)	8.47	11.38	0.044	5.4(-6)	3.9(-16)	
15.0	8640	5.77	1.59	0.30	3.5(-7)	1.2(-17)	1.23
	28800	6.92	4.86	0.23	1.5(-6)	7.4(-17)	
	86400	8.51	11.57	0.15	5.5(-6)	4.0(-16)	
	2.88(5)	10.40	23.64	0.07	1.7(-5)	1.8(-15)	

Table 4.6.1 Predicted wear rate of a CHL slipper design for different particle sizes and concentration.

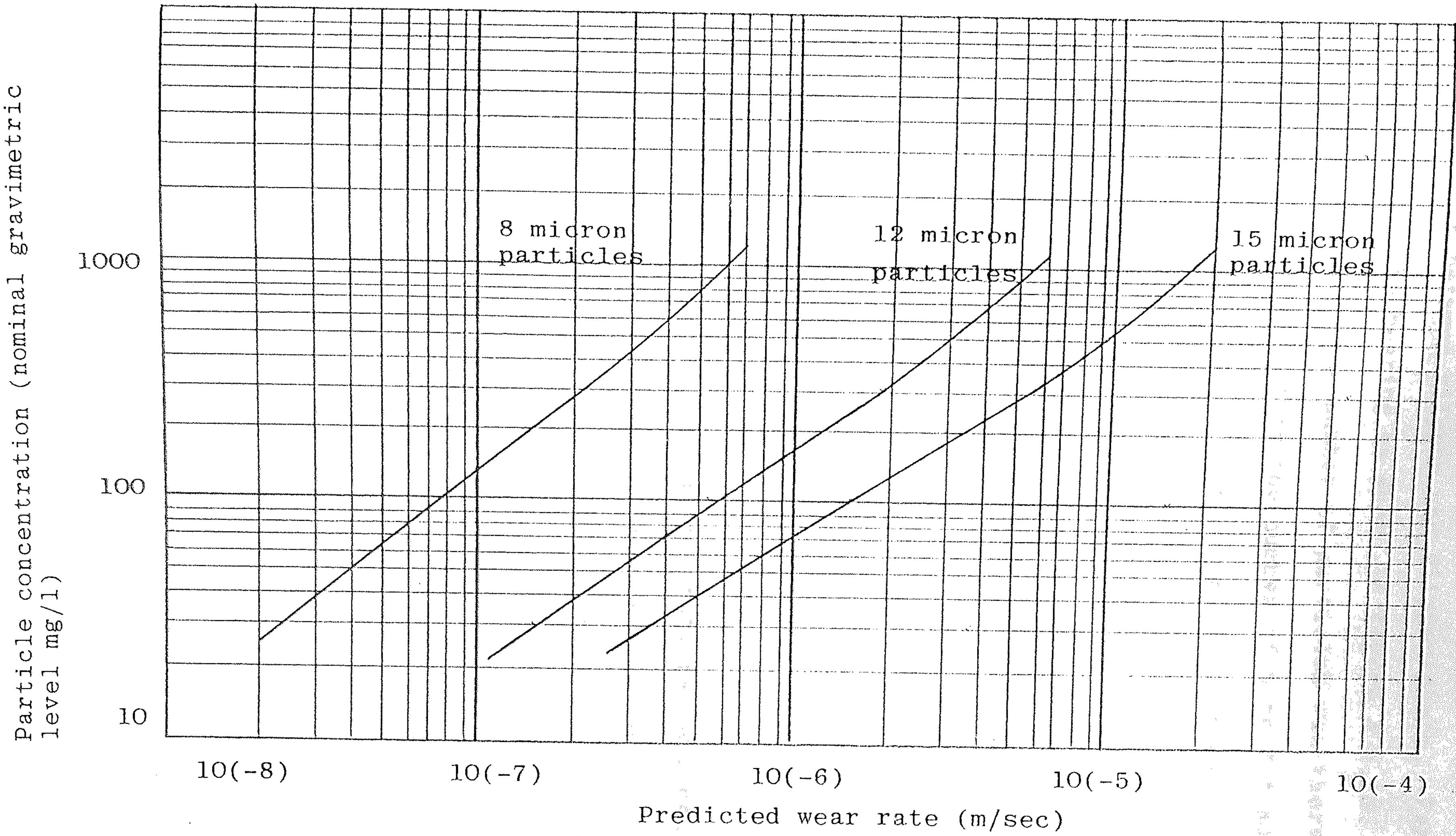


Figure 4.6.3 Predicted wear rates of a CHL slipper design for different particle sizes and concentrations. (Log scales)

could enter between slipper and slipper-plate. Particles could enter the bearing as shown in figure 4.5.1d. In the calculations, the number of particles was therefore taken to be equivalent to the number of 5 to 7.5 micron particles in the log-log^2 size distribution model of ACFTD proposed by Cole [21]. (This model is considered in section 4.8.)

Figure 4.6.3 shows that, as would be expected, predicted wear rates increase with both increasing fluid contamination level and with increasing particle size. (It must be remembered that, in an actual fluid, there would be fewer of the larger particles, so the situation would be more complex than is suggested here.) It can be seen that the rate of increase in wear rate starts to decrease at higher particle concentrations, and could be constant above a certain level. Reference to table 4.6.1 shows why this might be so. Although an increase in the particle concentration might be expected to increase wear rates, it can be seen that it also decreases the load on each particle. Reference to table 4.5.1 shows that, at a high enough particle concentration, the load on each particle might be insufficient to initiate plastic deformation in the pump surfaces, and no wear would occur. However, from figure 4.6.3 it can be seen that, below a certain concentration level, predicted wear rates obey a relation of the form

$$\frac{dz}{dt} \quad \alpha \quad n^k \qquad 4.6.11$$

where k is an index depending on the particle diameter. This is an important finding, because it relates to the OSU pump test. This relation will now be considered.

4.7 The OSU Pump Test Theory

The OSU pump test has been mentioned, but not described or discussed in any detail. The test is basically a method of predicting the life of a hydraulic pump operating on fluid contaminated to a specified level. The full test procedure is described in reference [72], and the test is discussed more fully in an internal report prepared for CHL [73]. The basis of the test is to operate a pump on high levels of abrasive contaminants, and to use the consequent reduction in pump output flow as a measure of the damage produced. Results obtained at the high test contamination levels must then be related to the levels of damage which would be expected at the lower levels typical of service conditions. OSU use a relation of the form

$$\frac{dQ}{dt} = -\gamma_i \, n_i^{2} Q \qquad 4.7.1$$

where Q is the pump output flow and γ_i is a "wear coefficient" for a given pump and a narrow band of particle diameters in range i. n_i is the number concentration of such particles per unit volume of fluid.

The usefulness of the OSU test rests largely on the validity of equation 4.7.1. This relation is an empirical one, derived from tests on gear pumps, and it has not been compared, as far as the author is aware, to any corresponding theoretical analysis. This will now be done.

For a pump of theoretical output Q_v

$$Q = Q_v - q_{tot} \qquad 4.7.2$$

where q_{tot} is the total pump leakage. Leakage calculations for most pump areas produce similar relations to that of equation 4.3.4, so that

$$q_{tot} = ch^3 \qquad 4.7.3$$

where c is a constant for a pump at a specified set of operating conditions. Therefore, from 4.7.2 and 4.7.3

$$\frac{dQ}{dt} = -3ch^2\frac{dh}{dt} \qquad 4.7.4$$

<u>If</u> changes in component dimensions produce corresponding increases in leakage path dimensions then, using the symbols of section 4.6

$$\frac{dQ}{dt} = -3ch^2\frac{dz}{dt} \qquad 4.7.5$$

Figure 4.7.1 shows values of h^2dz/dt calculated from the data of table 4.6.1. It can be seen that a relation of the form

$$h^2\frac{dz}{dt} \propto n^s \qquad 4.7.6$$

is appropriate, with s having a value of 1.07 to 1.54, depending on the size of the particles producing the wear. Combining 4.7.6 with 4.7.4 shows that

$$\frac{dQ}{dt} \propto n^s \qquad 4.7.7$$

For small changes in Q, it could well be that a relation of the form

$$\frac{dQ}{dt} = -\gamma n^s Q \qquad 4.7.8$$

will apply.

This analysis does therefore lend some support to the wear equation used by OSU in their test theory. However, the support has certain qualifications:

(a) In an axial piston pump, changes in component dimensions would not necessarily imply an increase in bearing clearances. For example, uniform linear wear of a slipper would not increase slipper leakage. It might be that component wear would

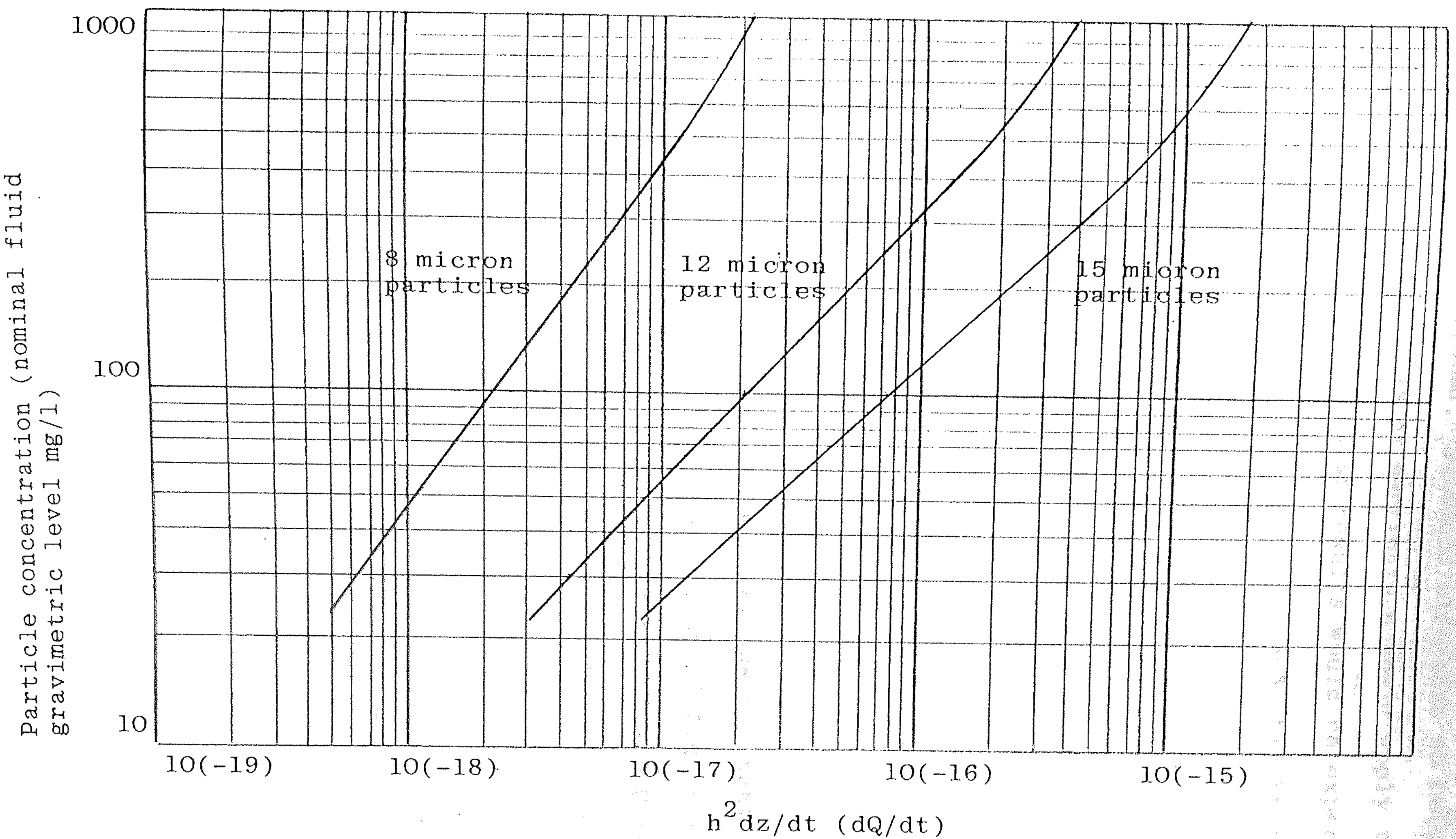


Figure 4.7.1 Predicted variation with contaminant concentration of pump output flow degradation. (Log scales)

increase clearances in a gear pump. This would have to be established. It would also have to be shown that the analysis of section 4.6 would apply to gear pumps, although similar results would be expected.

(b) The values of s predicted in the analysis were specific to one size of particle, to one pump design and indeed to one area in that pump. It might be possible to consider an 'average' value of s for a typical particle size distribution, but s would have to be established for each pump design and each bearing area in that design.

(c) The relation of equation 4.7.6 breaks down at high particle concentrations (and possibly at very low concentrations as well). One would have to be certain, when using the OSU test theory, that one was operating in the linear region of figure 4.6.3.

The conclusions to be drawn are that the OSU test theory may be applicable to gear pumps, but that more work needs to be done to establish its validity and to investigate variations in s between different designs. There is apparently no theoretical basis, on the analysis conducted here, for supposing that the theory will apply to axial piston pumps. Experimental evidence on this aspect of the work is given in section 6.5.

4.8 Reducing Abrasive Wear Rates

One possible use of the type of analysis presented in sections 4.6 and 4.7 is to suggest design improvements or material changes to reduce abrasive wear rates. Some general approaches would be to:

(a) Design less "stiff" bearings by reducing $\frac{dP}{dh}$. This would mean that particles would not be pressed so hard into pump surfaces.

(b) Try to design out the situations shown in figure 4.5.1, so that fewer particles enter bearings.

(c) Use through-hardened materials on both surfaces, or use a 'sacrificial' soft material on one surface to 'absorb' damage.

(d) Alter component dimensions to reduce calculated values of $dz/_{dt}$.

It must be emphasised that these approaches represent untested hypotheses. Ways in which the hypotheses could be verified are considered in Chapter 5.

Although no general redesign was conducted during this project, one method of reducing contamination sensitivity was studied in more detail. It seemed resonable to assume that the effects of contamination would be reduced by operating equipment at higher clearances. A greater proportion of damaging particles should then pass through critical areas without causing wear.

Two objections to this approach were foreseen. One is that it may not be possible to design equipment to operate at higher clearances, or indeed at any predetermined clearance. However, this is a separate problem, and does not affect the possible desirability of doing so. A second objection is that operation at higher clearances involves a substantial penalty in lost efficiency. As equation 4.3.4 indicated, leakage flows are approximately proportional to the cube of the clearance between components. But there may still be cases where reduced efficiency is an acceptable price to pay for increased reliability or longer life.

It has already been assumed that abrasive wear of a bearing will be caused by particles of the same order of size as the operating clearance h. A bearing 'sensitivity coefficient', S_A, can be defined as

$$S_A = qn(h) \qquad 4.8.1$$

n(h) represents the particle size distribution of the fluid contaminant, such that there are n(h)dh particles of diameter h to h+dh per unit volume of fluid. S_A is therefore a measure of the number of potentially damaging particles entering the bearing per second.

The analysis could not be continued without a suitable mathematical model for the size distribution of a fluid contaminant. Herdan [74] and others have suggested that some powders have a log-normal particle size distribution such that

$$n(h) = \frac{n(o)}{\sigma h\sqrt{2\pi}} \exp\left[\frac{-1}{2\sigma^2} \ln^2\left(\frac{h}{M_h}\right)\right] \qquad 4.8.2$$

M_h and σ are constants. (This distribution is a modified form of the normal distribution. If y has a normal distribution with mean M_h and variance σ^2, and $y = \ln(h)$, then h has a log-normal distribution.) Such a model might also be suitable for fluid contamination.

Cole [21] derived his log-log^2 model for ACFTD from a log-normal model by assuming that M_h = 1 micron. In fact his analysis seems to be in error, although his result may still be useful. His distribution is

$$N(h) = N(1)e^{-B\ln^2(h)} \qquad 4.8.3$$

where N(h) is the number of particles larger than diameter h per micro-gramme of ACFTD, and B is a constant. ACFTD is generally thought to be fairly representative of contaminants in hydraulic systems, so equation 4.8.3 could apply to a system contaminant. However, the author derived an equivalent mass distribution from Cole's model, and compared it to the published mass distribution of ACFTD. The results are shown in table 4.8.1, and it can be seen that there is some discrepancy.

To obtain a better correlation, a true log-normal model for ACFTD was derived from a "least-squared-error" fit to the published mass data. The calculations gave the results that, in equation 4.8.2, $\sigma = 1.17$ and $M_h = 0.133$ microns. The distribution is compared to published data, and to the log-log^2 model, in table 4.8.1. It can be seen that the log-normal model is not ideal, but that it does give a much closer 'fit' to published data than does the log-log^2 relation. (Further implications of these findings are considered in section 8.3.)

Both particle size models of ACFTD were used to calculate values of S_A from equation 4.8.1. The results are shown in figure 4.8.1, where it can be seen that the different particle size models give markedly different results. Remembering that S_A only represents the numbers of potentially damaging particles entering the bearing per second, the log-normal model suggests that S_A may be progressively reduced by operation at clearances above 2 microns. But the log-log^2 model suggests the exact opposite! It is not possible to draw any general conclusions about pump design from the data shown in figure 4.8.1. However the results do illustrate two important points.

(a) This type of analysis will not provide useful results unless and until a representative particle size distribution for actual system contaminants can be accurately determined.

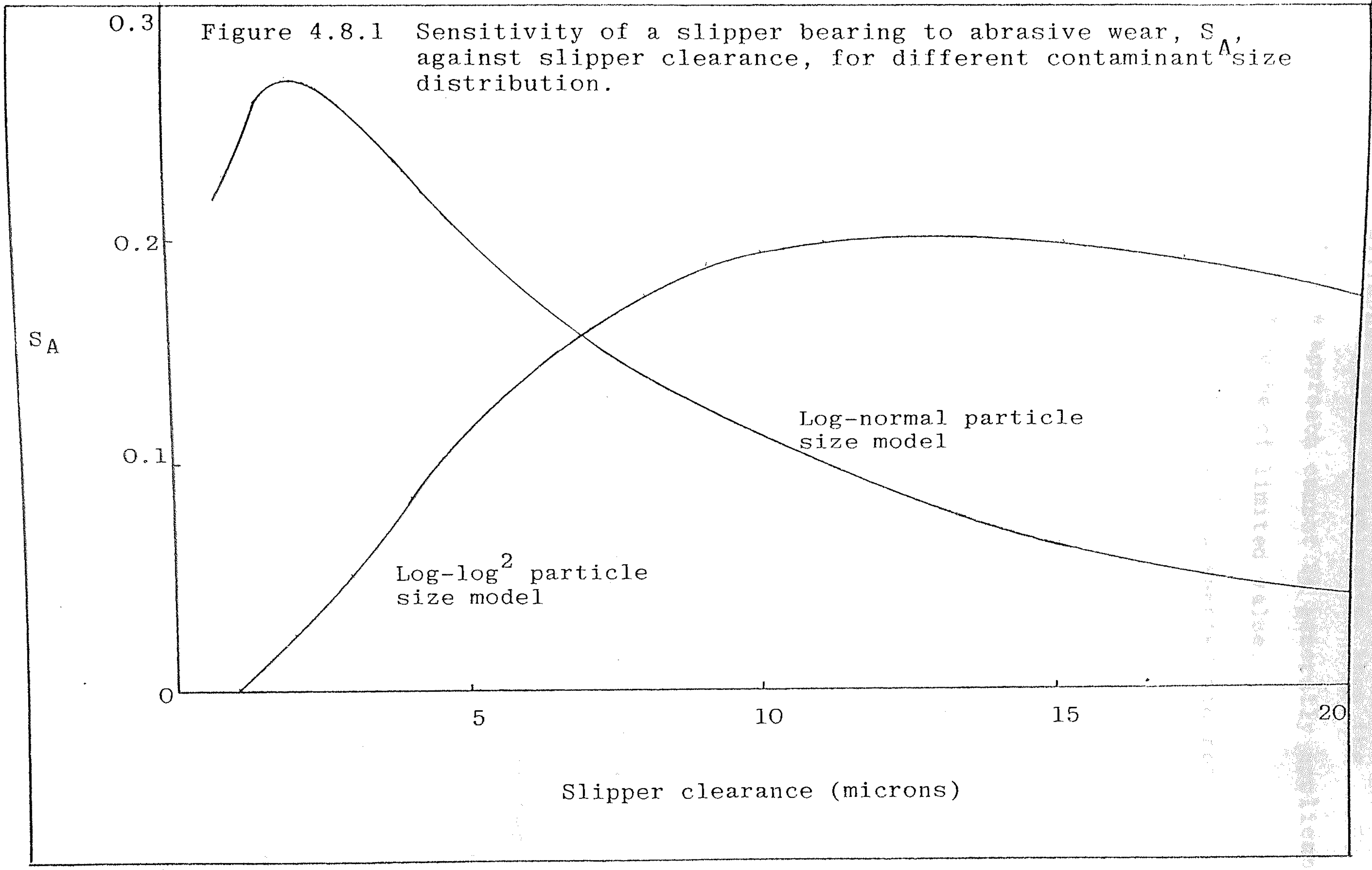
(b) If particle size distributions vary considerably

PARTICLE SIZE MICRONS		MASS DISTRIBUTION: PROPORTION OF DUST OF DIAMETER GREATER THAN SIZE INDICATED			PARTICLES OF DIAMETER GREATER THAN INDICATED SIZE PER MICRO-GRAMME	
LONGEST DIMENSION	STOKES DIAMETER	MANUFACTURER'S DATA	LOG-LOG^2 MODEL	LOG NORMAL MODEL	LOG-LOG^2 MODEL	LOG NORMAL MODEL
1.37	1.0		1.00	0.96	1672	106000
6.85	5.0	0.61	0.95	0.66	305.8	2419.0
13.70	10.0	0.43	0.82	0.43	69.3	277.5
27.40	20.0	0.27	0.58	0.22	10.0	23.0
41.10	30.0	0.15[1]	0.41	0.13	2.6	4.5
54.80	40.0	0.09	0.31	0.09	0.9	1.2
68.50	50.0	0.06[1]	0.23	0.06	0.4	0.7
82.20	60.0	0.04[1]	0.18	0.04	0.2	Results cannot be calculated
109.60	80.0	0	0.11	0.02	0.05	

NOTE 1: Manufacturer's data is not comprehensive. These are values interpolated from manufacturer's specification.

Table 4.8.1 Particle size distribution of Air Cleaner Fine Test Dust.

Figure 4.8.1 Sensitivity of a slipper bearing to abrasive wear, S_A, against slipper clearance, for different contaminant size distribution.



between different fluid contaminants, then the results of this approach cannot be generally applicable, and must therefore be of limited value.

Of course, the analysis of S_A presented did not consider the difference in the wear rates produced by different sized particles. Nor did it consider the changes in the load characteristics of a bearing which would be created by designing it for operation at a different clearance.

It is difficult to draw any general observations about the variation of wear-rate with slipper clearance, due to the complexity of the relations involved. But, to investigate the likely effects, a hypothetical slipper redesign was undertaken. The calculated theoretical operating clearance of the slipper considered in sections 4.3 and 4.6 was increased from 5 to 8 microns. This could be done in several ways in practice, but the easiest would be to reduce K_c by increasing the diameter of the slipper capillary. It was calculated that the modest increase in capillary diameter required would increase slipper leakage by a factor of four, which would produce a measurable reduction in pump efficiency.

The expected wear rate of the new design was calculated, using the relations derived in section 4.6. It was assumed that damage would be produced, in the new slipper, by 8 to 12 micron particles all behaving as 12 micron particles. Particle counts were determined from the log-log^2 particle size model. The results of the calculations are shown in table 4.8.2, and are illustrated in figure 4.8.2. It can be seen that, at least on the log-log^2 particle size model, operation at higher

NOMINAL FLUID GRAV. LEVEL MG/L.	SLIPPER OPERATING AT 5 MICRONS CLEARANCE				SLIPPER OPERATING AT 8 MICRONS CLEARANCE			
	5 - 8 microns particles per ml of fluid [1]	h mm	$\frac{dz}{dt}$ m/sec	$h^2\frac{dz}{dt}$ m^3/sec	8 - 12 micron particles per ml of fluid [1]	h mm	$\frac{dz}{dt}$ m/sec	$h^2\frac{dz}{dt}$ m^3/sec
30	8640	5.078	2.4(-8)	6.2(-19)	3978	8.159	7.91(-8)	5.26(-18)
100	28800	5.233	7.9(-8)	2.2(-18)	13260	8.452	2.51(-7)	1.79(-17)
300	86400	5.547	2.2(-7)	6.8(-18)	39780	8.979	6.54(-7)	5.27(-17)

1 Based on log - $\log^2$ model

Table 4.8.2 Predicted wear rates of different CHL slipper designs.

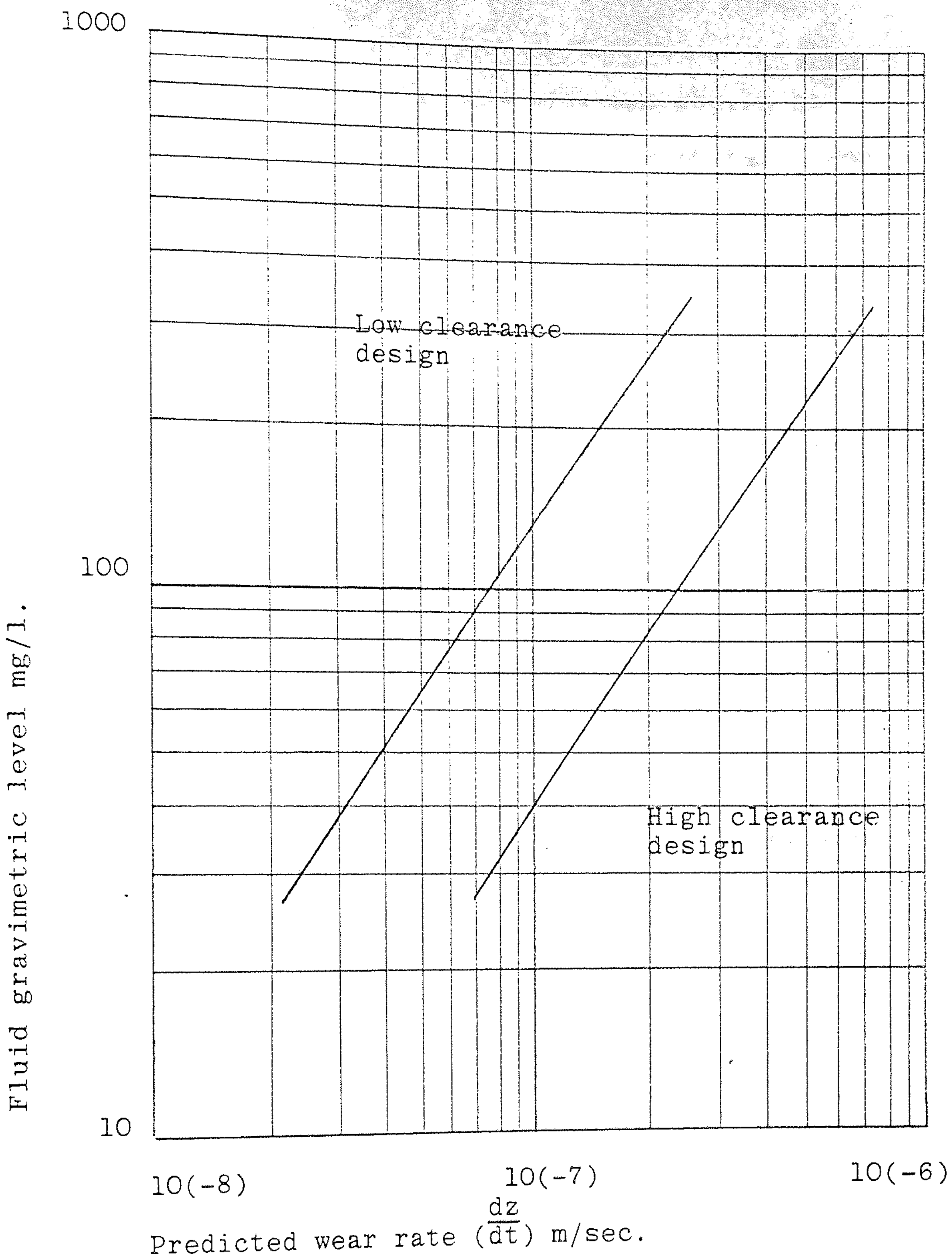


Figure 4.8.2 Predicted wear rates of high and low clearance slipper designs.

clearances would be expected to make matters worse rather than better.

It was not thought that the analysis should be continued at this stage. The work conducted had shown that this type of analysis is feasible, and it could be applied to other pumps and other pump areas. However, until the uncertainty about a suitable particle size distribution can be resolved, the results of such work must be open to question. Attention was therefore directed towards the problem of silting of fine clearances and the blocking of lubricant feed passages.

4.9 Silting of Fine Clearances

It may be assumed that an axial piston pump cannot operate satisfactorily unless moving components are separated by thin films of lubricant. It is known that the entrances to these clearances can silt up, as was shown in figure 4.5.1 (a). In a pump this could deprive an area of lubricant, or it could destroy the force balance on a component. A similar effect can occur in some types of valve, impairing valve operation.

Consider the slipper shown in figure 4.3.1. If the outer edge of the slipper pocket were to silt up, the slipper land would be deprived of lubricant, and this would produce an unbalanced load on the slipper. In the CHL design considered in section 4.6, this load would be 136N at a piston bore pressure of 400 psi, or 1020N at 3000 psi. If, as Kakoullis [4] says, the dominant mode of slipper lubrication is hydrodynamic, then the slipper could still operate satisfactorily once the pump is running. But it could not do so during and immediately after start-up. The experiments described in Chapters 6 and 7 suggest that silting is an important effect. Laurenson [60] also found that face seals silted up when stationary. The seals then closed up, with disastrous consequences when rotation started.

It might be thought that the rate at which a bearing silted up could be reduced by operating the bearing at higher clearances. It is true that the higher the operating clearance between two components, the less the proportion of contaminant which will be trapped at the clearance entrance. But operating at a higher clearance increases the total leakage flow, and hence the total amount of contaminant passing into the clearance. Increased leakage also represents a considerable efficiency loss, and is to be avoided wherever possible.

It is possible to define a mathematical sensitivity of a bearing to silting. If each particle of diameter x blocks a length x of the remaining bearing width w, then

$$\frac{dw}{dt} = -q \int_h^\infty x n(x) dx \qquad 4.9.1$$

where q represents the bearing leakage flow, h is the bearing clearance, and n(x)dx is the number concentration of particles of diameter x to x+dx per unit fluid volume.

This expression would obviously give an exponential reduction in the unblocked bearing width. The bearing's silting sensitivity could be defined as the half-life of w, and this could be maximised with respect to h for a given distribution of n(x).

This analysis was not continued, because of the uncertainty about a suitable form for n(x), the contaminant size distribution. Nau [75] made a similar observation in work into the silting of valves. Using the size distribution of NAS 1638 [23], he obtained the unexpected result that sensitivities to silting could be improved by reducing operating clearances.

This discussion, and Nau's work, merely reinforce the need for more work to determine actual contaminant size distributions, and to fit usable mathematical models to these distributions.

4.10 Blocking of Capillaries

The slipper bearing shown in figure 4.3.1 contains another area which might be affected by fluid contamination. This is the controlling capillary. The capillary affects three aspects of the slipper's operation: it partly determines the bearing clearance h (equation 4.3.1); it partly determines the leakage flow q (4.3.3); it also affects the bearing load characteristic, and particularly the bearing 'stiffness' dP/dh (from 4.6.8).

The usual slipper design approach is to reduce capillary diameters as far as possible. This helps to decrease slipper leakage, and also gives a stiff bearing (high dP/dh) which tends to be more stable in operation. However, reducing capillary diameters also increases the probability that the capillary will block up.

It is obvious that particles of a longest dimesnion larger than a capillary's diameter may obstruct the capillary flow. Once an obstruction has been formed, other particles may quickly form a 'log-jam' to completely block the capillary, and slipper seizure may follow. CHL's service engineer gave the author many examples of pump failures caused, apparently, by the obstruction of fine passages by PTFE tape, swarf, paint, rubber seal particles, etc. This type of occurrence cannot be designed out, but it should be prevented by care during maintenance and assembly. Sensitive equipment can also be protected by strainers.

Much greater conceptual difficulties are created by Laurenson's work [60], in which 500 micron diameter capillaries became completely blocked by roughly spherical, 34 micron diameter particles. Laurenson reported his results, but was unable to offer an explanation. Unfortunately, he conducted his experiments at only one, high particle concentration level (1200 mg/l), and under conditions which were unrepresentative of hydraulic system operation. An attempt was therefore made, in this work, to derive a theoretical explanation for the results so that they might be applied to hydraulic pumps.

A first approach was to try and calculate the velocity and radial position of a particle travelling down a capillary in a laminar-flow fluid. If larger particles travel faster than do small ones under these conditions, or if particles collect along the capillary walls, then these mechanisms might explain Laurenson's findings. However, the approach proved fruitless, the mathematics involved being beyond the scope of this work. A literature search produced only one relevant paper, by Segre and Silberberg [76]. They found, by analysis and experiment, that under steady, laminar conditions, macroscopic spheres (larger than 150 microns diameter), tended to travel down a capillary in an annular ring, the ring having a radius of 0.3 times the capillary diameter. However, the conditions under which they conducted their work were not representative of hydraulic pumps.

A second approach was to consider reducing the likelihood of capillary blockage by increasing capillary diameter. It can be shown, from equations 4.3.1 and 4.3.2, that, for a slipper bearing,

$$q = \frac{\pi P_s}{\mu K_c \left[1 + \dfrac{\ln(Ro/Ri)}{2\left\{ \dfrac{Ro^2 - Ri^2}{d_p^2} - 6 \ln(Ro/Ri) \right\}} \right]} \qquad 4.10.1$$

For a capillary,

$$K_c = \frac{128 L_c}{d_c^4} \qquad 4.10.2$$

where L_c is capillary length and d_c is capillary diameter. It can be seen that an increase in d_c will substantially increase leakage flow. This will bring more particles through the capillary, but at an increased velocity. It cannot, therefore, be assumed that increasing a capillary diameter will reduce the probability of that capillary becoming blocked.

It became clear, at this stage, that the probability of capillary blocking will be some function of n, d_c, L_c, μ, fluid temperature and capillary pressure differential. It might also be dependent on contaminant size distribution. It also became clear, that, without considerably more experimental data, a suitable relation could not be determined.

One final piece of work was conducted into the distribution of capillary failure times. The probability that a capillary will block in the time interval t to t+dt may be given the symbol f(t). The problem was to determine the form of f(t), and then relate it to fluid contamination levels.

It was assumed that a capillary would block if the number of particles (y) within a critical fluid volume v_c in the capillary exceeded some value n_c. For a constant average fluid contamination level n, the probability of this occurrence ($P(y>n_c)$) is given by

$$P(y>n_c) = \sum_{j=n_c+1}^{\infty} \frac{e^{-nv_c}(nv_c)^j}{j!} \qquad 4.10.3$$

If n_c is much greater than nv_c, this may be approximated by

$$P(y>n_c) = \frac{e^{-nv_c}(nv_c)^{n_c+1}}{(n_c+1)!} \qquad 4.10.4$$

It should be noted that this probability is not linear with n. Doubling the fluid contamination level will more than double the probability that a capillary will become blocked. This probability is also independent of time, implying that capillary failures will follow an exponential distribution such that

$$f(t) = \frac{1}{\theta} e^{-t/\theta} \qquad 4.10.5$$

θ will be related to $P(y>n_c)$, and possibly to q.

Unfortunately, Laurenson's results, reproduced in figure 4.10.1, do not support the exponential failure model suggested above. The data can be fitted to a normal distribution, with a mean failure time of 16.3 minutes, and a standard deviation of 9.9 minutes. A Pearson 'goodness of fit' test shows no evidence to reject such a distribution at a 90% level, although it

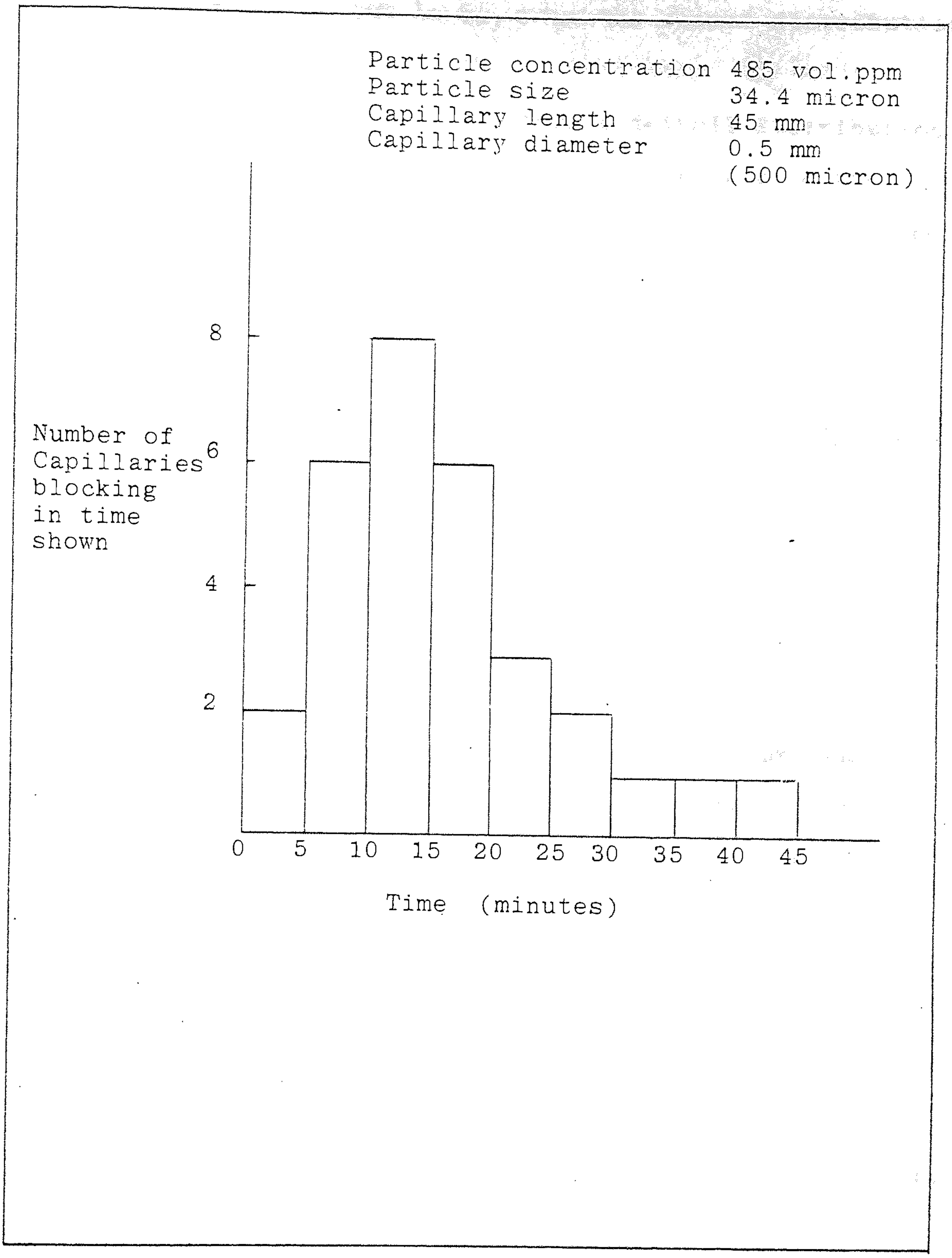


Figure 4.10.1 Ulster polytechnic data on blocking times for fine capillaries.

does suggest, at a 60% level, that the normal distribution may be inappropriate.

A better fit is obtained from a Weibull distribution, suggested by the pronounced 'skew' of the data to the left. Parameters for such a distribution were calculated, to give

$$f(t) = 0.0056t\, e^{-0.0028t^2} \qquad 4.10.6$$

A Pearson test showed no evidence, at a 60% level, that this distribution is inappropriate. The significance of the distribution, if it is correct, is that capillary failure rate z(t) is an increasing function of t such that

$$z(t) = 0.0056t \qquad 4.10.7$$

It is difficult to suggest why this should be so. It may be that particles collect along the capillary wall and so gradually reduce the capillary diameter. But Laurenson reported that capillaries operated at a constant flow rate until they suddenly blocked. It may be that experimental conditions took some time to stabilise, and that, once they did so, capillaries failed following an exponential distribution.

It was not possible to take the analysis further due to the lack of more research data. However, capillary blocking is an important effect, and more work in this area is needed. Although such work was outside the scope of this research, the analysis conducted does show the areas of insufficient knowledge. Future experiments should be conducted to:

(a) determine an appropriate failure distribution for capillary blocking;

(b) relate the parameters of this distribution to particle concentrations, and to capillary dimensions.

As a general point, the experiments should be conducted under conditions representative of those which obtain in an axial piston pump, and should use particles having a size range, rather than being of one diameter.

4.11 Conclusions on Theoretical Analysis of the Effects of Fluid Contamination.

The work described in this chapter certainly showed that theoretical calculations are a feasible way to investigate the effects of fluid contamination. The usefulness of the approach must be considered in relation to the three project aims given in section 3.7.

(i) Theoretical analysis has helped to increase CHL's general knowledge of fluid contamination and of its effects. The main outcomes of the work have been to illustrate the complexity of the subject, to identify some of the areas where current knowledge is lacking, and to develop theoretical techniques which could be applied to any piece of hydraulic equipment.

(ii) For a variety of reasons, theoretical calculations cannot yet be used to determine acceptable fluid contamination levels for CHL's pumps.

(iii) Theoretical analysis can be used to suggest design improvements to reduce the effects of fluid contamination on a pump. However, with present knowledge in this area, it cannot be guaranteed that these suggested modifications will improve matters. Any design changes introduced would have to be regarded as untested hypotheses.

The overall conclusion to be drawn from the work was that, whilst theoretical analysis was feasible and useful, it could not be used to satisfy all the research objectives. The next chapter will therefore consider alternative ways to pursue these objectives.

5.0 TESTING ON CONTAMINATED FLUIDS

5.1 Introduction

Let us summarise what has been presented so far. This research was conducted in pursuit of three objectives which were:

(a) to improve CHL's general knowledge and understanding of fluid contamination and its effects;

(b) to find ways of determining 'acceptable' fluid contamination levels for axial piston pumps;

(c) to examine ways of improving the ability of axial piston pumps to operate on contaminated fluids.

It was argued, in chapter three, that these objectives could be pursued in three ways:

(a) by theoretical analysis during pump design;

(b) through some form of test work;

(c) by recording and analysing data on field performance.

The work presented in chapter four showed that theoretical analysis can help to improve general understanding of fluid contamination and of its effects. It is especially valuable in identifying those areas where current knowledge is incomplete. However, the results of such analysis can only be, for the present, a set of untested hypotheses. It would be a brave or a foolish designer who would test these hypotheses in a new pump to be released for immediate sale. The conclusion to be reached is that theoretical analysis cannot yet be used with confidence to determine an acceptable fluid contamination level for a given pump, or to suggest successful design modifications to lessen pump contamination sensitivity.

Having reached this conclusion, the research could have proceeded in one of two ways.

(a) Theoretical knowledge could have been increased by testing some of the hypotheses suggested in chapter four.

(b) The research objectives could have been pursued through one or both of the remaining strategies of testwork or the analysis of service performance.

Referring to the pump design and development model of figure 3.8.1, , information from testwork is usually both easier to obtain and cheaper to act upon than is data from the field. This being the case, both possible approaches to subsequent research led to the same conclusion; some form of contamination testing would have to be conducted. Before deciding on the form of this experimental work, attention was given to the possible objectives of a contamination test, and to which of these objectives could be pursued with the prospect of obtaining results of value to CHL.

5.2 The Objectives of Industrial Tests

Industrial testing is somewhat broader in scope than 'scientific' testing. It was decided, after careful consideration, that an industrial test will be conducted in pursuit of one or more of five objectives. These are:

(a) to verify or refute a scientific hypothesis;

(b) to measure the performance of an industrial unit, component or system;

(c) to estimate equipment 'life' or reliability;

(d) to identify failure modes and 'weak' components or areas in equipment;

(e) to give a quantitative 'order-of-merit' rating to some aspect of a product.

It is important to identify the objectives of a specific test, because these will partly determine both the conduct of the test and the confidence which may be placed in the test results. Each possible test objective was therefore considered in relation to contamination testing.

5.3 Testing Scientific Hypotheses

The theoretical work presented in chapter four suggested several hypotheses which could have been tested in the 'classic' scientific manner. For example, the work described in section 4.6. led to the conclusion that the linear wear rate of a slipper bearing operating on contaminated fluid will be proportional to some power of the number particle concentration. Subsequent project work could have been directed to proving or refuting this hypothesis.

A major consideration with this type of work is whether it should be conducted on actual equipment or on specially-built, small-scale test rigs. Peterson [77] considers this point and concludes that testing on small, purpose-built rigs has four advantages over testing on the complete plant or unit.

(a) Variables may be more precisely-controlled and more easily varied in a small rig.

(b) Results may be cheaper to obtain.

(c) Results may be obtained more rapidly.

(d) Changes in materials or design may be introduced more easily.

Against these advantages must be set the problem of relating the results of such work to the conditions which exist in the actual equipment. This problem is particularly acute when dealing with research into lubrication and wear (tribology). Several authors stress the likely consequences of a change in any variable in a wear process. To quote Eyre [78]: "Friction and wear are not intrinsic material properties, but are characteristics of the engineering system. Any change in load, speed or environmental conditions may cause catastrophic changes in the wear rate of one or both of the surfaces in contact. <u>Great care must be exercised in applying general solutions to specific problems</u>." (Author's emphasis). Peterson [op cit] lists twelve variables in any wear process. With this number of parameters to consider, it becomes clear that, for work in this field to be directly relevant to industry, it must either be conducted on complete pumps or on very carefully designed test rigs, each one specific to one pump area.

CHL accepted that tests conducted to verify a scientific hypothesis would be better conducted on a specially built test rig simulating one pump area, such as the slipper bearing. This led the company to two observations. The first was that work of this type is more suited to a university, 'pure-research' project, and is somewhat outside CHL's province. The second was that, at this stage in the company's knowledge of fluid contamination, work of this type risked being irrelevant to the company's needs. There was a very real danager that a specialised, and very expensive rig might be constructed to investigate an effect which did not occur in equipment in service.

This is not to say that CHL cannot or should not contribute to this type of scientific research. CHL should actively suggest those areas and phenomena which merit further study of this type, and should give guidance as to the conditions under which experimental work will give the most useful results. One way to identify these areas and conditions is by the type of theoretical analysis presented in chapter four. A second way is by pump contamination testing, but the prime objective of such testing must be to identify failure modes and areas, and not to test scientific hypotheses.

5.4 Testing to Measure Performance

Industrial equipment is often tested to determine its technical performance. However, there is an assumption, within the hydraulics industry, that fluid contamination represents a reliability problem, rather than a performance problem. The strict validity of this assumption is considered in chapter six, but it was accepted at this stage in the project. No further consideration was therefore given to the possibility of determining pump performance on contaminated fluids, as an objective in itself. A series of such tests might, however, be used to estimate the effects of fluid contamination on pump reliability, as an indication of failure modes, or to identify pump components affected by fluid contamination.

5.5 Reliability Testing on Contaminated Fluids

It is a central tenet of this thesis that the reliability of a hydraulic pump is affected in some way by the contamination level of the fluid which it is pumping. So far, no formal definition of reliability has been given. This omission will now be rectified, but first, a brief explanation of probability density functions is required.

If the service lives of a large number of pumps were recorded, it would be found that individual pump failure times varied over a large range. Some pumps would fail shortly after being placed in service, whereas others would give many hours of trouble-free running. It would also be found that pumps were more likely to fail at some times than at others. Failures might, for example, be fairly frequent around 5000 hours of operation, with less frequent failures at times less than, or in excess of this value.

It is possible to formalise these concepts through the use of a failure probability density function, $f(t)$, for pump failures. The probability that a given pump will fail in the short interval of time t to $t+dt$ is given by $f(t)dt$. Figure 5.5.1 shows several probability density functions based on the Weibull distribution which was mentioned in section 4.10.

It is now possible to define reliability as a mathematical expression. The reliability of a piece of equipment at time t is the probability that the equipment will still be operating satisfactorily after time t. Given the symbol $R(t)$, it can be seen that

$$R(t) = \int_t^\infty f(t)dt \qquad 5.5.1$$

To talk of the 'life' of a piece of equipment is misleading. It is more correct to talk of the reliability of that equipment at a certain life. For example, a customer might require 90% reliability at 1800 hours. In practice, reliability can only be estimated, so a full reliability specification must contain a confidence level on the estimate. The customer might therefore need to

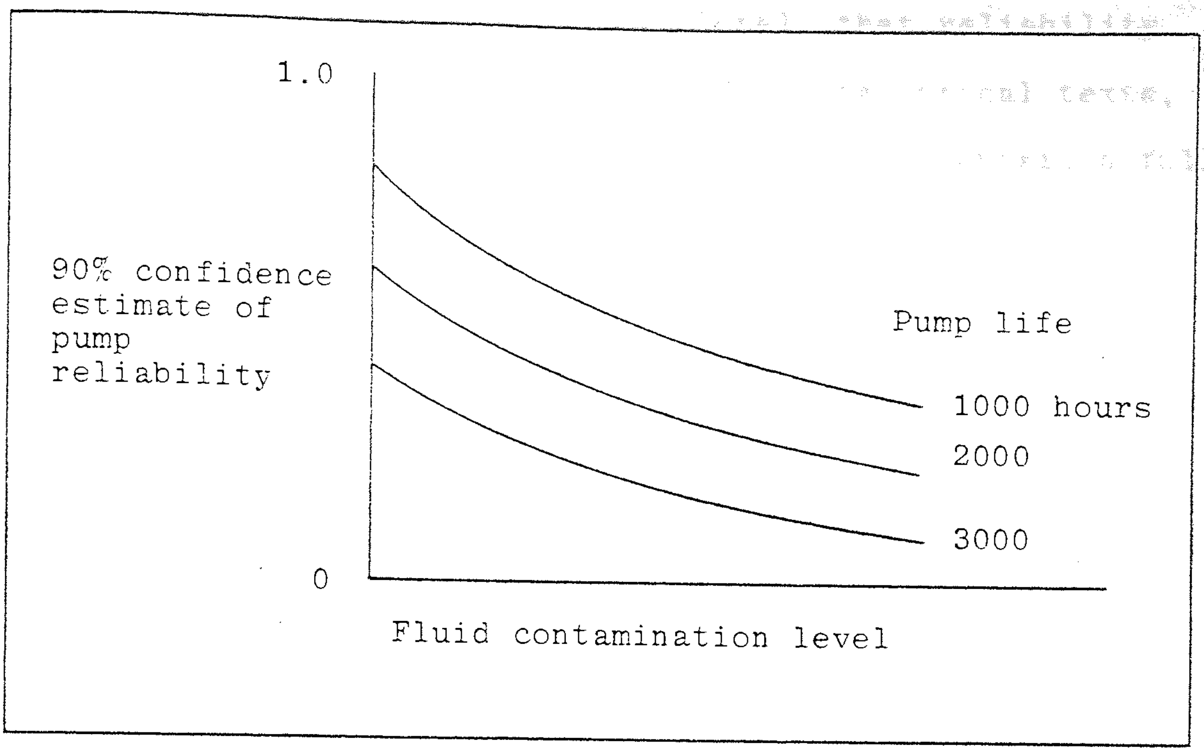


Figure 5.5.2 Possible variation of pump reliability with fluid contamination level

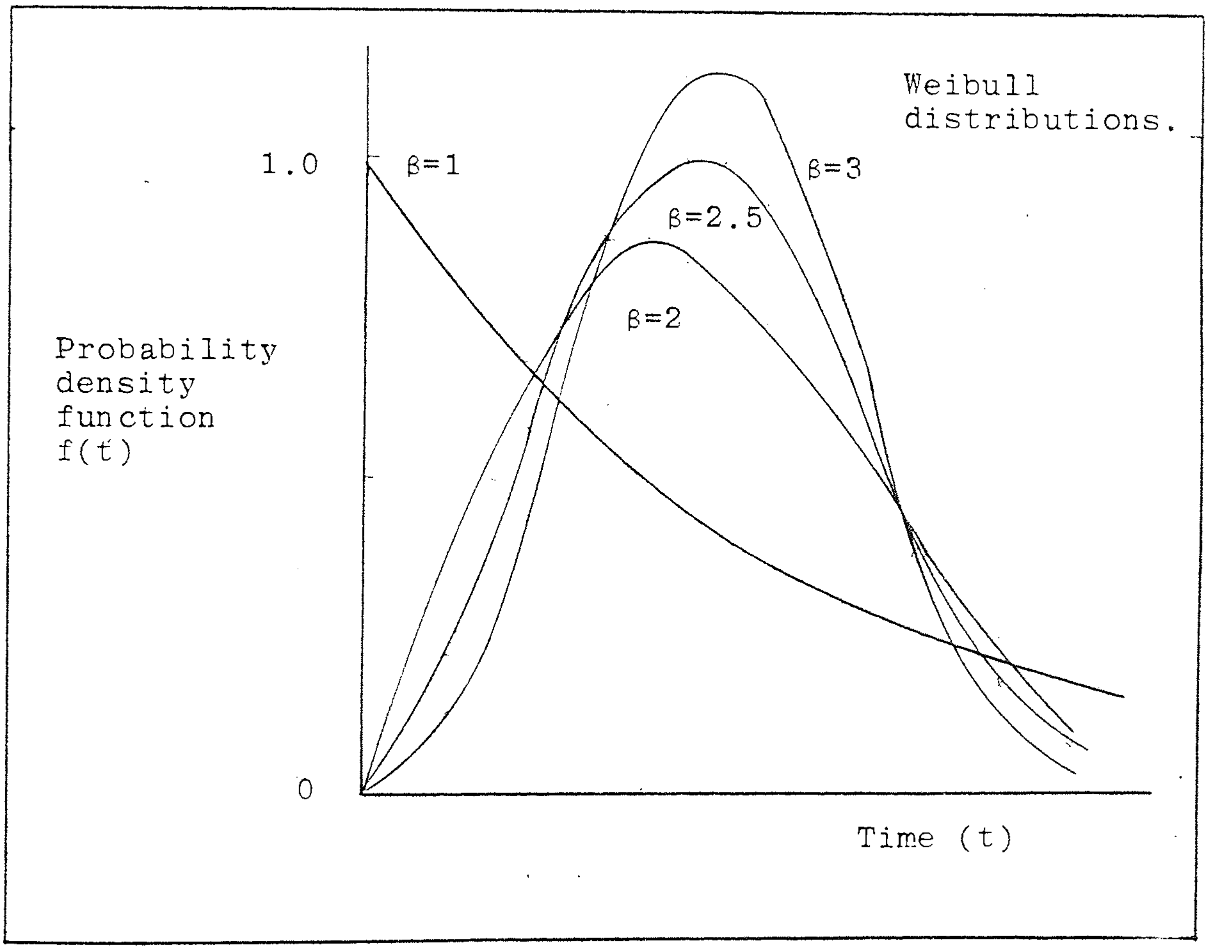


Figure 5.5.1 Pump failure probability density functions.

be certain, at a 75% confidence level, that reliability will exceed 90% at 1800 hours. (Many statistical texts, such as Meyer [79], or Lloyd and Lipow [80] contain a fuller discussion of reliability and of confidence levels.)

It would have been an excellent outcome to this project if graphs of the form shown in figure 5.5.2 could have been produced, relating pump reliability to fluid contamination. Such data could be used as a major step towards achieving the 'ideal solution' to fluid contamination outlined in section 3.6. and illustrated in figure 3.6.1. A graph of this form could be used to determine an acceptable fluid contamination level for a given pump. An aggressive company could use favourable data of this form to boost sales. Even a defensive company could use such data to indicate where reliability was unsatisfactory.

To pursue this approach, attention was given to the problems which would have to be overcome in estimating, by pump tests, the reliability of pumps operating on contaminated fluids.

The standard method of obtaining a reliability estimate has four stages. These are illustrated in figure 5.5.3 and summarised below.

(a) Determine the shape of the probability density function for the unit under test.

(b) Estimate the parameters which quantify the probability density function.

(c) Use the estimated probability density function to predict product reliability at the lives of interest, and place confidence intervals on the predictions obtained.

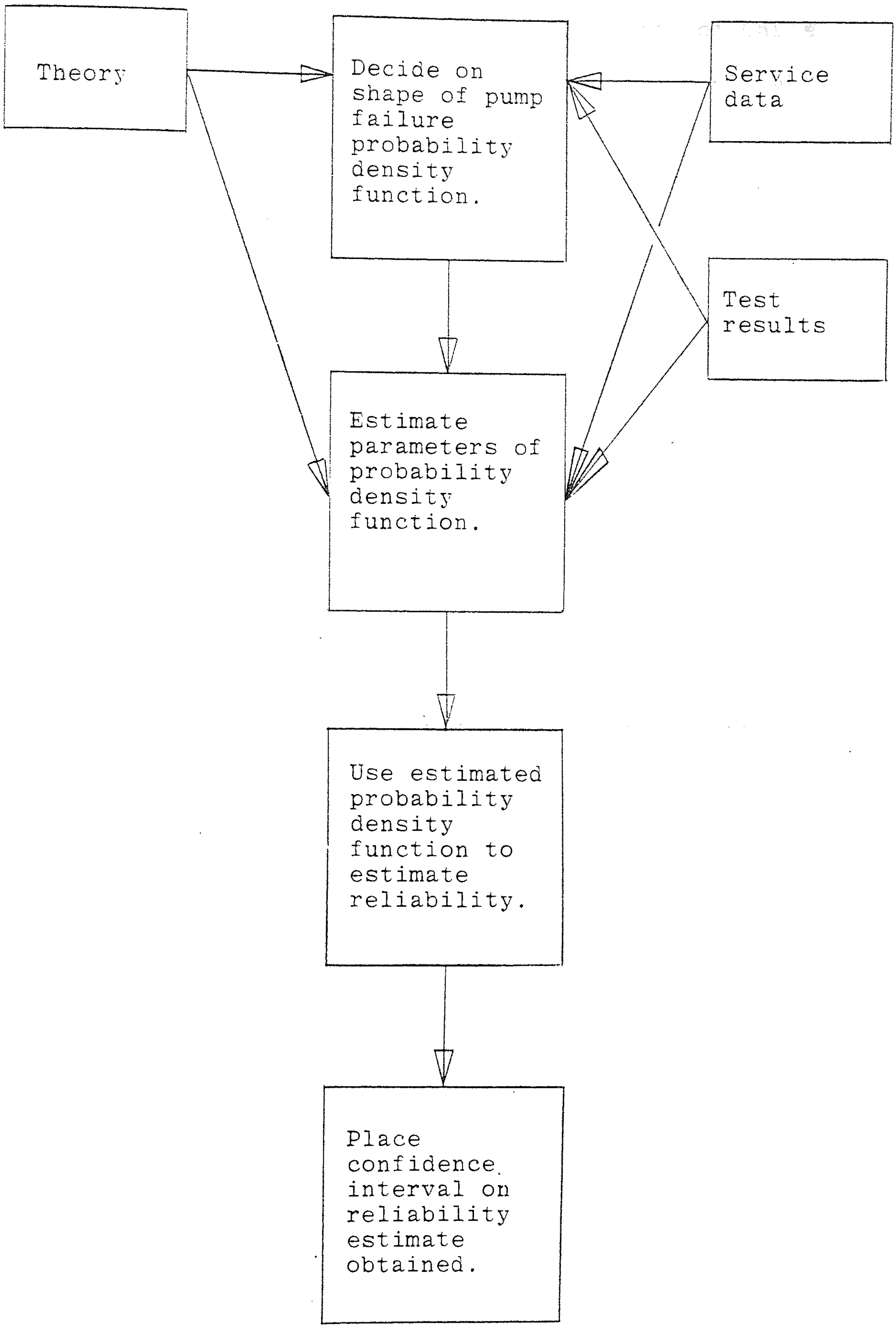


Figure 5.5.3. Reliability estimation.

The first problem encountered in an attempt to apply this strategy to estimating the reliability of CHL's pumps was that no data were available to suggest an appropriate model for the probability density function of pump lives. Such a model might be developed from theoretical considerations. But a similar procedure was tried in section 4.10, in relation to the blocking of capillaries by contaminants. An exponential failure distribution was suggested, but was found to be at variance with Laurenson's experimental results [60]. This approach does not therefore seem feasible at present.

Another possibility would be to derive the probability density function from a series of pump tests. However, it was shown in section 4.10 that, even with data from tests on thirty capillaries, no firm decision could be reached as to whether the underlying probability density function for capillary failures was a Weibull or a Gaussian distribution. It would not be feasible for CHL to test thirty (or more) pumps, and this approach can again be ruled out.

The only other possible approach would be to use data from service to suggest a suitable probability density function for pump failures. This is not possible at present, because CHL do not operate a suitable data recording system. However, it represents the only practical strategy if a probability density function for pump lives is to be obtained. The problem is therefore considered in more detail in chapter 8.

As far as this project was concerned, the lack of a model for pump failure lives effectively ruled out the possibility of a viable test to predict pump reliability.

However, the analysis was taken a little further. This was done to see if it would be worth trying to derive a pump failure distribution at some time in the future.

To continue the analysis, an exponential failure distribution for pump lives was assumed, partly because this distribution is appropriate to failures which occur due to a random event (such as an orifice blockage in a pump operating on contaminated fluids), but mainly because the exponential distribution is defined by a single parameter. This generally makes data derived from the exponential distribution less uncertain than data derived from other distributions, because only one parameter has to be estimated. The exponential distribution therefore represents the 'best case' which can be expected in this type of work.

The probability density function of the exponential distribution is given by

$$f(t) = \frac{1}{\theta} e^{-t/\theta} \qquad 5.5.2$$

θ is a parameter, which can be shown to be the mean life of the equipment concerned, and can be calculated given a reliability requirement at a certain life.

For the customer, an 'adequate pump reliability' must depend on the application. For example, adequate reliability for a pump used on a steel forge manipulator would probably be unacceptable if the same pump were used in the steering gear of a nuclear submarine. CHL's minimum reliability requirement should be that equipment warranty claims do not become excessive. CHL, like most pump manufacturers, give a one year warranty on their products. The company's sales department advised the author that it is standard practice to allow 2% of sales revenue to cover the cost of warranty claims. The company's service department estimated that a typical repair might cost 25% of the cost of a new pump. On this basis, 8% of pumps sold could be returned under warranty without costs exceeding budgetted levels. This is equivalent to a company reliability requirement of 92% at one year. For a pump operating 8 hours a day, 45 working weeks a year, this would imply 92% reliability at 1800 hours. (This requirement is not excessive. Articles in 'Machine Design' magazine [81] give overhaul times of 5000 - 19000 hours for pumps in aircraft hydraulic systems. The same articles suggest that mobile machinery should give 1500 to 9000 hours of "trouble-free" service.)

From equations 5.5.2 and 5.5.1, CHL's minimum reliability requirement would be achieved if θ exceeded 21600 hours. Remembering that θ represents, on the exponential model, mean pump failure time, θ could be estimated by running a number of pumps (say n) and recording their failure times, T_1, T_2, T_3 ..... T_n. θ would then be estimated as $\hat{\theta}$, where

$$\hat{\theta} = \frac{1}{n} \sum_{i=1}^{n} T_i \qquad 5.5.3$$

It is necessary to place confidence levels on θ. With data from a single pump test, CHL could only be confident, at a 90% level, that the true value of θ lay between $19.5\hat{\theta}$ and $0.3\hat{\theta}$. With data from two pumps, this confidence interval would reduce to $5.6\hat{\theta}$ and $0.43\hat{\theta}$. The interval obviously becomes narrower the greater the number of pumps tested, although the upper limit can never be lower than θ. But, as an example, with data from two tests, CHL would have to achieve a mean failure time in excess of 50,000 hours to be 90% confident that the original reliability requirement had been met. This would involve continuous testing for six years, which is obviously impractical, certainly so in terms of this project.

This type of analysis merely reinforced the original conclusion, that tests, conducted under realistic conditions, to place quantitative estimates on the reliability of a pump operating on contaminated fluids are impractical, due to both the excessive test times involved and the numbers of pumps which would have to be tested. It will also be shown, in chapter six, that it is impossible to devise a 'realistic' contamination test.

5.6 Accelerated Reliability Tests

Although tests at 'realistic' contamination levels had been shown to be impractical, it was thought that an 'accelerated' test might still be feasible. In such a test, the problem of excessive test length could be overcome, either by operating the test at a high contamination level, or by using an especially damaging test contaminant. OSU use both these tactics together in their pump test [72].

To implement this approach, the probability density function for pump failures would have to be determined at the accelerated conditions. This would require the testing of a large number of pumps (probably in excess of thirty). The results obtained would then need to be related to those which would be expected at more moderate conditions. For this to be feasible at all, tests conducted at the accelerated conditions would have to produce effects which differed only in degree, but not in type, from those which would be expected in service. If this could be shown to be so, then some quantitative relation would be needed between effects produced at different contamination conditions. Such a relation might be obtained in three ways.

(a) A suitable relation could be derived by theoretical analysis. This approach was attempted in chapter four, where it was shown that it is not yet a feasible proposition.

(b) Tests could be conducted at accelerated conditions and at more realistic contamination levels. But this would involve the same difficulties as were considered in section 5.5, so it is not a practical solution.

(c) Tests could be conducted at two or more high contamination levels, and the results could be extrapolated to lower levels. This approach would rest entirely on an assumption that the extrapolation was valid. It would also require an enormous amount of testwork if any confidence was to be placed in the results.

To illustrate some of the problems which might be encountered, a Monte Carlo analysis was conducted to simulate pump lives on a Weibull failure model. This type of analysis relies on the fact that a cumulative probability density function has a constant probability distribution. The cumulative probability density function, F(t), is related to the probability density function f(t) by

$$F(t) = \int_0^t f(t)dt \qquad 5.6.1$$

The Monte Carlo analysis therefore uses the fact that all values of F(t) are equally likely. The procedure is to generate a random number between 0 and 1, equate this to the same value of F(t), and read off a value of pump life. The values of pump life obtained will then come from the underlying distribution f(t)

This procedure was followed for two Weibull probability density functions where

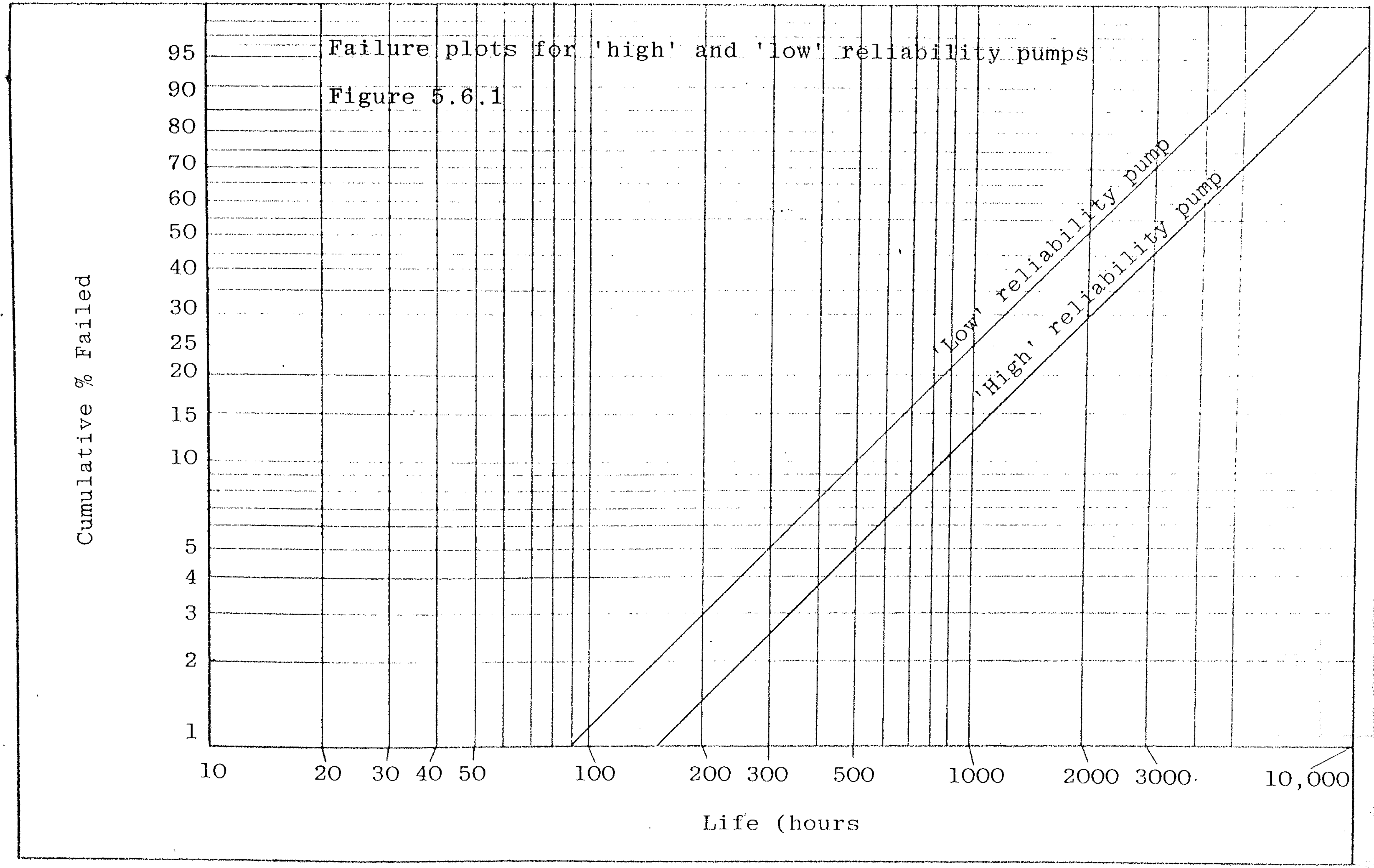
$$f(t) = \alpha\beta t^{\beta-1}e^{-\alpha t^{\beta}} \qquad 5.6.2$$

β was selected as 2 to correspond to the experimental data obtained by Laurenson [60] and described in section 4.10. The values of α were selected so as to model a high reliability pump operating on 'clean' fluid, and a lower reliability pump operating on contaminated fluids. The two distributions are shown (as cumulative failure distributions) in figure 5.6.1 and calculated pump lives are shown in table 5.6.1.

Table 5.6.1 illustrates two points. The first is the wide variation between individual pump lives. The second is that, even though one pump has a higher overall reliability than the other, some pumps could still last longer on the contaminated fluids than would some operating on clean fluid. There would be a danger, in a comparison test, of obtaining a completely misleading result.

The analysis was taken further, by using the values from table 5.6.1 to estimate the true underlying Weibull failure distribution. The results will not be presented here, but they showed that, using some sets of data agreement was quite close, whilst, with other sets, the estimated failure distribution was markedly different to the actual distribution.

The simulation exercise, and the discussion presented earlier in the section, led to the conclusion



Failure plots for 'high' and 'low' reliability pumps

Figure 5.6.1

PUMP REFERENCE NUMBERS	PUMP LIVES (HOURS)	
	'HIGH RELIABILITY' PUMP	'LOW RELIABILITY' PUMP
1	1638	2773 *
2	1260	941
3	1022	1282 *
4	3785	2043
5	2178	1310
6	2322	656
7	1020	895
8	1255	982
9	1223	1037
10	2272	1234
11	2950	1397
12	1078	1020
13	3726	1514
14	1954	2136 *
15	1092	634
16	4198	1823
17	2524	875
18	2074	2321 *
19	1826	1418
20	3125	439

* Case where a comparison test would give an incorrect prediction.

Table 5.6.1 Pump failure times generated from 'high' and 'low' reliability probability density functions (see Fig. 5.6.1).

that pump contamination tests at accelerated conditions are not a practical way to estimate pump reliability.

5.7 Qualitative Tests to Identify Problem Areas in Pumps.

Of the five objectives of industrial testing given in section 5.2, three have been rejected as being either inappropriate to CHL's needs and resources, or impractical to pursue with any chance of success. However, the remaining two objectives are feasible. The first of these is to use pump tests to identify failure modes in a pump, and to identify those areas of a pump most susceptible to the effects of fluid contamination. Such knowledge would be very useful. As mentioned in section 5.3, it could be used to direct the aim of more basic research by identifying areas where results would be of most use. It could also be used to suggest design or materials changes in a pump to render the pump less sensitive to the effects of fluid contamination. The aim would be to steadily improve a pump's resistance to the effects of fluid contamination. This leads to to the second practical objective of a pump contamination test.

5.8 Order-of-Merit Contamination Rating

Having argued that it is not possible to place an accurate estimate on the reliability of a pump operating on contaminated fluids, it would still be of value to know that one pump is better able to withstand fluid contamination than another. This type of 'semi-quantitative' rating could be used to compare competing pump designs, and to assess the likely benefit of any design or materials changes introduced to lessen fluid contamination effects. It must be stated that the usefulness of such a test depends on a convincing demonstration that a pump which performs well in the test will also perform well in the field. The problems involved in such a demonstration are discussed in chapter eight.

5.9 Conclusions

It was shown, in chapter four, that the three overall objectives of this research could not have been pursued through theoretical analysis alone, although such analysis can give useful results. This finding led to the conclusion that the research should proceed through some form of testwork, conducted either to extend current theoretical knowledge in this field, or to follow the overall research aims more directly.

In this chapter, it was argued that a contamination test would have one of five aims. A consideration of each possible aim in turn showed the following.

(a) Tests conducted primarily to verify or refute a scientific hypothesis are really outside the scope of CHL's activities. The results of such tests could also have been irrelevant at this stage, as the company did not yet have enough knowledge of fluid contamination to direct its resources onto the most important problems. (This does not mean that test results would not be used to support or reject a scientific theory, only that this should not be the prime aim of the work.)

(b) The performance of a pump operating on contaminated fluids is of little inherent interest, although a reduction in performance might be used as an indication of damage induced by contamination, or of impaired reliability. However, these two uses represent different overall test aims, and they are considered in (d) and (c) respectively.

(c) Pump contamination tests cannot be used to relate pump reliability to fluid contamination levels. For the results to be of value, large numbers of pumps would have to be tested for long periods. The numbers of pumps required and the test times involved were shown to be impractical. Accelerated tests are not a solution to the problem.

(d) Pump contamination tests could possibly be used to identify failure modes and pump areas affected by contaminants.

(e) An 'order-of-merit' pump contamination rating might also be feasible.

It was decided, on the basis of these conclusions, that a series of pump contamination tests would be conducted. Chapter six therefore describes work on a general pump contamination test, whereas chapter seven considers the problems involved in contamination testing a unit for a specific application. It is well to relate this work to the original research objectives summarised in section 5.1. Pump tests were conducted to:

(a) Improve general knowledge of fluid contamination and of its effects;

(b) Develop methods of identifying failure modes due to fluid contamination, and of identifying pump components affected by fluid contamination. This knowledge might be used to improve the ability of axial piston pumps to operate on contaminated fluids.

The pump tests were not conducted to determine acceptable fluid contamination levels for a piston pump design. Having shown that this cannot be accomplished by theoretical analysis, or by test-work, the only remaining approach is to use service data. The ways in which this may be done are considered in chapter 8.

6.0 A GENERAL PUMP CONTAMINATION TEST

6.1 Introduction

It was shown in chapter 5, that it is not practical to use pump teststo relate pump reliability to fluid contamination levels. The argument put forward was that very little confidence can be placed in the accuracy of a reliability estimate obtained in this way unless a large number of pumps are tested for a very long time. The times involved were shown to be excessive, and the number of pumps required for a complete test programme would make this approach prohibitively expensive.

It will also be shown, later in this chapter, that it is not possible to devise a test which will model, in a laboratory, the contamination exposure to which a pump will be subjected in service. It follows that any pump contamination test must be an essentially unrealistic process. As this type of test cannot model the contamination conditions which a pump will experience in the field, it is futile to use contamination tests to try and determine whether or not a pump will be affected by fluid contamination in service. Any conclusions reached on this basis must be misleading.

This is not to say that pump tests have no place in a programme of research into the effects of fluid contamination. However, the aim of such tests must be to <u>deliberately stimulate</u>, in the test pump, the types of effect (wear, silting, loss of lubricant, etc.) which are expected in service.

The next stages in this research were therefore:

(a) to determine the best way to conduct a pump contamination test, given the usual constraints of 'reasonable' resources;

(b) to determine what output data can be obtained from such a test;

(c) to determine what useful deductions can be made from the results of a pump contamination test programme.

It was envisaged that if suitable data could be obtained they would be used for one or more of three purposes.

(a) Test results would be used to assign a simple 'order-of-merit' rating to the ability of a pump to operate on contaminated fluids.

(b) A suitable test could be used to identify those areas of a pump which are susceptible to the effects of fluid contamination. This information could be used either to determine the priorities of future, more basic research, or to suggest pump design and material changes to reduce contamination - related damage in a given pump.

(c) The test results could be used to improve the general understanding of the behaviour and effects of fluid contamination.

This chapter proceeds with a discussion of the main considerations involved in setting up a pump contamination test. This leads to the conclusion that the best approach to contamination testing is that adopted by OSU for their pump test. However, it is shown that the data obtained during this test are of very limited value to a pump manufacturer, as they give no information as to what has been happening within the pump during the test. To see if more useful data could be obtained from tests of the OSU format, a series of such tests was conducted on small axial piston pumps. The tests are described, and the results are presented and discussed. The chapter concludes with a discussion of the usefulness of pump contamination tests in the light of the experience obtained.

6.2 Testing Pumps on Contaminated Fluids

6.2.1 The basis of a pump contamination test.

The basis of a pump contamination test is simply to operate a pump at specified conditions on fluid containing a known level and type of contaminant, and to determine what happens to the pump. Within this broad framework, decisions must be taken about the test contaminant, about its method of introduction, about the test length and operating conditions, and about the test measurements taken. These factors will be considered individually.

6.2.2 The introduction of test contaminants to the test system.

A pump contamination test may be run as a "multipass" or a "singlepass" process (OSU terminology). In a multipass test, be it on a pump or a filter, a batch of contaminant is injected into an otherwise unfiltered test circuit, and is then allowed to circulate continuously for the period of the test. In a singlepass test, contaminant is injected at a steady rate upstream of the test unit and is removed, as far as possible, downstream. The two approaches are illustrated in figure 6.2.1.

The main potential advantage of the singlepass

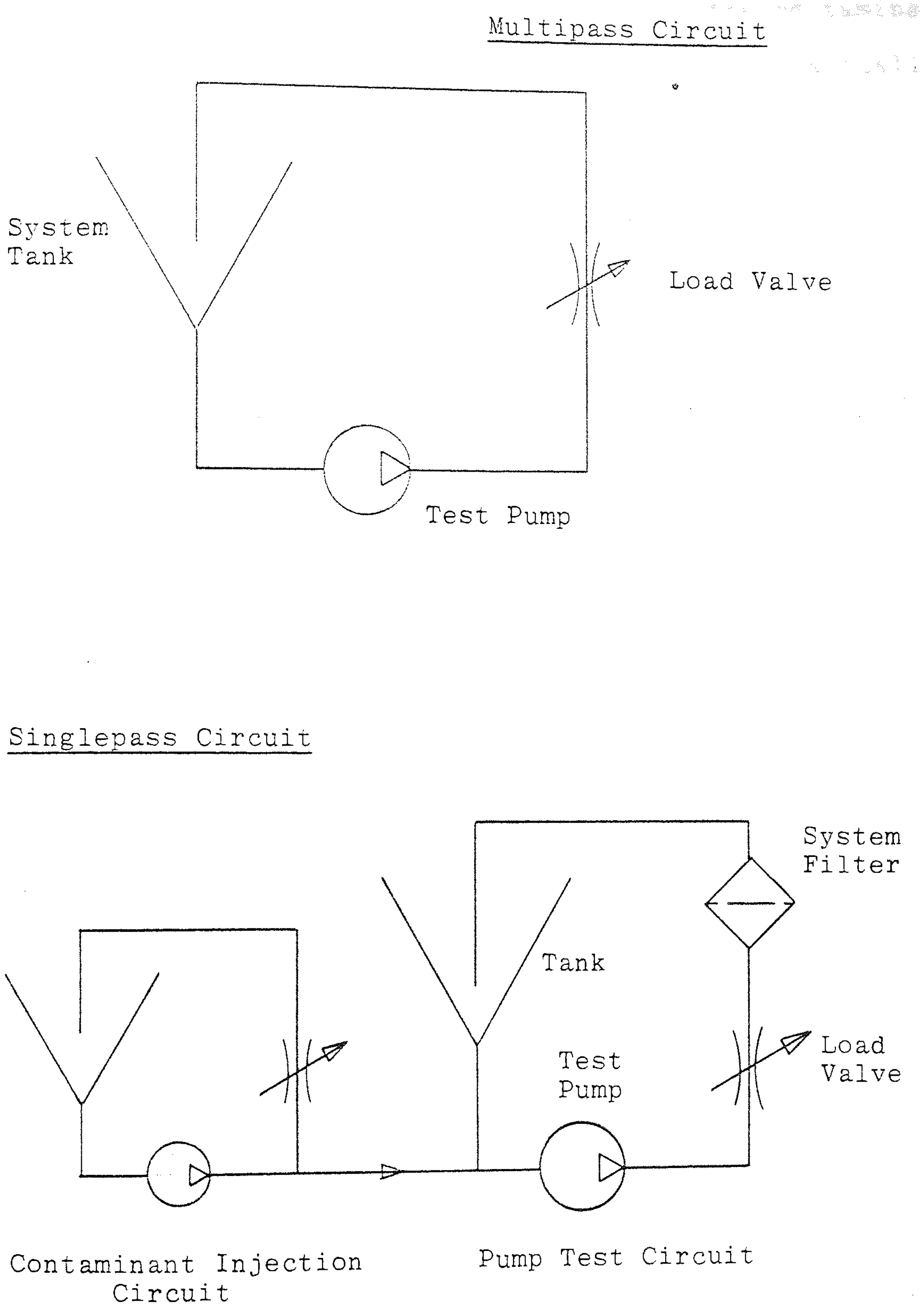


Figure 6.2.1

BASIC MULTIPASS AND SINGLEPASS PUMP TEST CIRCUITS approach is that, in an ideal test, one could be certain about the level, size and properties of the contaminant entering the test pump at any time. However, actually realising this advantage in practice is both difficult and expensive. As can be seen from figure 6.2.1, the singlepass test rig is more complex, and hence more costly, than the multipass rig. It is also expensive both to add and to remove the amounts of test contaminant required in a singlepass test. For example, to run just a ten hour test, at a contamination level of 30 mg/l, on a pump delivering 30 gpm would require the injection and removal of 2.5 kgs of contaminant. If this contaminant had been prepared into a special size range it could cost up to £1000. The author showed, in work at OSU [82], that the maximum dirt-holding capacity of a high-efficiency filter ($\beta 10 > 100$[1]) as determined in an OSU multipass filter test [83], is no better than 50 gms. In another paper [84], the author also showed that the standard multipass filter test gives an overestimate of filter dirt capacity. The test example given above would therefore require the use of at least fifty filter elements, at a cost probably in excess of £2500. Filter elements would also have to be changed every twelve minutes, which would be very difficult to accomplish. Even with very fine filters, there will be an inevitable build-up of small particles (of a size below the filter pore size) in a singlepass test.

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[1] See glossary.

Practical considerations therefore make it almost inevitable that any pump contamination test should follow a multipass procedure. Wolf [85] used a singlepass test in early work (1965) at OSU into the effects of fluid contamination on gear pumps. Significantly, OSU abandoned this approach in 1969 and have used multipass tests in almost all subsequent work.

The multipass test is not without its problems, and has two main difficulties.

The first is that, during such a test, particles worn from pump components may re-enter the pump to produce further damage. But this could well happen in service. The process would only be unrealistic if very large particles (above 25 microns diameter) were re-entering the pump. If such particles were being produced it is reasonable to assume, although the assumption requires verification, that the pump would be close to failure anyway.

The second complicating factor with a multipass test concerns the effects of the test pump on the fluid contamination itself. Bensch [86] found that, as a multipass pump test proceeds there is a steady reduction in the mean size of the fluid contamination present. He took this to indicate that particles are gradually broken down as they pass through the test pump. Bensch was studying abrasive wear effects and also concluded that there must be a gradual reduction during a test in the abrasive wear properties of the particles introduced at the start of the test. Rabinovicz [58] reports similar findings from more basic research into abrasive wear phenomena.

Bensch's overall conclusion was that the effective concentration of abrasive wear particles at time t, $n_a(t)$ reduces during a multipass pump contamination test, following a relation of the form

$$n_a(t) = n_a(o)e^{-t/\tau} \qquad 6.2.2.1$$

where τ is a "particle destruction time constant" whose precise value depends on parameters of the test system. Bensch calculated a value of 9 minutes for τ in the OSU pump test.

Bensch's findings imply that any multipass pump contamination test conducted to stimulate abrasive wear will last only a few hours. No additional abrasive wear damage will be produced over longer periods. Bensch's conclusion also suggests that a multipass pump test, even though it represents the best practical approach to contamination testing, cannot be used as an exact simulation of service contamination conditions. This can be seen by reference to work described in chapter 3.

In chapter 3 the variation with time of a system fluid contamination level was given as

$$A(t) = \frac{R}{Q_f\eta}\left(1-e^{-Q_f\eta t/v}\right) + A(o)e^{-Q_f\eta t/v} \qquad 6.2.2.2.$$

If $V/Q_f\eta$ is regarded as a time constant for the removal of contamination from the system, and given the symbol τ', 6.2.2.2 may be rewritten as

$$A(t) = \frac{R\tau'}{V}\left(1-e^{-t/\tau'}\right) + A(o)e^{-t/\tau'} \qquad 6.2.2.3$$

Bensch effectively showed that not all of the particles present in a system are capable of producing abrasive wear. It can be shown that, where abrasive wear is of most concern, the concentration, $A_a(t)$ of potentially damaging particles is less than $A(t)$, and is given by

$$A_a(t) = \frac{R\tau''}{V}\left(1-e^{-t/\tau''}\right) + A_o^{-} e^{-t/\tau''} \qquad 6.2.2.4$$

where $$\frac{1}{\tau''} = \frac{1}{\tau} + \frac{1}{\tau'} \qquad 6.2.2.5$$

To summarise, equation 6.2.2.1 represents the concentration of effective, abrasive wear particles which will be present at time t in a multipass pump contamination test. Equation 6.2.2.4 represents the concentration of such particles expected in service. It can be seen that even if the parameters involved in equation 6.2.2.4 could be determined, which would be extraordinarily difficult, equation 6.2.2.1 could not be used as a model of service conditions.

A further implication of Bensch's results, which he did not consider, is that the level of contamination present in a system is not the sole factor determining the severity of any abrasive wear effects produced. Of equal importance is the manner in which that level was achieved. These points and the validity of Bensch's conclusions, will be discussed later, in relation to the results of pump tests.

6.2.3. Test contaminants

The discussion of section 6.2.2. leads to the conclusion that a practical pump contamination test must follow a multipass procedure. This means that such a test will be relatively short in duration, and that it cannot be truly realistic. The aim of a pump contamination test must therefore be to induce, in a pump, the types of effect which are experienced in service. These effects can only be induced, in a short time, by a suitable choice of contaminant size, types, and concentrations.

One way to conduct a pump contamination test would be to run the test on supplies of contaminated fluids drained from hydraulic systems in service. This approach is not recommended for five reasons.

(a) It is not possible, when using supplies of old fluid, to specify the contamination level at which a test is to be run. This level is predetermined by the level which existed in the system(s) from which the fluid was drained.

(b) It is not possible, with this approach, to repeat a test.

(c) As discussed in section 6.2.2., "old" contaminant may produce less abrasive wear than will "fresh" contaminant. A test run on old fluid is therefore unlikely to produce measurable effects in an acceptable timescale.

(d) Any effect which might be produced by contaminant in used fluid will probably have occurred already in the system from which the fluid was taken. This approach is therefore unlikely to provide new data.

(e) It is extremely difficult to handle the contaminant in used fluid. Particles tend to settle to the bottom of the fluid container, and cannot easily be remixed into suspension.

A pump contamination test should therefore be conducted using hydraulic fluid which has been artificially contaminated through the addition of a dust or power. The important features of such a "test contaminant" are its size distribution, its physical and chemical properties, and the concentration at which it is used.

Although a completely realistic pump contamination test is not feasible, the size distribution of any test contaminants used in testwork can and should be chosen to correspond to the size distributions of contaminants found in actual systems. These tend to be of the form shown in figure 6.2.2, containing a broad range of particle sizes, but dominated, in number terms, by smaller particles. There is little point in running tests using excessively large particles, or particles of a uniform size, as these situations do not occur in practice. Having agreed that a completely realistic contamination test is not possible, it is still clear that more or less confidence can be placed in the usefulness of pump contamination test results, depending on how far test conditions depart from real-life.

Test contaminant materials, and the test contaminant concentration must be selected so as to deliberately produce damage in the test pump. There is no point in trying to devise a "typical" test contaminant, particularly if it is to be used at "typical" concentrations. Firstly, contaminant materials vary widely from one hydraulic system to another, and the concept of a "typical" contaminant is therefore a myth. Secondly, whereas actual contaminants produce gradual effects over many thousands of hours, a contamination test must be completed in a much shorter time.

The only feasible approach is therefore to deliberately select contaminants to produce the types of effect to be studied. If abrasive wear is to be studied, a hard abrasive contaminant such as silica should be used. If other effects such as silting, are of interest, then much softer contaminants are appropriate.

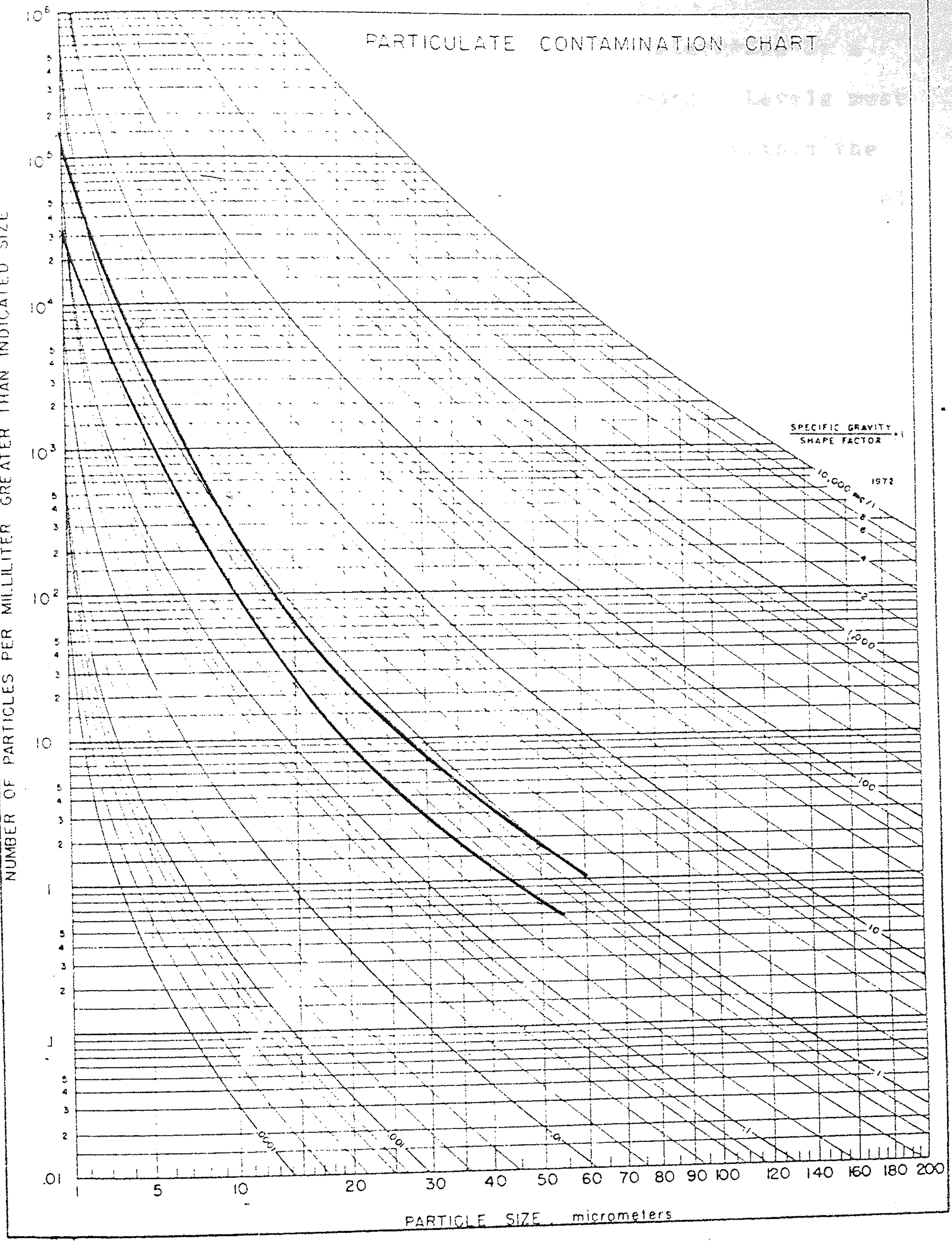

Figure 6.2.2. 'Typical' contaminant size distributions.

Test contamination levels must be determined by a compromise between conflicting requirements. Levels must be high enough to produce measurable effects within the restricted timescale of a multipass test. However, levels should be kept as low as possible, to try and ensure that any effects produced differ only in degree, but not in kind, from those produced in service. Suitable levels can only be determined by experience, and this was one aim of the tests described later in this chapter.

6.2.4. Test length and operating conditions

The selection of values of pump speed, output flow, and suction and delivery pressures to be used during a pump contamination test must be a matter for the individual or organisation conducting the tests. These values must be somewhat arbitrary, for there is no such thing as a "typical" pump duty. This was shown in a study by Lee [30], and was confirmed by the author in industrial visits made during the project.

It was originally thought that the choice of test length would be made on grounds of cost. Any test run for a period in excess of twelve hours will require either an automatic rig with emergency shutdowns, or continuous monitoring on a two-shift basis. Either alternative would be relatively expensive to implement. However, as Bensch [op cit] found, and as will be shown later in this chapter, any multipass pump contamination test is completed in a few hours, in the sense that no new effects are apparently produced after this time. A singlepass test could theoretically be run for much longer periods, but the problems with this approach have already been pointed out. The conclusion to be drawn is that pump contamination tests need last for only a few hours, and a simple rig is all that is required.

6.2.5 Test results

In the OSU pump test, pump output flow is used as a measure of the damage produced by operation on contaminated fluids. Other pump performance parameters, such as case leakage, volumetric or mechanical efficiency could be used as indirect measures of the effects of contamination on the test pump. Whilst this type of data might be used to assign a simple "contamination sensitivity rating" to a pump, it is of very limited value in providing information as to the type and location of any damage produced within the pump. In the tests described later, attempts were therefore made to determine what had been happening within the test pumps, by analysis of fluid samples taken during the tests, and by post-test examination of pump components.

6.2.6 The optimum approach to pump contamination testing.

The considerations given in sections 6.2.2. to 6.2.5 inclusive lead to the conclusion that the best available approach to pump contamination testing is to run fairly short, multipass tests at relatively high levels of fluid contamination. The fluid used in these tests must be artificially contaminated by the addition of dusts or powders having a particle size distribution similar to those shown in figure 6.2.2.

6.3 The OSU Pump Test

The basic test decided upon in section 6.2.6. is precisely that adopted by OSU in their pump contamination test, and one approach to this research would have been to have adopted the OSU test in its entirety. This was not done because, although the test method is an acceptable approach to pump contamination testing, the results obtained and the uses to which the results are put, are of little value to a pump manufacturer. The OSU test was eventually rejected for four reasons.

(a) The theory associated with the test is based on empirical data from tests on gear pumps. It remains to be shown that the same theory is applicable to axial piston pumps. Although the theoretical work presented in Chapter 4 lends some support to the OSU test theory much more work is required before the quantitative data from this type of test may be used with confidence.

(b) The test data are used, by OSU, to relate pump life to fluid contamination levels. It has already been shown that this is an over ambitious aim.

(c) Based on CHL's experience, the criterion of pump failure used in the OSU test, namely reduction in pump output flow, was not thought to be appropriate to axial piston pumps. CHL's experience has been that piston pumps tend to fail in a sudden, catastrophic manner, rather than through a slow, steady reduction in output flow. This point is considered later in relation to the results of the pump contamination tests conducted.

(d) The main shortcoming of the results of an OSU test is that an external performance parameter is used as an indication of what has happened inside the test pump. As stated above, this might be used to give a simple, order-of-merit "contamination rating" to the pump design, but knowledge of what has happened inside the test pump would be of far more value to a pump manufacturer.

It was no part of this research to decide whether or not the OSU pump test is "correct". The decision required was whether or not the test provided or could provide, enough information for it to be a useful tool for a pump manufacturer. The decisions reached were that, in its present form, the OSU test does not provide this type of information, but that it might be possible to obtain more useful results from the OSU test, or from a simpler version of it.

To investigate this matter, a series of pump contamination tests was conducted, and the tests and associated results will be described below. The test procedure used was of a similar, but simpler, form to that employed by OSU. The OSU test requires successive injections of test contaminant in nine different size ranges. This is to allow calculation of the values of γ_i used in the theoretical relation:

$$\frac{dQ}{dt} = -\gamma_i n_i^2 Q \qquad 6.3.1$$

As these values were not to be calculated, only one size range of contaminant was used in each test.

6.4 Pump Contamination Test Programme

6.4.1 Test pumps

Seven, nominally identical, fixed-displacement axial piston pumps were purchased specifically for the test programme, at a cost of £1200.

The pump chosen for the tests was the Reyrolle A70 design. This unit, which is illustrated in figure 6.4.1 and plate 6.4.2, has a nominal displacement of 0.7 in^3/rev., giving a theoretical delivery of 3.8 gpm at a swash-angle of 20° and speed of 1500 rpm.

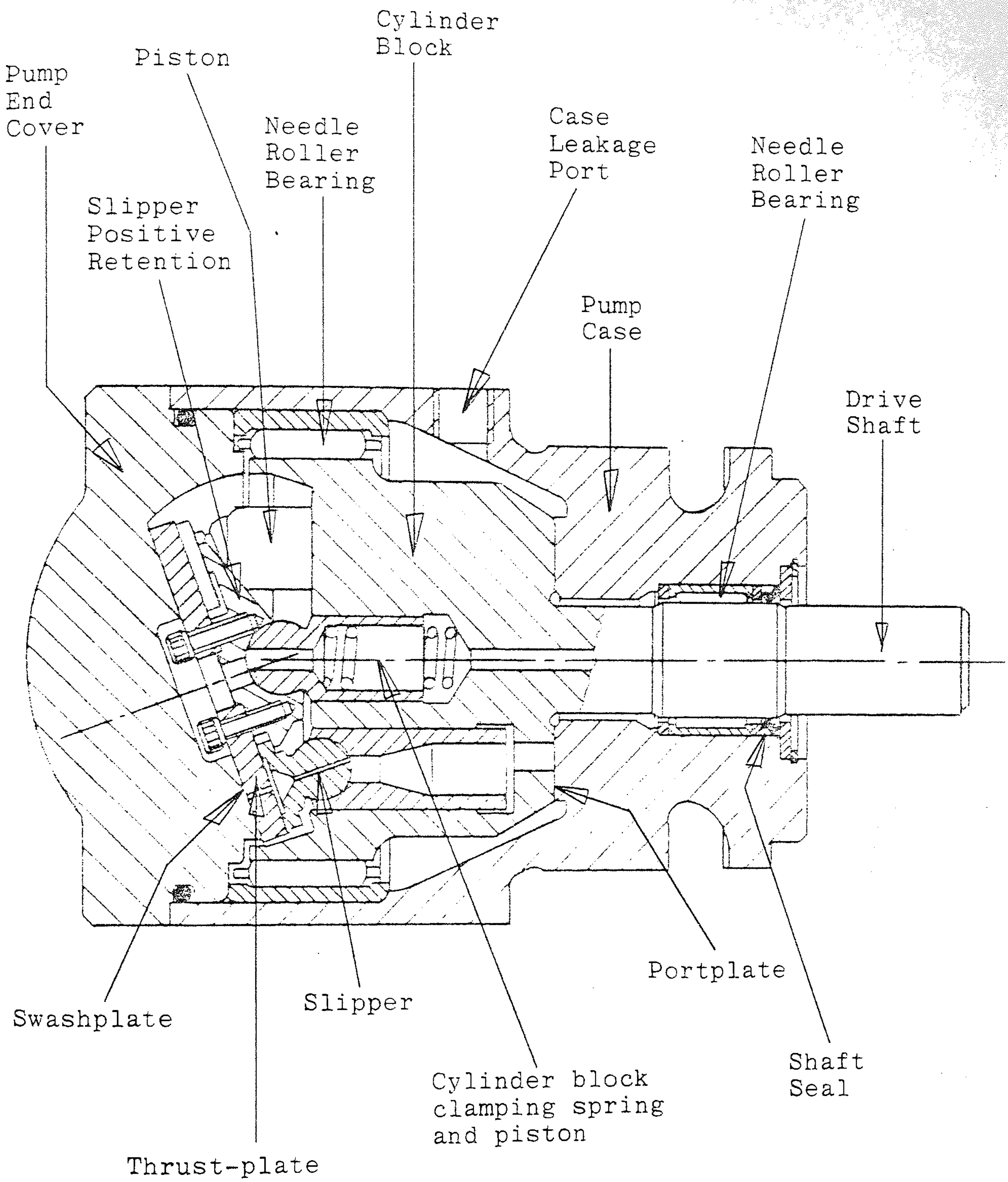

Figure 6.4.1 General arrangement of Reyrolle A70 pump.

Aston University

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Plate 6.4.2. Reyrolle A70 pump components

The A70 pump was selected because it is a fairly simple design, and is relatively cheap. Being of modest output, it required only a small, low-power test rig. It also had the advantage of running unboosted.

The A70 is a fairly conventional single-bearing, axial piston pump. The only novel feature is the use of a thrust-plate between the slipper bearings and the swash-plate. (Shown in figure 6.4.1.) The thrust plate spins at pump speed, so that the slippers are almost stationary with respect to the thrust-plate, as opposed to sliding around the slipper-plate in a more conventional design.

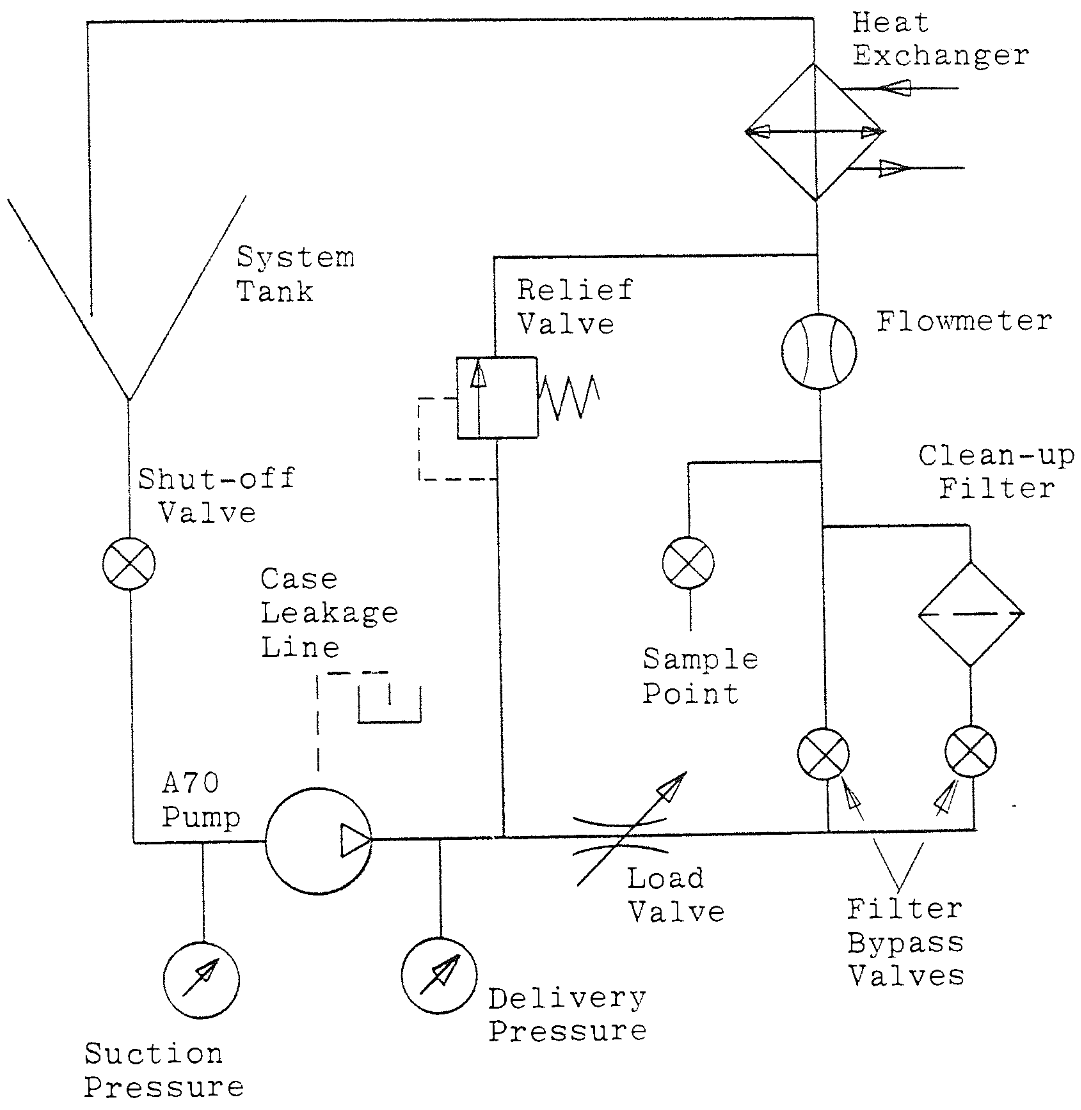
6.4.2. Test rig.

A test rig was conducted at Aston University specifically for this test programme. The rig design finally adopted is shown in figure 6.4.3 and photographs of the rig are shown in plates 6.4.4 and 6.4.5.

The rig specification was that it should allow operation of the A70 pump at a nominal speed of 1500 rpm and delivery pressure of 1500 psi, at specified fluid contamination levels up to 300 mg/l. Shell Tellus 37 hydraulic fluid was used in all tests at a bulk system fluid temperature of $40^{o}C$.

The only unusual rig design features were those incorporated to keep test contaminant in suspension. The majority of these features drew on experience which the author gained while studying at OSU.

The system tank was made of aluminium alloy, to reduce contaminant generation through corrosion. The tank had smooth sides, and was conical in form, with the tank outlet at the base of the cone, to minimise the possibility of contaminant settling out in the tank. The



Aston University Circuit for Contamination Testing
A70 Pump

Figure 6.4.3

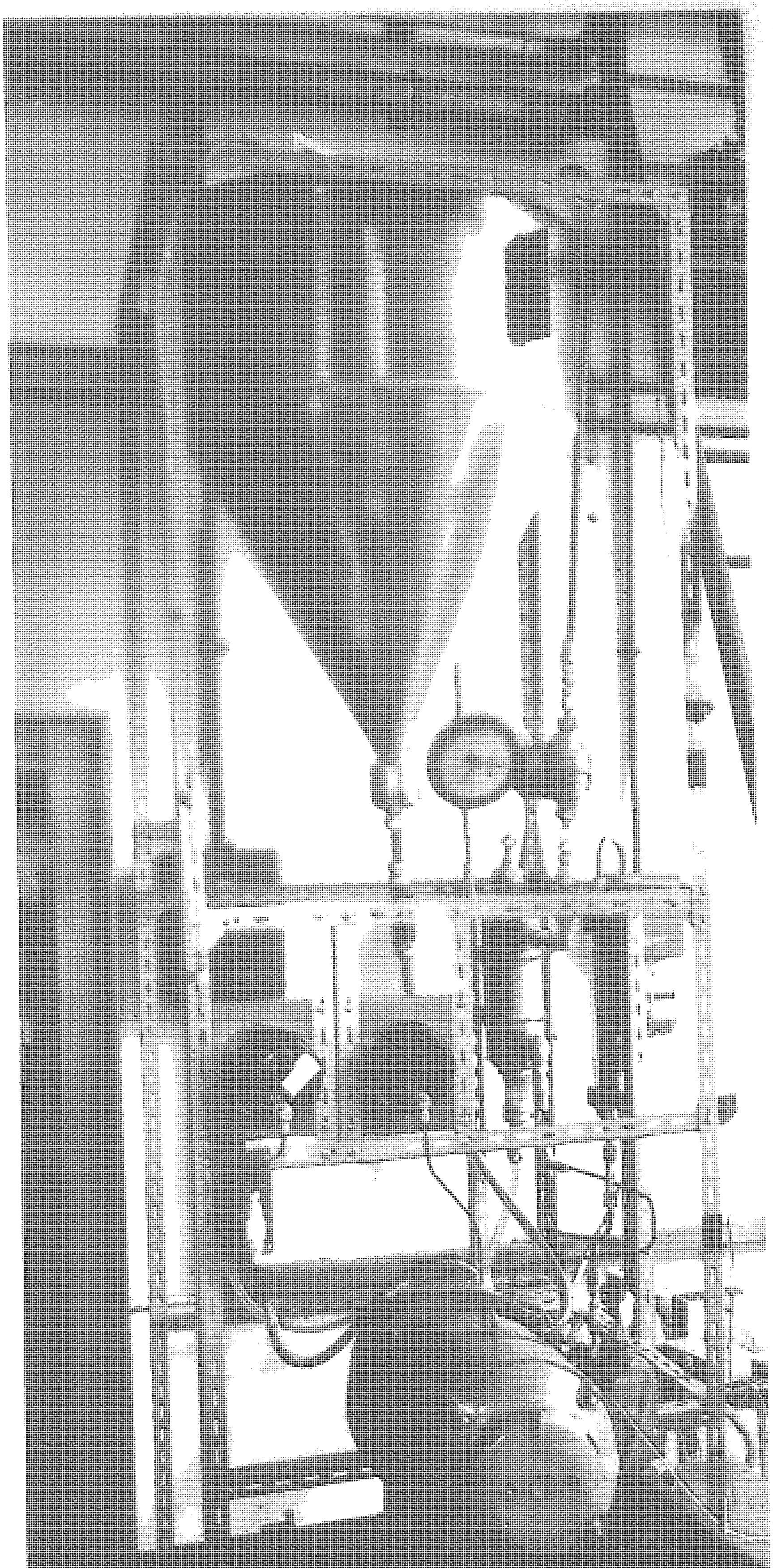


Plate 6.4.4. General view of Aston University test rig

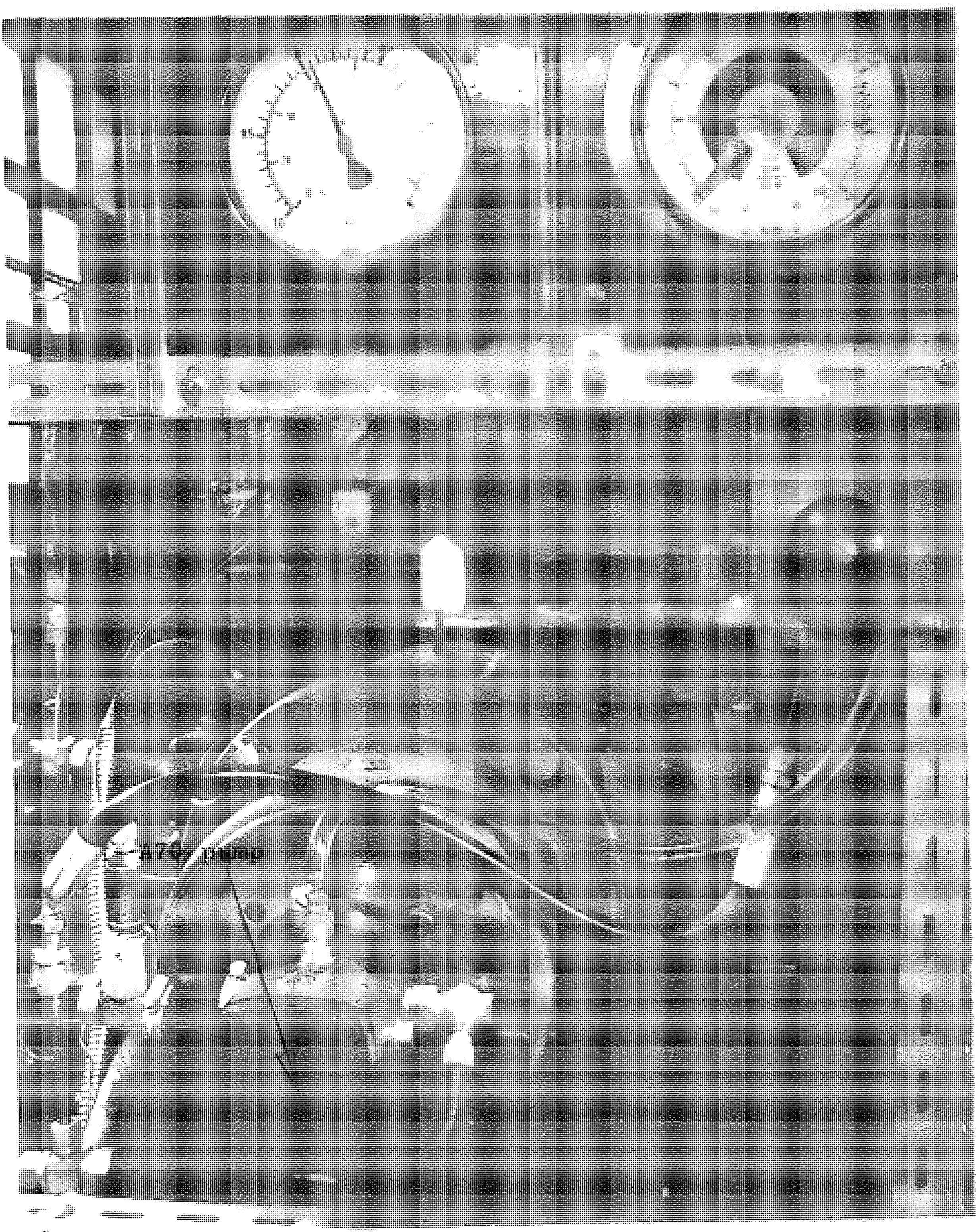


Plate 6.4.5. A70 pump mounting on Aston University contamination test rig

system return line was also arranged to impart a mixing action to the tank fluid, and the tank was kept covered at all times.

The test pump, shown in plate 6.4.5, was mounted vertically below the tank outlet, to prevent particles settling out in the pump section pipework. All rig pipework was of steel or plastic. Rubber hose was not used in case it should shed particles into the test fluid.

OSU recommend that all rig pipework should be sized to produce turbulent flows. (This is thought to keep contaminant in suspension.) Pipes were initially sized in this way but, at the low pump output flow involved, this imposed unacceptable back pressures on the system cooler and flowmeter. Pipe diameters were therefore increased, and it is probable that laminar flow existed in parts of the rig. Day and Lee [14] reported similar problems in the design of a filter test rig, but found, in a series of careful measurements, that there was no tendency for particles to settle out in quiescent areas.

Tee-junctions in rig pipework were positioned with the normally 'dead' leg vertically upwards.

Ball valves, were used for the rig shut-off valves and for the fluid sample valve, as this type of valve is reported to be the least likely to trap contaminant. The fluid sample valve was located near to an elbow fitting, following the recommendation of Day and Lee [op cit] to ensure that fluid was sampled from an area of turbulent flow.

The system oil cooler was mounted vertically, with oil entering at the base of the cooler and leaving at the top. Pipe runs were kept vertical as far as possible.

A 'Fairey' filter of "one micron" nominal rating and generous capacity was installed as a "clean-up" filter. The requirement for the filter was that it should be capable of cleaning the system fluid to a contamination level less than one tenth of the lowest level to be used in the test.

Rig instrumentation was deliberately kept as simple as possible. This helped to keep down project costs and rig construction time, and also served to make the test rig reliable and easy to operate.

Pump suction and delivery pressures were measured on 'Wika' pressure gauges previously calibrated on a deadweight tester. Pump speed was determined by a magnetic pick-up situated so as to detect the passage of a small bolt on the pump/electric-motor coupling. The pick-up output was converted to an equivalent time-period and displayed on a digital meter. Bulk system temperatures were measured using a thermistor probe inserted into the system tank. (Bulk system fluid temperatures were controlled by regulating the water flow to the oil cooler.)

Some problems were expected in measuring fluid flow. CHL use small turbine flowmeters for this purpose. These devices are quite delicate and it was expected that their reliability and accuracy might be seriously impaired by operation on heavily contaminated fluids. To prevent any such problems, a Fischer and Porter tube and float meter was used. This device had a quoted accuracy of ±2% of full scale flow. The meter was subsequently calibrated against a displacement type flowmeter, the calibration curve being shown in figure 6.4.6.

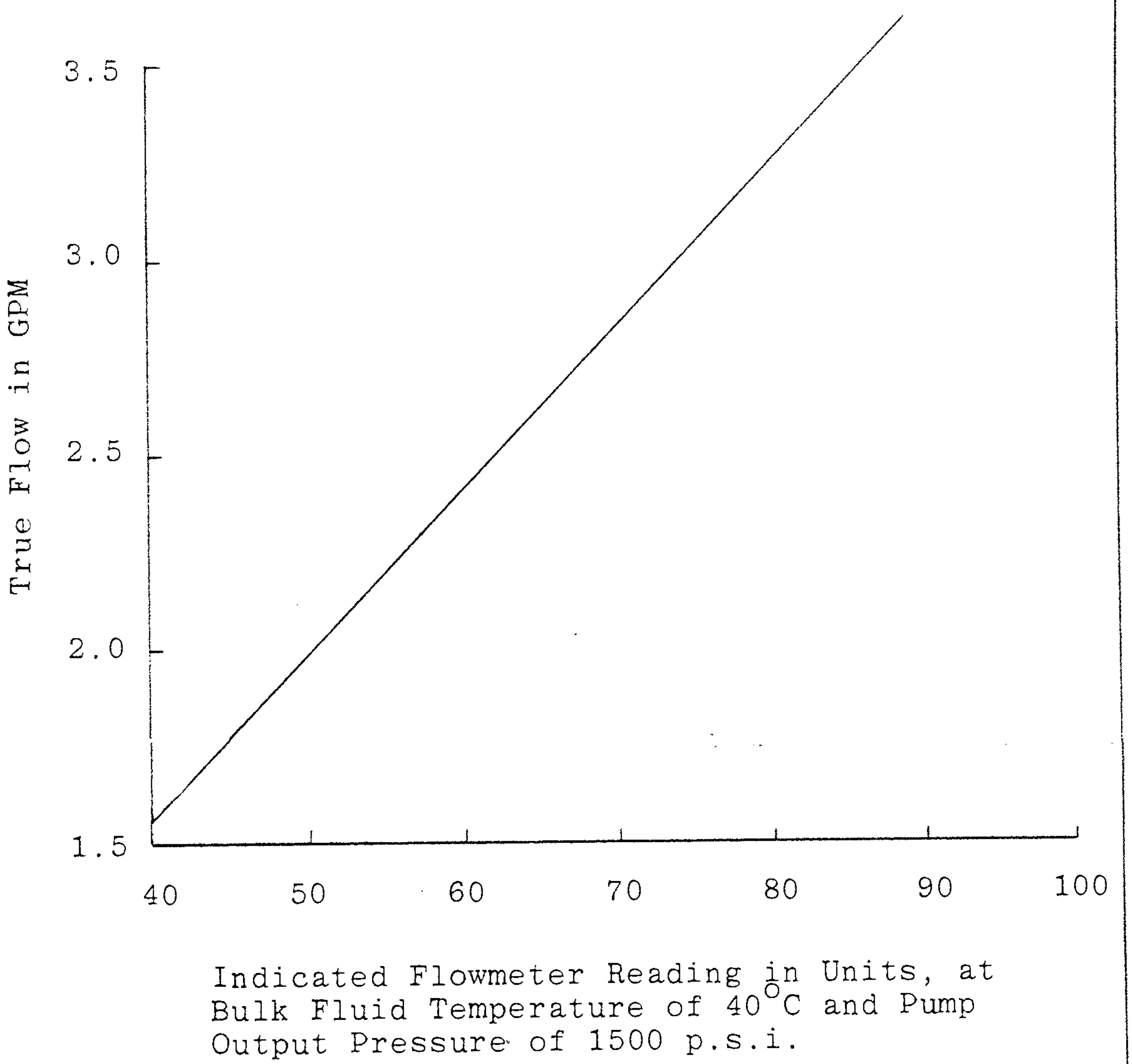
6.4.3. Rig Commissioning

No major problems were experienced with the commissioning of the rig, but one point should be mentioned. It is standard practice, with this type of rig, to determine how well the rig can maintain contaminant in suspension. The usual procedure is to contaminate the rig system fluid to a level of 300 mg/l, and then determine that this level does not vary by more than 10% over a one hour period. This procedure was not followed, for two reasons. Firstly, tests were to be run for seven hours, and it was not expected that a level could be maintained for this length of time anyway. Secondly, the author knows of cases where it has taken eighteen months to commission a contamination rig in the approved manner. This sort of time was not available in this research.

It was therefore decided that fluid samples should be taken during the tests, for subsequent gravimetric analysis. The aim was to determine how contamination levels varied during the tests, on the assumption that, even if the test procedure was imperfect, it should at least be consistent.

6.4.4. Test procedure

The basic pump test involved operating a pump on contaminated fluid, for a period of seven hours at a mean delivery pressure of 1500 psi and a nominal speed of 1500 rpm. Pump inlet pressure was 0 psi (±2) and bulk system fluid temperature was maintained at 40°C (±2).



Flow in GPM = 0.118 + 0.0374 x Indicated Flow in Units

Figure 6.4.6

Calibration of Aston Rig Flowmeter

Before mounting on the rig, the test pump was stripped down and was examined visually for manufacturing defects and incorrect assembly, either of which would have invalidated test results. Pump components were then cleaned for two minutes in an ultrasonic bath filled with Genklene[1], and were subsequently allowed to dry in air for thirty minutes. Parts were then coated with clean oil, and the pump was reassembled. The cleaning procedure was designed to remove any internal contamination left in the pump from manufacturing and assembly operations.

After mounting the pump on the test rig, five gallons of new hydraulic fluid was added to the system reservoir. A new batch of fluid was used for each test, to prevent a build-up in the fluid of fine particles from previous tests. The volume of fluid used is important because, as will be shown later, it affects the value of τ, the "particle destruction time constant". OSU recommend that the ratio of fluid volume to pump output flow per minute should be 0.25. However, this requires a very small system fluid volume, and can lead to substantial aeration of the fluid. Excessive fluid aeration can damage a pump, so the ratio of V/Q in these tests was increased to a nominal value of 1.32.

After adding oil to the rig, the test pump was "run-in" for ten minutes at each of 500, 1000 and 1500 psi. This running-in period was included to allow components to "bed-in" with each other, and to give time for surfaces to be hardened and smoothed. It is fairly well established that, during this process, substantial.

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1 'Genklene' is an ICI brand name for inhibited 1.1.1 trichloroethane, a non-flammable industrial solvent.

numbers of wear particles are produced. The pump was therefore run for a further thirty minutes at 500 psi. During this period, and during the running-in process, the hydraulic system fluid was passed through the clean-up filter. This was done to remove much of the contaminant present in the fluid in its "as-delivered" condition, and also to remove wear particles from the running-in. In fact, the Ferrography[1] results presented later suggest that this running-in period should be longer in any future test work.

At the end of the running-in process, a fluid sample was taken for subsequent analysis, and the system filters were switched out of circuit. Pump delivery pressure was increased to 1500 psi and the required amount of test contaminant was injected into the test circuit. Contaminant was prepared by mixing it to a slurry with clean oil taken from the rig. To disperse the contaminant and break up any agglomerations of particles, the slurry was shaken by hand for one minute, agitated ultrasonically for five minutes, and then shaken by hand for a further minute. (The author verified this procedure at OSU, by preparing a sample of ACFTD in this way and then measuring the particle size distribution of the sample contaminant on a HiaC 320 counter. The results are shown in figure 6.4.7 compared to the widely accepted log-log[2] distribution for ACFTD.) After preparation, the contaminant slurry was poured into the test system reservoir. After the injection of contaminant,

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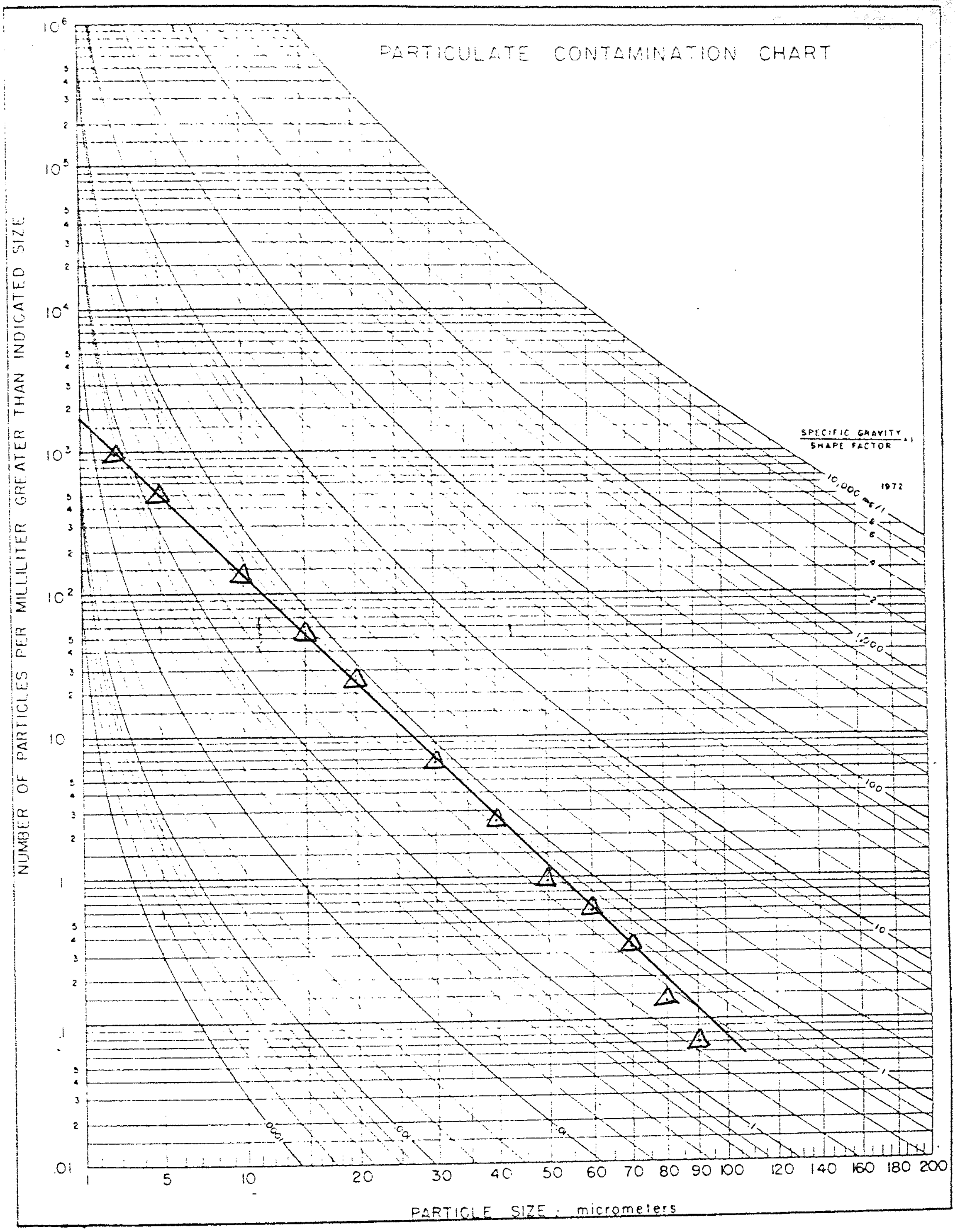
1 See glossary and appendix.



Particle counts on ACFTD prepared as in A70 contamination tests (△), compared to log-log² distribution of ACFTD (solid line on graph).

Figure 6.4.7

each test was run for seven hours, this period being the longest that could be achieved in a single operating shift. Pump output flow was monitored throughout a test. This was done, partly to give some warning of any disastrous changes occurring within the test pump, but mainly to assess the usefulness of this external parameter as an indicator of the effects produced in an axial piston pump by fluid contamination.

Fluid samples were also taken during each test, each sample being collected into a specially cleaned bottle. At the completion of a test, the test pump was shut down and the rig fluid was drained. This was to prevent residual contamination of subsequent tests.

6.4.5 Basic test programme and results

The basic test programme, summarised in table 6.4.1. consisted of ten pump tests. These were conducted to answer four questions.

(a) Is the procedure adopted a viable method of contamination - testing hydraulic pumps, and does it produce measurable effects?

(b) What is the test repeatability?

(c) Do the test contamination level, size and type have an effect on the results obtained?

(d) Is it possible to identify the nature and location of any damage produced during a test?

Each test is referenced by two numbers. The first is the serial number of the pump used in the test (1 to 7 inclusive). The second is the reference number of the test conducted on that pump. The different tests are described below.

Test 1.1 was conducted as a "baseline" against which

TEST REF NO.	PUMP	CONT. LEVEL mg/l	CONTAMINANT TYPE	RESULT FIGURE
1.1	1	-	System	Table 6.4.3
2.1	2	30	0-50 μ ACFTD	Fig. 6.4.8
3.1	3	100	0-50 μ ACFTD	Fig. 6.4.8
4.1	4	100	Graphite	Fig. 6.4.11
5.1	5	100	0-50 μ ACFTD	Fig. 6.4.10
5.2	5	100	0-50 μ ACFTD	Fig. 6.4.10
5.3	5	-	System	Fig 6.4.10
6.1	6	100	0-5 μ ACFTD	See text
7.2	7	300	0-50 μ ACFTD	Fig. 6.4.12
7.3	7	-	System	Fig 6.4.12
2.2	2	Varying	0-50 μ ACFTD	Fig 6.6.1

Table 6.4.1 Summary of test programme.

to compare the results of other tests. No "artificial" contaminant was added during the test, nor was the circuit fluid filtered. Fluid samples were taken at the start and finish of the test, resulting particle counts being shown in table 6.4.2. Table 6.4.3. gives values of pump output flow measured at the beginning and at the end of the test. (All values of pump output flow have been converted to equivalent flows at 1500 rpm, to allow for variations in the speed of the induction motor drive to the test pump. Values of pump output flow have also been left in flow meter 'units', because relative changes and comparisons between pumps were more important than absolute values. These could be derived from the calibration curve of figure 6.4.6)

Tests 2.1, 3.1 and 7.2 were run on fluid contaminated with ACFTD[1] in a nominal size range of 0-50 microns. Test contamination levels were 30, 100 and 300 mg/l respectively. Pump output flow fell during all three tests, the results being shown in figure 6.4.8. Test 7.2 was started as test 7.1, but an error was made in the latter test, and the test contaminant was introduced at 300 mg/l before the system filters were taken out of circuit. The test ran for only a few minutes before the filters blocked, and the test was therefore abandoned. If nothing else, this error graphically illustrated the sorts of problems which would be encountered in a singlepass test.

Samples of fluid from tests 2.1, 3.1 and 7.2 were subjected to gravimetric analysis, to determine fluid

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1 ACFTD - see section 4.8

Sample Number	No. of particles larger than indicated size (in microns) per ml. of fluid (1)					Thermal Control Class	Grav. Level mg/l (2)
	5	10	15	25	50		
2	9131	2960	1144	207.5	7.3	>9	11.0
4	3986	681.5	223.9	43.9	3.0	8	3.0

NOTES: 1. All counts are the average of two 50 ml. samples. Counts were determined on a Hiac PC 320 calibrated on ACFTD.

2. Counts were converted to an equivalent gravimetric level.

Table 6.4.2

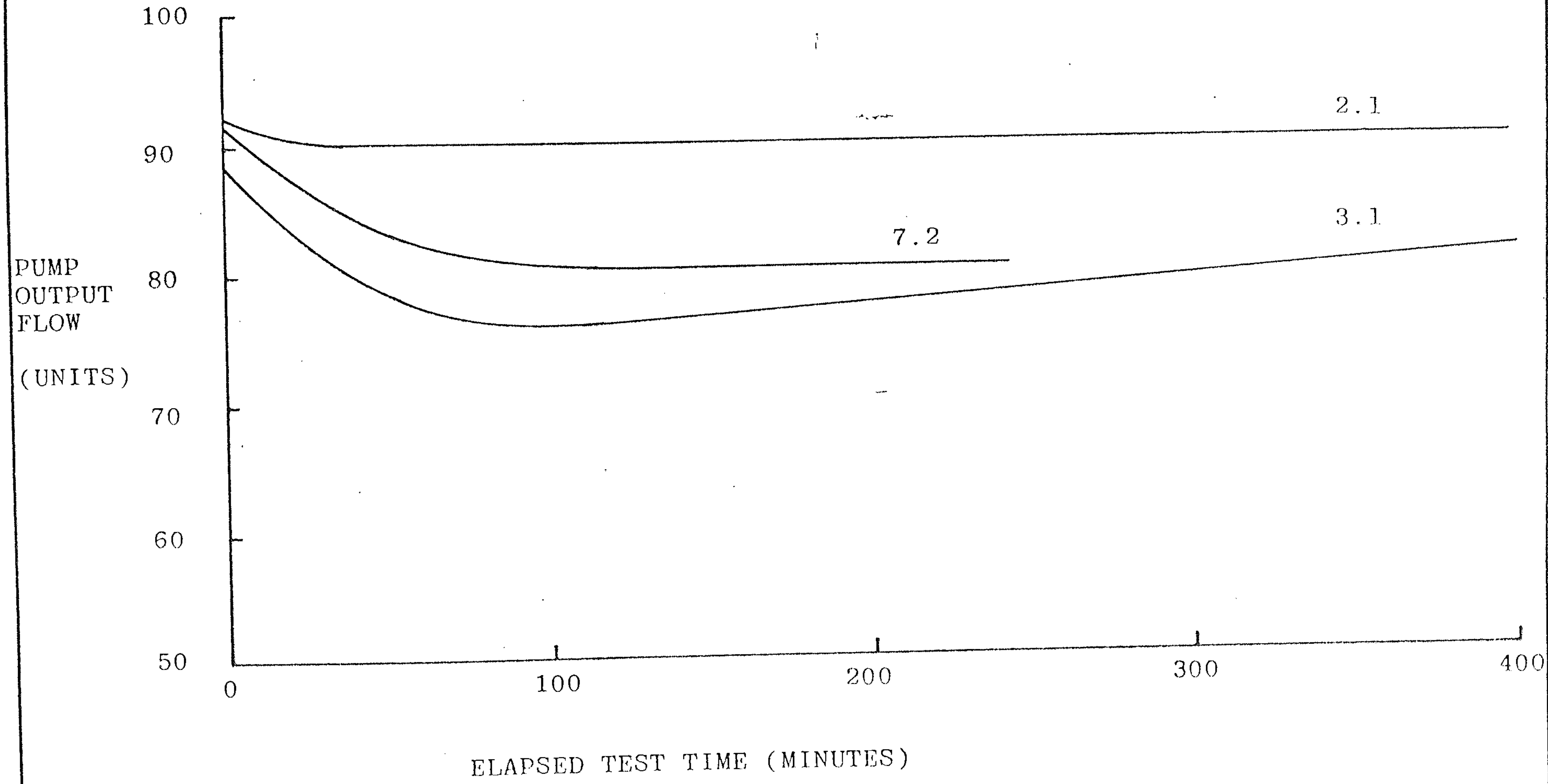
Particle counts on fluid samples taken at beginning (2) and end (4) of test 1.1

Test Period	Pump output flow (units) at 1500 rpm, at delivery pressure shown		
	500 psi	1000 psi	1500 psi
Start of Test	94.2	92.5	91.0
After 7 hours	94.4	92.8	91.4

Pump output flow during test 1.1

Table 6.4.3

Figure 6.4.8
Output flow of A70 pumps running on 30 mg/l (2.1), 100 mg/l (3.1) and 300 mg/l (7.2) of 0-50 micron ACFTD.



contamination levels at various times during the tests. The results of these, and other gravimetric analyses, are shown in table 6.4.4.

Test number 5.1 was conducted in an identical manner to test 3.1, to compare the repeatability of the test procedure. Values of pump output flow obtained during both tests are shown in figure 6.4.9.

Test 5.2 subjected pump number 5 to a repeat test on 100 mg/l of 0-50 micron ACFTD, to assess the cumulative nature of any damage produced. The results of tests 5.1 and 5.2 are shown in figure 6.4.10, as are the results of test 5.3, which was run on filtered oil to determine whether observed reductions in the output flow of pump 5 were temporary or permanent.

Test number 4.1 was run using a soft graphite contaminant at 100 mg/l. Values of pump output flow in this test are shown in figure 6.4.11, compared to the results of test 5.1.

Test number 6.1 was run on 100 mg/l of ACFTD that had been especially prepared ('classified') into a nominal size range of 0-5 microns. Pump output flow during this test remained constant, to ±0.5%.

Finally, pump number 7 was run on filtered fluid, in test 7.3, to assess the permanence of damage produced in test 7.2. The results of the two tests are compared in figure 6.4.12.

Before discussing the results of the main tests, two more general results will be described. During the early stages of a test, considerable quantities of air were visible in the pump case drainage line (which was of clear plastic). While this air was present, particles

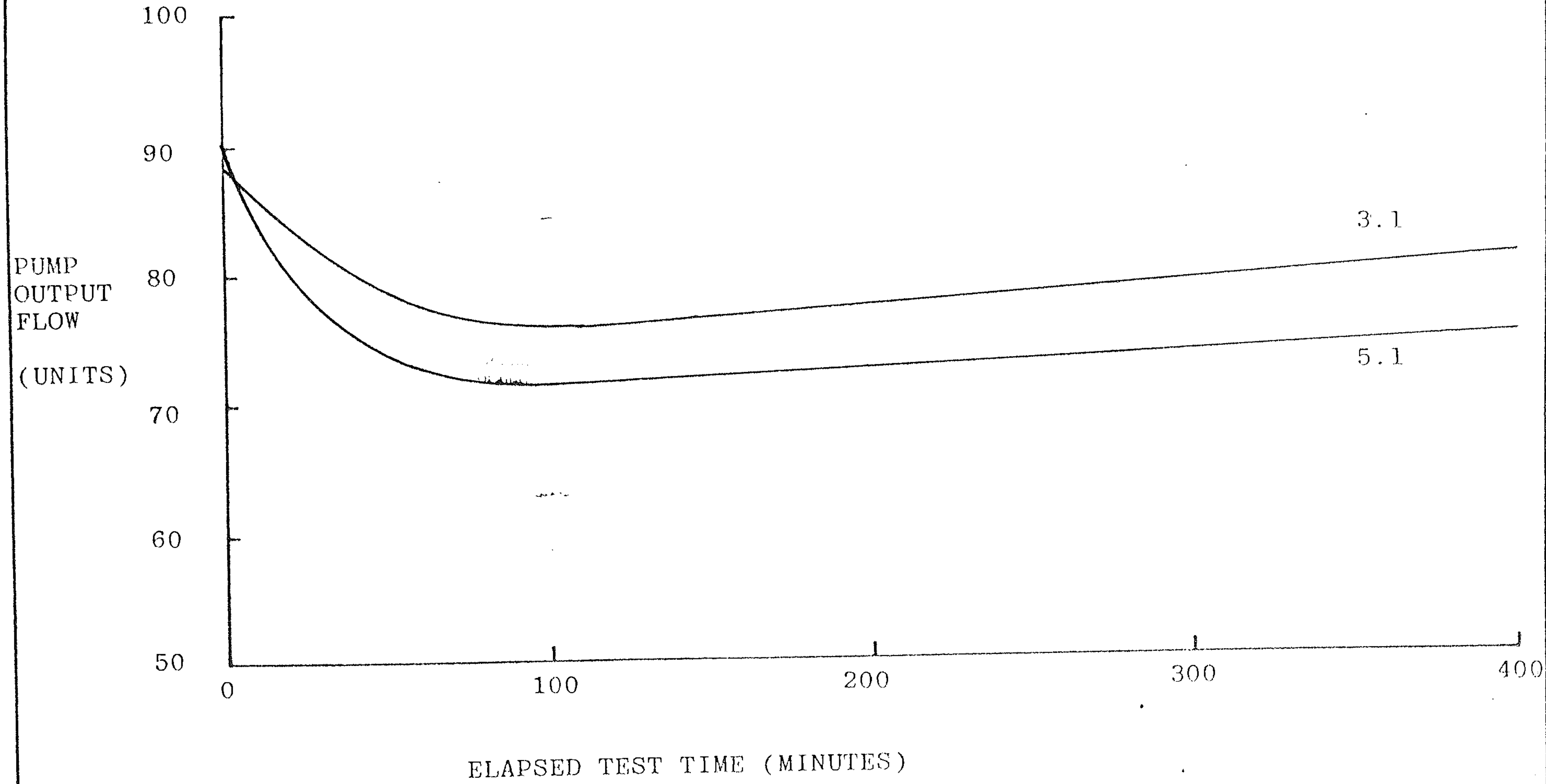
Test Number	Elapsed test time (Minutes)	Gravimetric Level mg/l	
		Expected	Measured
2.1 2.1 2.1	35 290 420	30 30 30	28 49 44
3.1 3.1 3.1	30 260 410	100 100 100	164 - (1) 98
5.1 5.1 5.1	26 236 420	100 100 100	144 124 92
5.2	237	100	136
6.1 6.1 6.1	30 267 420	100 100 100	152 80 120
7.2	86	300	228

NOTE: 1. Gravimetric level of this sample could not be determined.

Table 6.4.4

Gravimetric Levels of System Fluid during Contamination Test Programme

Figure 6.4.9
Output flow of A70 pumps running on 100 mg/l of 0-50 micron ACFTD.



Output flow of A70 pump run on 100 mg/l of 0-50 micron ACFTD (5.1);
5.2 shows subsequent performance on same contaminant.
Dashed line shows final performance on filtered oil.

Figure 6.4.10

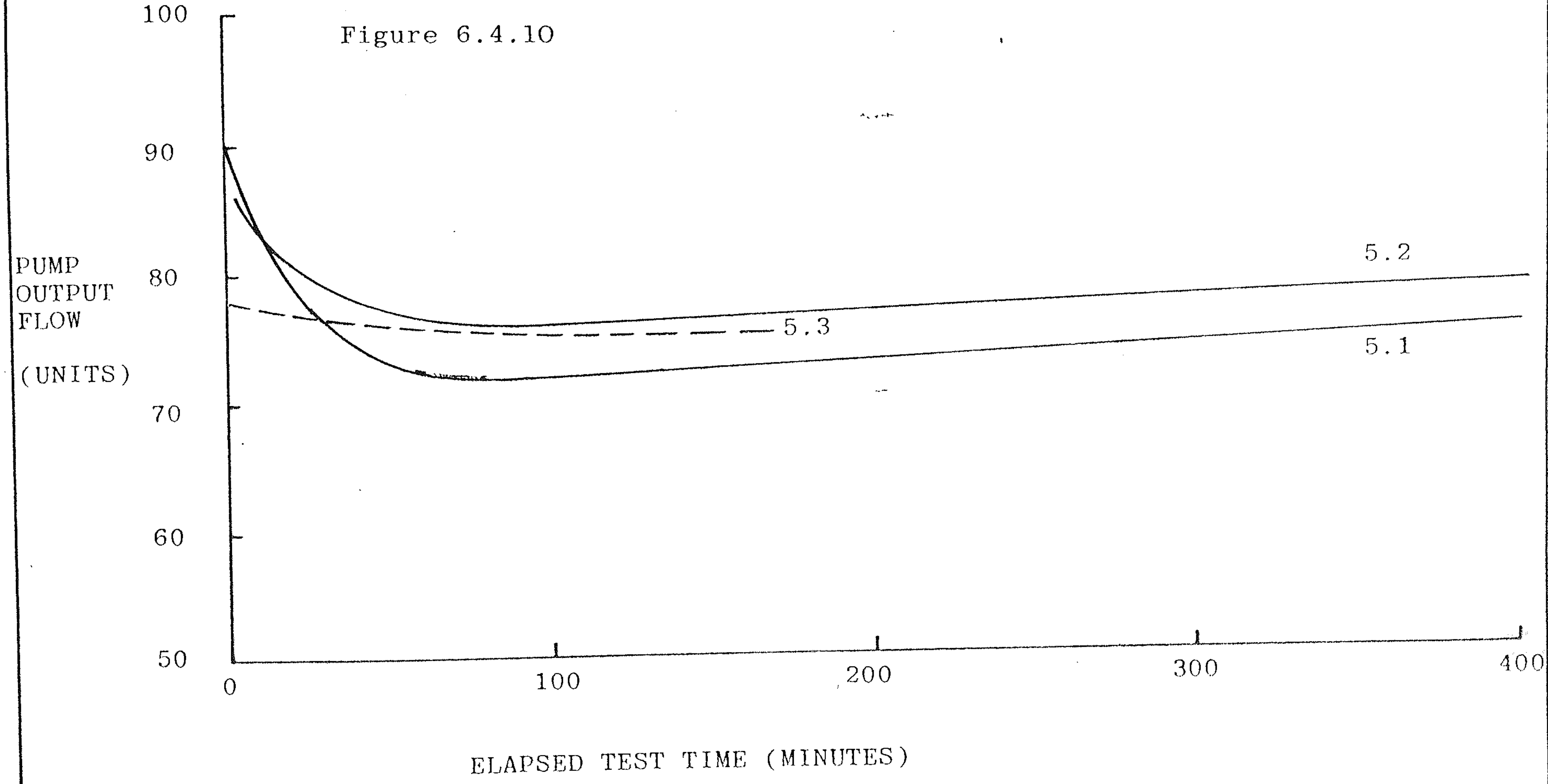


Figure 6.4.11
Output flow of A70 pump run on 100 mg/1 of graphite (4.1) and 0-50 micron ACFTD (5.1)

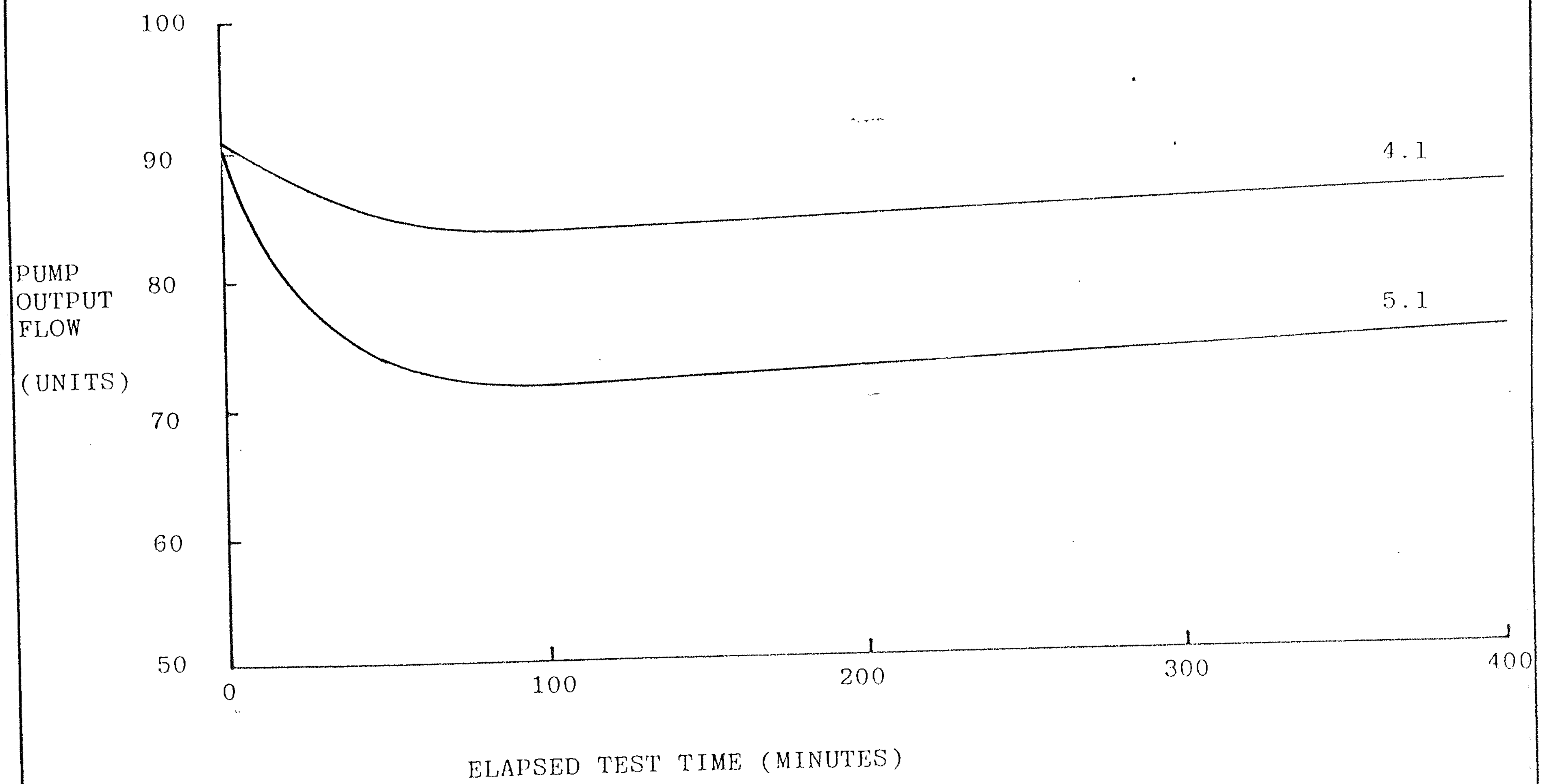
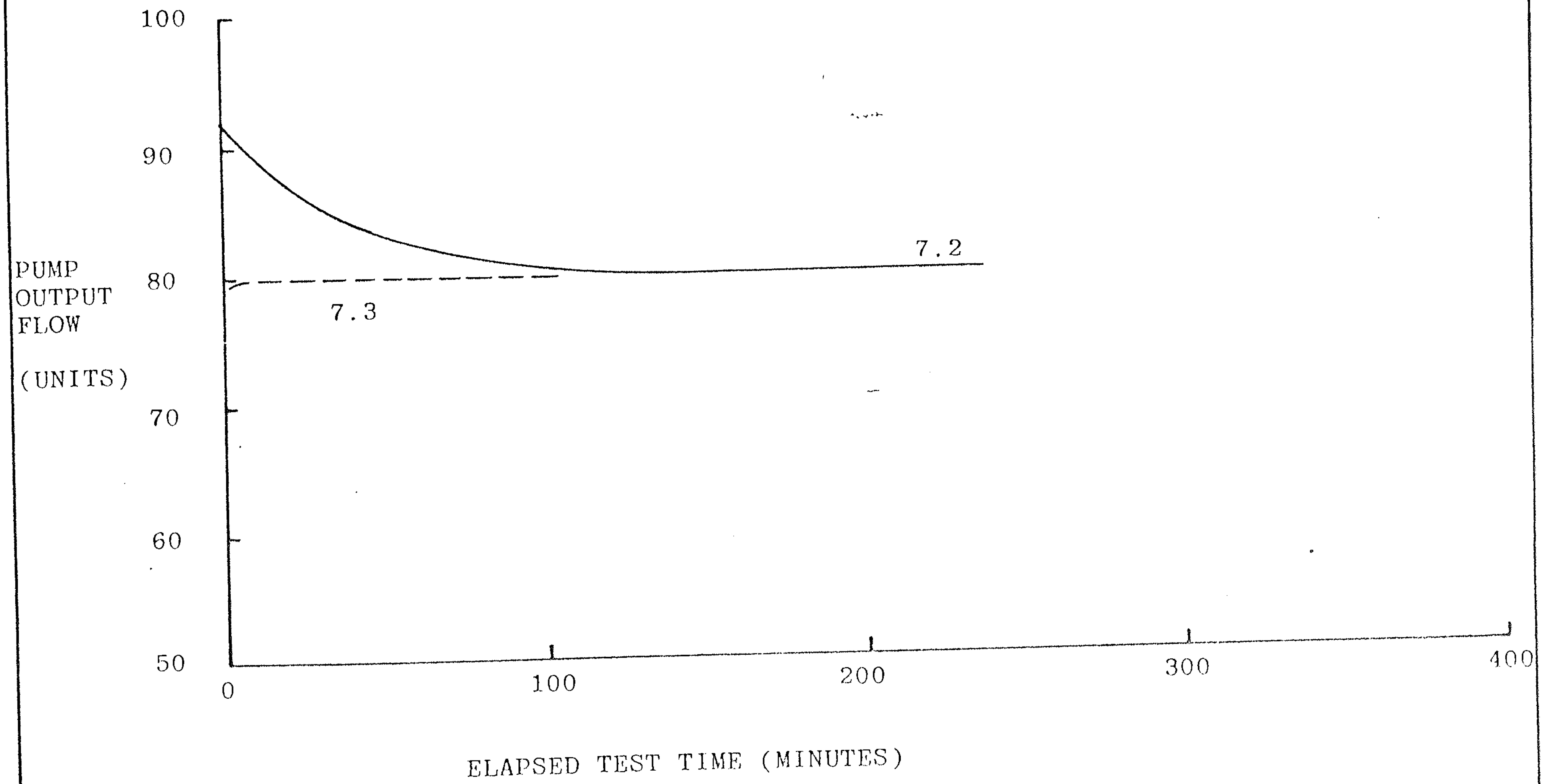


Figure 6.4.12
Output flow of A70 pump on 300 mg/l of 0-50 micron ACFTD (7.2).
Dashed line shows subsequent performance on filtered oil (7.3).



of metal could also be seen in the drain line. These particles had the appearance of flat platelets, and were estimated to be up to 500 microns across. No more particles were visible once the air cleared from the drain line.

The second effect concerned pump delivery pressures. When pumps were running on filtered fluid, fluctuations on the output pressure gauge were of the order of ±15 psi. When contaminants were added to the system, fluctuations increased steadily for five minutes, before stabilising at around ±100 psi.

6.4.6 Preliminary discussion of main test results

The results presented so far are incomplete; details of post-test pump examination and of Ferrographic analysis of fluid samples have not yet been given. However, enough data has been presented to allow discussion of questions (a) and (b) posed at the start of section 6.4.5. These were:

(a) Is the test procedure a practical one to use, and does it produce measurable effects on the A70 pump?

(b) Is the test procedure repeatable?

Conducting the tests was relatively straightforward, and no great problems were experienced during the test programme. The gravimetric analysis data given in table 6.4.4 shows that test contamination levels were of the required order throughout each test, although considerable variation is evident. This is probably not due to experimented error in the fluid analysis. The author showed in a study at OSU [87] that gravimetric analysis can be an extremely accurate technique. It is therefore clear that high levels of fluid contamination, of roughly the required level, were maintained throughout each test. This is not to say that the contaminant present in the later stage of a test was the same as that which was added at the start. Some of the contaminant added at the beginning of a test could well have settled out, or silted out in fine clearances during a test to be replaced by wear particles generated from the test pump, or by particles generated within the rig itself.

This possibility was checked by conducting optical microscope counts on fluid samples from tests 5.1. The particle counts obtained are shown in table 6.4.5. These counts support the OSU theory that particles are gradually broken down during a test. However, a large proportion of the particles counted were obviously not silica, and the counts represent only 18% of those expected from the gravimetric analysis.

It would be dangerous to deduce too much from this last observation, as microscope counts are notoriously inaccurate. It is also likely that silica particles were present on the microscope slide but were not counted. ACFTD particles are very difficult to count by optical microscope techniques, as the particles are transluscent. Two pieces of more qualitative evidence suggest that substantial quantities of ACFTD were present throughout tests run on this contaminant. The first is that substantial numbers of silica particles were found during Ferrographic analyses of fluid samples taken at the beginning and end of tests 2.1, 3.1, 5.1 and 6.1. The second is that ACFTD leaves a very characteristic, sandy-coloured residue on the surface of a membrane used

ELAPSED TEST TIME (MINS)	NUMBER OF PARTICLES/ML OF DIAMETER GREATER THAN INDICATED SIZE (MICRONS)		
	5	19	33
30	8796.6	570.6	76.6
240	9941.0	335.0	44.0
420	14189.0	352.0	8.0

Table 6.4.5 Microscope particle counts on fluid samples from test 5.1.

for gravimetric analysis. This type of residue was seen on the surface of all membranes used during gravimetric analysis of samples from tests run on ACFTD, although samples taken later in a test produced a "dirtier" residue. This would suggest the presence of some non-silica contaminant, which probably came from the test pump itself.

The conclusion to be drawn is that the tests did allow operation of piston pumps at contamination levels close to those specified, and that at least some of the original test contaminant was present throughout each test. It would obviously be of interest to know how much of the original test contaminant is present in the later stages of a test, but the difficulties of such an estimation would be considerable.

Having established that the tests did subject axial piston pumps to substantial levels of fluid contamination, the next question to be answered was whether or not such operation damaged the pumps. This question will be considered first in relation to the measured reductions in output flow.

The results of test 1.1, shown in table 6.4.3. indicate that operation on low levels of "system" contamination produced no measurable reduction in pump output flow over a seven hour period. Contamination levels in this test are shown in table 6.4.2. Contamination levels measured at the start of other tests, before test contaminants were added, are shown in table 6.4.6. It can be seen that these levels

Sample No.	Test	Number of particles larger than indicated size (microns) per ml. of fluid (1)					Thermal Control Class	Grav. Level mg/l (2)
		5	10	15	25	50		
7	2.1	411.6	86.6	30.8	3.9	0.3	4	0.36
13	3.1	192.6	34.3	7.07	4.2	0.8	3	0.25
19	4.1	180.4	35.1	15.3	5.6	0.6	3	0.29
25	5.1	844	181.1	69.5	18.9	1.5	6	0.96
31	5.2	778	168.3	66.0	15.8	1.6	6	0.92
37	6.1	144	22.8	7.9	1.8	0.2	2	0.16
43	5.3	471	60.9	23.6	7.8	1.2	4	0.53

NOTES:

1. All measurements are the average of two 50 ml samples. Counts were obtained on a Hiac PC-320 calibrated on ACFTD.
2. Counts were converted to an equivalent gravimetric level.

Table 6.4.6 .

Contamination Levels at Start of Pump Contamination Tests.

were all below those which existed in test 1.1, and it can be concluded that any effects noted in subsequent tests were due to the introduction of "artificial" contaminants.

(The data of table 6.4.6. indicate two further points. The first is that, even though a "one micron" filter was installed in the circuit from which these samples were taken, many particles larger than one micron in diameter were present in the system fluid. This supports the author's view, given in chapter 4, that the widely accepted concept that a filter is similar to a rigid mesh is misleading. The second point concerns the variation present between different samples. This variation is much greater than can be accounted for by counting or sampling errors. This illustrates the point that installation of a specific filter cannot guarantee that a required contamination level will be maintained.)

The most obvious result of all the tests run on contaminated fluids, except test 6.1, was a reduction in measured pump flow. These data are shown in figures 6.4.8. to 6.4.12. Of course, the fluid contamination could have affected the flowmeter, so giving misleading results. However a direct measurement of case leakage, taken during test 5.1, showed that this had increased to an amount equivalent (to within 5%) to the measured reduction in pump output flow. The results obtained therefore represent genuine reductions in pump output due to the presence of fluid contamination. Again, these reductions could have been purely temporary, caused by an effect of the contaminant on fluid viscosity, or by particles "wedging" components apart and thus increasing leakage flows. However, the results of tests 5.3 and 7.3, which were both run on filtered fluids, indicate that induced damage was permanent.

The conclusions to be drawn are that the tests conducted induced permanent changes within the test pumps, that these changes were caused by fluid contamination, and that at least some of the effects manifested themselves as a reduction in pump delivered flow.

Deductions should not be made from the test results without some consideration of test repeatability. This is especially true if a "pump contamination rating" is desired. The results shown in figures 6.4.8. to 6.4.12. certainly indicate that, as far as pump output flow is concerned, the test adopted is qualitatively repeatable. Wherever output flow fell during a test, the results obtained were of the same general form. However, the results of tests 3.1 and 5.1, shown in figure 6.4.9. indicate that quantitative repeatability is not so good. The output flow of nominally identical pumps, subjected to the same test procedure, fell by quite different amounts.

There are many reasons why quantitative test repeatability was poor. For example, although the pumps used in the tests were nominally identical, there would inevitably have been variations between them. Table 6.4.7 below shows the output flow of the pumps in their "as-delivered" condition, and this indicates that no two pumps are exactly alike.

PUMP	1	2	3	4	5	6	7
OUTPUT FLOW (UNITS)	91.0	92.1	88.3	90.6	90.2	92.6	92.4

Table 6.4.7. Output flow of A70 pumps at 1500 rpm, 1500 psi and 40°C system temperature.

There will also have been differences between batches of contaminants, and in the way in which different batches were introduced to the test circuit. Atmospheric conditions may have played a part. Rabinowicz [58] found, in experiments into abrasive wear, that wear rates were affected by as much as 20% by the ambient humidity.

The conclusions to be reached about test repeatability were that it was good enough to allow deductions to be made about the effects of fluid contamination. However, the level of repeatability obtained would have to be taken into account when making such deductions. In addition, repeatability was not good enough to allow the derivation of a "pump contamination rating", and this point will be considered in more detail.

6.4.7. A pump contamination sensitivity rating.

A pump contamination sensitivity rating might be used to give an "order-of-merit" figure to the ability of a pump to operate on contaminated fluids. Such a rating would be valuable to users in helping them to select dirt-tolerant equipment. It could also be used by pump manufacturers to test the potential benefit of any design or materials changes introduced in a pump to reduce the effect of fluid contamination on that pump.

The type of test described in this chapter might be used to derive such a rating, but such use would be constrained by four things.

(i) Because the test described here is, and can only be, an unrealistic process, it should only be used to derive a pump contamination rating if it can be shown that a pump which performs well in the test will also perform well, in its tolerance of fluid contamination, in service. Such a demonstration would not be easy, and this point is considered, fairly briefly, in chapter eight.

(ii) The test parameter used to rate the pump must be relevant to those failure modes most likely to produce failure in service. For example, the OSU test, and the tests described here, use pump output flow as a measure of the damage produced by fluid contamination. But Johnson [88] reports that many users experience rolling element bearing failures as the most common effect of fluid contamination. Pump output flow would not indicate this type of problem, although overall pump efficiency or pump noise emission might do so. More work is therefore needed to determine the most common pump failure modes which can be attributed directly to fluid contamination, and to decide on pump performance parameters which will indicate these problems. This work is discussed, again, in chapter eight.

(iii) Test repeatability will have to be improved before any contamination rating will be of much value. The results of tests 3.1 and 5.1, given in figure 6.4.9. show the level of variation obtained in output flow data from identical tests. By how much better than the A70 would a pump have to perform before one could say that it was a "less contamination sensitive" pump? The question cannot be answered at present. It must also be noted that the test variation exhibited in figure 6.4.9. does not include any variation caused by operation on different test rigs. If a contamination test was to be adopted for widespread use, this would have to be taken into account.

(iv) The test results indicate a continual variation in pump output flow over the lives of some tests. Tests produced an initial fall in delivered flow, but many pumps exhibited a steady improvement in output in the later stages of a test. If a contamination rating is to be assigned, at what point in the test should pump performance be measured?

All these points may form the basis of future work, but their present implication is that CHL is still some way from being able to devise a pump contamination rating. This leaves two uses of the test results: to improve general understanding of the effects of fluid contamination; to identify the nature and location of fluid contamination damage. These uses will now be considered. The first task is to explain why pump output flows behaved in the observed manner.

6.4.8 Observed changes in pump output flow.

The pump output flow curves shown in figures 6.4.8. to 6.4.12. consist of two sections. The initial phase is an apparently exponential reduction in pump output flow over the first 100 minutes of a test. The second phase is one of operation at constant, or gradually increasing output.

The exponential reduction in output flow may be explained by one or more of three possible processes.

(i) Contaminant could have been settling out or silting out during a test so that, after a time, not enough contaminant was present to produce further pump damage. This possibility was considered in section 6.4.6, where it was shown that substantial levels of fluid contamination were maintained throughout each test, and that at least some of the contaminant introduced at the start of a test was still present at the end. The observed results cannot therefore be explained in this manner.

(ii) The contaminant introduced at the start of a test could have been gradually broken down in size, or could have been altered in some other way, so that, after a time, no more damage was produced. The data shown in table 6.4.5. indicate that the average size of particles did reduce as a test progressed, and the results of test 6.1 also show that smaller particles (0-5 micron ACFTD) produce less damage than do larger ones. This effect could therefore explain some of the observed results. OSU would ascribe the observed changes in pump output flows to this factor alone. From the OSU equations relating to pump contamination tests,

$$\frac{dQ}{dt} = -\gamma n^2 Q \qquad 6.4.8.1$$

and

$$n_a(t) = n_a(o)e^{-t/\tau} \qquad 6.4.8.2$$

it can be shown that pump output flow in a multipass test should follow a curve of the form

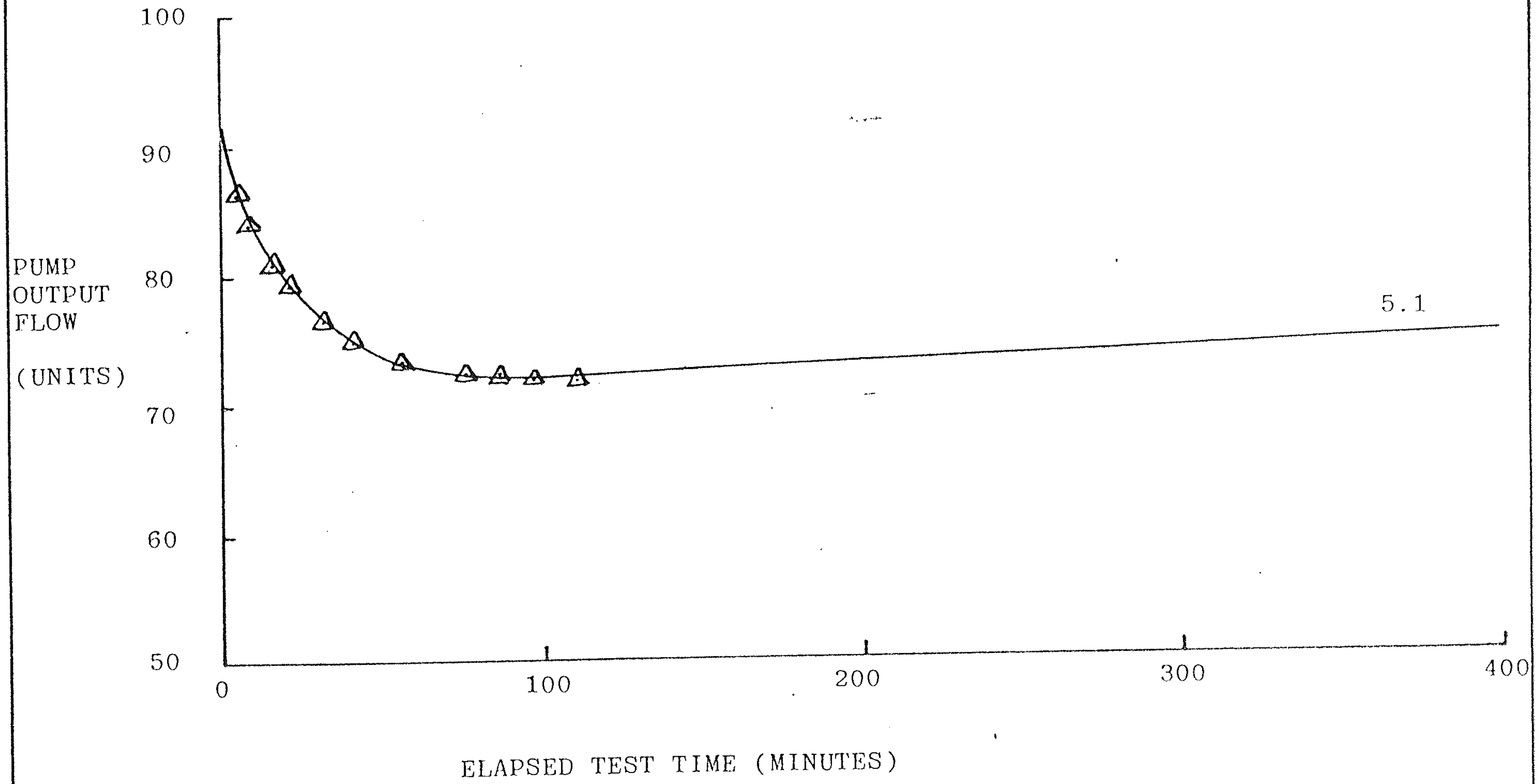
$$\ln\left(\frac{Q}{Q_o}\right) = \left[\ln\left(\frac{Q_f}{Q_o}\right)\right]\left(1-e^{-2t/\tau}\right) \qquad 6.4.8.3$$

where Q represents pump output flow at time t, and Q_f and Q_o are final and initial values of pump output flow respectively. OSU suggest a value of 9 minutes for τ, the "particle destruction time constant". However, they use a ratio of pump flow/system volume (Q/V) equal to 4, whereas, in the rig built for this work, this ratio was 0.72. This would mean that each particle would pass through the test pump less frequently than in the OSU test, and τ would therefore have had a value closer to 50 minutes.

Equation 6.4.8.3 was used, with the results of test 5.1, to check the OSU relation. The value of Q_f/Q_o was taken from the test results, and corresponding values of Q were calculated for various times during the test. The results are shown in figure 6.4.13, and it can be seen that agreement is excellent. This certainly seems to support the OSU theory. However, other test results obtained show that the overall situation must be more complicated than the OSU theory suggests. These results will be considered next.

(iii)The test pumps themselves could have undergone

Figure 6.4.13
A70 pump output during test 5.1. Solid line shows experimental data. Data points are calculated values from the O.S.U. theory.



internal changes which rendered them immune to further contamination-induced effects, or at least to further effects producing changes in output flow. Evidence for this hypothesis comes from the results of three tests.

(a) Equation 6.4.8.2, and hence equation 6.4.8.3, are only strictly applicable to the case of abrasive contaminants. But, in test 4.1, graphite contaminant, which has no abrasive wear potential, produced much the same effect on pump output flow as was induced by operation at the same level of silica. Some other explanation is required, at least for this test, than a change in the contaminant. Such an explanation would be that gradual changes took place within the pump and that, once this had occurred, no further damage could be measured.

(b) Test 5.2 subjected pump number 5 to a repeated exposure to 100 mg/l of 0-50 micron ACFTD. On the OSU theory, pump output flow should have fallen considerably during this test, to a level much below that measured at the end of test 5.1. In fact, as shown in figure 6.4.10, output flow at the end of test 5.2 was no worse than it was at the end of 5.1. This suggests that pump number 5 had been rendered immune to further contaminant-induced damage, or that such damage as was produced did not affect pump output flow.

(c) The reduction in pump output flow measured during test 7.2, run on 300 mg/l of 0-50 micron ACFTD, was no more (in fact slightly less) than that produced by operation on 100 mg/l of the same contaminant. This result casts some doubts on the applicability of the OSU theory to axial piston pumps. It also suggests that the A70 pump may suffer a certain level of damage due to fluid contamination and that, once this has occurred, no further reduction in pump output flow will be produced. If this is so, then fluid contamination may represent a performance problem as much as it does a reliability problem.

The second phase in each test was a period of operation at constant, or slightly improving output flow. The discussion given above suggests that operation at constant flow would occur because

(i) some change had occurred in the size or properties of the test contaminant and/or

(ii) changes had occurred within the test pumps themselves to render them immune to further contaminant-induced reductions in pump output flow.

Where a steady improvement in output flow was measured, this could have been due to any one of three factors.

(i) Some of the increased leakage flow produced during a test could have been temporary, caused by particles wedging components apart. If, as OSU suggest, and as this work confirms, the average size of particles decreases during a test, this temporary leakage should reduce as a test progresses. The test results are inconclusive on this point. The delivery pressure fluctuations noted after contaminant was introduced may be evidence that particles were wedging components apart (see section 6.4.5).

But this could also have been caused by the effects of particles on the load valve. It will also be remembered that the reductions in the output flow of pumps 5 and 7 persisted, when these pumps were operated on filtered fluids, at the levels measured when operating on contaminated fluids. This tends to rule out a temporary effect.

(ii) Turvey [66] measured a 20% reduction in the viscosity of hydraulic fluid contaminated to 300 mg/l with coal particles. Such an effect would, if it occurred in a pump, increase leakage and decrease output flow. If this effect did occur it must be size-dependent because output flow in test 6.1, run on 0-5 micron ACFTD, remained constant. Therefore, any increased leakage from this effect would become less significant as a test progressed and the mean size of particles decreased.

(iii) It is possible that abrasive contaminants can act as a coarse "lapping" compound within an operating pump. As the mean size of an abrasive contaminant reduces during a test, the particles may start to improve the pump by re-machining damaged surfaces. This could gradually increase pump performance. The flaw in this argument is that the output of pump number 4, run on graphite, also improved towards the end of the test, and this result cannot be explained by the possible effect of an abrasive contaminant.

It is not possible to decide which, if any, of the above effects were responsible for the observed improvements in pump output flow. The significance of the result is that some of the damage caused to a pump by fluid contamination may be only temporary. This would have to be considered in assigning any pump contamination rating, and it might also be possible in the future, to identify and incorporate pump features to improve this "healing" process.

The discussion to date has been entirely concerned with deductions which can be made from external pump performance. Nothing has been presented about the actual effects of contaminants on the pump components. It is this type of information which should be of most importance to a pump manufacturer.

6.4.9 Pump analysis

The most useful information which a pump manufacturer could derive from the type of test described here would be to know which components had been affected by contamination, and in what ways. To recap, this information could be used for two purposes.

(i) It could suggest methods of reducing the effects of contaminants.

(ii) It could suggest the most urgent priorities for theoretical analysis or more basic research.

In this research, information on these aspects was sought in three ways. The first was to strip down each pump after test and examine it. The second was to analyse wear particles from fluid samples taken during the tests, whilst the third was to measure the improvement produced in the performance of a damaged pump when undamaged components were substituted for worn parts. The first approach will be described in this section.

Stripping down a pump for examination after a test may seem to be the obvious way to find out what has happened during the test. However, there are several drawbacks with this practice, and these will be mentioned before actual results are presented.

(i) Dismantling a pump can easily destroy the evidence which is being sought. For example, a blocked orifice may be cleared, or a concentration of wear particles may be washed away.

(ii) Examination of a pump can only give information about the condition of components at one time. If a pump fails before it is examined it will probably be impossible to determine the initial cause of failure. The fluid will also be contaminated with wear debris, and it becomes impossible to decide whether contamination caused the original problem.

(iii) If the pump is reassembled for a subsequent test, it may have different characteristics to those which it originally possessed. This problem was encountered with tests on pump number 5 because the A70 pump has no means to accurately locate the swashplate in respect to the portplate. This affects the pump timing, and, for the tests described in section 6.4.11 the pump case and cover had to be dowelled together.

(iv) It is impossible to take measurements from some components without destroying the components themselves. For example, to take Talylin[1] traces of the portplate and cylinder block of pump 7, the components had to be cut up. This makes any retest impossible.

Having described some of the problems involved in this approach, it must also be stated that visual and metrological analysis of pump components did give very useful evidence of the effects of fluid contamination.

All pumps subjected to contaminants showed much the same visible signs of wear. These were polishing of the upper and lower faces of the thrust plate, marking of pistons, scratching of rolling element bearing surfaces and scratching of the portplate face of the cylinder block. Some of these effects can be seen in plates 6.4.14, 6.4.15. and 6.4.16. The cast-iron portplate and swashplate showed no visible signs of damage.

Three specific points were that:

(i) The pump run on 0-5 micron ACFTD showed much less signs of wear than did any pump except number 1 (which did not run on artificial contaminants).

(ii) No pump showed evidence that it had been concentrating contaminants in any area, or that fine clearances had been blocked up.

(iii) Pump number 4, which ran on graphite, still showed signs of wear.

This type of visual examination can indicate areas of gross wear, but it can give only limited information. More specifically, it did not indicate

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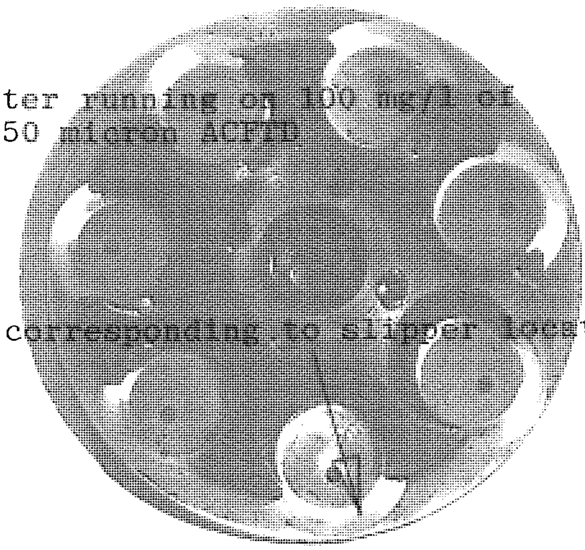
[1] See glossary

Plate 6.4.14. A70 pump thrust-plates (lower faces) after test



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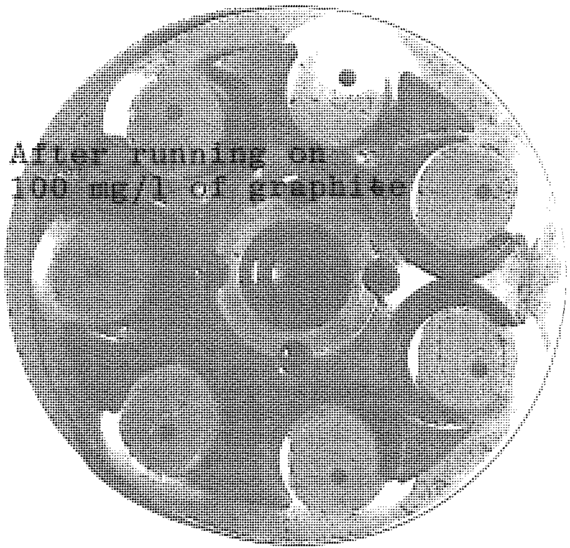

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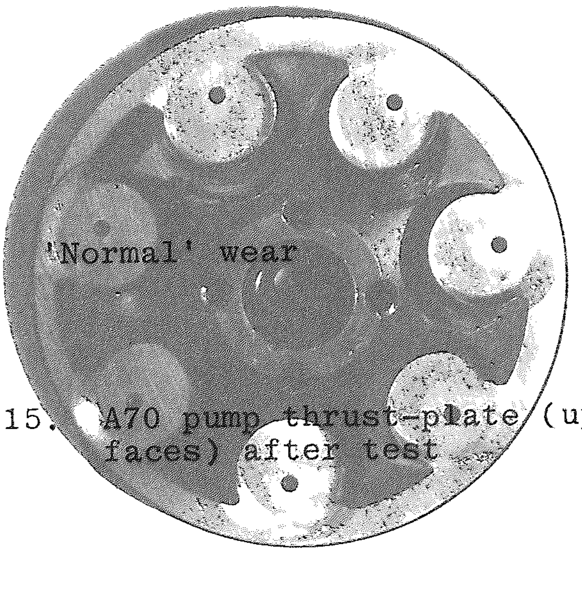


1

Plate 6.4.15. A70 pump thrust-plate (upper faces) after test

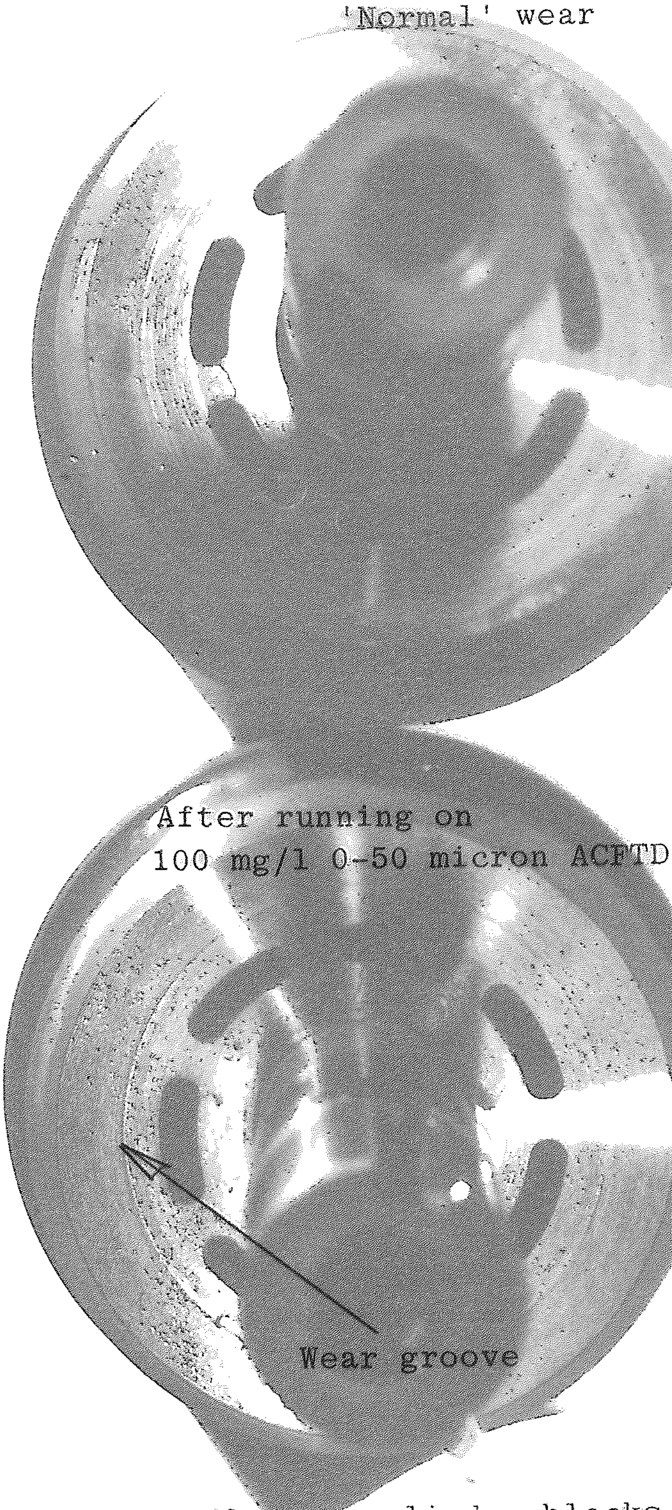


Plate 6.4.16. A70 pump cylinder blocks after test

which components suffered most damage, or which area was responsible for the measured decrease in pump output flow. More sophisticated methods are obviously needed to derive this information.

Many types of surface analysis could have been attempted, but, in the event, analysis was restricted to surface finish and surface profile measurement. The surface finish measurements taken are shown in table 6.4.8 ; the areas from which measurements are taken are shown in figure 6.4.17. Taking pump number 1 as the standard of a pump subjected to "normal" running-in, it can be seen that operating on contaminated fluids did produce a severe deterioration in some surfaces of most pumps subjected to contaminants. The exception was pump number 6, which ran on 0-5 micron ACFTD.

No clear pattern emerges from the data of table 6.4.8., but three points do stand out.

(i) The cylinder blocks of all pumps subjected to contaminants other than 0-5 micron ACFTD suffered surface damage.

(ii) Pump number 4 suffered most of its damage on the cylinder block.

(iii) Pump number 5 produced a larger reduction in output flow than did pump number 3, and also suffered more surface damage.

It is reasonable to conclude that damage to the cylinder block was produced by an effect other than abrasive wear. Damage to pump pistons, thrust-plates and swashplates seems to have been caused by abrasive contaminants of a size above 5 microns in

Pump No.	Surface finish (micron CLA) of component indicated.					Contamination Exposure
	Cyl. Block	Piston	Thrust-plate		Swash-plate	
			Upper Face	Lower Face		
1	0.13	0.10	0.05	0.05	0.40	No artificial contaminant
2	0.49	0.10	0.04	0.10	0.70	30 mg/l 0-50 micron ACFTD
3	0.30	0.15	0.05	0.12	0.65	100 mg/l 0-50 micron ACFTD
4	1.10	0.12	0.05	0.02	0.60	100 mg/l graphite
5	1.40	0.17	0.25	0.32	0.80	Two tests on 100 mg/l of 0-50 micron ACFTD
6	0.19	0.20	0.05	0.05	0.70	100 mg/l 0-5 micron ACFTD
7	-(1)	0.51	0.09	0.32	1.30	300 mg/l 0-50 micron ACFTD

NOTE: 1. Component was too badly marked to take reading.

Table 6.4.8

Surface Finish of A70 Pump Components after Running on Contaminated Fluids.

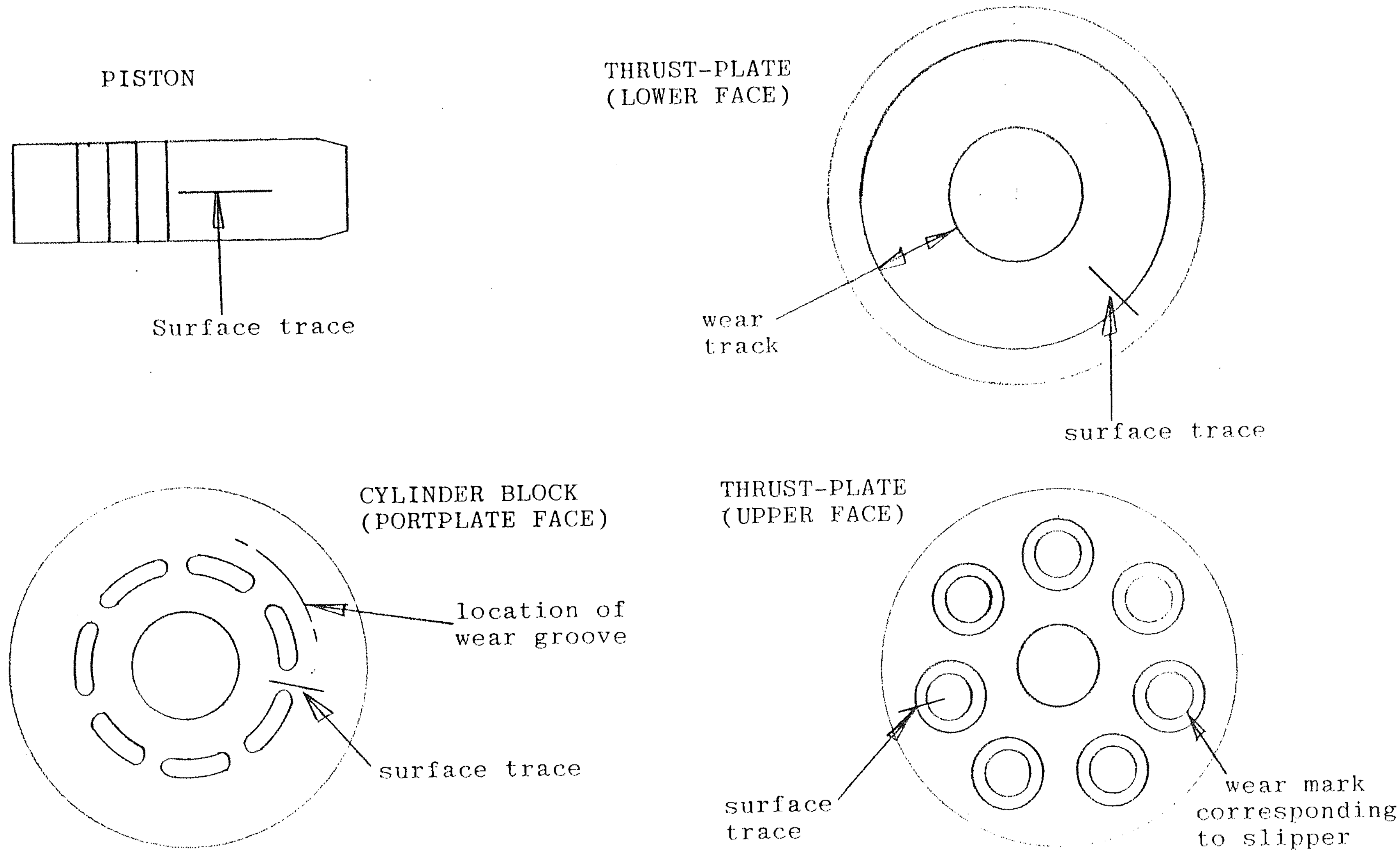


Figure 6.4.17. A70 pump areas from which surface finish traces were taken

diameter.

These results do not yet explain why pump leakage flow increased to the extent that it did. It may be that there is a direct relation between leakage flows and surface finish. Surprisingly little work has been conducted on this aspect of lubrication theory. Dowd and Barwell [89] conducted experiments into the interaction between piston and cylinder of an axial piston pump, but they did not investigate leakage as such. They explained their findings on the basis of "micro-asperity hydrodynamic lift", and showed that the degree of contact between piston and cylinder was dependent on component surface finishes. The same could also be true of leakage flows and more work in this area would be useful.

In cases where pure hydrostatic lubrication exists, surface finish changes are unlikely to increase leakage flows. This was shown by calculating the flow expected between parallel plates having (hypothetical) sinusoidal wear tracks, as shown in figure 6.4.18. The leakage flow per unit plate width, q_w, was calculated as

$$q_w = \frac{1}{12\mu} \cdot \frac{dP}{dr} \cdot \left[h_o^{\,3} + \frac{3h_o y_o^{\,2}}{2} \right] \qquad 6.4.9.1$$

compared to the case of flat plates where

$$q = \frac{1}{12\mu} \cdot \frac{dP}{dr} \cdot \left[h^3 \right] \qquad 6.4.9.2$$

Because dP/dr must be the same in both cases to support a given load, the fluid pressure at the clearance entry must also be the same. This pressure will be determined by a restrictor (as, for example, in a

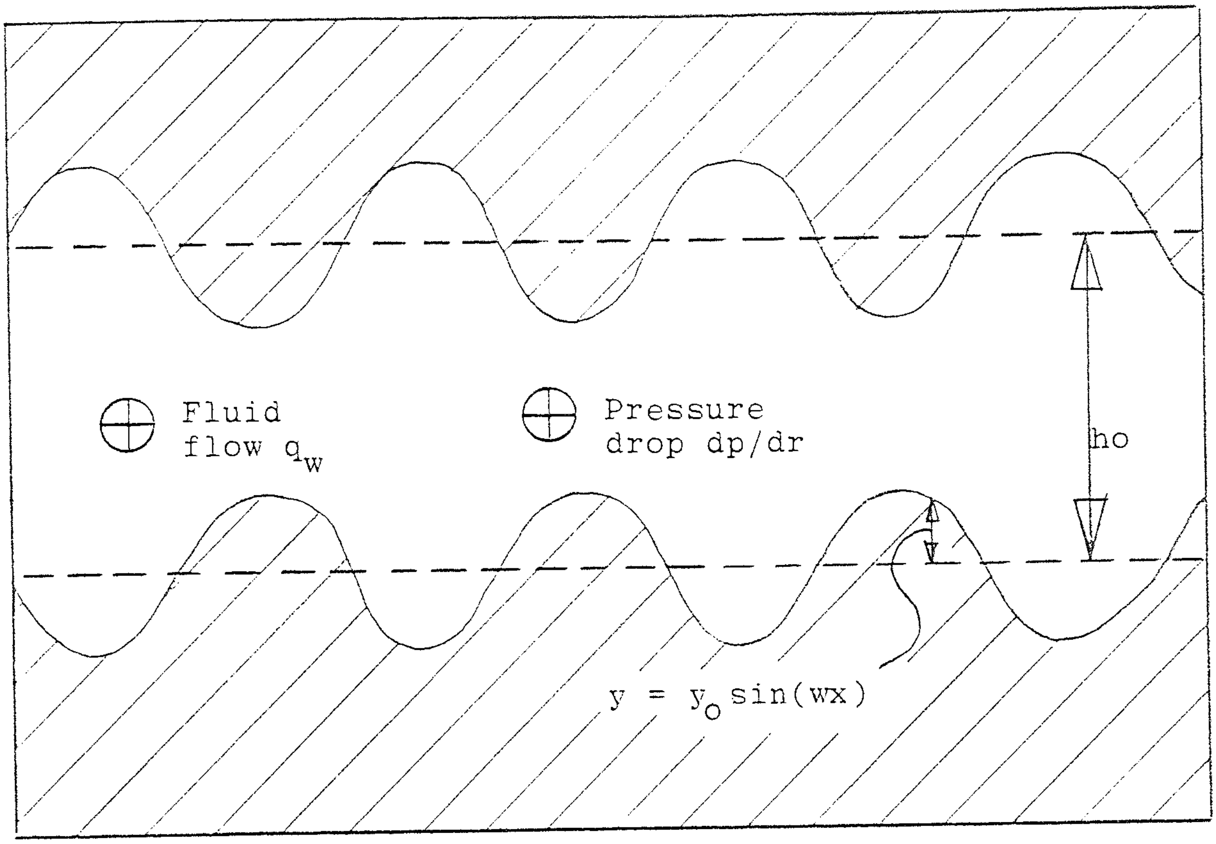


Figure 6.4.18

Fluid flow between plates having sinusoidal wear tracks.

slipper). For clearance inlet pressures to be the same, leakage flow through the restrictor, and hence through the bearings, must be equal whether the plates are flat or worn. Therefore $q_w = q$ and hence

$$h_o^3 + \frac{3h_o y_o^2}{2} = h^3 \qquad 6.4.9.3$$

The conclusions to be drawn are that surface finish will not affect the leakage flow through a hydrostatic bearing, but that it will affect the mean bearing clearance. In fact, for values of h around 6 microns and values of y_o around 1 micron, as was the case in the pumps tested here, h_o would be of the order of 5.9 microns, and surface finish has little influence at all.

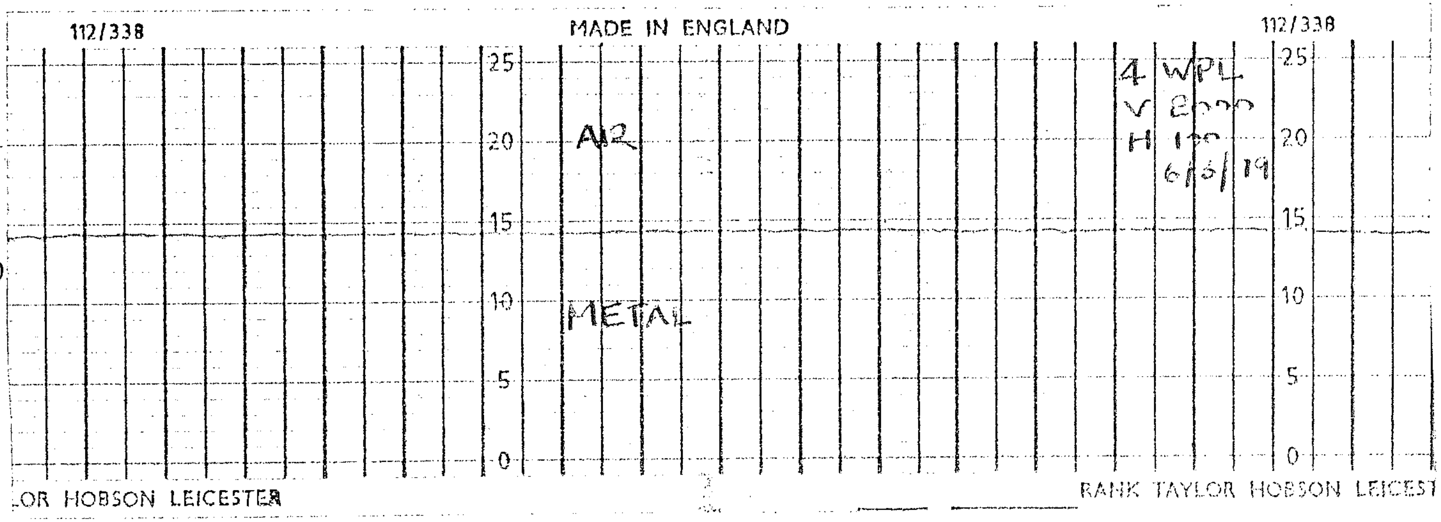
Changes in the form of surfaces might affect pump leakage. Such changes occurred in three pump areas, and were identified by taking surface profile traces. These results will now be described.

(i) Lower face of thrust plate.

Figures 6.4.19 and 6.4.20 show traces taken, at the same location, across the lower face of the thrust-plates of pumps 1, 4, 5 and 6 after tests 1.1, 4.1, 5.2 and 6.1. (Pumps 2, 3 and 7 produced similar traces to those shown by pump number 5.) It can be seen that pumps 1, 4 and 6 suffered little wear in this area, whereas pump number 5 does show a marked wearing effect. It is interesting to note that pump number 4 exhibited a drop in output flow, but did not suffer much thrust-plate wear. Pump number 6 (run on 0-5 micron ACFTD) also shows little wear in this area. It is possible to conclude that wear

Surface traces across lower face of thrust plates of pumps 4 (upper trace) and 5 (lower trace). Pump 4 had been run on 100 mg/l of graphite; pump 5 had run two tests on 100 mg/l of 0-50 micron ACFTD.

Horizontal magnification x100
Vertical magnification x2000



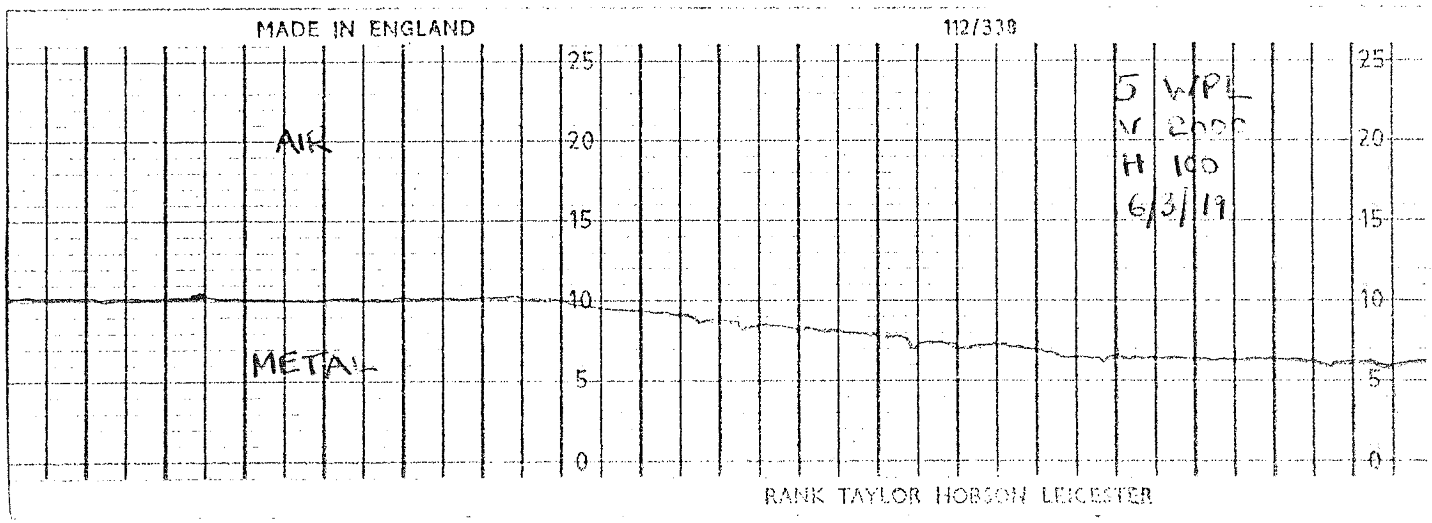


figure 6.4.19

Surface traces across lower face of thrust plates from pumps 6 (upper trace) and 1 (lower trace). Pump 6 had run on 100mg/l of 0-5 micron ACFTD. Pump 1 had been exposed to lower levels of 'natural' system contaminants.

Horizontal magnification x100
Vertical magnification x2000

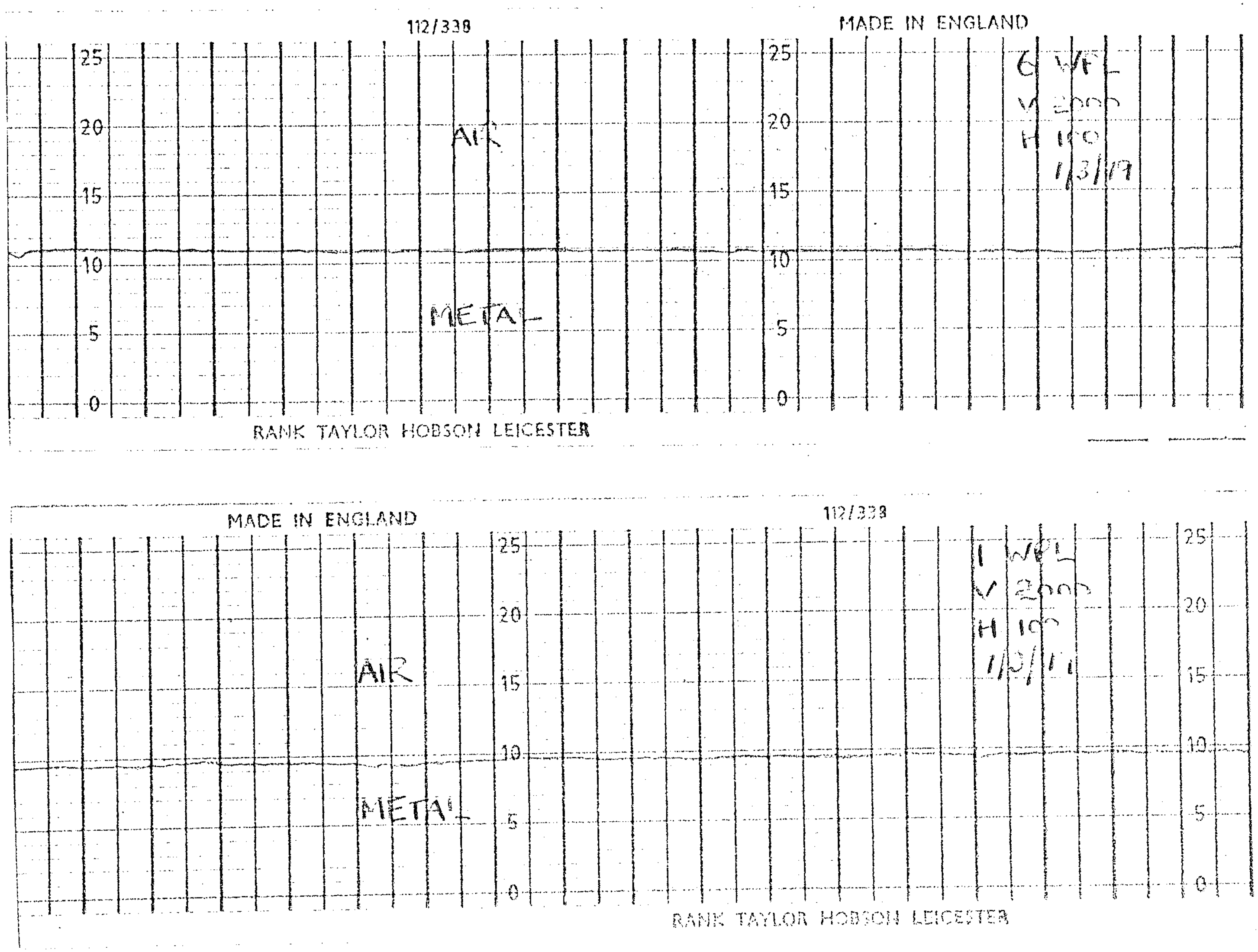


figure 6.4.20

in this area was produced by large, abrasive particles, but that this wear was not entirely responsible for measured reductions in pump output flow.

(ii) Upper face of thrust-plate.

Figures 6.4.21 and 6.4.22 show traces taken across the upper face of pump thrust plates at a location beneath the slipper position. It can be seen again that pump number 5 suffered considerable wear in this area. Pump number 4 suffered some wear, pumps 1 and 6 very little at all. The conclusion is, again, that wear in this region was caused primarily by larger abrasive particles.

The wear exhibited by both upper and lower faces of the thrust-plates of pumps run on 0 -50 micron ACFTD may be described as a "dishing" of the surface. (Note that the scales used on surface traces magnify this effect.) The possible significance of this effect was considered by calculating the leakage through a flat slipper bearing onto a dished plate, as shown in figure 6.4.23.

It can be shown that the pressure profile across a slipper is given by

$$P_r = \frac{6\mu q}{\pi h^3} \ln\left(\frac{Ro}{r}\right) \qquad 6.4.9.4$$

where all symbols have the same meanings as in chapter 4. A plot of $\ln(Ro/_r)$ for typical slipper dimensions also showed that this relation may be approximated by a straight line.

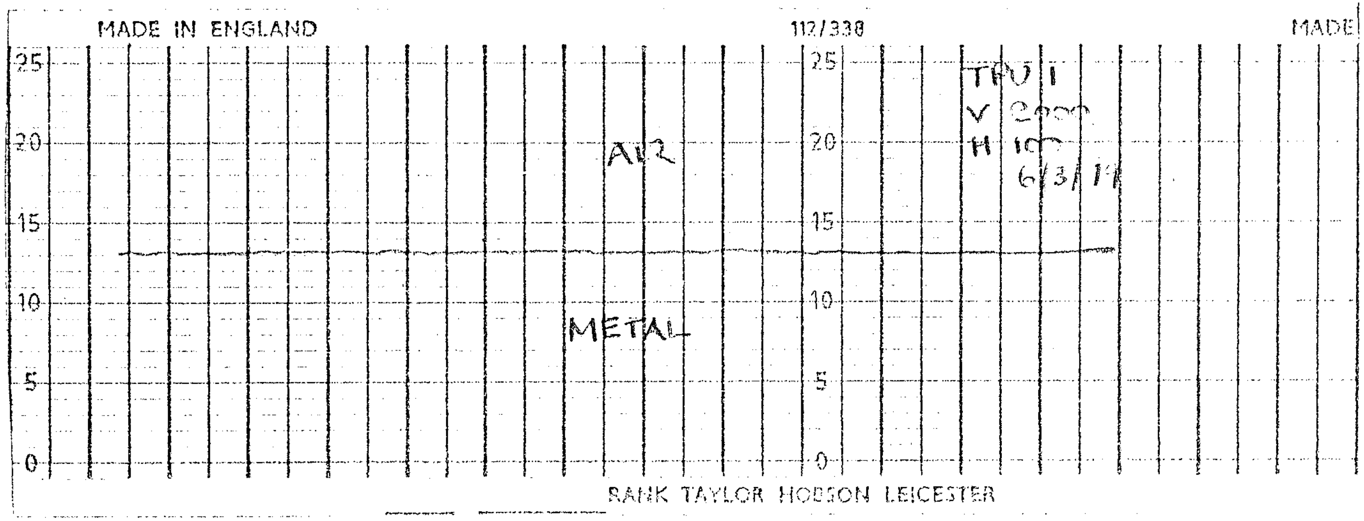
If the dishing shown in figure 6.4.23 is modelled by the equation

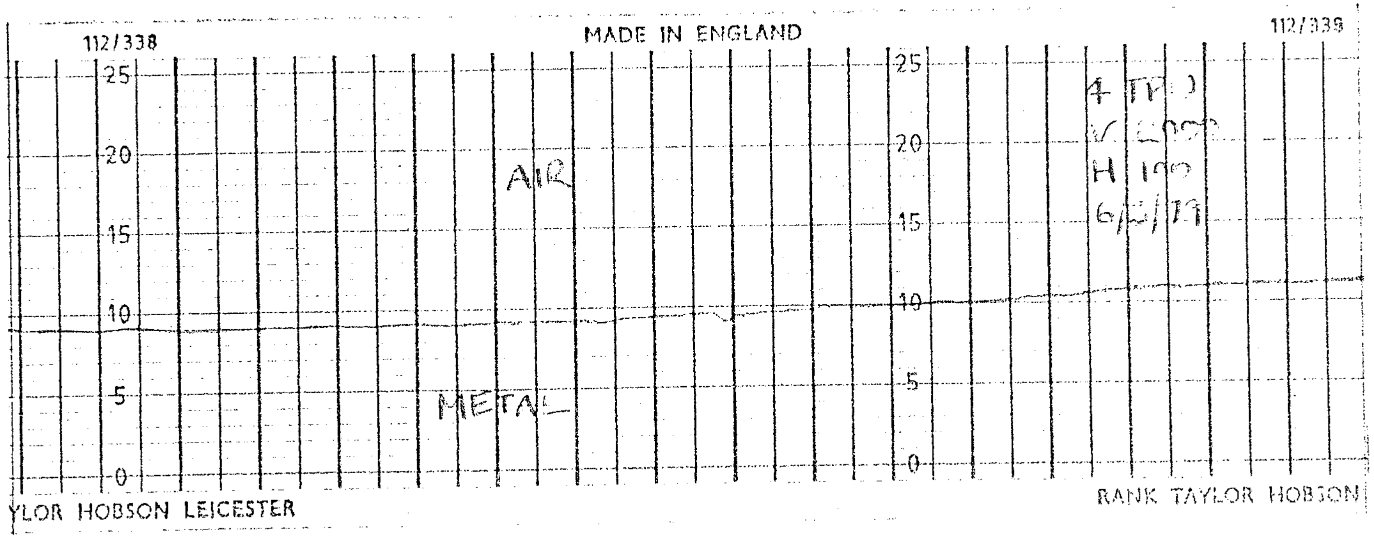
$$h'^3 = ho^3(1-\alpha r) \qquad 6.4.9.5$$

Surface traces across upper face of thrust plates from pumps 1 (upper trace) and 4 (lower trace). Pump 1 had run on relatively low levels of 'natural' contaminants. Pump 4 had run on 100 mg/l of graphite contaminant.

Horizontal magnification x100
Vertical magnification x2000

figure 6.4.21





Surface traces across upper face of thrust plates of pumps 5 (upper trace) and 6 (lower trace). Pumps had been exposed to two tests on 0-50 micron ACFTD at 100 mg/l (5), and one test on 100 mg/l of 0-5 micron ACFTD (6).

Horizontal magnification x100
Vertical magnification x2000

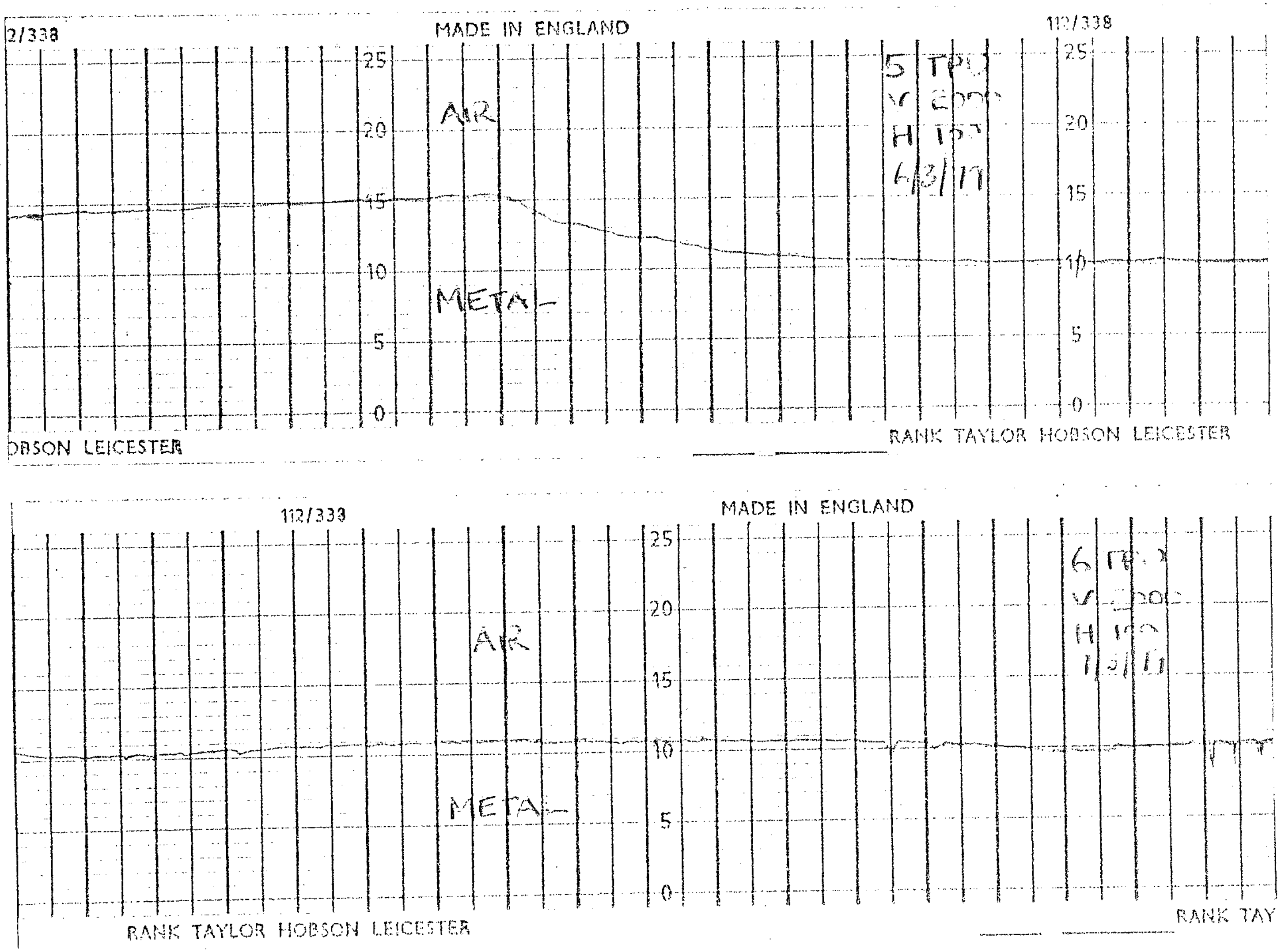


figure 6.4.22

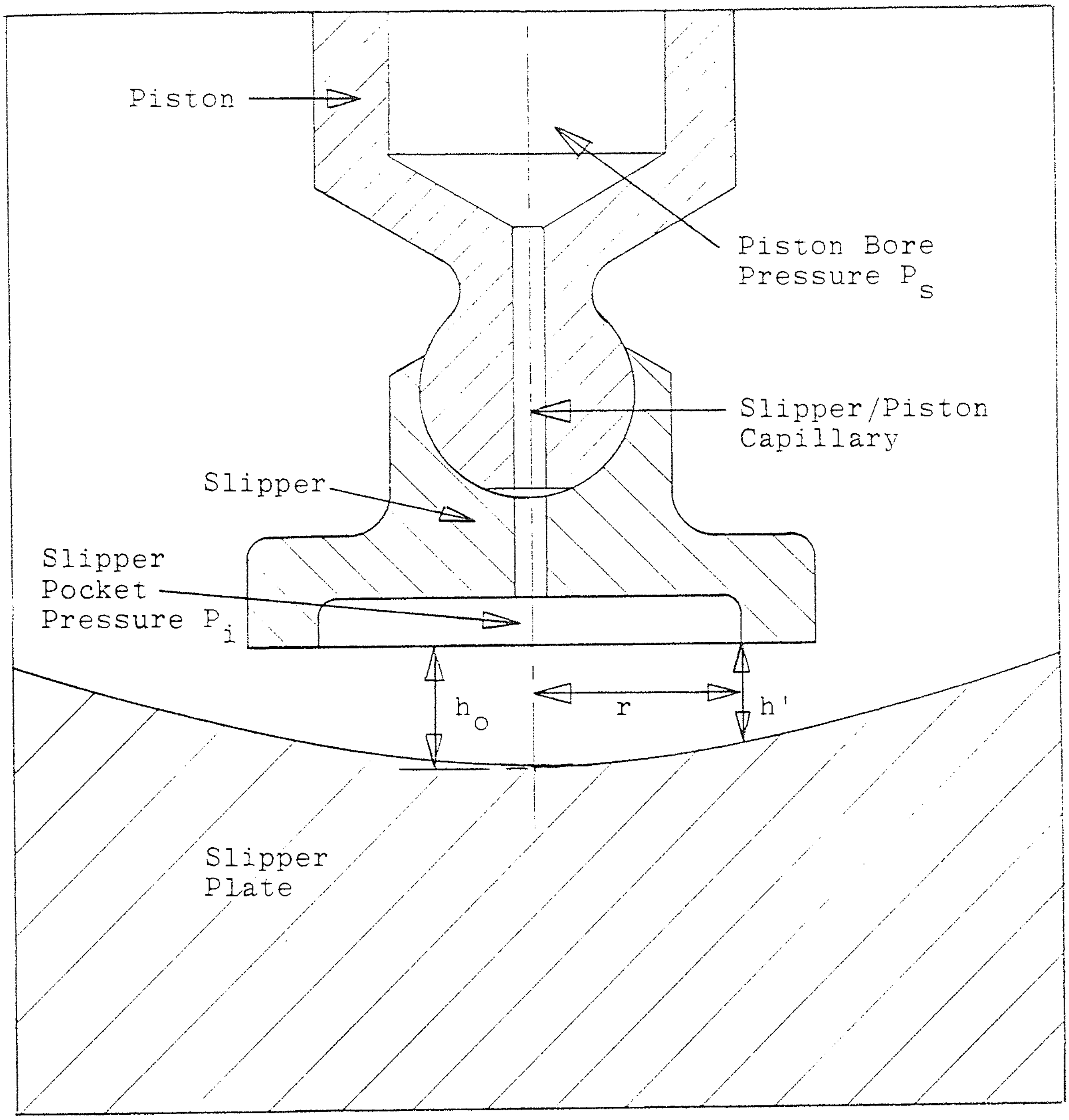


Figure 6.4.23

Lubrication of a Flat Slipper on a "dished" Slipper-Plate then

$$P_r = \frac{6\mu q}{\pi h_o^3} \ln\left\{\frac{r(1-\alpha Ri)}{Ri(1-\alpha r)}\right\} \qquad 6.4.9.6$$

For the amount of curvature shown by the A70 components examined, equation 6.4.9.6 can also be modelled as a straight line variation of pressure. This again means that the load carrying capacity of a slipper on a dished plate will be determined by the bearing pocket pressure P_i. To support the same loads, the slipper on a flat plate and the slipper on a dished plate must have the same inlet pocket pressure, and this again requires that they have the same leakage flow.

The conclusion to be drawn is that the change of surface form present cannot account for the increased leakage.

(iii) Cylinder block portplate face.

The third area where wear had obviously occurred was on the portplate face of the cylinder block. Surface finish deteriorated in this area in all pumps exposed to 0-50 micron ACFTD, and in the pump run on graphite. Although this was a general deterioration, the major effect was the wearing of a pronounced groove on the cylinder block face. This groove can just be seen in the photograph shown as plate 6.4.16. It is clearly visible in the surface finish traces shown in figures 6.4.24 and 6.4.25. The groove corresponds to the location of the outside land of the portplate shown in figure 6.4.26.

It is possible to make some deductions about the cause of this wear, and about its importance for

Surface traces across cylinder blocks of pumps 4 (upper trace) and 7 (lower trace). Pump 4 had run on 100 mg/l of graphite. Pump 7 had been exposed to 300 mg/l of 0-50 micron ACFTD.

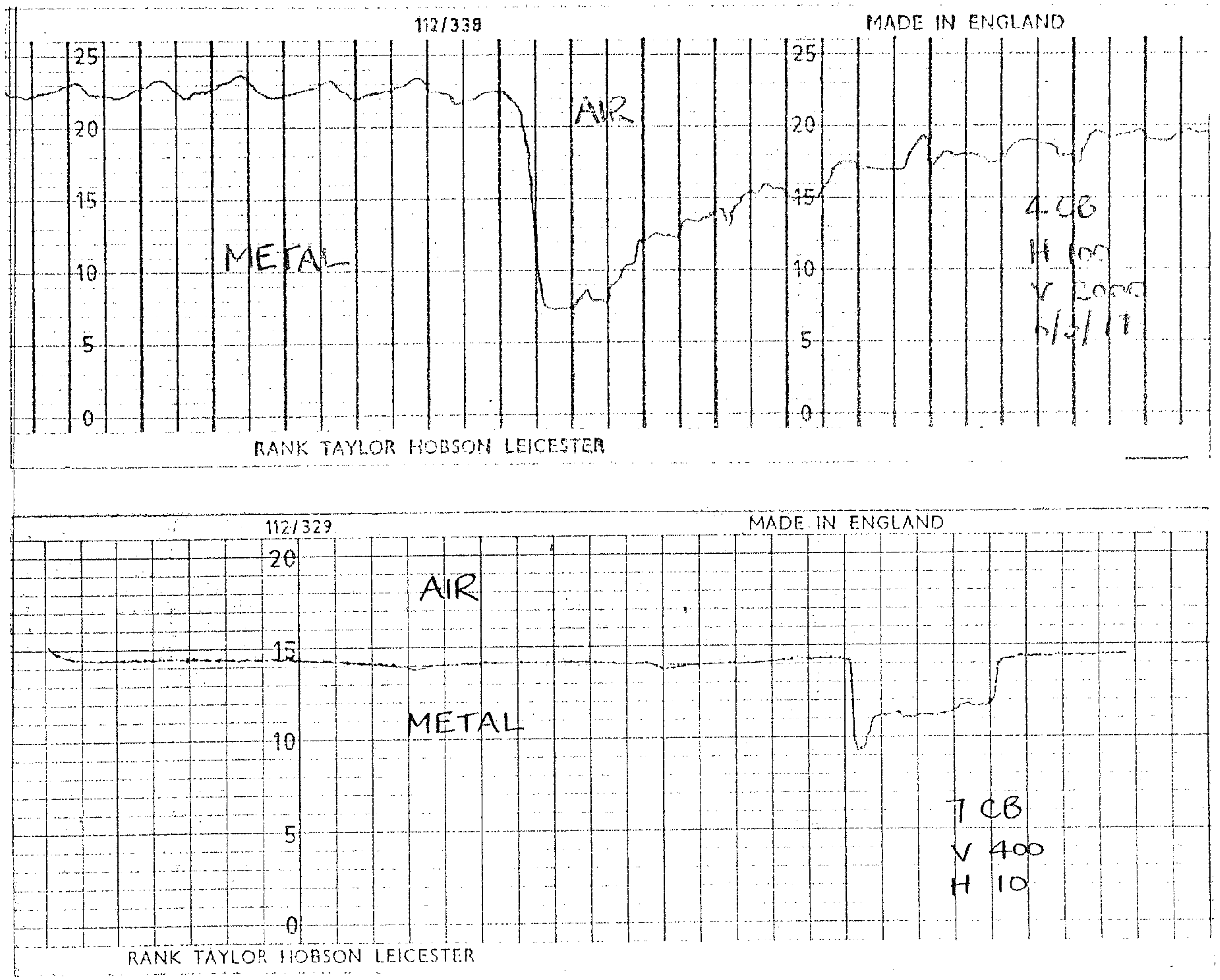
Upper trace:

Horizontal magnification x100
Vertical magnification x2000

Lower trace:

Horizontal magnification x10
Vertical magnification x400

figure 6.4.24



Surface traces across cylinder block of pumps 6 (upper trace) and 1 (lower trace). Pump 6 had run on 100 mg/l of 0-5 micron ACFTD. Pump 1 had run on relatively low levels of 'natural' contamination.

Horizontal magnification x100
Vertical magnification x2000

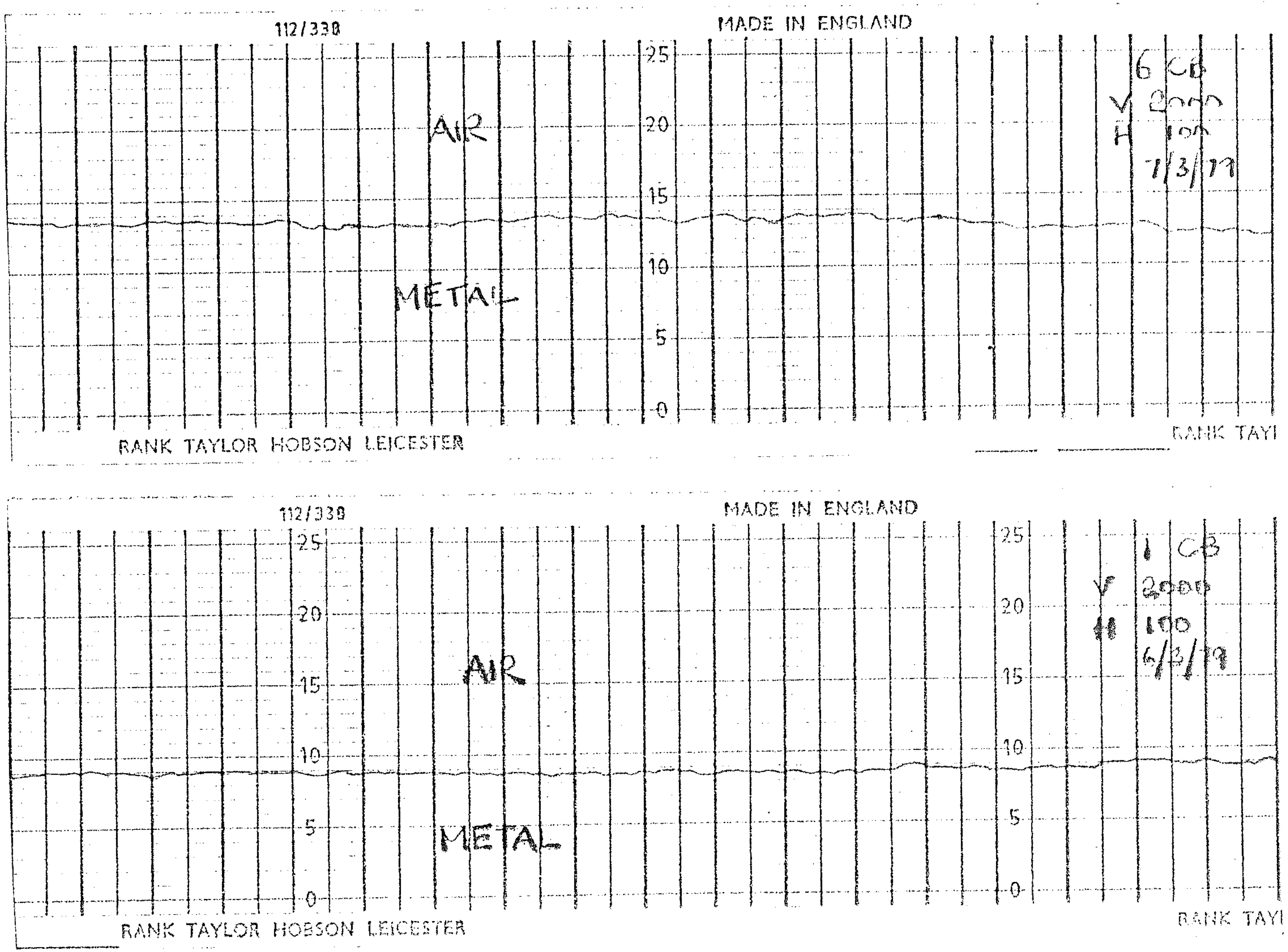


figure 6.4.25

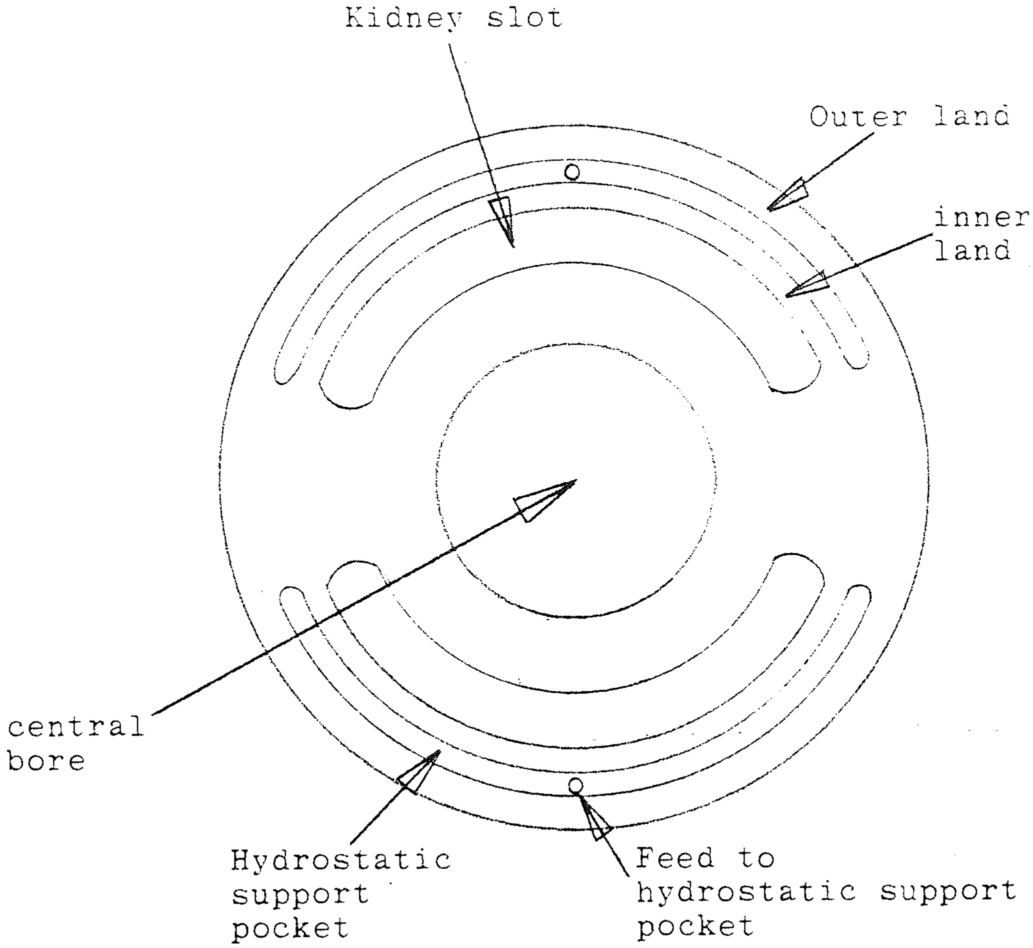


Figure 6.4.26 Portplate of A70 pump.

leakage. Because the groove appeared in pump number 4, which ran on graphite, the wear cannot have been abrasive. It was more probably due to silting of the hydrostatic support pocket shown in figure 6.4.26. This would destroy the force balance on the portplate, and could allow wear to take place. The groove is also probably responsible for much of the increased pump leakage observed. This is because it was the only drastic change which occurred in <u>all</u> the pumps which suffered a significant fall in output flow.

The surface finish traces indicate that, once the groove had been worn, the surface effective clearance at the outside land of the portplate would have increased from around 5 microns to 30 microns. Modelling the land as a pair of parallel plates, the leakage flow would be given by q_p, where

$$q_p = \frac{P_i}{b} \cdot \frac{wh^3}{12\mu} \qquad 6.4.9.7$$

where w is the bearing width, b is the bearing depth, and P_i the pressure at the hydrostatic pocket.

Assuming (realistically) that the A70 pump was designed to operate at 5 microns clearance, and that P_i in the tests would be around 1400 psi, this would imply an increase in leakage from 1.17×10^{-3} gpm to 0.25 gpm. This last figure represents 7% of the nominal pump output, and is of the same order of magnitude as the measured reduction in pump output flow.

The presence of the groove can also explain other findings. Once the groove was worn, two things would tend to happen.

(a) Any increase in groove depth would not tend to increase leakage, because this portplate area would already offer little resistance to flow and leakage would then be determined by the restrictor in the hydrostatic support. This could explain why, even though pump 7 suffered most damage (see table 6.4.8 and figure 6.4.8.) it exhibited no significantly bigger reduction in pump output flow than pumps 3, 5 and 4.

(b) Once the clearance in this area had risen to 25 microns, the majority of potentially damaging particles would pass through the portplate without producing wear. This would explain why, once a pump had been damaged in this way, it seemed to be immune to further operation on contaminated fluids (but see the Ferrography results quoted below).

It seems fairly clear that the cylinder block/ portplate bearing is one area of the A70 pump sensitive to fluid contamination. A final attempt was made to verify this suggestion by rebuilding and retesting pump number 5. This work will now be described.

6.4.10 Pump component changes

An attempt was made to identify through performance measurements those pump components which had experienced wear which could be detected as a decrease in pump delivered flow. The approach adopted was to run pump number 5, which had undergone two tests at 100 mg/l of 0-50 ACFTD, and replace various components by parts from pump number 1, representing a nominally undamaged pump.

The results of the exercise are summarised in table 6.4.9. It would be unwise to make too many deductions on the basis of these results, but they do indicate that the main sources of increased leakage in damaged pumps lie in the cylinder-block and thrust plate. This agrees with the results and discussion of section 6.4.9. It will also be remembered that pump number 4 (run on graphite) experienced no significant thrust-plate wear compared to that suffered by pump number 5, (See figure 6.4.5.), and that pump number 5 underwent a greater reduction in output flow, so it seems probable that extra leakage in pump 5 was largely from the thrust-plate/swashplate bearing.

6.4.11 Conclusions on the effects of contamination on the A70 pump.

On the basis of the work described above, it is possible to state that:

(i) Operation on 100 mg/l of <u>fresh</u> contaminant produced severe damage in the A70 pump, except where this contaminant was of a small size (0-5 micron).

(ii) At least some of this damage could be detected as a reduction in pump delivered flow.

(iii) The major areas of damage seem to be the port-plate face of the cylinder block, and the lower face of the thrust-plate.

(iv) The portplate bearing is damaged by operation on both hard and soft contaminants of a Stoke's diameter above 5 microns; the thrust-plate face seems to be affected only by operation on abrasive contaminants.

(v) Damage seems to occur primarily in areas which

COMPONENT(S) REPLACED	PUMP OUTPUT FLOW (UNITS) AT 1500 RPM INPUT SPEED
NONE	84.8
CYLINDER BLOCK	85.8
THRUST PLATE	86.1
PISTON/SLIPPERS	85.0
CYLINDER BLOCK THRUST PLATE PISTON/SLIPPERS	88.4

Table 6.4.9 Performance of pump number 5, after exposure to two tests on 100 mg/l 0-50 micron ACFTD, with component replacement from pump number 1.

rely on hydrostatic lubrication.

(vi) Damage to the cylinder-block face is sufficiently severe to suggest that major seizure might follow operation on high levels of contaminated fluids.

6.5 Ferrographic Analysis

Although the analysis described in sections 6.4.9 and 6.4.10 gave valuable information about the effects of contaminants on the A70 pump, it left several questions unanswered. For example, did operation on graphite produce the same type of wear as was produced by operation on ACFTD? In addition, the drawbacks of dismantling pumps for examination have also been mentioned. These problems are made worse when the pump is in service rather than under test.

One way to obtain more data is to analyse wear particles from fluid samples. This approach has several potential advantages for equipment in service. It can be used with operating equipment, is fairly cheap, and can give advanced warning of impending failures.

To assess the usefulness of the approach, and to obtain test data, samples of fluid taken from tests 1.1, 2.1, 3.1, 4.1, 5.1, 5.2 and 6.1 were analysed by Ferrography. Ferrography is described in appendix A, but it is basically a means of preparing a microscope slide of wear particles (with some other particles as well). An experienced operator, examining such a slide, can determine, from the size, number and characteristics of the particles present, what types of wear have been occurring in the system from which the sample was taken. He can also estimate the severity of the wear.

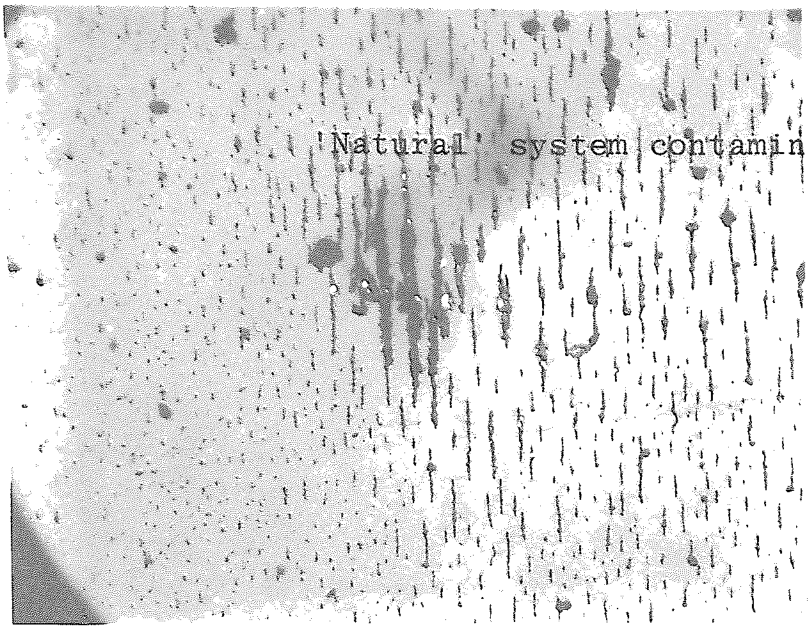
Because neither CHL nor Aston University had much experience in this field, the author took fluid samples to the Tribology Centre at the University College of Swansea, where he assisted staff in preparing and analysing Ferrography slides. There would be little point in describing the appearance of every slide examined. Instead, several specific findings will be presented and discussed.

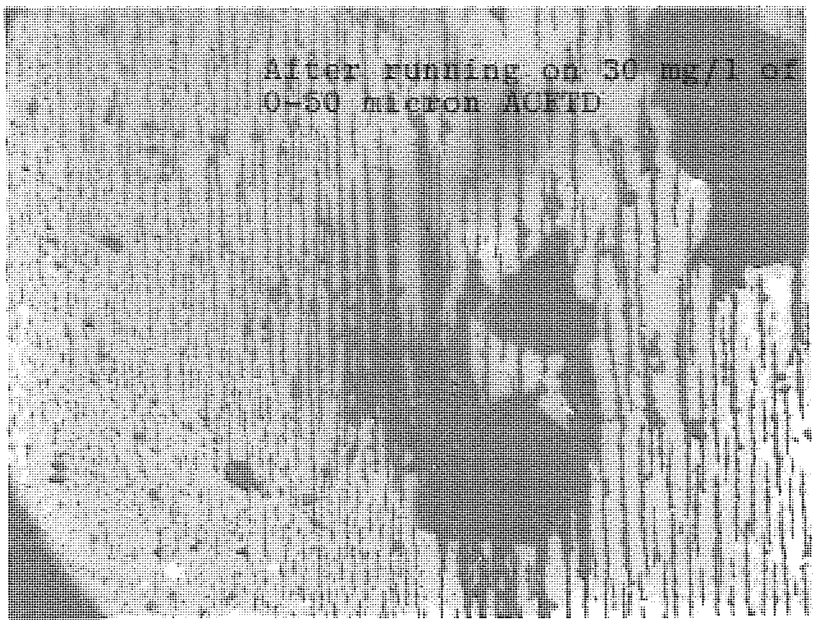
(i) Without prior knowledge of the test conditions relating to a specific fluid sample, the staff at Swansea reported a change in the appearance of a Ferrogram whenever there had been a corresponding change in experimental conditions. This finding supports the usefulness of Ferrography as a diagnostic tool. It also suggests a possible use of pump contamination tests. This would be to build up a reference library of Ferrograms relating to known conditions. These could then be used to diagnose contamination-related problems in service.

(ii) In all cases, fluid samples taken later in a test showed a higher concentration of wear particles than did samples taken in the initial test period.

(iii) Samples from test 5.1 showed a higher concentration of wear particles than did samples from test 3.1, which in turn showed a higher concentration than did samples from test 2.1. This agrees with the levels of output flow reduction produced in each test.

Plate 6.5.1. Wear particles (mag x400)





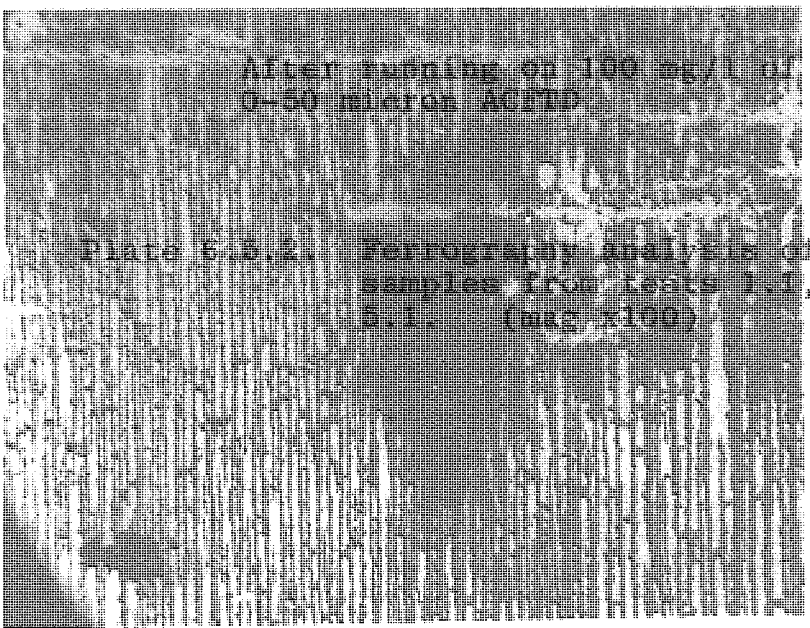
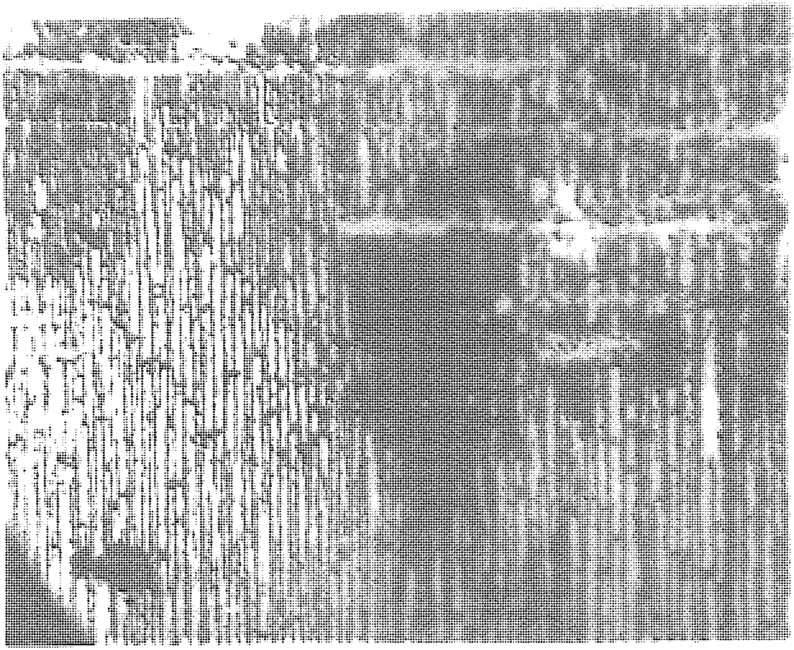
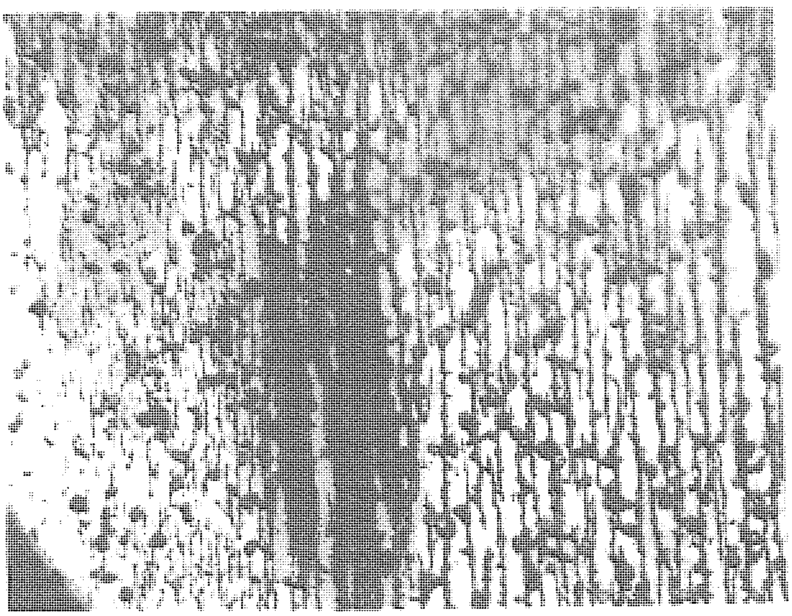


Plate 6.5.2. Ferrography analysis of fluid samples from tests 1.1, 2.1, and 5.1. (mag x100)





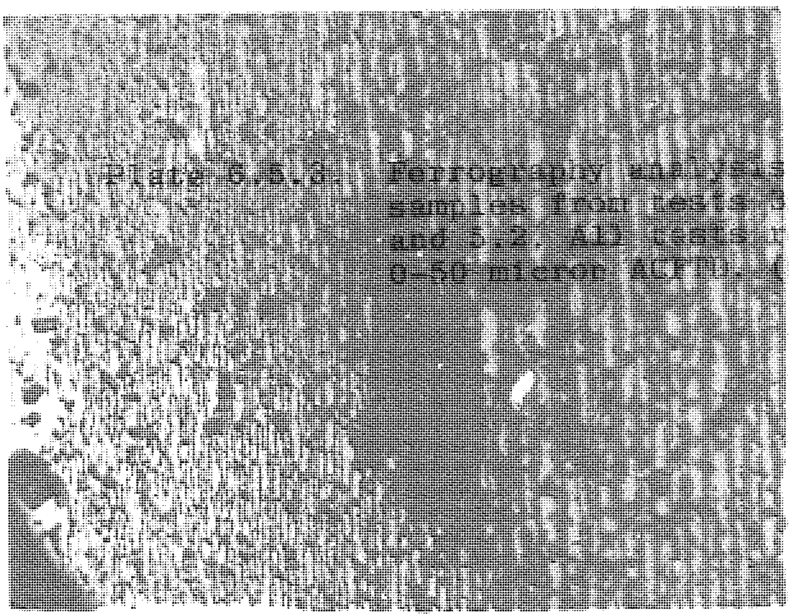
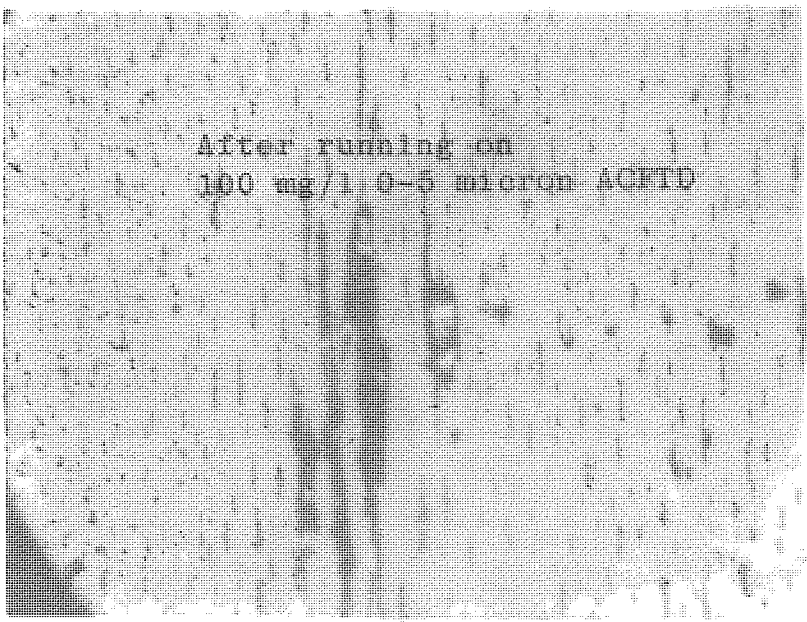
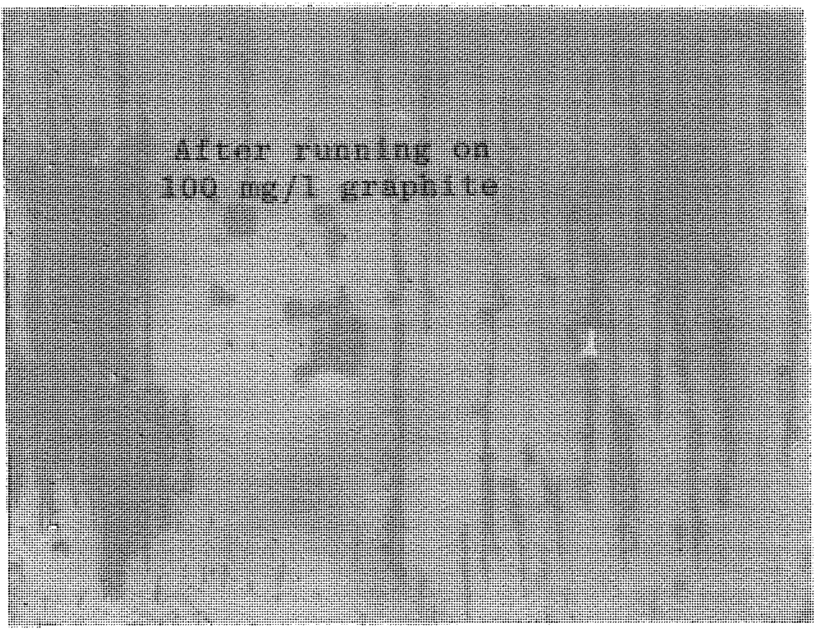


Plate 5.5.3. Ferrography analysis of fluid samples from tests 3.1, 5.1, and 5.2. All tests run on 100 mg/l 0-50 micron ACFTD. (mag x100)





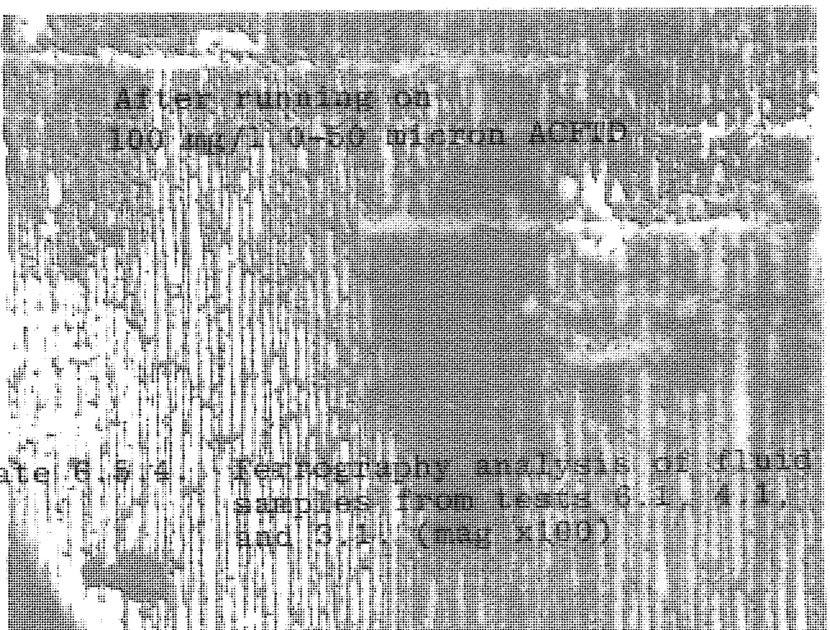


Plate 6.5.4. Ferrography analysis of fluid samples from tests 6.1, 4.1, and 3.1. (mag x100)

(iv) Samples of fluid from all tests run on ACFTD showed the presence of silica particles. The majority of wear particles present were indicative of "cutting" or "abrasive" wear. Typical particles are seen in plate 6.5.1.

(v) Particles taken from test 1.1 were of a lower concentration and generally of a different type to those from tests run on ACFTD, and were indicative of normal "running-in" wear. Plate 6.5.2 gives an indication of the concentration of particles in samples from tests 1.1, 2.1 and 5.1. (The majority of fluid samples showed some "running-in" wear particles, suggesting that the test running-in process should be extended in any future work.)

(vi) Plate 6.5.3 shows slides taken during tests 3.1, 5.1 and 5.2, all of which were run on 100 mg/l of 0-50 micron ACFTD. The slides show the repeatability of the Ferrography technique. They also show that substantial wear was taking place in test 5.2, even though pump output flow in this test did not fall below the level of test 5.1. The conclusion to be drawn is that output flow is not a reliable indicator of the degree of wear which is taking place in a pump.

(vii)Plate 6.5.4 shows slides prepared from samples taken in tests 6.1 (0-5 micron ACFTD), 4.1 (100 mg/l graphite) and 3.1 (0-50 micron ACFTD). The differences are obvious. Significantly, wear particles produced by the graphite were completely different to the majority of those produced by ACFTD, being indicative of a "surface-fatigue" type of failure. The majority of particles from the test run on 0-5 micron ACFTD were, again, not typical of abrasive wear.

As shown above, Ferrography can certainly give improved insight into what has been happening during a pump contamination test, and it must have considerable potential for use as a valuable diagnostic aid to monitor equipment in service. It can also give increased understanding of the general effects of fluid contamination. For example, the wear particles shown in plates 6.5.1 to 6.5.4 are predominantly metallic. Examination of plate 6.5.2 shows that wear particle concentrations in samples from tests run on contaminated fluids are much higher than in samples from test 1.1, run on "uncontaminated fluid". As these particles are metallic, they cannot have come from the artificial contaminants added, and the conclusion to be drawn is that contamination is both a cause of wear, and a symptom of it. Of course, if new particles produce wear themselves, an unstable condition can develop. Using the symbols of chapter three, this can be expressed as

$$R = R' + \emptyset(A) \qquad 6.5.1$$

Looking at plate 6.5.2, which shows particle concentration produced at different contaminant levels, it is feasible that

$$R = R' + \rho A(t) \qquad 6.5.2$$

where R' is particle ingression from the environment and ρ is a constant for the pump's generated contaminant.

If this relation is substituted in the analysis of chapter 3 the result obtained is that:

$$A(t) = \frac{R'}{Q_f \eta - \rho} \left[1 - e^{\left\{ \frac{-(Q_f \eta - \rho)t}{v} \right\}} \right] + A_o e^{\left\{ \frac{-(Q_f \eta - \rho)t}{v} \right\}} \qquad 6.5.2$$

Unless $Q_f \eta > \rho$, the contamination level in such a system will never stabilise.

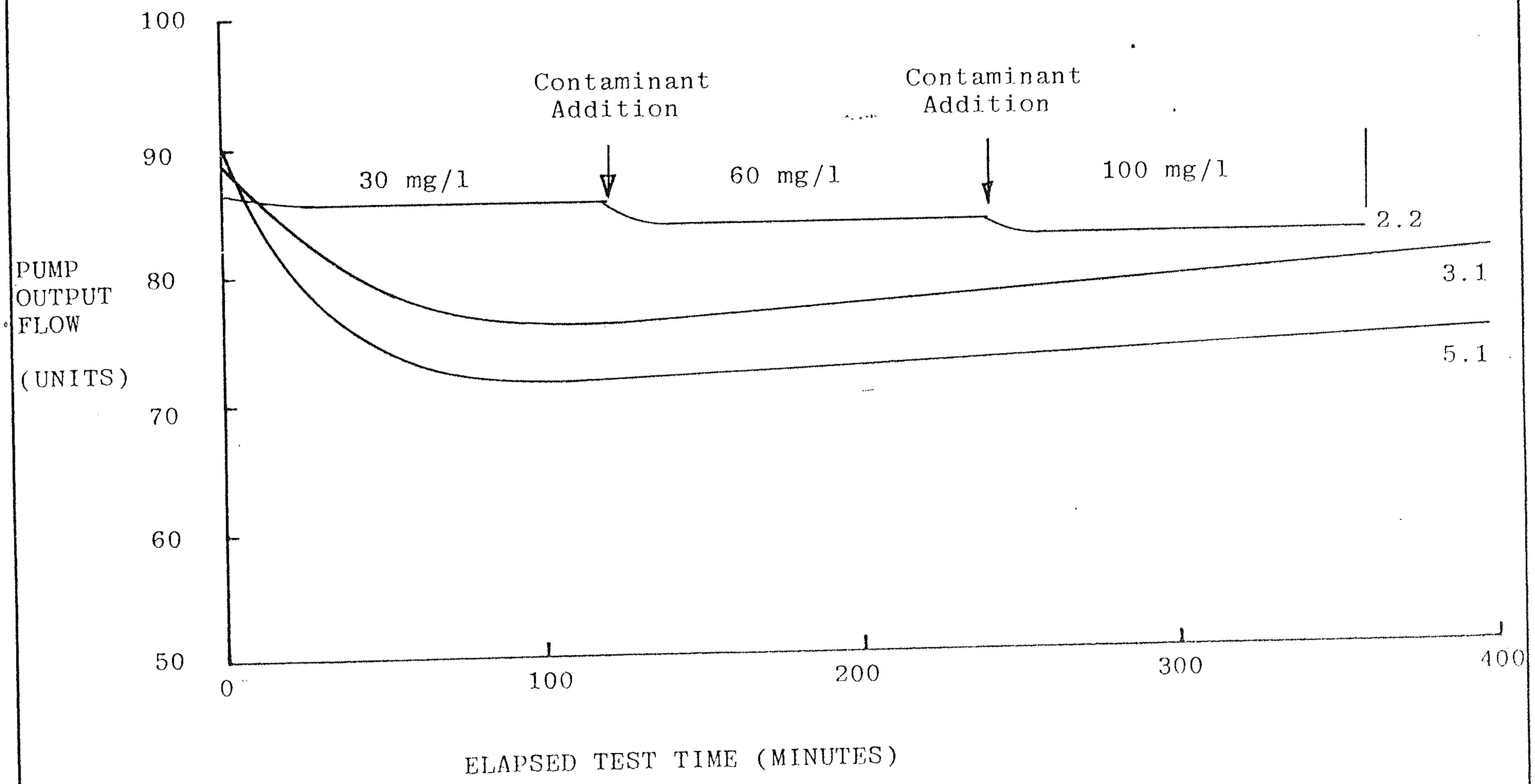
No further Ferrography analysis was conducted, but one more pump test was run, to examine a suggestion presented in section 6.2.2.

6.6 The Effect of Contamination Build-up on Test Results.

In section 6.2.2. it was shown that, if the OSU theory about particle degradation is correct, then the manner in which a contamination level is achieved will partly determine the effects of that contamination. This was felt to be important enough to require testing. Pump number 2 was therefore subjected to a test (2.2) on 100 mg/l of 0-50 micron ACFTD. However, instead of injecting the test contaminant in one batch, the final level was achieved gradually, by injecting batches of contaminant at two hour intervals. The theory was that the output flow drop produced by such a procedure would be less than that produced by a single addition of contamination.

The test results are shown in figure 6.6.1 and it can be seen that the expected result was obtained. The degradation of output flow exhibited by pump 2 was less than that shown by pumps 3 or 5. However, the difference may not be significant. It would be worth extending this aspect of the work in future.

Figure 6.6.1
Effect of build-up of contaminant (0-50 micron ACFTD) in test 2.2, compared to previous test results. (See section 6.6.)



6.7 General Conclusions about the use of Pump Contamination Testing.

The experimental work described in this chapter could easily have been extended. For example, pump surfaces could have been examined by electron microscope to obtain more information about the type of wear which had been taking place. However, enough information had been gathered by this stage to satisfy the research objectives. These were described at the beginning of this chapter, and will now be considered in relation to the results of the work.

(i) How should a pump contamination test be conducted?

The work shown in this chapter leads to the conclusion that a pump contamination test must be a relatively short, multipass procedure. It should be conducted at a contamination level of 30-300 mg/l, and must be run on artificially contaminated fluid. The contaminants used must be selected to produce the type of problem to be studied, or to allow useful deductions to be made about the effects of different contaminants.

(ii) What output data can be obtained from a pump contamination test?

The amount and types of data obtained are limited only by the experimental techniques available to the researcher. There is no doubt that some contamination-induced damage can be detected as a drop in pump performance. However, such external parameters can only give limited information. Fluid analysis and the examination and measurement of pump components can greatly increase understanding of the effects of fluid contamination on a pump. One must also choose any external performance parameters to monitor the most likely cause of pump failure. For example, pump output flow would not identify a potential bearing failure. It has also been shown that output flow is an unreliable indicator of pump component wear.

(iii) For what purposes may pump contamination test results be used?

There are five possible uses.

(a) Predicting pump "life" or reliability.

This is not feasible. The work described in this chapter merely reinforces the conclusion of chapter five.

(b) Assigning pump contamination ratings.

This possibility was considered at some length in section 6.4.7. The overall conclusion reached was that it is not yet feasible, but that it could become so with further work.

(c) To identify pump areas susceptible to fluid contamination and to suggest improvements in these areas.

The work described in this chapter indicated fairly clearly that this is a feasible use of contamination test results. The A70 pump can suffer damage to the cylinder-block/portplate bearing through operation on high levels (100 mg/l) of both hard and soft contaminants. In addition, the lower face of the thrust-plate is affected by abrasive contaminants. Possible improvements would be to redesign the portplate and use a much harder thrust-plate material.

(d) To improve general understanding of the effects of fluid contamination.

The work presented in this chapter has certainly helped to increase general knowledge of fluid contamination and of its effects.

(e) To determine the priorities of future work.

Again, this is a practical use of contamination test results. If more basic theoretical and experimental work were to follow from this research, it should probably be concentrated on situations similar to the A70 portplate outer land and hydrostatic support pad.

The overall conclusion to be reached is that pump contamination tests can give useful output data. It must be mentioned that it is not possible to devise a completely realistic test and it therefore remains to be shown that the effects produced in such a test are those most frequently encountered in service. The question to be answered by CHL is whether or not the type of test described in this chapter can provide enough useful information to make a test programme on CHL pumps economically worthwhile. This question is considered in chapter 9.

7.0 CONTAMINATION TESTING FOR A SPECIALISED APPLICATION

7.1 Introduction

Two of the limitations of the type of generalised pump contamination test described in chapter 6 are that the test contaminants used, and the test profile in terms of pump speed, pressure, flow, etc. are unlikely to represent any one, specific pump application. This does not invalidate the results of such tests and it can still be very useful to know, for example, that a particular design of pump is likely to suffer severe portplate damage if the pump is operated on fluid containing abrasive contaminants. However, a pump manufacturer will often supply a large proportion of his output for use in one type of system. In these cases, a more specialised contamination test may be both desirable and feasible.

CHL were in this position at the start of this research, when the company's main product was a hydrostatic steering transmission for a military vehicle. Information was required on the likely effects of fluid contamination on the steering unit, and a specialised contamination test was therefore devised and implemented. The main considerations involved in setting up the test were the level of contamination likely to be encountered in service, the nature of this contamination, and the test profile. These aspects of the work will be considered in turn, but the chapter will commence with a brief description of the hydrostatic transmission, and of the reasons why a fluid contamination problem was anticipated. The chapter also contains a section (7.4) describing a series of filter tests conducted in connection with the work. The chapter closes with a discussion of the test results obtained, and of the usefulness of this overall approach.

7.2 Closed-Circuit Hydrostatic Transmissions

In many mechanical engineering devices, a variable-speed drive must be obtained from the output of a prime mover. Where the prime mover is a petrol engine, output speed can be varied by a throttle. However, this does mean that, even with a gearbox, the engine usually operates away from its most efficient speed. If the prime mover is an electric motor, the equipment required to vary the motor speed will be both expensive and bulky.

In many cases it is preferable to use a hydrostatic transmission, such as the one shown in figure 7.2.1. The prime mover drives the main hydraulic pump at constant speed. However, the pump output flow can be varied over an infinite range from zero to maximum delivered flow, simply by varying the pump swashplate angle. As the hydraulic motor is of fixed displacement, variations in the flow through the motor will cause the motor to rotate at different speeds, and an infinitely variable output speed is thereby obtained.

The system shown is a closed-circuit transmission, in which fluid from the hydraulic motor is returned to the main pump inlet. However, some fluid is lost from the main circuit by leakage in both the main pump and motor, and this fluid has to be replenished by the boost pump. The boost pump also raises the pressure at the main pump inlet, and this improves the system performance.

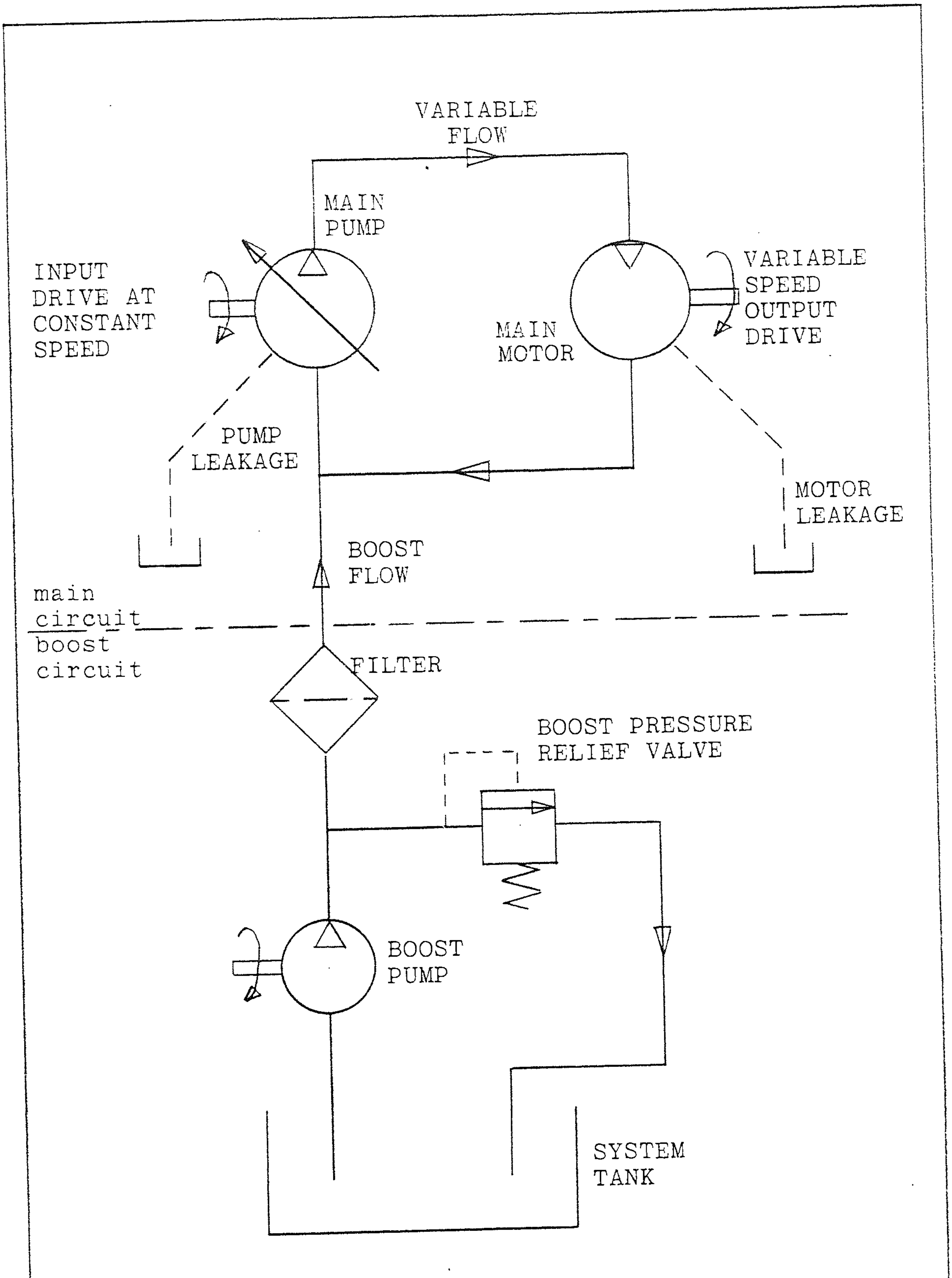


Figure 7.2.1 Simple hydrostatic transmission.
(This circuit is very similar to
CHL's steering transmission).

The transmission designed by CHL was used to steer a tracked, military vehicle, and, as in many military applications, two of the main design constraints were to keep overall size and weight to the minimum acceptable levels. These constraints were responsible for the two design features which created a potentially serious fluid contamination problem.

(i) To save space, the steering transmission used the same fluid, from the same sump, as was used for the vehicle's main gearbox and transmission. It was suspected that this fluid would be heavily contaminated, mainly with particles worn from the 'wet' brake and clutch plates used in the vehicle.

(ii) Again because of space limitations, all circuit filtration was installed in the boost circuit. Very little volume could be devoted to the system filters, so the dirt capacity and the performance of the filters used were fairly low.

The situation was therefore one in which the hydrostatic transmission would draw its fluid from a relatively dirty supply, and where the fluid could probably not be filtered to a recommended level. Both the customer and CHL were anxious to know if the expected fluid contamination levels would be such as to seriously impair the reliability or performance of the steering unit. The customer was also looking for an estimate of the expected unit "life" in the vehicle.

It was decided that a series of tests would be conducted to simulate, as closely as possible, the contamination conditions which would be encountered by the hydrostatic transmission. The first step was to determine how fluid contamination levels in the vehicle would vary with time, and to estimate the maximum contamination level likely to be experienced.

7.3 Fluid Contamination Build-up in a Closed-Circuit Hydrostatic Transmission

In chapter three, an expression (equation 3.2.1) was derived for the variation with time of the contamination level in a simple hydraulic circuit. The circuit model used was shown in figure 3.2.1.

The hydrostatic transmission shown in figure 7.2.1 will have a more complex contamination behaviour than that of the circuit considered in chapter three. This is because any particles generated within the main system circuit, or introduced to that circuit from the boost circuit, can only escape through the leakage paths in the main pump and motor. In the boost circuit, particles can and will be removed by the installed filter. These features imply that contamination levels in the main and boost circuits need not be the same.

To investigate this possibility the hydrostatic transmission was modelled as in figure 7.3.1, showing a continuous interchange of particles between main and boost circuits. For particles of diameter smaller than leakage path dimensions in the main pump and motor, t_a will equal t_b. For larger particles, t_b will have a finite value, but t_a will equal zero. All contaminant addition and removal was assumed to take place in the boost circuit and, under these conditions it was calculated that:

$$B(t) = B_1 e^{g_1 t} + B_2 e^{g_2 t} + R/Q_{B}\eta \qquad 7.3.1$$

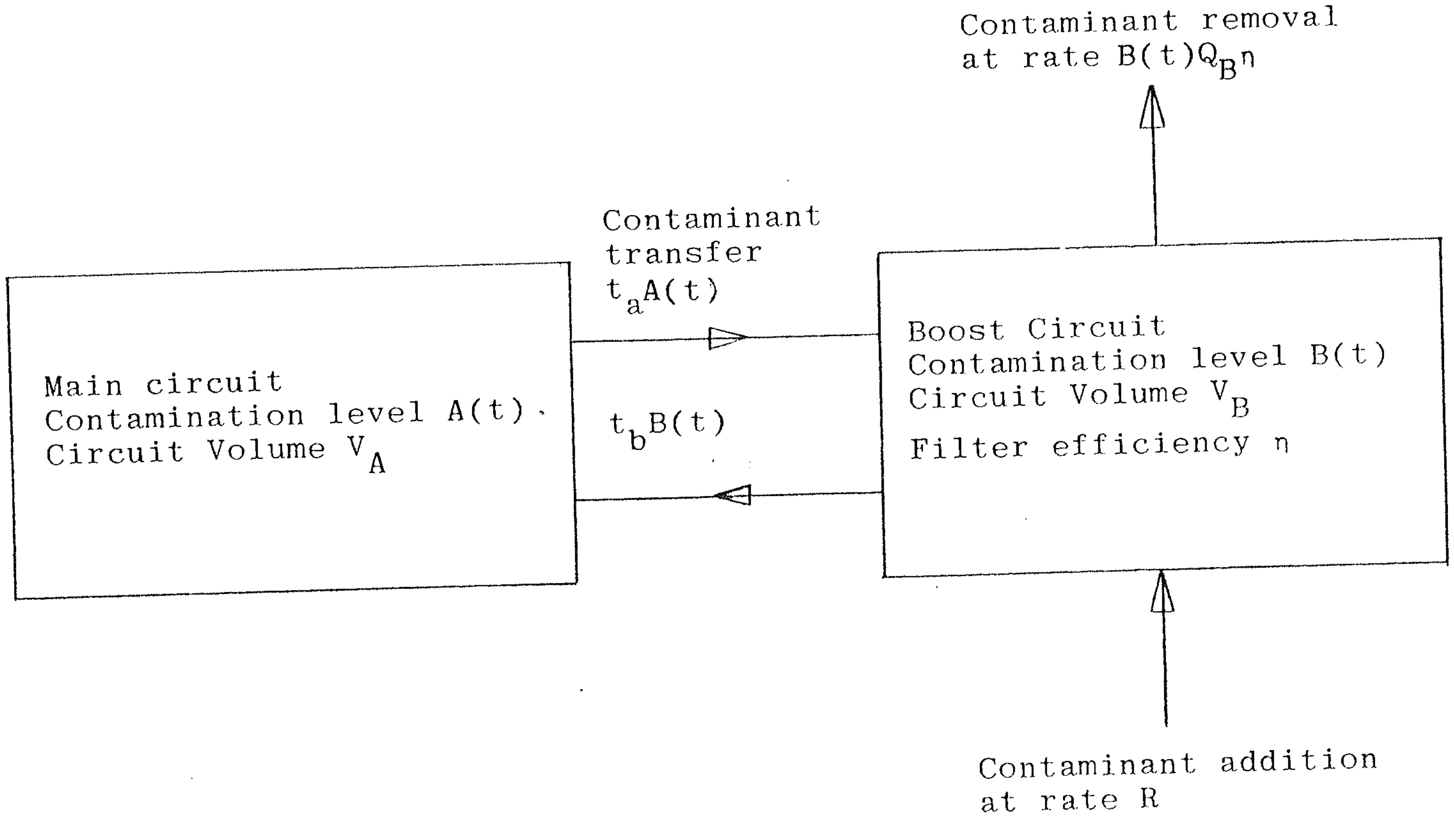


Figure 7.3.1

Contaminant conditions in a closed-circuit hydrostatic transmission.

$$A(t) = \frac{t_b B_1 e^{g_1 t}}{(g_1 V_A + t_a)} + \frac{t_b B_2 e^{g_2 t}}{(g_2 V_A + t_a)} + \frac{R t_b}{t_a Q_B \eta} \qquad 7.3.2$$

where g_1, g_2 are the roots of

$$\frac{V_B g^2}{t_a} + \left\{ \frac{Q_B \eta + t_b}{t_a} + \frac{V_B}{V_A} \right\} g + \frac{Q_B \eta}{V_A} = 0 \qquad 7.3.3$$

and B_1, B_2 are constants. The same assumptions were made about the system ingression rate R, and filter performance η, as were used in the analysis of chapter three.

These formulae could not be used without some quantitative estimate of the parameters involved. This data was easy to obtain, except in the case of filter efficiency. The work conducted on this problem is described in the next section, but, even without quantitative data, the equations give useful information. They indicate that:

(i) Boost circuit contamination levels will stabilize at a level $R/(Q_B \eta)$.

(ii) Main circuit contamination levels will stabilize at a level $R/(Q_B \eta)$ when $t_a = t_b$ (for small particles).

(iii) Contamination levels for larger particles will not stabilize ($t_a = 0$) in the main circuit.

(iv) Contamination levels in the main and boost circuits will not be the same.

7.4 Determining Filter Performance

Some of the problems involved in determining filter performance have already been briefly mentioned. Although the subject of filter testing may seem to be outside the scope of this research, it had to be considered in the work. This was partly to allow quantitative use of the equation in section 7.3, and partly to help in the selection of the best available filter for use in the vehicle.

A survey of filter manufacturers' catalogues showed that no manufacturer publishes efficiency data for his product. Instead, filter performance is usually specified as a "micron rating", which is a very approximate measure of the pore sizes in the filter medium. Two such ratings are in common use. The first is the 'nominal' rating of a filter. This is variously described as the mean pore size, or as the particle size at which the filter is 98% efficient. Neither definition is of much value. A mean pore size is of little use without some knowledge of the standard deviation about the mean, and it is impossible to accurately determine efficiencies above 85% due to the limits on the attainable accuracy of particle counts.

Some manufacturers use an 'absolute' filter rating, which is usually defined as the diameter of the largest, hard glass sphere which will pass through the filter. Absolute filter ratings are typically 2 to 3 times an equivalent nominal rating, and this may be one reason why they are not so popular with filter manufacturers as are nominal values.

Although published filter performance data cannot be used to determine filter efficiencies, it was thought that this might be possible if more information was available on the ways in which filters were tested. A questionnaire was therefore prepared and circulated to most of the major UK filter manufacturers, to determine how they assessed the performance of their products. The questionnaire is shown in appendix B.

Nine completed questionnaires were returned. The results showed two things.

(i) Filter manufacturers' published data cannot be used to determine filter efficiencies at different particle sizes.

(ii) No two filter manufacturers among those returning questionnaires test and rate the performance of a filter on the same basis. This makes it impossible to compare alternative filters without further testing.

The next stage in the research was therefore to commission a series of filter tests at the British Hydromechanics Research Association at Cranfield. Seven filters were tested, to assess both their efficiencies at different particle sizes, and their dirt-holding capacities. The test procedure was basically the multipass filter test method developed by OSU and now accepted as an American National Standard [83]. In the OSU test the test filter is installed in a circuit as shown in figure 7.4.1. ACFTD contaminant is then injected continuously upstream of the filter so that, if the filter were 100% efficient at all particle sizes, the upstream fluid contamination level would be maintained at 10 mg/l. Fluid samples are taken upstream and downstream of the filter, and the filter efficiency can be calculated from particle counts obtained on these samples. As a test progresses, the filter becomes gradually 'clogged' with contaminant, and the pressure drop across the element rises. The test is usually terminated when this pressure drop reaches 40 psi, and the filter's dirt-holding capacity can be estimated from

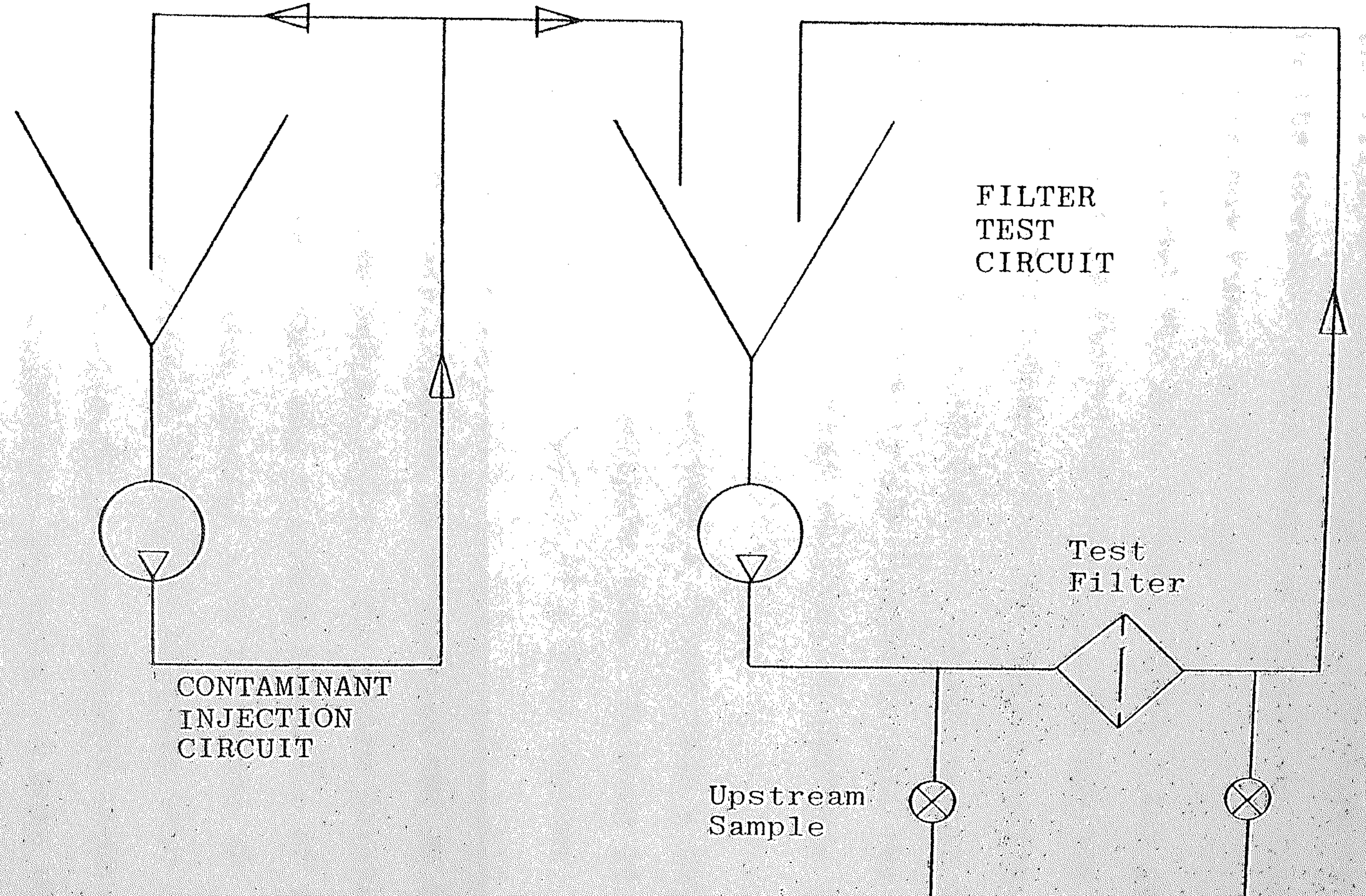


Figure 7.4.1

BASIC MULTIPASS FILTER
TEST CIRCUIT

the amount of contaminant added to the system up to this time.

Table 7.4.1.shows the efficiencies calculated for the filters tested, and figure 7.4.2 shows the increases in the pressure drops measured across the filter elements at various times in the test.

Although the data shown in table 7.4.1 gave some indication of filter efficiency, BHRA advised CHL that the calculated values cannot be very accurate. The reasons why this should be so stemmed partly from the high fluid dilution ratios needed to make the counts, and partly from the limited accuracy of particle counting techniques.

For example, consider a filter with a true efficiency of 5% at a certain particle size. Because of the saturation characteristics of a Hiac counter, the actual count measured in a test on this filter would be around 40 and 600 particles/ml at 50 microns and 20 microns particle size respectively. The author determined, while at OSU, that the standard deviation of such counts measured on a Hiac counter will be around 4%. It is therefore reasonable to expect that the true contamination level will be within 8% of the measured count. (This is a very simple statistical analysis. A more sophisticated approach is possible, but it would not significantly affect the overall conclusions obtained.) In the filter test considered above, for a true upstream count of 600 particles, the actual downstream count would be 570 particles. But the upstream level could be measured as anything between 552 and 648 particles, the downstream as 524 to 616 particles. The derived filter efficiency could therefore be anywhere between 19% and -11.6%.

FILTER REF	FILTER EFFICENCY (IN %) AT PARTICLE SIZE SHOWN (MICRONS				
	5	10	15	25	50
A 10%*	3.5	5.1	5.0	6.4	20.4
20%	9.7	23.2	14.3	10.2	27.7
40%	2.4	3.2	6.9	12.1	45.7
B 10%	15.8	26.2	39.8	56.2	63.1
40%	6.6	13.8	19.3	22.8	51.6
C 10%	10.7	28.4	42.7	63.1	95.8
20%	9.6	16.4	18.0	21.6	72.8
40%	5.4	12.9	22.8	35.8	61.1
D 10%	12.7	11.1	8.5	7.6	-30.0
20%	18.9	21.8	43.5	-53.0	-48.0
40%	5.0	8.7	10.0	10.5	29.5
E 10%	0.5	8.1	12.8	17.7	55.4
20%	-6.5	-5.3	-3.3	5.8	12.8
40%	-5.8	0.4	5.1	21.4	70.0

Table 7.4.1

Filter efficiencies calculated directly from upstream and downstream particle counts for BHRA filter test data.

* These values represent the pressure drop across the filter as a % of the maximum allowable pressure drop (usually 40 psi) See figure 7.4.2.

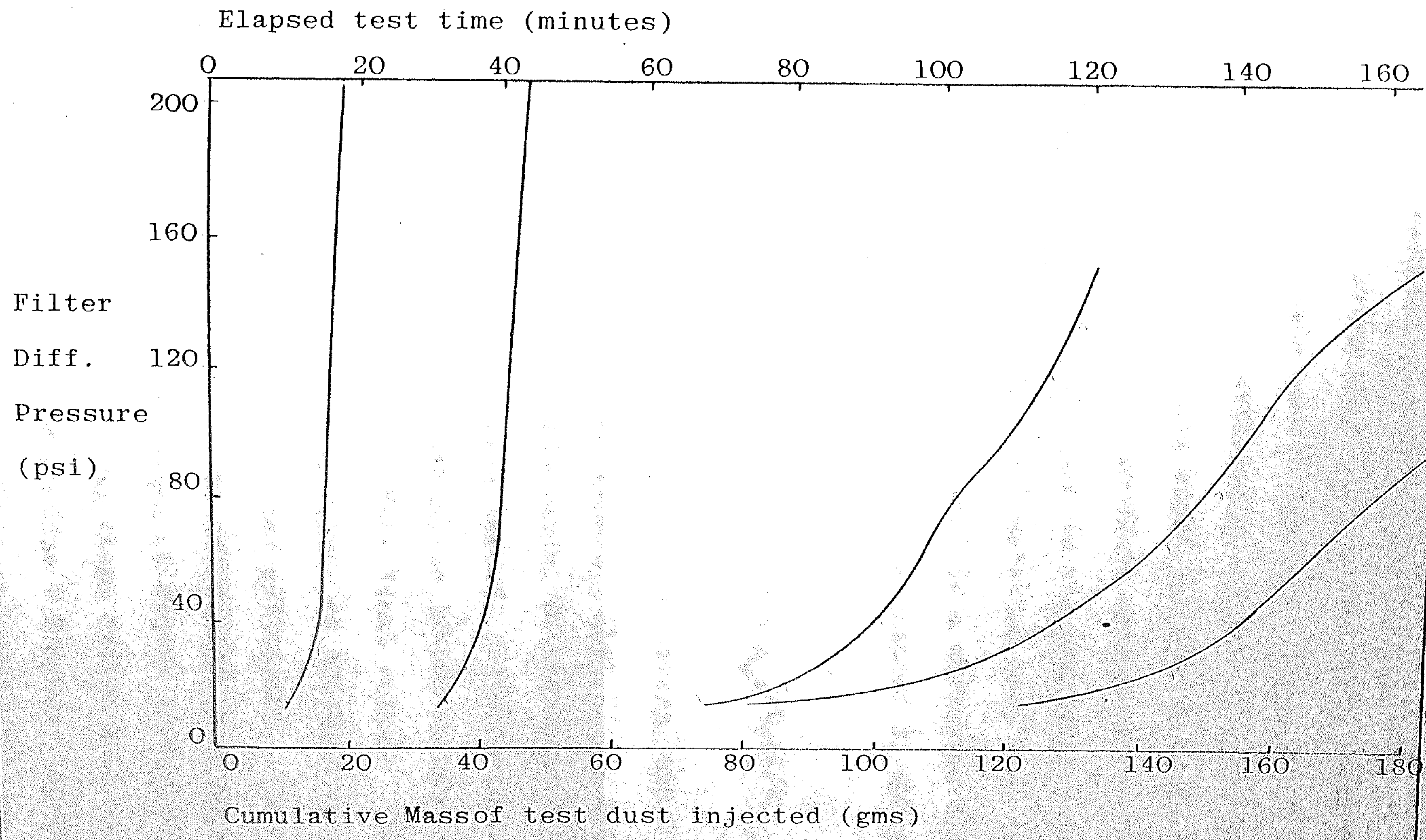


Figure 7.4.2 Dirt capacity of various filters (data from BHRA tests)

Similar problems are encountered in estimating very high filter efficiencies, although these are caused primarily by the very low downstream particle counts obtained.

BHRA recommended, and subsequently conducted, a series of single-pass filter tests to give more accurate values of filter efficiency. Unfortunately, the test results were not quite ready in time to be used in this project. The author therefore attempted to overcome the problem of count accuracy in the available data by calculating the expected variation with time of the upstream contamination level in a multipass filter test. This problem was considered in chapter 3, where it was shown that

$$A(t) = \frac{R}{\eta Q_f} \left[1 - e^{-Q_f \eta t/v} \right] + A_o e^{-Q_f \eta t/v} \qquad 7.4.1$$

Knowing final contamination levels, and R and Q_f, it was possible to estimate η. When filter performances were calculated on this basis, it was found that, for the majority of elements tested, filter efficiencies were so low that contamination levels in the tests would not have stabilized at their final values. This problem was overcome by plotting values of $A(t)Q_f/R$ against time for various values of filter efficiency. This allowed filter efficiencies to be determined as an upper limit.

The calculated values of filter efficiency are shown in table 7.4.2, and these values were used in the analysis described in section 7.5 below. It must be emphasised that these values are still suspect due to the dilution ratios (around 800 to 1) used to make particle counts, although they represent the best data available during the project.

FILTER REF.	FILTER EFFICIENCY (IN %) AT PARTICLE SIZE SHOWN (MICRONS)				
	5	10	15	25	50
A	0.0	0.0	0.0	0.0	0.0
B	0.0	2.0	3.0	6.5	30.0
C	0.0	<0.1	<0.1	<0.1	0.5
D	0.0	0.0	0.0	0.0	0.4

Table 7.4.2

Filter efficiencies calculated from equation 7.4.1 for BHRA filter test data.

The more accurate values of filter efficiency obtained from the single-pass tests were consistently higher than those shown in table 7.4.2. This illustrates the point, made by BHRA, that the multipass filter test is unsuitable for testing coarse filters.

7.5 Service and Test Contamination Levels

Having estimated values of filter efficiencies, equations 7.3.1 to 7.3.3 were used to estimate the maximum contamination levels likely to be experienced by the steering unit in service. It was assumed that filter efficiency would be zero for particles of a diameter below 10 microns, and 0.2% for particles above this size. The calculations showed the following results.

(i) The steering transmission would be unprotected against particles below 10 micron diameter, and the concentration of such particles would increase over the life of the vehilce, checked only by processes such as sedimentation in the vehicle sump.

(ii) The concentration of larger particles (above 10 microns diameter) would stabilise. The rise in the numbers of such particles would be dominated by terms in exp $(-8.5x10^{5}t)$ where t is measured in seconds, so that contamination levels would take a long time to reach stable values.

(iii) The concentration of larger particles in the main circuit would be approximately ten times that in the boost circuit.

If a series of fluid samples could have been taken from an operating system in the field, it would have been possible to estimate final system contamination levels, and to devise a test programme on this basis. Unfortunately, this could not be done, for two reasons.

(i) Fluid sampling points had not been incorporated in the vehicle steering transmission at the design stage and it was not possible to have them fitted during this project. To have taken fluid samples from an operating unit, other than from properly designed and installed sampling points, would not only have given misleading results, it would also have been dangerous, as in operation fluid temperatures exceeded 100°C.

(ii) The point was made, in chapter three, that fluid contamination is as much a problem of management and communication as it is a technical problem. The long lines of communication in this project made it almost impossible to obtain reliable data about fluid contamination levels.

Only one sample was eventually made available for analysis, and this sample was taken from the fluid drained from a vehicle's sump. The contamination level measured is shown in figure 7.5.1, and it can be seen that the indicated level is very high compared, for example, to Thermal Control class 6, which is one recommended level for hydrostatic transmissions. However, the sample was probably unrepresentative, due to the manner in which it was taken, and, as it was effectively drawn from the boost circuit, it would be below the level expected in the main pump and motor.

Whilst making a recommendation that much more data was needed on typical system contamination levels, it was

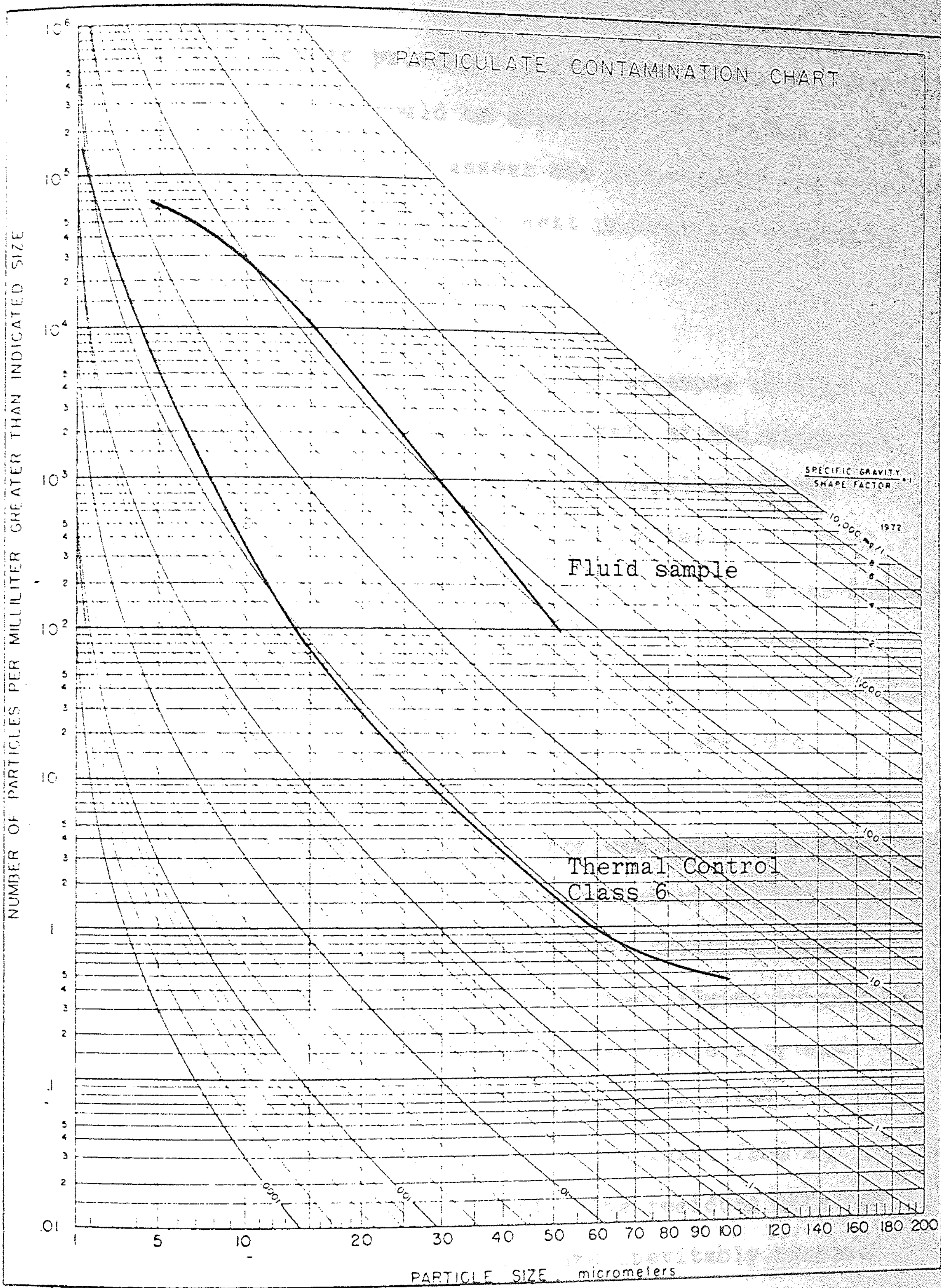


Figure 7.5.1. Particle count data on fluid sample from hydrostatic transmission.

still necessary to proceed with the tests. It was therefore decided that tests would be conducted at a number of fluid contamination levels to assess the severity of the effects produced at each level. The next problem was obtaining a suitable test contaminant.

7.6 Test Contaminants

Two approaches were adopted in attempts to find a suitable test contaminant. The first, at the suggestion of the vehicle customer, was to use supplies of oil drained from vehicles which had run on test.

This approach was tried, but without any great success. The problems of using supplies of "old" fluid were discussed in section 6.2.3 and they will not be reiterated here. Suffice it to say that these problems were discovered during this section of the work. One possible way to circumvent some of the problems would have been to remove the contaminant from supplies of old oil, and to then wash and dry the particles to obtain a powder. This might then have been added to test fluids to produce required contamination levels. This possibility was examined carefully, but was eventually rejected. It is impossible to remove all the contaminant from a large volume of hydraulic fluid. The residues obtained from filtration or centrifuging are inevitably biassed towards the larger particles, and this makes the resulting dust unsuitable for use in contamination tests.

An alternative approach was to determine the nature of the contamination present in the system and then to model this contaminant with a suitable blend of powders. One difficulty with this approach has already been mentioned, and this was that it was not possible to obtain representative fluid samples from an operating system. The samples which were obtained from the vehicle sump were analysed by X-ray diffraction and by electron probe micro-analysis, but neither technique could give a quantitative estimate of the relative proportions of different materials present as contamination.

It was suspected that a considerable proportion of the fluid contamination would have come from the sintered bronze brake and clutch plates used in the vehicle. Spectrographic analysis of contaminant and of a clutch plate confirmed this suspicion, and a series of powders was therefore obtained to model the contaminant.

This proved more difficult than was anticipated. The plate manufacturer was able to give details of the plate materials, but not of their relative proportions, as this latter information was a commercial secret. Nor was he able to supply samples of the constituent plate materials, although he did suggest suitable sources for some of these materials. These materials were obtained, in powder form, and were used to blend a test contaminant. This contaminant was not an ideal representation of what would be found in the vehicle. However, it was composed of materials which were likely to be experienced in service, and it had a size distribution (verified by analysis on a HiaC counter) which was similar to those of particles found in hydraulic fluid.

At this stage, the contaminant was still lacking in one respect. The eventual vehicle application required that it should operate in hot, dry, dusty conditions where the main system contaminant expected would be silica. The vehicle was never in fact tested in its intended environment, so it was not possible to estimate how much silica the steering transmission would have to contend with. (The problem of obtaining a representative sample would have still remained.) In the absence of any data, an arbitrary amount of an abrasive dust (approximately 7% by weight) was added to the test contaminant.

7.7 Test Circuit and Test Profile

The main design criterion for the test circuit was that it should model the vehicle's hydraulic steering system as closely as possible. The vehicle system contained many components (pumps, valves, servo controls) which might have been affected by fluid contamination, and the aim was to test as many of these components as possible. An additional factor was that it would have been impossible to have tested the main pump without the majority of the other components being present as well.

One way to have modelled the vehicle hydraulic system would have been to conduct the tests in an actual vehicle. However, this would have been very difficult because, once installed, the steering unit is inaccessible and it would have been almost impossible to add contaminant, take fluid samples and obtain useful data on how the unit was performing.

The solution adopted was to test a main pump, dropping its output pressure across a load valve, rather than across a main hydraulic motor. This was done largely for economy, because it removed the requirement to absorb the output power from the hydraulic motor. As the hydraulic motor was basically a pump running in reverse, it was thought that any effects which would be produced in service in the motor would be identified in the pump during the test.

The rig was originally built with the circuit of figure 7.7.1. The rig design incorporated most of the features which were described in chapter 6, with reference to the test rig constructed at Aston University. Whilst the circuit modelled the vehicle steering system circuit shown in figure 7.2.1, it was found that, with this configuration, fluid temperatures could not be controlled within the required limits. This was because in the vehicle only efficiency losses were transferred to the fluid as heat whereas, in the test circuit, the full output power of the test pump was converted to heat as the fluid dropped pressure across the load valve.

The problem was overcome by rebuilding the test rig with the circuit of figure 7.7.2. Whilst this cured the problem of temperature control, it meant that the contamination behaviour of the test rig ceased to model that of the vehicle circuit. Referring to equations 7.3.1 to 7.3.3, the values of t_a and t_b in the test rig were not the same as those expected in the vehicle. Without more data on fluid contamination behaviour in the vehicle, there was no way to determine the possible significance of this fact. This emphasised, again, the need for properly designed and installed fluid sampling points in the vehicle.

It has been shown [90] that the results of a pump contamination test can be significantly affected by the type of hydraulic fluid used in the test, and by the fluid's temperature. The test rig was therefore designed to run on the same fluid, and at the same temperature, in excess of $100^{o}C$, as would be used in the vehicle.

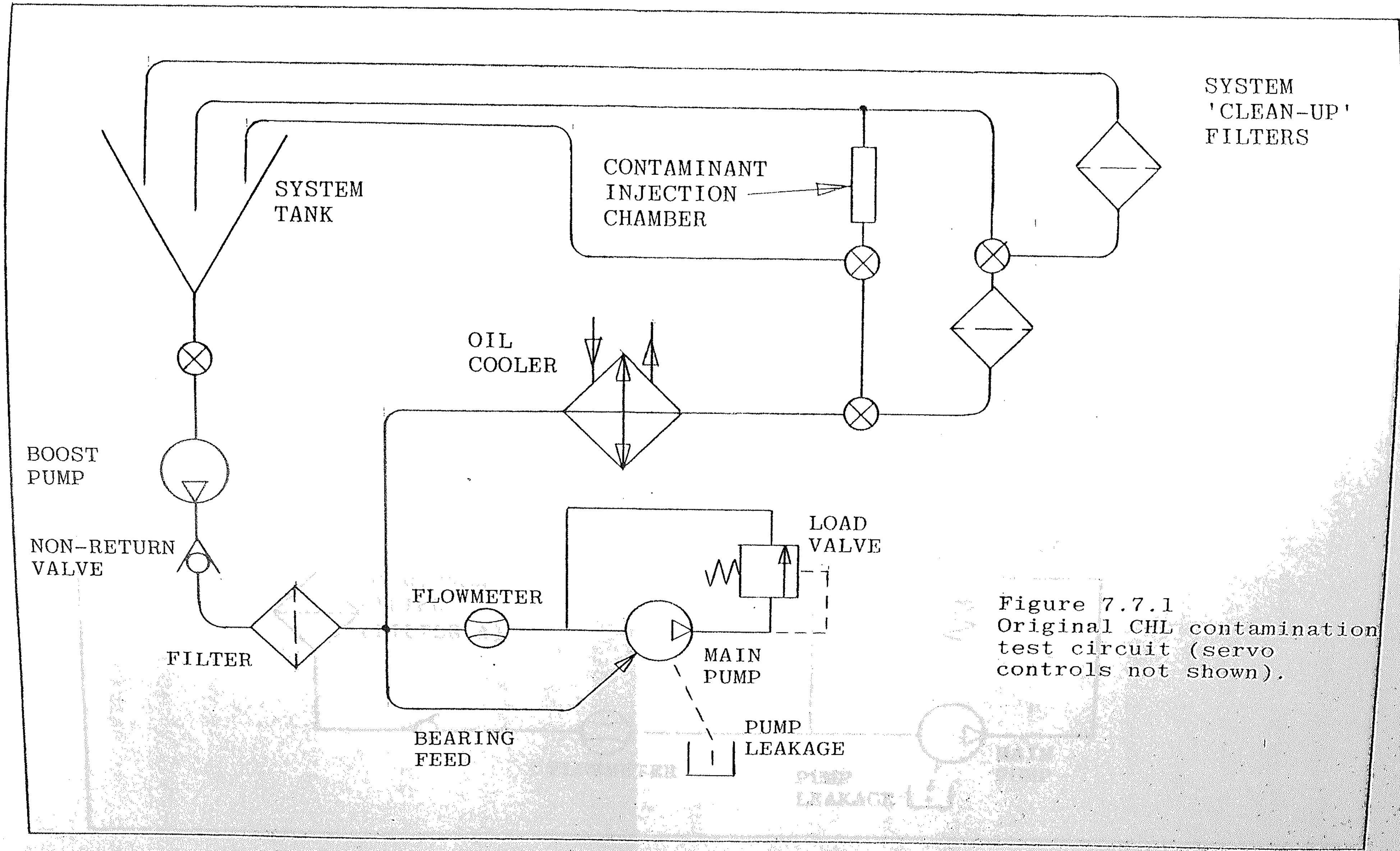
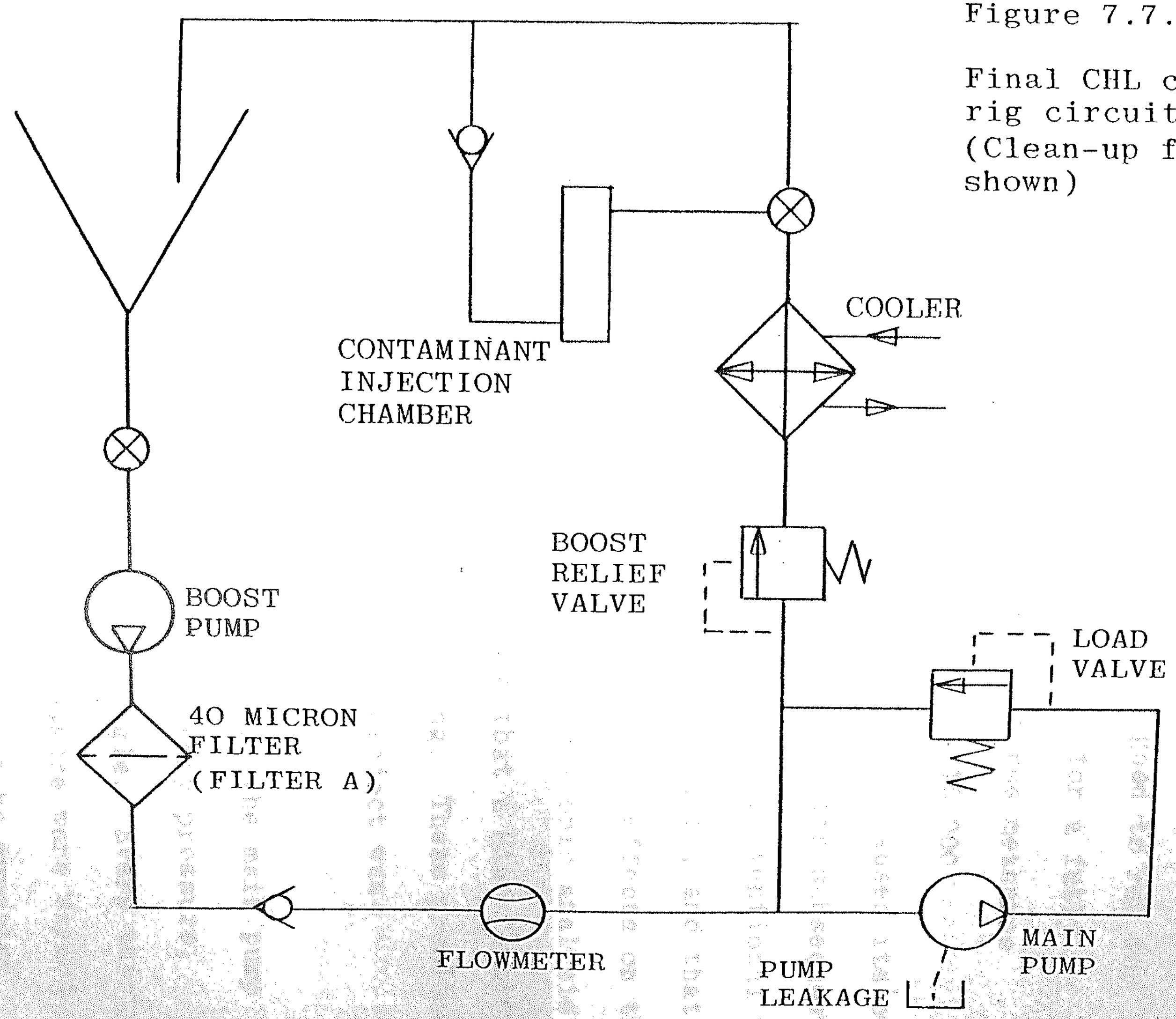


Figure 7.7.1
Original CHL contamination test circuit (servo controls not shown).

Figure 7.7.2

Final CHL contamination test rig circuit.
(Clean-up filters not shown)



This required that, for operator safety, the rig was built in a sealed cell, all instrument lines being brought to a remote control console. Fluid sample lines were built into a steel case, inside which a sample could be taken without risk to personnel. Because of the power levels used in the rig, the high system temperatures and pressures, and because tests were designed to run for long periods, the rig was instrumented for a fully automatic shutdown. The cost of all these measures pushed the eventual rig price to above £15,000.

The type of fluid used in the vehicle caused its own problems, not with running the tests but with subsequent fluid sample analysis. It was found that conventional solvents were not 100% effective on the fluid, and that one solvent which did work had undesirable effects on the membranes used for gravimetric and microscopic analysis. The fluid also turned so dark in use that a Hiac counter could not be used for particle counting. These problems were never fully resolved before the project was cancelled (see below).

The test profile adopted was to run the main pump at nominal conditions of 3000 psi delivery pressure, 3000 rpm input speed and full swash angle. Every thirty minutes, swash angle and delivery pressure were varied to give some representation of what might be expected in the vehicle. The original aim was to run tests for 50 hours, or until it was evident that some damage had been caused to the test pump.

7.8 Test Programme and Results

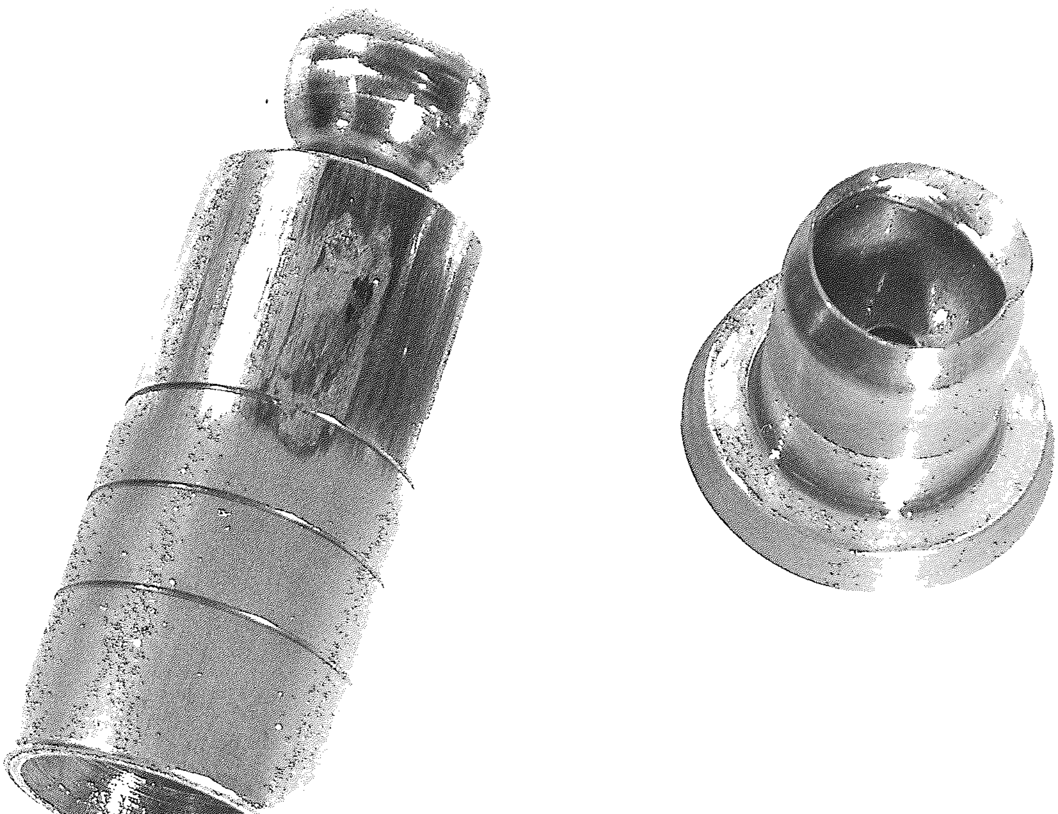
A full test programme was initially planned, with tests at several, different contamination levels, but only one test was actually completed. This was initially due to a change in priorities instigated by the vehicle manufacturer, and finally to the cancellation by the customer of the entire project.

The one test conducted was in two parts. The first involved running the pump for ten hours on used fluid drained from a vehicle. (The problems with this approach were detailed in section 6.2.3. However, at the time when this section of the work was conducted, these difficulties were not fully appreciated.) The system ran with no sign of any problems, but the rig had to be shut down after ten hours to repair some leaks. The fluid had to be drained to allow these repairs to be conducted, and it could not be recovered.

The second part of the test was therefore run using the "artificial" contaminant described in section 7.6, injected into the system to give a contamination level of 600 mg/l. After 25 minutes running, system boost pressure was lost and the main pump delivery pressure started to fluctuate wildly. The rig was immediately shut down and the main pump was stripped for examination.

The examination showed that the pump had failed in a catastrophic manner. Two pistons had completely seized in their bores, and had been pulled out of their corresponding slipper sockets. (See plate 7.8.1). The cylinder block showed scratching on the journal bearings and on the portplate face (see figure 7.8.2 and plate 7.8.3). Contaminant had been concentrated in the central bore of the cylinder block. Chemical tests showed that the surface finish treatment of the block (discussed in section 4.5) had been completely worn away from areas

Plate 7.8.1. CHL pump piston and slipper after test



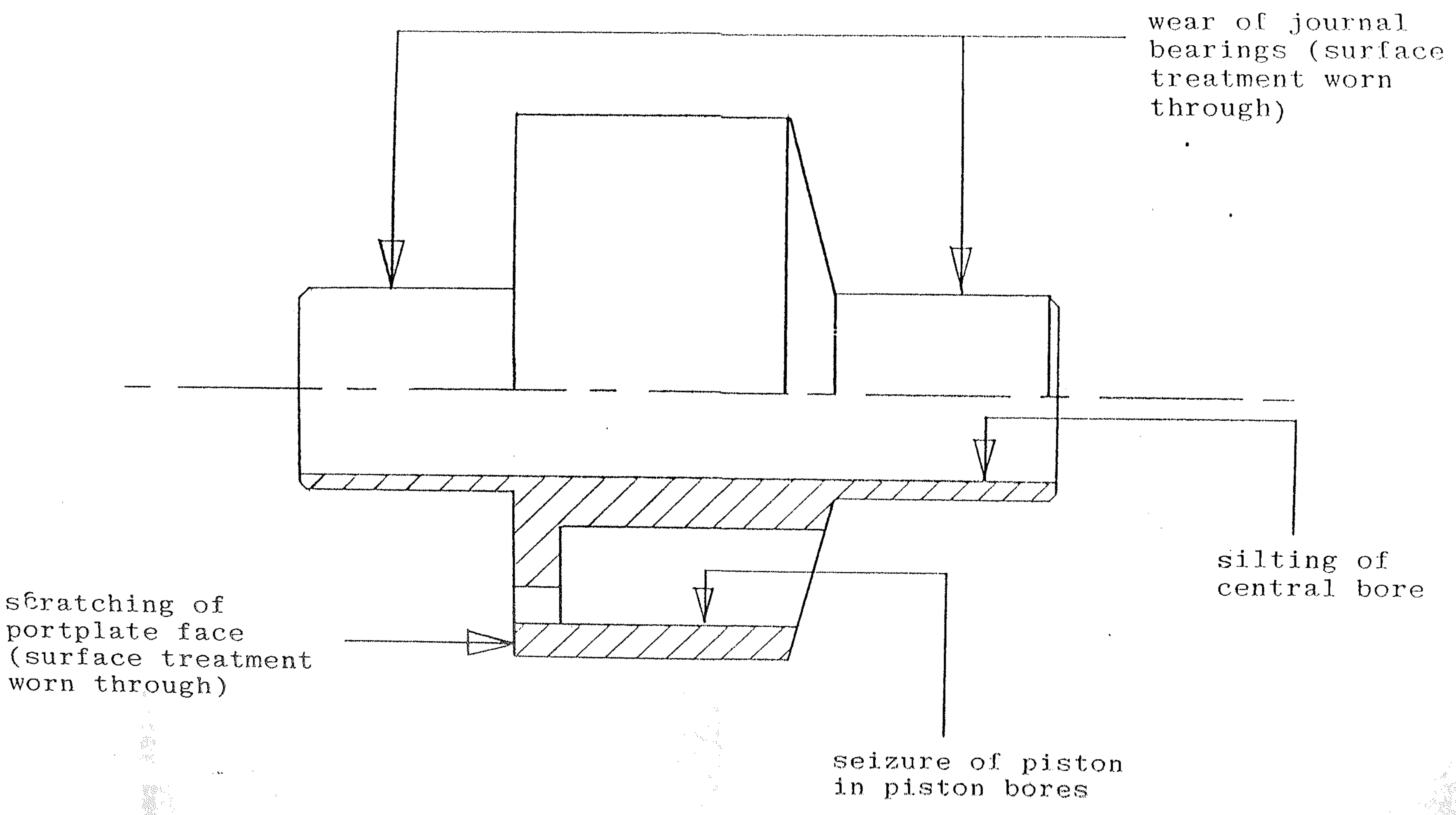


Figure 7.8.2 CHL pump cylinder block, showing location of contaminant - induced damage.

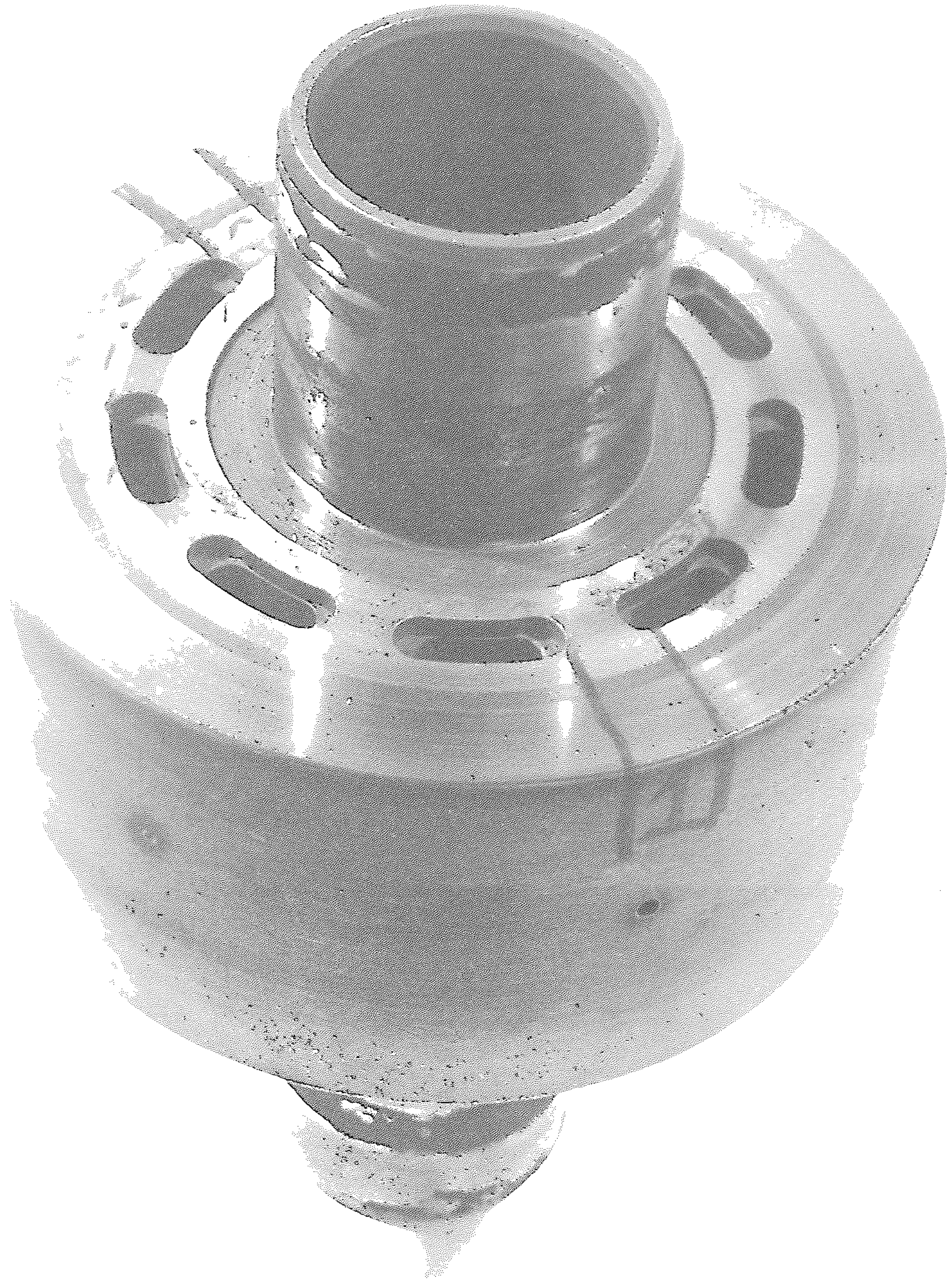
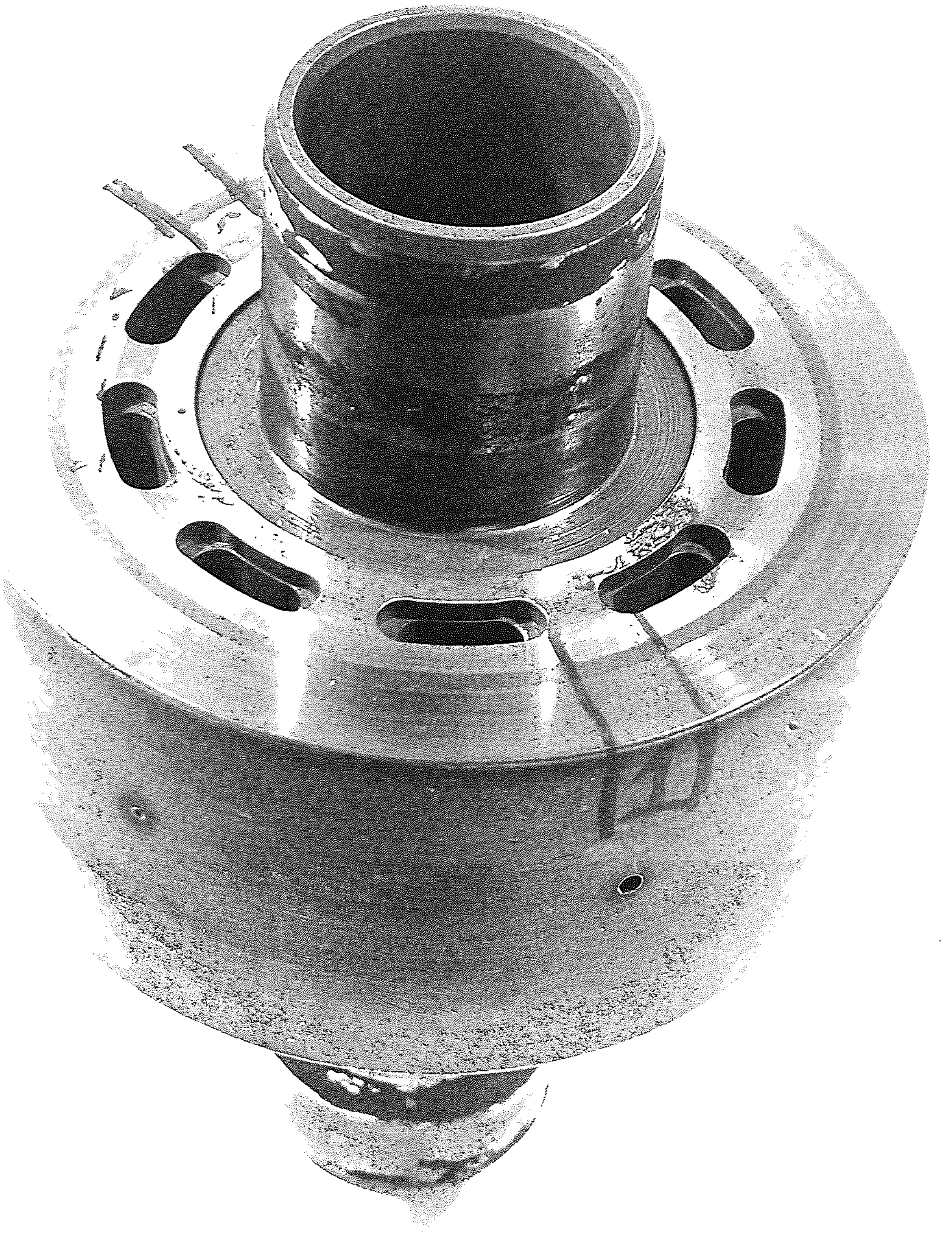


Plate 7.8.3. CHL pump cylinder block after test



of the block journal bearings and portplate face.

The pump slipper-plate showed scratch-marks, but, more interestingly, it also showed circular marks (plate 7.8.4.) corresponding to the position of the slippers when the pump was stationary (which it was at some times during the test). The possibility of slipper silting was considered in section 4.9, and the marks shown in plate 7.8.4 may be evidence that this was happening.

7.9 Discussion

It is impossible to draw any firm conclusions from the results of the very limited test programme conducted. It is however interesting to note that the pump failed ultimately through piston seizures. It will be remembered that this was not a problem in the A70 pump tests described in chapter 6. It was also noted that the pump slippers were still in good condition at the end of the test, suggesting that this pump area may not be as sensitive to the effects of fluid contamination as CHL thought.

Although the test provided very limited data about the likely effects of fluid contamination on the steering transmission, the whole exercise is a useful case study on this approach to contamination testing. On the basis of the work, the following points should be emphasised.

(i) It is possible to tailor a pump contamination test towards one, specific application. But the problems involved should not be underestimated, particularly those of determining and modelling the fluid contamination likely to be found.

(ii) It is still not possible to devise a completely realistic contamination test. Some of the problems

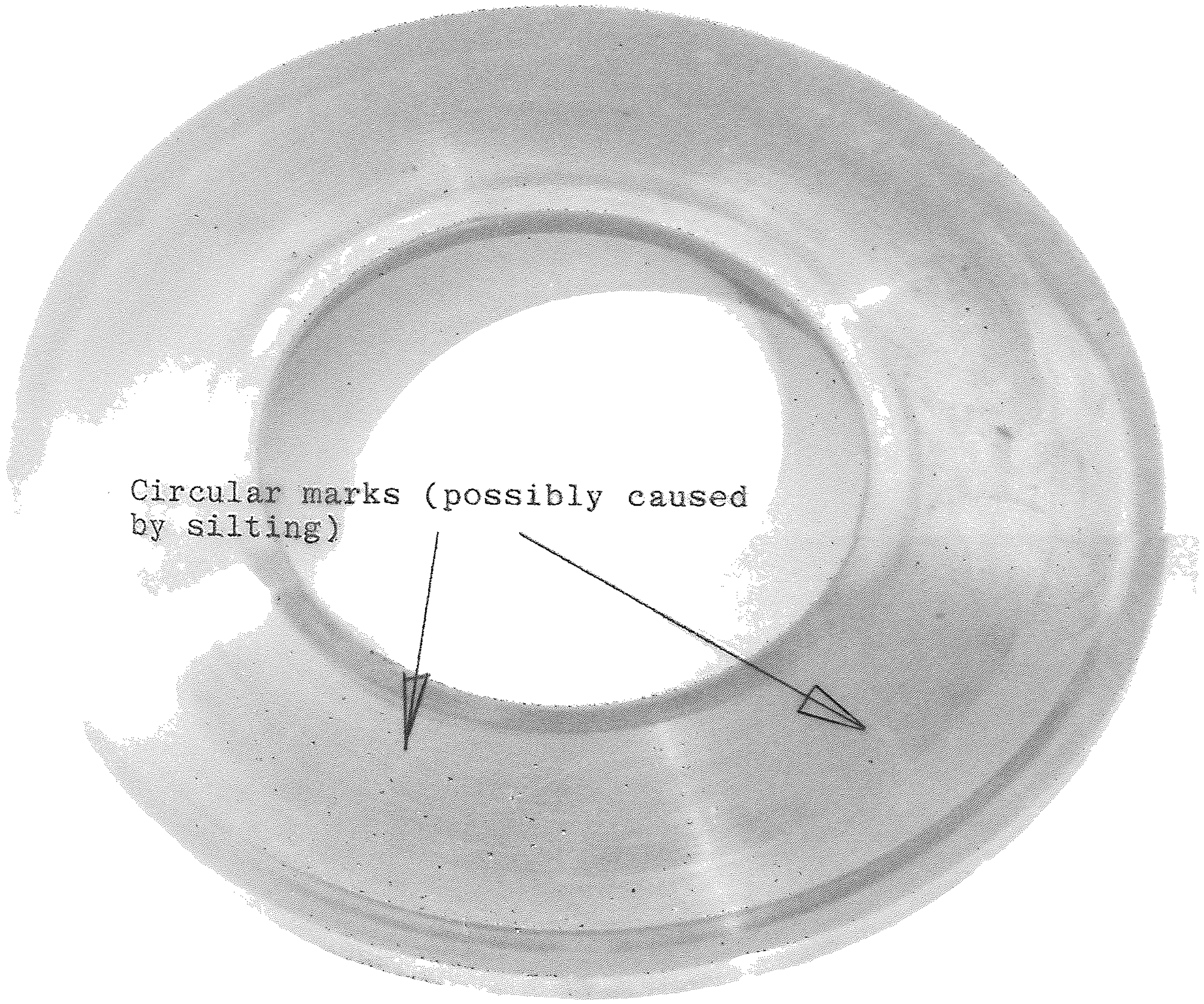


Plate 7.8.4. CHL pump slipper plate after test

involved were mentioned in this chapter, but the factors discussed in section 6.2 also apply. Any test has to be a multipass procedure and this immediately makes it unrealistic. It also means that the test will be short, and it is interesting to note, in this respect, that the test pump failed only 25 minutes after the injection of the artificial contaminant. The significance of the multipass procedure was not communicated adequately to the vehicle manufacturer, with the result that a vehicle filter (filter A in figure 7.7.2) was included in the test circuit. The only reason that the test was not cut short by this filter blocking up was that the filter was so inefficient (see table 7.4.2) that it made very little difference whether it was installed or not.

(iii) Tailoring a test to a specific application does not remove the considerations about reliability estimation which were discussed in chapter five. Life prediction is still impractical. Both the customer and CHL were looking for a relation of unit life to fluid contamination level. The author was not able to convince them, at the time, that this is not possible.

(iv) It is vital that the objective of a test be specified at the start of the test programme. The initial objective of this test was to determine the predicted "life" of the steering transmission when it was operated on fluid contaminated to certain levels. But, even if this could have been done, there was no way to monitor contamination levels in the vehicle, so that the information obtained could not have been acted upon. It might have been found that the unit would operate successfully if contamination levels were maintained below a certain value, but these levels could not have been measured in service so the knowledge would have been of little use.

(v) Many, if not most, of the problems encountered were those of management and communication. The discussion of chapter three was largely based on the author's experiences during this section of the work. It is vital that pump manufacturers communicate the importance of fluid contamination to all other parties involved in a hydraulics project. The contribution of each party to the overall problem must be clearly identified. Above all, it is important that consideration be given to fluid contamination in the early stages of a project. The analysis of section 7.4 would have given advance warning of the potential problem and, at the very least, adequate fluid sampling points could have been provided.

(vi) It is quite plain that a great deal more work is required into the problems involved in measuring filter performance.

By this stage in the project, it had become clear that theoretical analysis and pump testwork must be seen as part of a long-term effort to improve understanding of fluid contamination and its effects, and to increase the resistance of pumps to fluid contamination. The work described in chapters 4, 5, 6 and this chapter represents a useful contribution to this long term effort.

However, this still left contamination as a short-term problem requiring immediate attention. No way had yet been found to determine 'acceptable' fluid contamination levels for CHL's pumps. The next chapter therefore deals with this subject.

8.0 SERVICE EXPERIENCE AND RECOMMENDED FLUID CONTAMINATION LEVELS

8.1 Introduction

This fairly brief chapter contains a series of recommendations for a programme of work to increase CHL's knowledge of the fluid contamination levels encountered in service, and about the effects of this contamination. The chapter commences with a discussion of the need for the work, and continues with detailed recommendations as to how the work should be conducted. The chapter closes with a set of recommended fluid contamination levels for CHL's pumps, these recommendations representing the best available short-term "answer" to the problem of fluid contamination.

8.2 The Need to Monitor Service Experience

It was argued, in chapter three, that the problems caused by fluid contamination could only be tackled through a concerted effort by the entire hydraulics industry. Pump manufacturers would contribute to such an effort by:

(i) trying to improve the ability of their pumps to operate on contaminated fluids and

(ii) specifying "acceptable" fluid contamination levels for their products.

These would have been over-ambitious aims for a single Ph.D. research project, so three more restricted research objectives were selected. These were:

(i) to improve CHL's general understanding of fluid contamination and of its effects;

(ii) to recommend acceptable fluid contamination levels for CHL pumps;

(iii) to identify ways to improve the ability of CHL pumps to operate on contaminated fluids.

The pump design and development model shown in figure 3.8.1 suggested that the research could proceed through one or more of theoretical analysis, testwork, or the collection and analysis of data on service performance. There are two reasons why service experience was considered last as a research strategy.

(i) Service data is difficult to obtain, largely because it can only be collected through close co-operation between pump manufacturers, system designers and system users.

(ii) Service data can only be obtained relatively late in the development of a pump, by which time it may be of limited value in attempts to improve that specific pump. It may still, of course, be of use in developing new pumps.

Despite the greater attractiveness of theoretical analysis and pump testing as research strategies, it should be clear, by now, that they cannot provide all the data which will be required before the problem of fluid contamination can be fully solved. It is worth summarising the work presented to date, to illustrate this point.

The theoretical analysis presented in chapter four developed several ways of predicting the effects of fluid contamination. The work is useful, but it is lacking in certain areas:

(i) Theoretical analysis cannot yet be used to determine an acceptable fluid contamination level for a given pump.

(ii) The analysis considered only monosize particles. It was shown that the current mathematical model of contaminant size distributions is open to question. More data is needed on typical particle size distributions, and of the variation between such distributions in different fluids.

(iii) Theoretical analysis risks being irrelevant. More information is needed on the precise effects of fluid contaminants, and on whether contamination is primarily a cause or a symptom of pump unreliability.

(iv) Although theoretical analysis can be used to suggest design or materials improvements to reduce the effects of fluid contamination on a pump, these suggestions represent untested hypotheses which must be verified. This can be done either through a representative contamination test, or through monitoring the service performance of the modified pump design.

Pump testing can be used to improve the general understanding of contamination. For example, the test results of chapter six showed, quite conclusively, that contamination can be a cause of serious pump damage. They also showed that the quantity of wear debris from a pump can be a function of the existing level of abrasive contaminants.

But pump testing is not without its limitations.

(i) It is not practical to use pump tests to relate fluid contamination levels to pump reliability.

(ii) Pump test results cannot be used to recommend acceptable fluid contamination levels for a specific pump design.

(iii) It was shown, in chapter 6, that a pump contamination test is, and can only be, an unrealistic process. The usefulness of such a test procedure must depend, in part, on a demonstration that the effects produced by contamination in the test are similar to those produced by contamination in service.

All the above points lead to the conclusion that more data is needed from the field, both on actual fluid contamination levels, and on what damage, if any, can be directly attributed to this contamination. It was not possible, during the project, to collect fluid samples from operating systems, or to record data on pump reliabilities. However, it was possible to recommend how these activities should be pursued, and these recommendations are given below.

8.3 Measuring Contamination Levels in Service

It might be thought that it would be relatively easy to obtain accurate data on the fluid contamination level existing in a hydraulic fluid. This is not so. If CHL are to obtain such data, the following actions will be required.

(i) The company must encourage designers of hydraulic systems incorporating CHL pumps to incorporate fluid sampling points, at the system design stage.

(ii) CHL must train staff in the correct way to take a fluid sample.

(iii) CHL must make available a supply of laboratory cleaned bottles in which samples may be collected.

(iv) CHL must secure the co-operation of system users, so that fluid samples may be taken from operating systems.

If all these actions can be taken, then representative fluid samples may be obtained. The next problem would be to determine the contamination content of each sample. There are many techniques for determining the contamination level of a fluid. These were described in chapter two, and some have been used during this work. The difficulties involved in measuring the contaminant content of a fluid are primarily those of obtaining repeatability, both within a technique and between different techniques. The author determined, while at OSU, that the Hiac counter and gravimetric analysis can both give very repeatable results. But Day and Saunders [15] have shown that correlations between particle counts obtained using different techniques, and between results obtained with the same technique used at different locations, are not good. This leads to the following recommendations.

(i) CHL should standardise its methods of contaminant analysis. The author recommends that gravimetric analysis and an electro-optical counter (Hiac or Royco) should be used exclusively to measure contamination levels.

(ii) CHL should entrust all its fluid sample analysis to one laboratory. It would be preferable if this laboratory were owned and operated by CHL itself. The cost of setting up a laboratory would be in the region of £20,000.

The method adopted to report the results of fluid contamination analysis is also of some importance. The system adopted must be simple and unambiguous, and it is better if the results are quoted as a single number. This immediately suggests the use of one of the available systems of contamination class.

The choice of system is complicated by the fact that the four commonly-used systems have different "shapes" when plotted as cumulative oversize counts. For example, figure 8.3.1 shows Thermal Control class 6, NAS 1638 class 7, SAE 749 class 5 and MOD standard class 6300. It can be seen that the classes have different profiles, especially below 15 microns. One reason for this is that the Thermal Control and MOD standards are based on optical microscope counts, which tend to give higher counts at small particle sizes than do other techniques such as the Hiac counter. It is generally thought that this is due to experimental error in the microscope counting technique, mainly caused by the introduction of dirt during the microscope slide preparation. However, the author suggests that the differences are due to the characteristics of electro-optical counters.

The standard method of calibrating an electro-optical counter is to use small, latex spheres, of known uniform size to set the voltage levels corresponding to particles of three different diameters. The smallest of the spheres used is about 16 microns across, and the calibration must then be extrapolated down to three microns. It has never been adequately demonstrated that this procedure is valid.

An alternative approach, developed by OSU, is to set up the counter to give a predetermined size distribution when measuring ACFTD. This procedure helps to give repeatability between counters calibrated on this basis. However, the size distribution used for ACFTD is the log-log^2 model described in section 4.8. It was shown, in the same section, that this model underestimates the

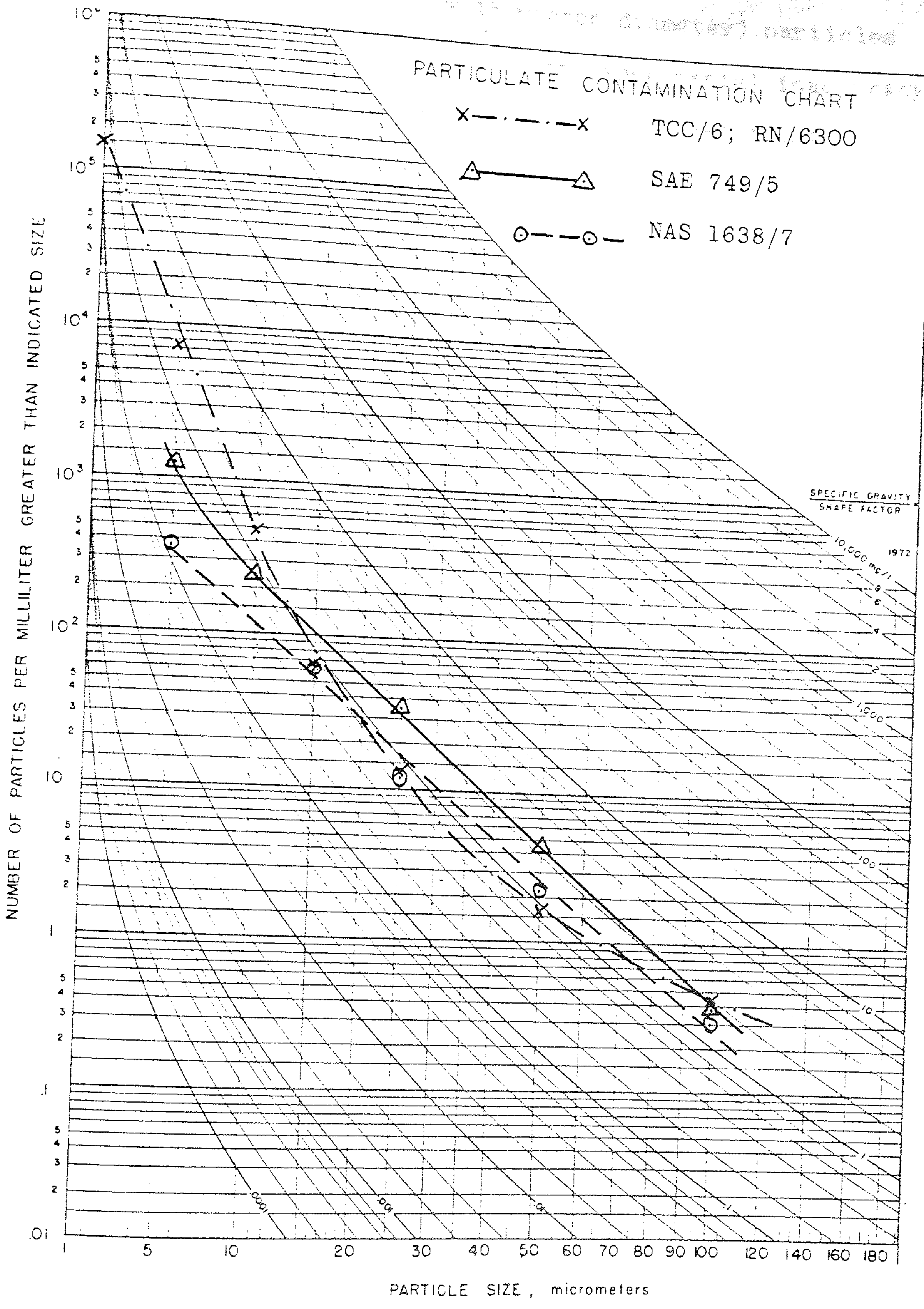


Figure 8.3.1. Comparison of different contamination class systems.

numbers of smaller (below 15 micron diameter) particles present. It is this, rather than experimental inaccuracy, which produces the differences noted between counts measured by optical microscope and electro-optical counters.

The implications of these findings are that:

(i) The Thermal Control and MOD systems of contamination class are appropriate to counts determined by optical microscope techniques. They could also be used to describe counts measured by an electro-optical (Hiac) counter if the counter were calibrated on the log-normal model of ACFTD developed in section 4.8

(ii) NAS 1638 and SAE 749 are appropriate to particle counts determined by a Hiac counter calibrated against the log-log^2 model of ACFTD.

Sampling would also have to be done on a regular basis and not just after a failure had occured, when contamination levels would be expected to be unusually high.

If all these recommendations and findings were implemented, CHL would obtain reliable data on the contamination levels experienced by its pumps in actual operation. (More detailed recommendations will be given to the company in a separate report [91]). The next step would be to relate these results to pump reliability, and to identify failure modes.

8.4 Recording Pump Reliability

There is no point in adopting an over sophisticated programme of recording unit reliabilities, at least in an initial approach to the problem. For example, it is difficult to compare the lives of pumps in different systems, because the pump duties vary so much. An initial programme should therefore concentrate on recording the areas in which problems occur (pumps, valves, fittings, etc.), and the type of problems (wear, seizure, mal-operation). It would also be useful to have some idea of the frequency of occurrence of each type of problem.

Even at a very simple level such a programme could produce useful general information. It might, for instance, become clear that systems with contamination levels below a certain value have noticeably fewer problems than do systems with higher contaminant concentrations. A demonstration that this was so would be based on non-parametric statistics. (These are covered in Fraser [92]) The work might also identify pump areas which are likely to be affected by fluid contamination. If these areas were the same as those affected during the type of contamination test described in chapter six, then increased confidence could be placed in the use of such tests to identify pump areas requiring improvements.

A programme of analysing service performance, if it were adopted, could have one further use. If it could be shown that one pump design was better able than another to operate on contaminated fluids, then examples of the two pumps could be subjected to the type of test described in chapter six. If the pump which performed better in service also performed better in the test, this would suggest that the test might be used to assign a contamination rating to pump designs. This would be of considerable value, particularly if it could be used to test pump design or materials improvements.

This discussion will not be taken any further, because, without implementation of this type of programme, it is impossible to decide what data can be obtained, or the analytical tools which will be required to extract useful information. Because the work would need considerable co-operation between CHL and pump users, it cannot proceed without a commitment at a fairly senior level in the co-operating organisations. This will need, in turn, an assessment of this work by CHL's management. It will also require that the reality of fluid contamination as a real problem be "sold" to pump users. It may be difficult to convince, say, a production manager of a large steelworks that invisible particles in his hydraulic systems could cost him a considerable amount of money.

Although the above discussion contains firm recommendations for future work, and although the work described in chapters 4 to 7 looks promising as the basis of a long term approach to the problem of fluid contamination, this still leaves fluid contamination as an immediate problem requiring fairly rapid action.

8.5 Recommended Fluid Contamination Levels for CHL's pumps

An ideal solution to the problem of fluid contamination was given in chapter three. In such a solution, CHL would recommend an acceptable fluid contamination level for its pumps. It would then be the responsibility of pump users to maintain contamination levels below this value, although CHL should be prepared to act in an advisory role, or even to conduct fluid sampling and analysis on a commercial basis.

With rare exceptions, pump manufacturers specify protection against fluid contamination as a required filter rating in microns. This practice is completely unsatisfactory, for four reasons.

(i) As was shown in chapter 3, filter performance is only one of several factors which together determine the contamination level of a fluid, and it is this contamination level which is of importance in determining the severity of any effects produced by contamination.

(ii) The analysis of section 7.3 showed that the location of a filter within a hydraulic system can affect the contamination level which the filter will maintain in different parts of the circuit.

(iii) The results of the questionnaire sent to filter manufacturers, which were summarised in section 7.5, showed that no two filter manufacturers test and rate the performance of their products on the same basis. This makes published "micron ratings" of little value.

(iv) Although filter test methods are available, the results can be very misleading, due to the limited accuracy of particle counting techniques. This was shown in section 7.4. It was also shown that some of the problems involved can be overcome, and that more realistic estimates of filter efficiency can be derived from existing test data. However, it must be emphasised that the performance in service of a filter may be considerably below that which is measured in a laboratory test, and it is service performance which is important.

What CHL therefore needs is a set of simple, unambiguous recommendations on acceptable fluid contamination levels for the company's pumps. These can only be of practical use to industry if they consist of recommended contamination classes, rather than comprehensive (and complex) particle size distributions.

The four commonly-used systems of contamination classes have already been mentioned and it was shown, in figure 8.3.1, that they are not strictly comparable, especially for particles below 15 microns diameter. However, it will be remembered that, in the tests described in chapter 6, 0-5 micron ACFTD produced no significant damage to the A70 pump. The figure of 5 microns refers to the Stoke's diameter of the particles used, so that the longest dimension of the largest particle present would be around 8 microns. It is this figure which would be used in any contamination analysis. The implication is that only particles above 8 microns diameter need be considered in describing the contamination level of a fluid. This has two implications:

(i) Examination of figure 8.3.1 shows that, above 10 microns particle size, the different contamination classes do have similar profiles when plotted as cumulative oversize distributions. This allows an approximate comparison between the different systems, the results being shown in table 8.5.1.

(ii) Above 10 microns diameter, the log-log^2 particle size model for ACFTD does not differ appreciably from the alternative log-normal model developed in section 4.8. This allows either calibration to be used for particle count analysis, providing that particles of 10 micron diameter and above are of primary concern.

Several organisations (Thermal Control Company, Hiac, the Royal Navy) publish recommended contamination levels for different types of hydraulic system. The author

THERMAL CONTROL	NAS 1638	SAE 749	ROYAL NAVY STANDARD	CETOP CODE	ISO CODE	SPERRY-VICKERS CODE	APPROX. GRAVIMETRIC LEVEL MG/L	
							SILICA	METAL
1	3	0-1	400	12/9	EQUIVA-LENT TO CETOP CODE	1/4	0.12	0.36
2	4-5	1-2	-	15/9		1/7	0.20	0.60
3	5	2-3	800	16/10		2/8	0.31	0.94
4	7	4	2000	18/12		4/10	1.03	3.11
5	8-9	5-6	-	19/13		5/11	2.40	7.25
6	9	6	6300	20/13		5/12	3.70	11.17
7	9-10	NOT AVAILABLE	-	20/13		5/12	4.37	13.19
8	10		-	21/14		6/13	5.74	17.33
9	10-11		15000	21/14		6/13	7.06	21.31

Table 8.5.1 Comparison of Different Contamination classes at approximately equal contamination levels.

used table 8.5.1 to collate the available data, the results obtained being summarised in table 8.5.2.

Until such time as CHL can derive its own recommended contamination levels from service experience, the figures given in table 8.5.2 represent the best available recommended contamination levels for CHL's pumps. Because not all users are familiar with the Thermal Control class system, CHL should therefore advise its customers to monitor and maintain fluid contamination levels below one of the following:

(i) Thermal Control class 6.

(ii) NAS 1638 class 9.

(iii) SAE 749 class 6.

(iv) MOD standard 05-42 class 6300.

(v) ISO contaminant code 20/13.

(vi) Sperry-Vickers' contaminant code 5/12.

(vii) A gravimetric level of 3.5 mg/l to 12.0 mg/l, depending on the system contaminants present.

TYPE OF SYSTEM	RECOMMENDED CONTAMINATION LEVEL (THERMAL CONTROL CLASS).
Small, high-pressure control systems, such as missile control systems.	1 to 3
Small/medium capacity, medium/ high pressure systems. e.g. aircraft controls, machine-tool hydraulics.	4 to 5
Medium/large capacity, medium/ high pressure systems. e.g. hydrostatic transmissions, any system with high pressure pumps, commercial aircraft hydraulic systems.	6 to 7
'Industrial systems'. Winches, elevators, surface ship systems, low pressure systems.	8 to 9

(Data collated from recommendations published by the Thermal Control Company, Hiac Inc., and the Royal Navy).

Table 8.5.2. Recommended Contamination classes for different types of hydraulic system.

9.0 PROJECT REVIEW AND CONCLUSIONS

9.1 Introduction

This chapter contains a brief review of the research described in the thesis. The work is first placed into the context of an overall programme of research into fluid contamination and its effects. The conclusions of the work are then summarised and related to CHL's needs and to the original project objectives. The chapter concludes with a discussion of possible future work.

9.2 Fluid Contamination Research

Any industrial project oriented towards the solution of a problem can be divided into five stages. These are:

(i) Precise problem definition.

(ii) Definition of the qualities of an acceptable problem solution.

(iii) Identification of research strategies which might lead to a solution.

(iv) Feasiblity studies to assess the viability of each possible strategy.

(v) Implementation of viable strategies to achieve a problem solution.

Of course, not all projects proceed through all five stages. Many are terminated early in their lives because it becomes clear that the potential benefits of a problem solution cannot justify the expense involved in achieving that solution.

The work described in this thesis was dominated by the fact that fluid contamination represented a new field of study for CHL. The research was therefore structured to take the company through the first four stages described above, with some work into stage (v).

The chief value of the work, both to the sponsoring company, and to other pump manufacturers who might study the thesis, is as a feasibility study into work on fluid contamination. This is quite normal with work in a new area of study. As Twiss [93] says: "The early stages of an R and D programme ... can be thought of as an investment of resources ... to determine the viability of the project". This research should be regarded as the first step along a (potentially) very long road, rather than as a completed project. A partial solution to the problem of fluid contamination has been suggested, but this solution could be refined in the future if the potential benefits were worth the cost of the work.

The conclusions of the work can be summarised in terms of the five project stages described above.

9.3 The problem of Fluid Contamination

The initial project objective, as perceived by the company, was to determine whether or not it is possible to rate the ability of a pump to operate on contaminated fluids. In fact, a solution to this problem, although inherently worthwhile, only has true value within a broader context. This was shown in chapter three, where the basic problem of fluid contamination was identified as being to determine and maintain an 'economic' contamination level for a given hydraulic system.

9.4 A 'Solution' to the Problem of Fluid Contamination

Having clarified the problem of fluid contamination, it was possible to identify an 'ideal solution'. This was shown in figure 3.6.1. Within such a solution, it is the task of CHL (and other manufacturers of hydraulic equipment) to:

(i) Determine acceptable contamination levels for their equipment.

(ii) Strive to improve the ability of their products to operate on contaminated fluids (and hence increase the levels recommended for their equipment).

The work described in this thesis has been largely directed towards the initial stages of these tasks. But the research will only make complete sense in the context of an industrial approach to the problem. This therefore requires that CHL press its views on all other parties involved in the design, operation and construction of hydraulic systems. A third aspect of the problem solution was therefore that CHL must:

(iii) Improve its own knowledge of fluid contamination and its effects. This knowledge must now be disseminated to users of hydraulic pumps.

9.5 Research Strategies

Three possible research strategies were identified in chapter three. These were to pursue the project objectives through theoretical analysis of pump design; testwork (either on complete pumps or on specialised rigs), or by the collection and analysis of data from the field. At the start of the work, these strategies were regarded as separate options, any one of which could be conducted in complete isolation from the other two. While this was a useful concept in helping to bring some form of order to a confused process, it is now clear that it is an oversimplification. A complete programme of work must use each approach. For example, much of the theoretical work conducted could not proceed without more data as to what conditions were expected in service. This problem also arose when trying to relate an essentially unrealistic pump contamination test to field experience. Each research strategy is feasible in pursuit of some, but not all objectives of the work. The next section will therefore consider the potential uses of each approach.

9.6 The uses of each Research Strategy

9.6.1 Theoretical pump design analysis

Design analysis of an axial piston pump can be used to indicate the sizes of contaminant particles likely to produce damage. However, it was shown in chapter four that such knowledge is of limited value, especially in relation to the problem of filter specification. The value of design analysis is in suggesting pump design or materials improvements to reduce the effects of fluid contamination. At present, such work is hampered by the lack of a realistic mathematical model for contaminant size distributions, although a suitable model was suggested. This lack suggests the need for more data from the field and for more work on the accuracy of particle counting.

To summarise:

(i) Design analysis cannot be used to determine an acceptable fluid contamination level for an axial piston pump.

(ii) Theoretical analysis can be used to suggest design improvements to reduce the effects of fluid contamination on a pump. However, more work is needed in this field, and one must also remember that any suggested improvement must be verified in some way. This problem is considered in section 9.6.2 below.

(iii) Theoretical analysis has helped to improve CHL's knowledge of fluid contamination and of its effects.

9.6.2 Pump testing

The problems of pump contamination testing were considered in chapters 5, 6 and 7. One important conclusion reached was that it is not possible to use pump tests to estimate pump reliability. This is a conclusion with widespread implications outside the bounds of this research. It means that a manufacturer of an engineering product cannot be absolutely confident, when releasing a new product onto the market, that his equipment will have an acceptable reliability in service. This is an unpalatable conclusion, but it must be faced. It has two major implications. The first is that the early service performance of any new product must be carefully monitored to identify problems and take remedial action where necessary. The second is that the objective of development testing should not be the estimation of equipment 'life' or reliability. It must instead be to identify equipment failure modes and to pinpoint weak components or areas in new products. This knowledge may be used to suggest immediate improvements, or it may be used to determine the priorities and direction of theoretical analysis or more basic research.

The pump tests described in chapters six and seven demonstrated both the feasibility and limitations of a pump contamination test. Such tests can be used to improve the general understanding of the effects of fluid contamination. They can also be used to study effects such as abrasive wear. However, a pump contamination test can only be an unrealistic process, and the doubt still remains that the results may not be relevant to what will happen in service. Future work will have to deal with this problem if tests are to be used to predict what will happen in service.

The question of a pump contamination rating was considered during the work. The prime value of such a rating would be to verify the beneficial effects of any design improvements made to reduce the effects of contamination on a pump. In this role, a pump contamination test would be a powerful and useful tool. However, this use is not yet feasible. Although a suitable test can be suggested on the basis of this work, two problems remain:

(i) The test would be unrealistic and it would have to be clearly demonstrated that a pump which performed well in the test would also perform in service.

(ii) More data is needed on the precise effects of fluid contamination in service. Only then can a suitable test parameter be selected to quantify the effects of contamination on a pump.

Thus, to summarise the feasible uses of pump testing:

(i) Pump tests cannot be used to determine an acceptable fluid contamination level for a given pump.

(ii) Pump tests can be used to suggest design improvements to reduce the effects of contamination on a pump, although it is difficult, at present, to verify the benefits of any such improvement.

(iii) Pump tests have helped to improve CHL's knowledge of fluid contamination and of its effects.

9.6.3. Service Experience

The true test of the performance of an industrial product is the performance of that product in service. All other design and development work is an attempt to obtain an early prediction of that performance and to improve it wherever possible. The conclusions of this work are that:

(i) Service experience is the only way to determine acceptable contamination levels for a given type of hydraulic pump.

(ii) Much more service data is needed on the effects of fluid contamination on pumps. In particular, data is needed on the levels of contamination to which CHL's pumps are exposed, and on the effects of this contamination. Only with such data can other work be directed towards the most pressing problems.

Although data is needed on service conditions, such data is difficult to obtain, largely because its collection requires co-operation between different organisations. Indirect service experience was all that could be obtained in this project, but this was used, in chapter eight, to derive recommended contamination levels for CHL's pumps.

9.7 Project Outcomes

It has already been stated that the usefulness of this work to CHL is primarily as a feasibility study for work on fluid contamination. However, the work has had more specific outcomes. As a result of the research, CHL have:

(i) A much improved knowledge of fluid contamination and of its effects. This knowledge is held partly in this thesis, partly in reports to the company, and will be summarised in a booklet to be prepared in the near future. (This will be discussed below.)

(ii) The company has a set of practical, recommended contamination levels for its products. These levels should be included in the company's sales literature and in its operating and maintenance instructions for users.

(iii) The company has a set of theoretical design approaches which it can apply to its products. However, more service data is required before these methods can give truly worthwhile results.

(iv) The company has an appreciation of the uses and limitations of pump contamination test methods.

(v) The company has a much better understanding of filter testing and specification.

9.8 Future Work

CHL's management will need time to study and assess the work described in this thesis before they commit the company to any further study into fluid contamination and its effects. However, the author would suggest that the following should be the priorities of any future work, both for CHL and for any other pump manufacturer contemplating research in this area.

(i) Education and customer service.

Many of the problems caused by fluid contamination have their root cause in ignorance of the damage that fluid contaminants can cause. Where the danger is appreciated, system users often need help in identifying specific problems and taking remedial actions. As a result of this research, CHL has a great deal of expertise in this field, and it is now in the company's own interests to disseminate this knowledge, and to provide service and assistance to its customers to help them with contamination-related problems. This is so for two reasons:

(a) "Good communications" and "after-sales-service" are major factors influencing hydraulics users in their choice of suppliers. (See reference [1]).

(b) The work conducted to date only makes sense in the context of an 'industrial approach' to the problem of fluid contamination. For this research to be truly useful, CHL must now press its views on the hydraulics industry as a whole, with a plea for some concerted action.

The chief educational point to emerge from this work is that the current method of specifying protection against contamination needs to be changed. A pump manufacturer should not specify protection as a required filter rating; he should, instead, specify an acceptable fluid contamination level for his product. It is the task of the user, with the active assistance of the pump manufacturer, to monitor and maintain this level. For this approach to be feasible, system designers must provide correctly-located fluid sampling points.

Of course, a Ph.D. thesis is not an acceptable vehicle for a process of industrial education. The author will therefore prepare a booklet, describing fluid contaminatin, its effects and the ways to reduce them. The booklet will be written for an audience of practical engineers and skilled fitters.

(ii) Service experience.

The next priority must be to obtain more data from the field. Without such data, it is not possible to direct theoretical analysis and testwork towards the most pressing problems, simply because the hydraulics' industry does not know what these problems are. CHL needs to know:

(a) What contamination levels do the company's pumps experience in service?

(b) What are the effects of fluid contaminants on CHL's pumps? The company needs to know the types and location of contaminant-induced damage, and also how often contamination is a symptom, rather than a cause of problems. This latter work could be usefully extended to include a more thorough investigation of the potential uses of Ferrography.

(c) If the company wishes to pursue the type of theoretical analysis developed in chapter four, then a mathematical model of the size distribution of system contaminants will be required. Although a suitable model was suggested in section 4.8, this model must be verified against the size distribution of contaminants from operating systems.

The above suggestions represent a considerable programme of work. The programme could be extended to include tests on CHL pumps, or on more basic research rigs to simulate one pump area. It is the author's opinion that this type of work should, for CHL at least, take second place to the priorities outlined above.

It is clear that the entire field of fluid contamination research is too great to be adequately investigated by one company, and CHL may prefer to engage, in future, in a collaborative programme of research. Alternatively, the company may concentrate on a few aspects of the problem which are deemed to be of over-riding importance. Possibly the most useful outcome of this work is that CHL are now able to make a rational and well-informed choice of future programmes.

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APPENDIX A

Ferrography

There are many occasions when information is required on the wear processes which have been taking place inside hydraulic or lubricating systems. One common method of gaining such information is to dismantle the plant involved and examine component parts for wear.

While this approach can give useful information, it does have limitations.

(i) Dismantling plant is inherently expensive, and costs are also incurred due to lost production while equipment is being examined.

(ii) Physical examination of equipment components can only give information about their condition at one instant of time. Selecting a time for examination is difficult and plant is usually studied only after it shows signs of problems. By this stage it is often both too late to prevent failure, and too late to identify the original problems.

(iii) Components often have to be cut up or prepared in some way before they can be analysed for wear. This is expensive.

(iv) Maintenance engineers are quite naturally loathe to dismantle plant that appears to be operating satisfactorily. This makes it difficult to obtain information as to what constitutes 'normal wear'. Also, plant can never be reassembled in the same state that it was in before examination. A series of studies on equipment wear might therefore be misleading if the data was obtained in this way.

What is needed is a means of diagnosing wear without dismantling equipment. One obvious approach is to examine the wear particles present in samples of fluid from hydraulic and lubricating systems.

Wear particles can be examined by simply filtering a fluid sample through a membrane and then examining the membrane under an optical microscope. However, the numbers of wear particles tend to be small compared to other contaminants from the system environment. They also tend to be randomly distributed as regards their size, and this makes it difficult to detect trends in wear particle size and/or concentration.

Ferrography is one way to overcome these difficulties. The technique of Ferrography is simply a method of preparing a microscope slide of ferrous wear particles classified into a size order. The procedure is illustrated in figure A1. A small sample of contaminated fluid (usually 2 mls) is mixed with a solvent. The sample is forced, by a peristaltic pump, down a plastic tube, and then flows down an inclined microscope slide placed in a strong, vertical, magnetic field. The fluid, and the majority of non-ferrous particles, flow down the slide, are collected, and pass to waste. Ferrous particles are pulled down onto the slide surface, larger particles towards the upper end of the slide, smaller ones lower down.

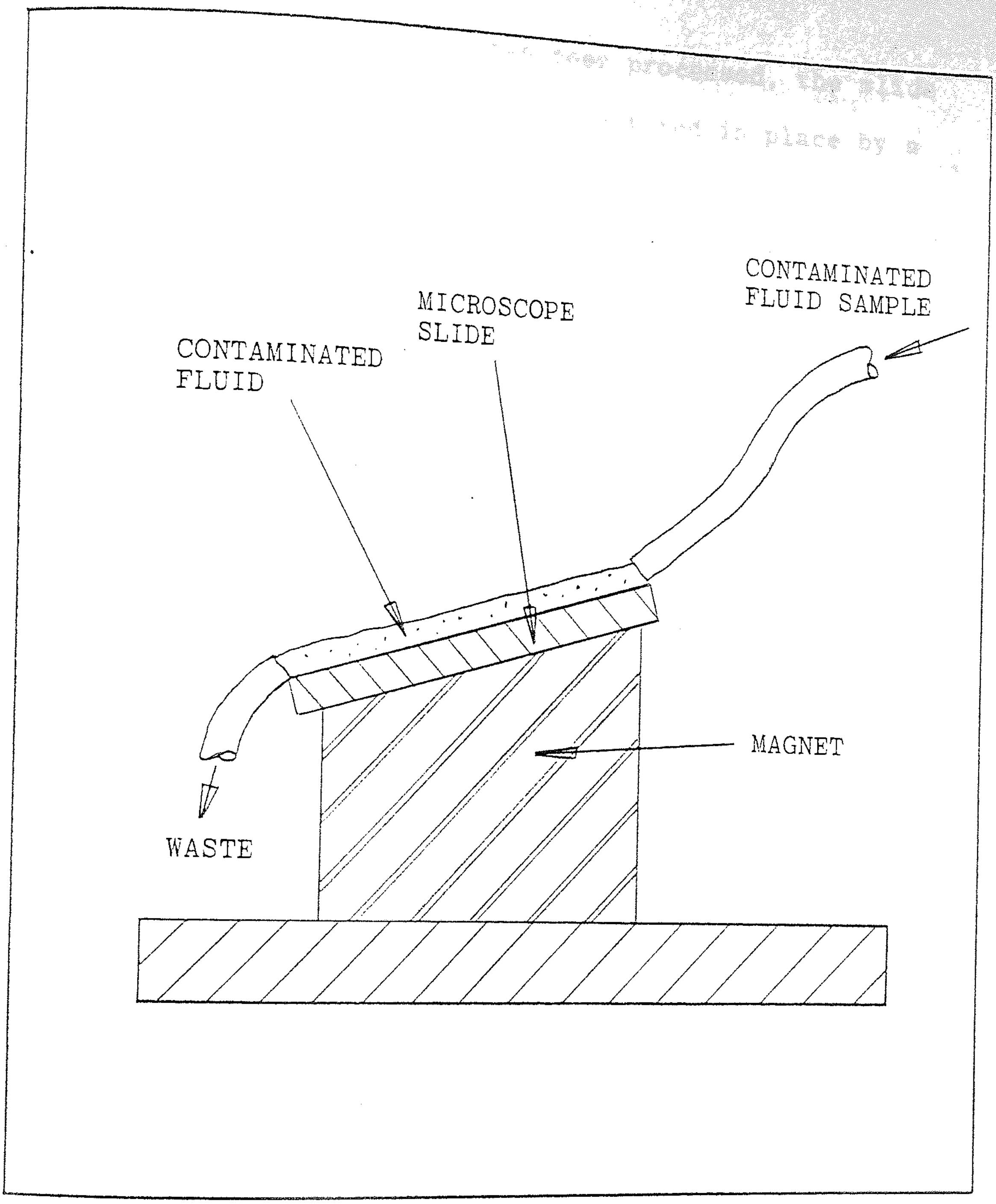

Figure A1. Ferrography slide preparation.

When the entire sample has been processed, the slide is washed and the particles are fixed in place by a chemical.

Ferrography slides can be examined using a number of techniques, although the most common are optical and electron microscopes. The aim of such analysis is to determine the size, shape, appearance, concentration and materials of the particles present. An experienced operator can use such information to diagnose the type(s) of wear which has been taking place in the system from which the sample was taken. He can also give some indication of the severity of the wear and, most importantly, he can give advance warning of an impending failure. It is this latter feature which makes Ferrography a potentially valuable tool for use in industry. (Further details of Ferrography are given in references [44] and [36]).

APPENDIX B : Questionnaire on filter testing

During the course of the research, information was needed on the performance of several hydraulic filters. This raised questions as to how filter manufacturers test and rate the performance of their products. To answer these questions, a questionnaire was prepared and distributed to most of the major UK filter manufacturers, of whom nine returned completed forms. The questionnare is reproduced below:

IHD questionnaire on testing and rating hydraulic filters

Would you please complete as much of this questionnaire as you are able and/or willing to do. Most questions require only short answers, but please amplify any points that you wish.

1. Which types of filter do you manufacture?
   (Please tick as appropriate).
   High pressure (To . psi)
   Return line
   Suction line
2. Please indicate the range of fluid flows covered by your products.
3. Do you apply a "nominal" rating to your products?
4. Is this rating based on a test method?
5. How do you define a nominal rating?
6. Do you apply an "absolute" rating to your products?
7. How do you define an absolute rating?
8. Are you aware of work being carried out at Oklahoma State University in the field of filter testing?

9. If so, do you test filters in accordance with the OSU multipass filter test? (ANSI B93. 19-1973)

10. If your answer to question 9 was "yes", do you publish $beta_{10}$ ratios for your products?

11. Are you aware of work on filter testing being carried out at the British Hydromechanics Research Association?

12. If you have an "in-house" test procedure, would you please answer the following:-

12.1 What test contaminant do you use?

Air cleaner fine test dust.

Latex spheres.

Glass spheres.

Other. (Please Specify).

12.2 Do you test at constant pressure?

12.3 How do you select this pressure?

12.4 How do you select test flow rates?

12.5 Do you conduct tests at pulsating pressures?

12.6 Do you conduct tests at pulsating flows?

12.7 At what fluid contamination levels do you conduct your tests?

12.8 What hydraulic oil do you use in filter testing?

12.9 If you test over a period of time, at what point do you measure filter performance?

13. How do you measure fluid contamination levels?

Hiac counter ; Royco Counter ;

Coulter counter ; Optical microscope ;

Gravimetric .

14. If you use a Hiac/Royco instrument, do you calibrate it using Air cleaner fine test dust?

15. Which definition of particle size do you use?

Longest dimension.

Martin's diameter.

Feret's diameter.

Projected Area (Hiac).

Equivalent sphere (Coulter).

Other.