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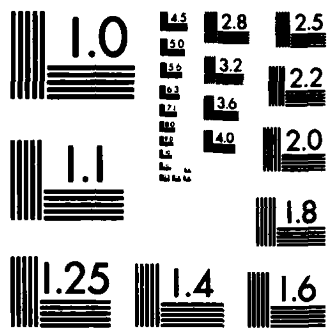

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LUBRICANTS

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1. INTRODUCTION

A lubricant is any material used to separate two surfaces in relative motion which can be readily sheared while adhering to the surfaces. In the process of acting as a lubricant, the material protects the solid surface from unacceptable wear while dissipating an accepatably small amount of energy. Any deformable media is a potential lubricant. The lubricant might be a deformable media readily available in the manufacturing process (that is, a process fluid lubricant) or a material specifically selected and introduced for its lubricating quality.

This chapter will discuss the range of materials used as generic lubricants (solids, liquids, gases, and greases), relevant physical and chemical properties, typical lubricants, major selection criteria, and an overview of specification procedures. Chapter emphasis will concern itself with an introduction to lubricants and with the literature on lubricants.

2. FUNCTIONS OF A LUBRICANT

Lubricants are functional materials in the mechanical system. The primary functions to be performed by the lubricant are to separate the surfaces, control wear, reduce friction, and reduce pitting fatigue. Frequently the function of separating surfaces is referred to as the load-carrying capacity or ability of the lubricant. Most of these primary functions of the lubricant are really not functions simply of the lubricant itself, but of the complete mechanism, and therefore depend not only on the lubricant properties and characteristics but also on the kinematics and dynamics of the system being lubricated.

Secondary, yet very important functions of the lubricant in tribological systems, are the scavenging of heat, dirt, and wear debris from the contact region, the prevention of corrosion throughout the system, and sealing, when mechanical seals as opposed to bearing surfaces are being lubricated.

These primary and secondary functions of the lubricant are related to a number of physical and chemical properties of the lubricant and the lubricant bearing material system. Table 1 consists of a list of a number of the lubricant and system/lubricant properties which are frequently discussed when considering a particular selection.

TABLE 1

LUBRICANT/SYSTEM PROPERTIES AFFECTING LUBRICANT FUNCTIONAL PERFORMANCE

DENSITY	SOLVENCY	VOLATILITY
VISCOSITY	PITTING	PAINTABILITY
POURABILITY	CORROSION	MISCIBILITY
FLAMMABILITY	RUST	EMULSIBILITY
FILTERABILITY	WETTABILITY	SURFACE TENSION
COMPATIBILITY	THERMAL COND.	SLUMPABILITY
STABILITY	ELECTRICAL COND.	DROPPING POINT
THERMAL	PENETRATION	VISCOELASTICITY
OXIDATIVE		FOAM
HYDROLYTIC		AIR RELEASE
BULK MODULUS		
HEAT CAPACITY		

The selection of a lubricant for a particular application is frequently very complex. The decision requires balancing the physical property requirements of the system, the chemical property requirements of the system, and the life requirements for the particular application, as well as the considerations of availability and cost. The Tribology Handbook presents a very useful but somewhat arbitrary initial selection procedure for lubricants based on bearings, loads and speeds as shown in Figure 1[1]. These selection limits are related to the thermal limits of materials as well as to considerations of feeding and traditional lubrication mechanisms used for the different range for the variables plotted. The first estimate

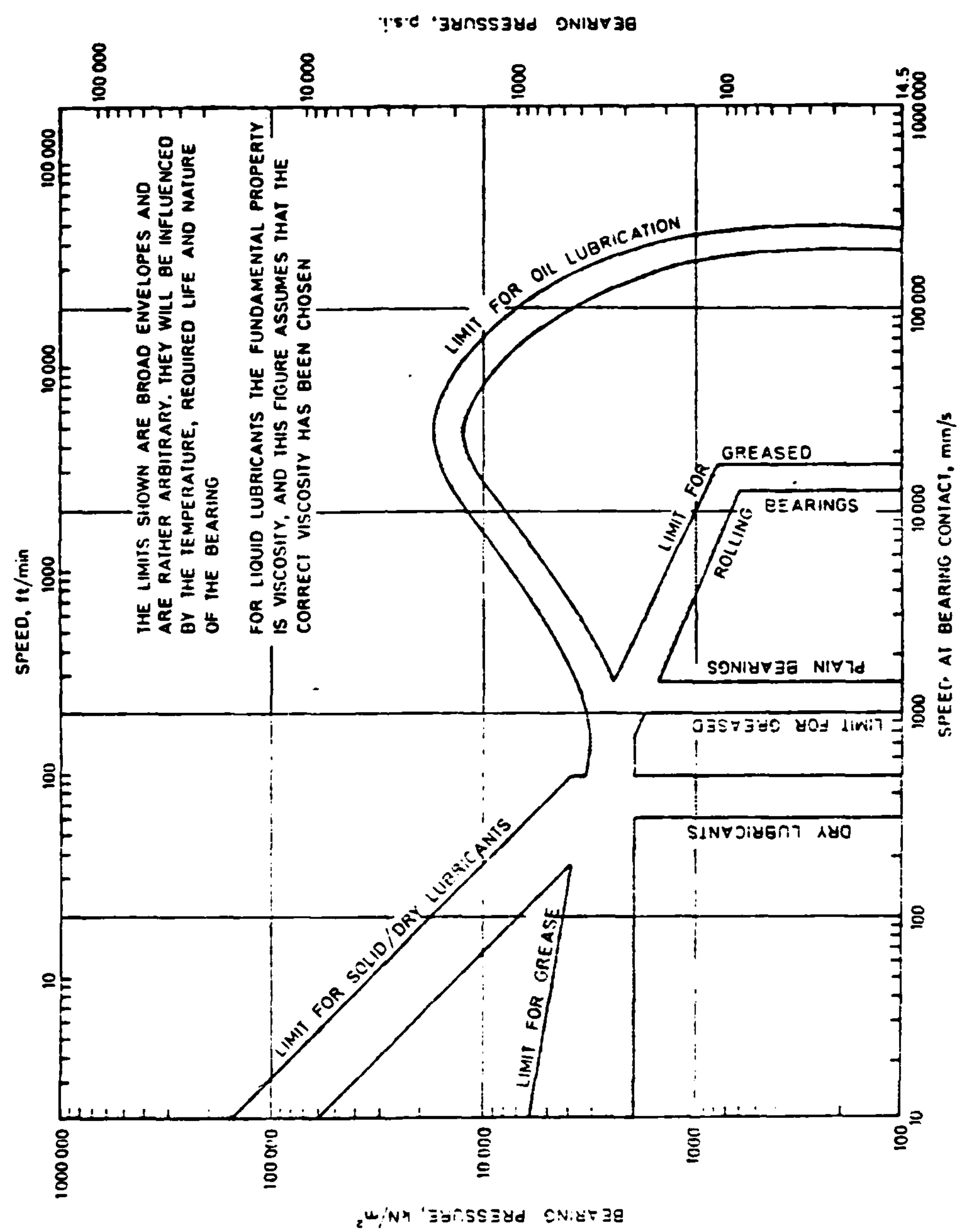


FIGURE 1 SPEED/LOAD LIMITATIONS FOR DIFFERENT TYPES OF LUBRICANT (REF. 1)

selection procedure shown in Figure 1 is to be modified and qualified by additional considerations presented throughout this chapter.

3. PHYSICAL AND CHEMICAL PROPERTIES OF LUBRICANTS (NON-RHEOLOGICAL)

Among the many physical properties of a lubricant that are of interest, the rheological properties are the most significant and will be discussed separately. In general, the remaining physical properties of interest vary over a relatively narrow range when considering a given chemical class of lubricants. This can be readily seen in Table 2 from the Tribology Handbook which presents the typical physical properties for highly refined mineral oils[1]. As we see from examining Table 2, the physical properties such as density, bulk modulus, thermal capacity, thermal conductivity, and vapor pressure, each have a relatively narrow range of property values for all of the mineral oils presented in the table. However, the viscosity has a rather wide range of values both among the different materials and at different temperatures for the same material. The different oils in the table represent a range of average molecular weights. The molecular weight has a strong influence on the viscosity as well as the pour point, vapor pressure, and the flash point for the material. Similar generalizations concerning the variation of physical properties within classes of synthetic lubricants can also be made. Properties for synthetic lubricants similar to those for mineral oils shown in Table 2, can be found in the literature, Tribology Handbook, Lubrication Handbook, and Synthetic Lubricants[1,2,3].

Of the chemical properties mentioned in Table 1, by far the most important is the oxidation stability of the lubricant which normally determines the life/temperature limitations of the lubricant in a particular application. In most lubrication applications there is adequate oxygen present in the system such that the life/temperature limitation is determined by the oxidation deterioration of the lubricant. In those special situations, where oxygen is not present, the thermal stability or the chemical reactivity of the lubricant at elevated temperature is the limiting characteristic.

Oxidation or thermal decomposition of the lubricant results in changing reactivity of the lubricant towards the system as well as changing rheological properties of the lubricant. In most situations, the result of oxidation or thermal degradation is an increase in viscosity which can inhibit the functioning of the lubricant as well as the circulation of the lubricant in the system.

Typical physical properties of highly refined mineral oils (*courtesy:* Institution of Mechanical Engineers)

		Naphthenic oils			Paraffinic oils		
		Spindle	Light machine	Heavy machine	Light machine	Heavy machine	Cylinder
Density (kg/m^3) at	25°C	862	880	897	862	875	891
Viscosity (mNs/m^2) at	30°C	18.6	45.0	171	42.0	153	810
	60°C	6.3	12.0	31	13.5	34	135
	100°C	2.4	3.9	7.5	4.3	9.1	27
Dynamic viscosity index		92	68	38	109	96	96
Kinematic viscosity index		45	45	43	98	95	95
Pour point, °C		−43	−40	−29	−9	−9	−9
Pressure – viscosity coefficient ($m^2/N \times 10^8$) at	30°C	2.1	2.6	2.8	2.2	2.4	3.4
	60°C	1.6	2.0	2.3	1.9	2.1	2.8
	100°C	1.3	1.6	1.8	1.4	1.6	2.2
Isentropic secant bulk modulus at 35 MN/m^2 and	30°C	—	—	—	198	206	—
	60°C	—	—	—	172	177	—
	100°C	—	—	—	141	149	—
Thermal capacity (J/kg °C) at	30°C	1880	1860	1850	1960	1910	1880
	60°C	1990	1960	1910	2020	2010	1990
	100°C	2120	2100	2080	2170	2150	2120
Thermal conductivity (Wm/m^2 °C) at	30°C	0.132	0.130	0.128	0.133	0.131	0.128
	60°C	0.131	0.128	0.126	0.131	0.129	0.126
	100°C	0.127	0.125	0.123	0.127	0.126	0.123
Temperature (°C) for vapour pressure of 0.001 mmHg		35	60	95	95	110	125
Flash point, open, °C		163	175	210	227	257	300

TABLE 2 TYPICAL PHYSICAL PROPERTIES OF HIGHLY REFINED MINERAL OILS (COURTESY: INSTITUTION OF MECHANICAL ENGINEERS)

The rate at which oxidation degradation occurs in the lubricant is influenced by a number of factors. The temperature level is a major contributing factor with the oxidation rate doubling with approximating every 8° to 10°C temperature of the lubricant system. The oxidation rate is also a function of the access to oxygen which is related to the degree of agitation of the oil with air and the presence of oxygen in the ambient. The presence of catalytic materials, particularly iron and copper with large surface areas exposed to the oil, also contribute to the oxidation. The type of oil, e.g., the chemical makeup of the base lubricant itself, as well as the presence of various additives including antioxidant additives, are also factors which determine the life limiting behavior of the oil.

A final influencing factor frequently overlooked is the rate of addition of lubricant to the system. The addition of fresh lubricant replenishes oxidation inhibitors as well as dilutes the buildup of oxidation products in the bulk material. Figure 2 presents some general guidelines concerning the time and temperature limitations of mineral oils[1]. As indicated in this figure, the lower limit of temperature application is primarily the result of rheological property changes at low temperature resulting in too high a viscosity or pour point difficulties inhibiting the flow of the lubricant. These low temperature rheological properties can also be influenced by additives which will be discussed.

Similar data to that shown in Figure 2 for mineral oils can be found in the literature for synthetic oils and greases, Reference 1, Tribology Handbook.

4. ADDITIVES

Although the rheological and boundary lubrication properties of a lubricant are crucial to its primary functions as a lubricant, the total and rather complex additive package used in lubricants is necessary for the successful functioning of lubricants in many applications. When examining the literature on additive technology, one finds at least fourteen different additives, named on the basis of their function, which are used in various lubricants. Probably the largest number of additives used in any lubricant system is that used in internal combustion engine applications where up to ten different additives can be found in a single lubricant.

Table 3 lists fourteen different additives presented by function performed and indicates in which type of machinery applications additives are utilized. In two cases, namely

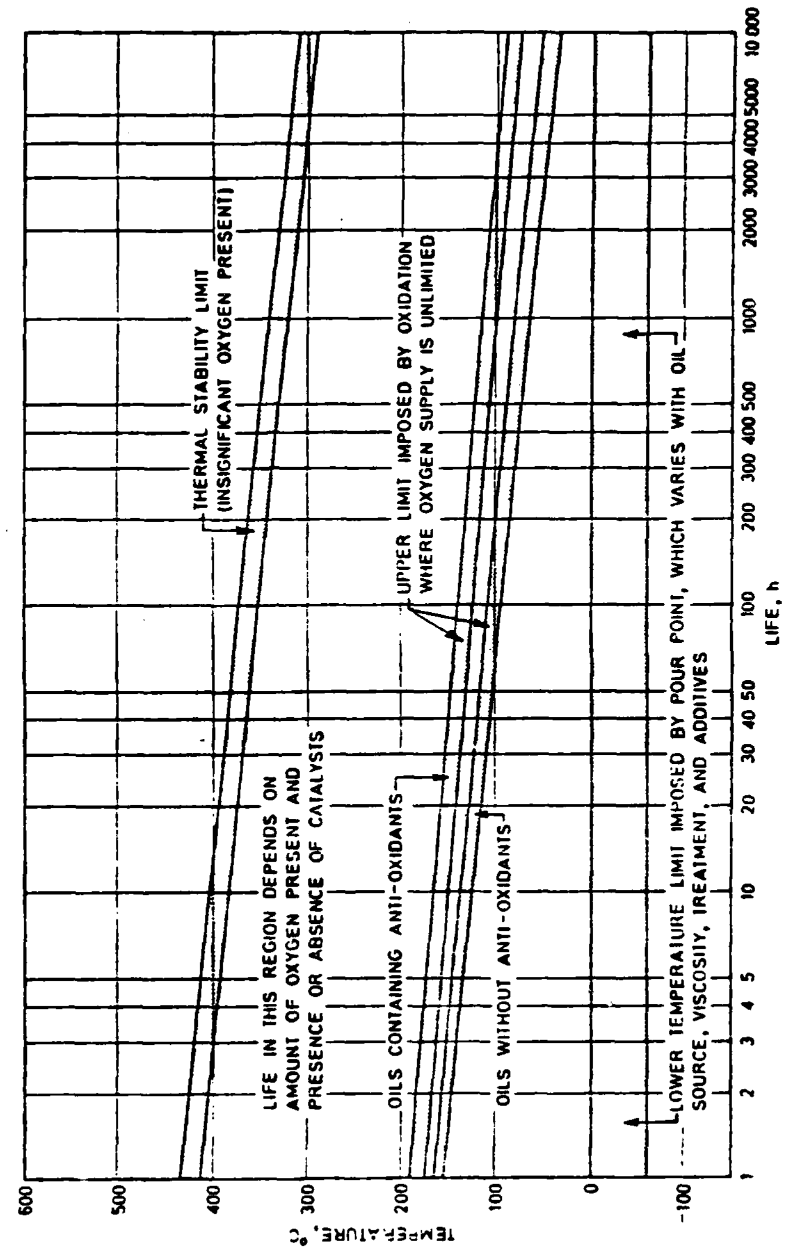


FIGURE 2 TEMPERATURE LIMITS FOR MINERAL OILS (REF. 1)

Additives / Machinery	Food Processing	Hydraulic	Steam & Gas Turbines	Steam Engine Cylinders	Air Compressors	Gears (Steel/Steel)	Gears (Steel/Bronze)	Machine Tool Slideways	Seal Refrigeration Compressors	Diesel & Otto Cycle Engines
Corrosion Inhibitor										x
Acid Neutralizer										x
Dispersant										x
Detergent										x
Tackiness								x		
Oiliness							x	x		
Fatty-oil				x						
Anti-foam		x				x				x
Viscosity Index Improver		x								x
Pour Point Depressant		x				x				x
Extreme Pressure						x				
Anti-wear		x				x				x
Anti-rust		x	x		x					x
Anti-oxidant		x	x		x	x	x			x

TABLE 3 TYPICAL ADDITIVE TYPES FOR MINERAL OIL LUBRICANTS IN VARIOUS TYPES OF MACHINERY

food processing machinery and sealed refrigeration compressors, it is likely that no additives will be found. This situation is prevalent in the food processing industry because the primary consideration is to avoid contamination of the product. In the case of the sealed refrigeration compressors, the lack of oxygen in the system and the design of the compressors permits operating in many cases, without additives.

In some cases, the additives listed are not particularly well defined and in fact the terminology used for some of them, e.g., oiliness and tackiness additives, is frequently objected to by some people in the field. These additives, however, will still be found in promotional literature. Oiliness additives are somtimes referred to as boundary lubrication additives where their function is to reduce friction under boundary lubrication conditions. These additives may increase load-carrying capacity by forming a solid lubricant film on the surface. Tackiness additives, on the other hand, are materials used to prevent the drainage of lubricant from surfaces either due to gravitational or centrifugal force fields.

Table 4, taken from Lubrication Magazine (a Texaco publication), shows examples of additives used in lubricating oils with typical chemical compositions of materials used for the additive type as well as a summary of the purpose of the particular additive. The chemical compositions listed should be considered only as typical and not all inclusive. For example, under viscosity index improvers, many other types of polymers are used in addition to the methacrylate and butylene polymers listed in the table.

Additive chemistry and additive formulation is a complex art in the field of lubrication. Many of the additives are chemically reactive under the conditions of application and several of them are surface reactive as well. The competition among additives for functioning is a major consideration in the formulating of lubricants. In addition, a seldom recognized complication is that there are many additives employed in the earlier stages of lubricant production as well as in other parts of a lubricating system which can interact with the additives in the lubricant itself.

There are appproximately twelve additive types used in the production of crude oil and refinery processing which might be found as carry-overs into the final product of a mineral oil lubricant. The behavior of these additives, in conjunction with those added to the lubricant for lubricating

Additive Type	Chemical Composition	Purpose
Viscosity index improver	Methacrylate polymers, Butylene polymers	Lower the rate of change of viscosity with temperature.
Pour point depressant	Alkylated naphthalene	Decrease pour point of oil.
Detergent-dispersant	Alkyl P_2S_5 products, metal sulfonates, alkylpolyamide, metal alkyl phenolates	Keep insolubles in suspension and maintain cleanliness.
Oxidation inhibitor	Zinc dialkyldithiophosphate	Retard oxidation of oils.
Rust inhibitor	Alkylamines	Prevent rusting of ferrous metals.
Corrosion inhibitor	Basic metal sulfonates	Prevent acidic materials from attaching to metal surfaces.
Extreme pressure agent	Sulfurized olefins, chlorinated paraffins	Prevent seizure of metal surfaces.
Foam inhibitor	Silicone polymers	Decrease tendency to foam.
Anti scuff-wear agent	Metal salts of alkyl acid phosphates.	Provide chemical polishing and reduce wear.

TABLE 4 EXAMPLES OF ADDITIVES USED IN LUBRICATING OILS
(COURTESY TEXACO MAGAZINE LUBRICATION)

characteristics, can be a complicating factor in the final performance of the lubricant.

In addition, in the application of internal combustion engines there may be up to ten additives used in the engine fuel which, as a result of piston blow-by of the combustion gases into the engine sump, may end up in the lubricating oil. The presence of these fuel additives in the lubricant can be either detrimental or beneficial depending on the additive, its interaction with the lubricant, and the environmental conditions. For example, if tetraethyl lead antiknock compound is used in the fuel, lead will be found in very high concentrations in engine oils after a significant number of miles have been accumulated on the oil. These lead compounds frequently act as lubricants in highly loaded contact areas such as valve-trained mechanisms. The removal of tetraethyl lead from the fuel of vehicles in the United States put additional requirements on the engine oil additives to make up for the loss of the lubricating performance of the lead compounds in the lubricating oil.

Lubricating greases are also likely to contain a substantial number of additives to improve functional performance. Table 5, taken from Lubrication Magazine, is a typical example of the list of types of additives that may be found in lubricating greases. Like the base oils themselves, the additive life and degradation is also a function of system temperature. Additive degradation or depletion is frequently the major cause for lubricant change requirements.

5. SOLID LUBRICANTS AND BOUNDARY LUBRICANTS

As discussed in the previous chapter on Lubrication, solid lubricants and boundary lubricants can be classified conceptually in the same category because they both function as the result of the formation of a solid film surface. That solid film on the surface, to function acceptably, must adhere to the surface and be readily sheared by the motion of the two solid surfaces.

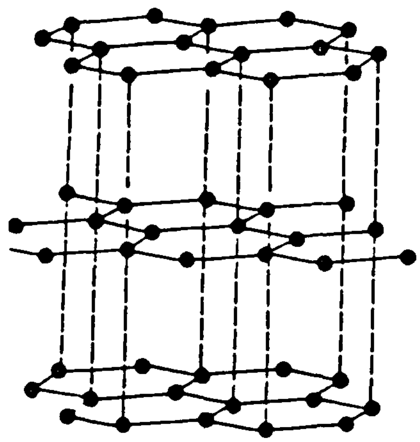
Solid lubricants are typically solid materials which are directly applied to the surface by a number of processes or put into solution in the oil or as a filler in the grease in order to be attached to the surface during the lubrication process. Boundary lubricants, on the other hand, are generally liquids or chemicals which are soluble in the lubricant as an additive and form a solid film on the surface by one of a number of possible interactions with the surface.

Additive Type	Chemical Composition	Purpose
Thickening agent	Metal soaps	Hold fluid by adsorption.
Fillers	Metal oxides	Add bulk to grease.
Oxidation inhibitor	Phenyl-beta-naphthylamine	Inhibit oxidation.
Metal deactivator	Mercaptobenzothiazole	Prevent catalytic effect of metals.
Corrosion inhibitor	Ammonium dinonyl naphthalene sulfonate	Arrest corrosion.
Anti-wear agent	Dibenzyl disulfide	Reduce wear.
Extreme pressure agent	Chlorinated wax Lead naphthenate	Reduce friction.
Dropping point improver	Fatty soaps	Increase dropping point.
Stabilizer	Fatty acid esters	Increase usable temperature.
Tackiness agent	Polybutylenes	Provide adhesiveness on metal surface.

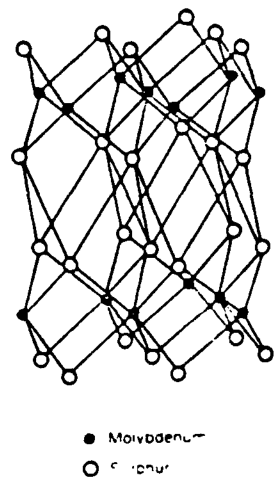
TABLE 5 EXAMPLES OF ADDITIVES USED IN LUBRICATING GREASES (COURTESY TEXACO MAGAZINE LUBRICATION)

The solid film of the boundary lubricant can be formed on the surface by physical adsorption, which is a relatively weakly bonded film somewhat similar to the condensation process of a gas onto a surface, or by chemical adsorption which is a somewhat more strongly bonded film on the surface resulting from a chemical bonding between the boundary lubricant additive and molecules in the surface. It can also be formed by chemical reaction which is similar to a mild form of corrosion in which surface molecules are removed to form new chemical compounds with the boundary lubricant. The resulting compound material remains on the surface. The films formed by reaction are generally the more strongly adhered films. The mechanism of film formation is dependent upon the composition of both the boundary lubricant additive and the surfaces being lubricated as well as the temperature and pressure of the environment.

Solid lubricants are frequently thought of as being lamellar solids, such as molybdemum disulfide and graphite, the structures of which are shown in Figure 3. These two materials are thought to function well as solid lubricants because of the relatively low shear strength planes separating the lamellar structure of the molecule. Although this concept has some basis, it is a gross over-simplification of the performance of these molecules. In particular, the graphite performance is very much dependent upon the atmosphere in which lubrication takes place because the low shear strength of the graphite crystal structure is apparently dependent upon the presence of condensable materials at the edges of the crystal structure. The graphite is not a suitable lubricant under very high vacuum conditions where there are no condensable materials present, whereas the molybdemum disulfide is more suitable under these conditions.



THE CRYSTAL STRUCTURE OF GRAPHITE



THE CRYSTAL STRUCTURE OF MOLYBDENUM DISULPHIDE

FIGURE 3

Although solid lubricants are frequently thought to be lamellar solids, all lamellar solids are not good solid lubricants, and conversely many good solid lubricants are not lamellar crystal structure materials. In fact, many amorphous solid materials possess the necessary characteristics of low shear strength and good adhesion to solid surfaces. In the case of the lamellar solids, orientation of the crystal structure on the surface influences the resulting friction and wear when these materials are used.

A typical schematic of the way boundary lubricants are thought to function is shown in Figure 4. These chemically formed boundary lubricant films are thought to perform several functions. They are thought to fill the valleys and cushion the surfaces through the smoothing of the surfaces permitting micro-elastohydrodynamic lubrication to occur at the asperities. They may also change the mechanical properties of the surface layers of the solids through such phenomena as the Rehbinder effect. The boundary lubricant fillms can range in thickness from a few angstroms to 1 μm or more.

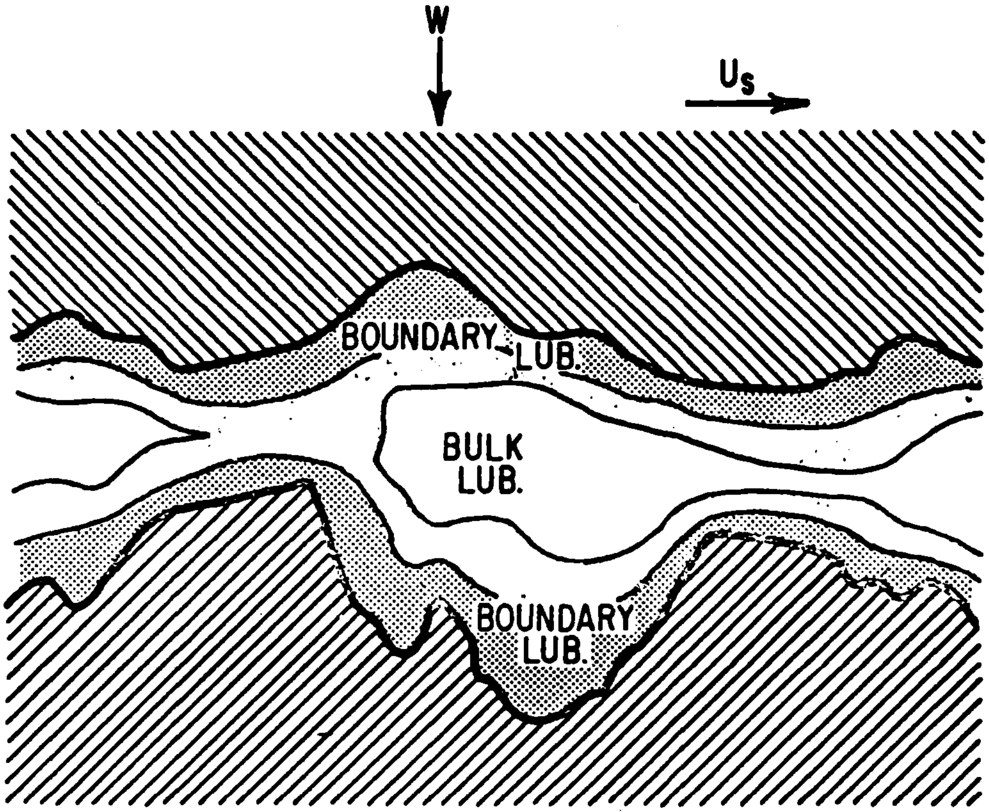


FIGURE 4 SCHEMATIC OF BOUNDARY LUBRICANT FILMS

Even gases which condense on and/or react with the surface can act as boundary lubricants in the sense that they reduce or inhibit seizure or adhesion between the surfaces and can reduce friction compared to the absence of any film on the surface. Figure 5 is an example from Bowden and Tabor of early studies on the presence of gases (H_2S and Cl_2) in vacuum environments on the friction of clean iron surfaces[4]. At room temperature with clean surfaces, there is a seizure between the two iron surfaces resulting in coefficients of friction in excess of two. When gas is present, the coefficient of friction drops to about 1.2 in case of H_2S and to about 0.4 in the case of Cl_2. As the temperature rises, the gas film is eventually desorbed resulting in seizure.

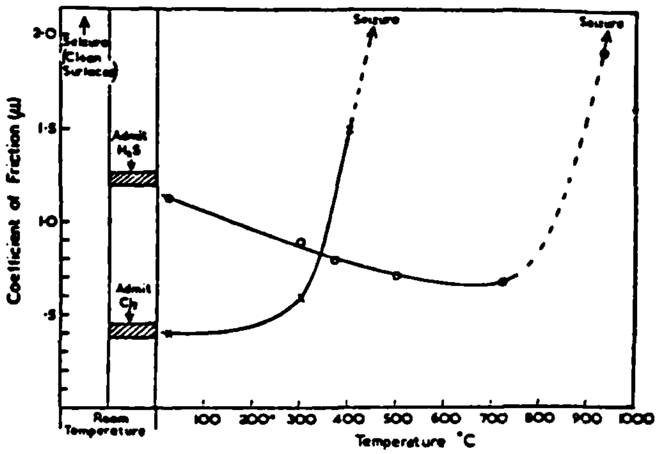


FIGURE 5 EFFECT OF TEMPERATURE ON FRICTION OF CHLORIDE AND SULPHIDE FILMS FORMED ON CLEAN IRON SURFACES. THE CHLORIDE GIVES THE LOWER FRICTION, BUT BREAKS DOWN AT A MUCH LOWER TEMPERATURE (REF. 4)

In most applications however, the primary concern related to boundary lubricants is the influence on either adhesive wear or pitting wear. Figure 6 shows the influence of typical boundary lubricant additives within various classes of chemicals, on the wear coefficient in four ball tests. The four ball tests consist of one rotating ball loaded against three stationary balls and is a common lubricant test device.

Figure 6 shows that some additives can reduce wear by up to three orders of magnitude as compared to that exhibited by the base oil, and in some cases additives can increase the wear compared to the base oil wear. Table 6 shows a range of wear coefficient values for four ball tests on 52100 steel with various lubricants and boundary lubrication additives. The ambient atmosphere, as well as the lubricant type and additive, have significant influence on the wear coefficient. This influence, for the conditions shown in Table 6, can vary by more than eight orders of magnitude from that existing in a dry argon environment to that existing with a fully formulated engine oil in an air environment.

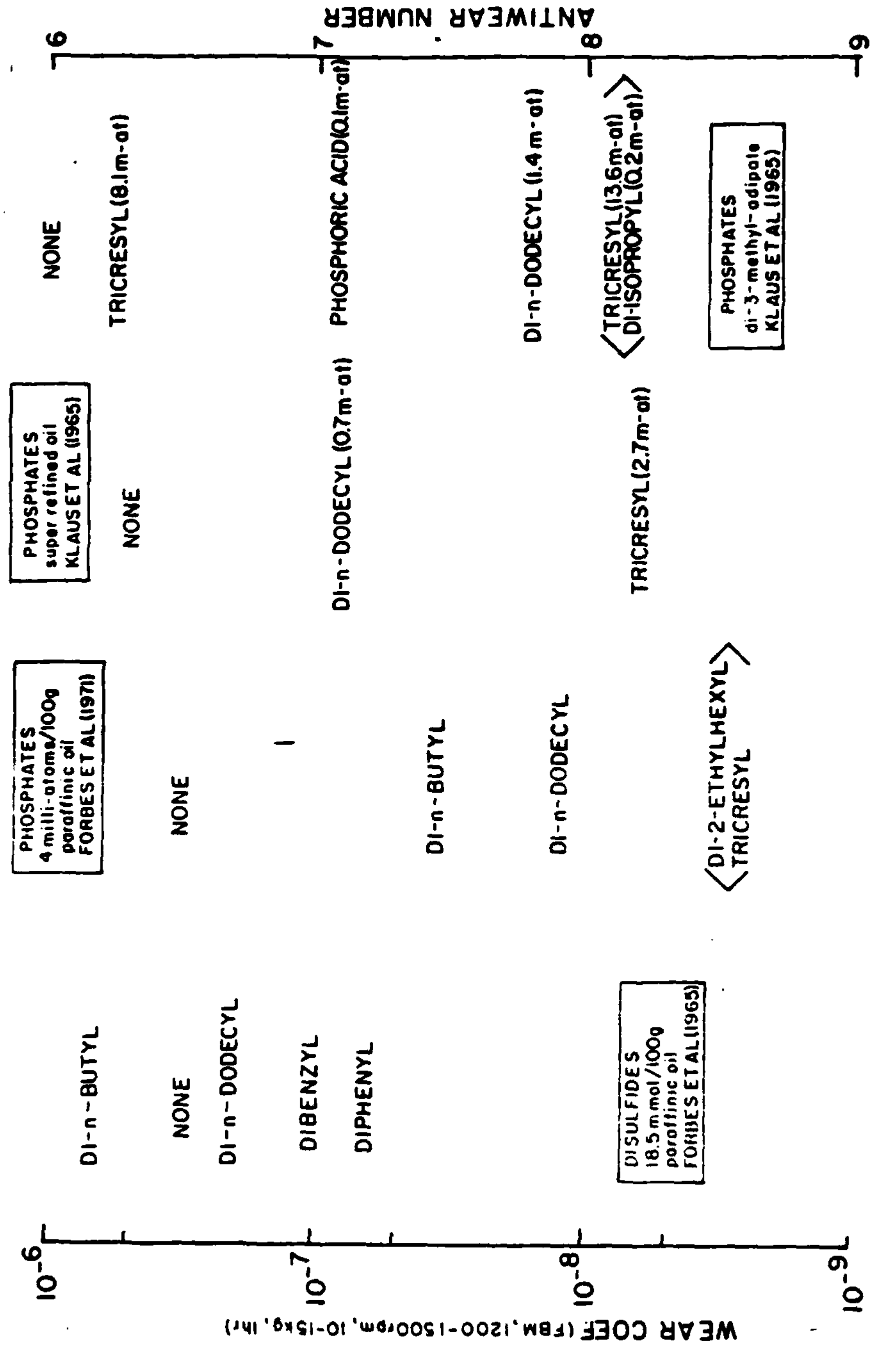


FIGURE 6 INFLUENCE OF TYPICAL BOUNDARY LUBRICANTS ON WEAR IN FOUR BALL MACHINE (FEIN, REF. 5)

AISI 52100 STEEL

Lubricant	Atmosphere	K
None	Dry Argon	1.0×10^{-2}
None	Dry Air	1.0×10^{-3}
Cyclohexane	Air	8.4×10^{-6}
Paraffinic Oil	Air	3.2×10^{-7}
Tricresylphosphate/ Paraffinic Oil	Air	3.3×10^{-9}
Engine Oil	Air	$<2.0 \times 10^{-10}$

TABLE 6 COMPARISON OF K VALUES (REF. 5)

For any given type of boundary lubrication additive, the chemical reactivity with the surface is important to the performance of the additive and is heavily dependent upon the surface temperature. Figure 7 shows the influence of surface temperature on the wear in a four ball machine for a one percent organophosphite additive in mineral oil[6]. The different surface temperatures shown in the curve resulted from changes in not only ambient temperature but also load and speed in the four ball test.

Another measure of the performance of boundary lubrication additives is the load-carrying capacity of the resulting lubricant in a mechanism. A load-carrying capacity, although not a well-defined property, can be thought of as that load at which severe surface distress occurs resulting in the inoperability of the tribo-contact. Figure 8 shows a plot of wear versus work transmitted in an FZG gear test rig for a base oil, a base oil plus a mild EP additive, and a base oil plus a strong EP additive. The stronger the EP additive, e.g., the greater the reactivity between the EP additive and the surface, the more boundary lubricant film that is formed and thus the greater the amount of energy that can be transmitted through the gear mesh.

Figure 9 contains data showing the relative improvement of load-carrying capacity as a result of different chemical reactivity in two types of EP additives. Not only can the difference be large between classes of lubricants, but the range of load-carrying capacity is also large within a given lubricant class for different chemical structures.

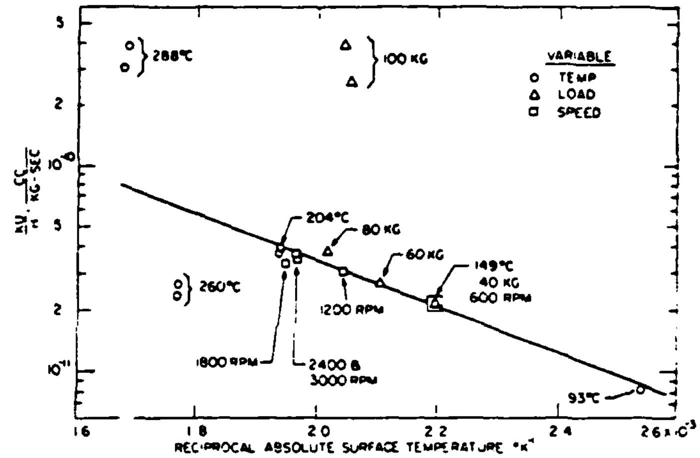


FIGURE 7 CORRELATION OF WEAR WITH SURFACE TEMPERATURES: FOUR BALL MACHINE, 1% ORGANOPHOSPHITE ADDITIVE IN MINERAL OIL (REF. 6)

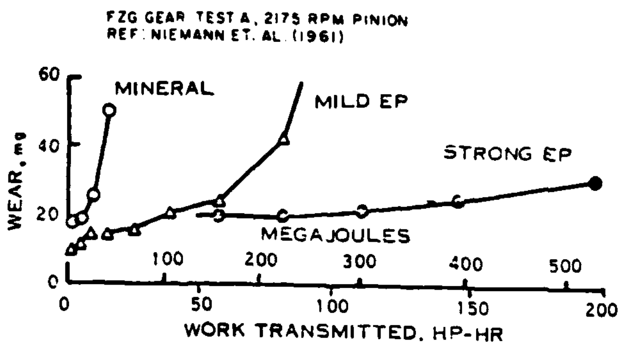


FIGURE 8 LOAD CARRYING LIMIT FOR VARIOUS LUBRICANTS

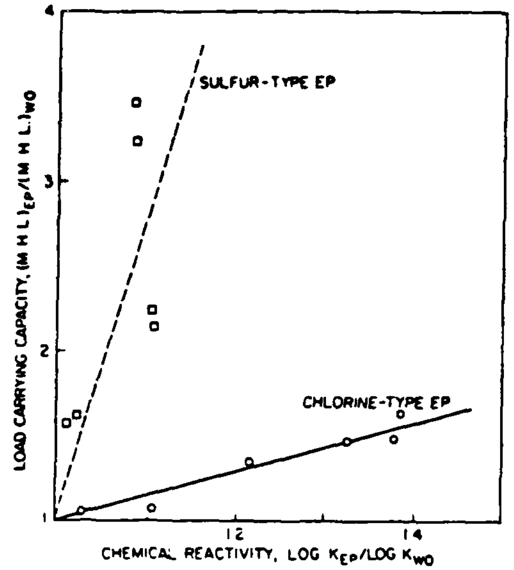


FIGURE 9 DEPENDENCE OF LOAD CARRYING CAPACITY ON CHEMICAL REACTIVITY (REF. 7)

In addition to their influence on adhesive wear coefficients and load-carrying capacity, boundary lubrication additives will also influence fatigue life of bearings. Even the presence of small quantities of water can change the fatigue life of bearings as shown in Figures 10 and 11.

The additives discussed above may be present in lubricants from as low a concentration as a few parts per million up to approximately ten percent by weight. The very low concentration in the parts per million level might be the intentionally added anti-foam agents or the unintentionally present water, both of which have significant effects on the performance of the lubricant. At the higher concentration levels are such additives as detergents, dispersants, and V.I. improvers in automotive engine oils. The more common concentration level for most additives is in the range of one percent or less by volume.

6. RHEOLOGY OF LUBRICANTS

Rheology is the study of the flow and deformation of materials. The rheology of lubricants is primarily concerned with the shear stress shear rate behavior of lubricants or more commonly the viscosity of lubricants. The viscosity or effective viscosity of lubricants is probably the most

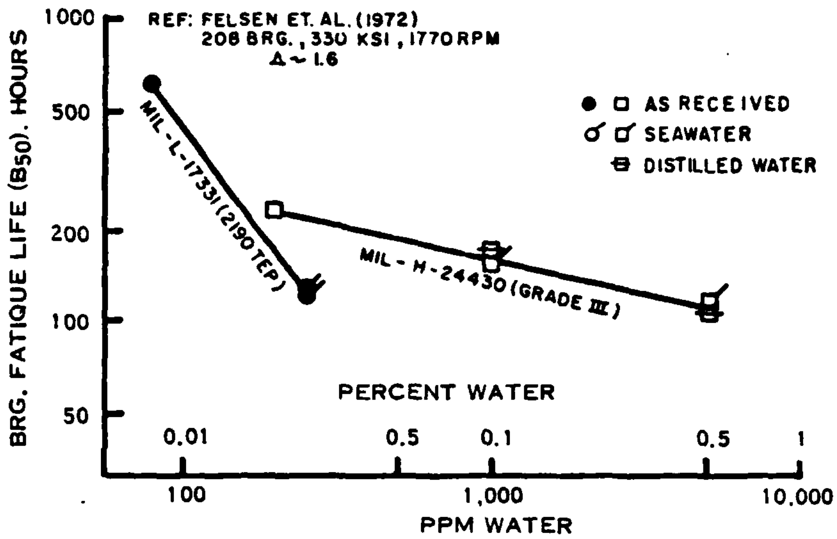


FIGURE 10 INFLUENCE OF WATER ON BEARING FATIGUE LIFE

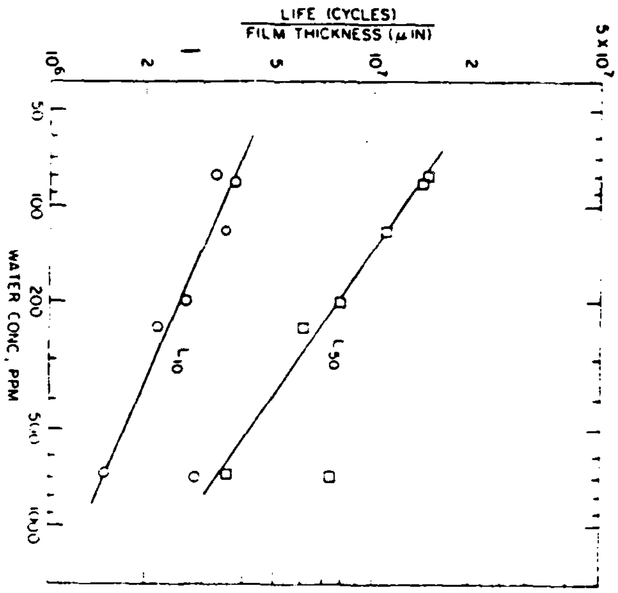


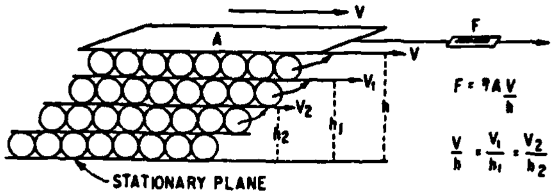
FIGURE 11 DEPENDENCE OF FATIGUE LIFE ON WATER CONTENT OF HYDRAULIC OILS AS PURCHASED (REF. 8)

commonly used property for specifying lubricants. The viscosity of the lubricant can be a function of temperature, pressure, and rate of shearing of the material. In addition to viscosity, the shear properties related to liquid-amorphous-solid transitions and the yield shear properties of lubricants in the amorphous solid regime are also important and will be discussed.

The viscosity is defined as the material property which is the ratio of shear stress to the rate of shearing strain in the material. An idealized experiment for measuring the viscosity is shown schematically in Figure 12. This figure depicts the relative motion between two parallel plates enclosing the fluid whose viscosity is being measured. The shear stress occurring is equivalent to the force required to move the plate divided by the area of the plate, and the rate of shearing strain is equivalent to the velocity gradient of the fluid between the two plates, which in steady flow for a Newtonian fluid, is simply the ratio of the relative velocity divided by the thickness of the film between the two surfaces. This viscosity is sometimes referred to as the absolute or dynamic viscosity and, given its definition, has dimensions of force times time divided by a length squared, which in the S.I. system of units is

$$\frac{N}{m^2}\,s \text{ or Pas}$$

In the more traditional English units, this might be in units of pounds per force seconds per square inch.



F = FORCE IN N

η = ABSOLUTE VISCOSITY, $\frac{NS}{m^2}$

A = AREA, m^2

V = VELOCITY, m/s

h = PLANE SEPARATION, m

FIGURE 12 SCHEMATIC FOR DEFINING VISCOSITY

Because of convenience in measuring techniques and usefulness in other fields, such as heat transfer and fluid mechanics, a second measure of viscosity is frequently reported. This second measure is referred to as the kinematic viscosity and is simply the ratio of the viscosity described above to the density of the material. The kinematic viscosity has dimensions of length squared per unit time which in the S.I. unit system is usually

$$\frac{m^2}{s} \text{ or } \frac{mm^2}{s}$$

Both the absolute viscosity and the kinematic viscosity can be reported in many different units which can be quite confusing even to the seasoned worker in the field. Probably the most common additional units used for these properties are the Poise or centipoise for absolute viscosity where the centipoise is a mPas/sec. In the case of the kinematic viscosity, a common unit is the centistoke which is the same as mm^2/s in the S.I. system.

The rheological behavior of lubricants, more particularly the viscous behavior of lubricants, is frequently divided into two broad categories referred to as Newtonian and non-Newtonian behavior. The non-Newtonian behavior is subsequently divided into a number of special cases many of which are appropriate for descriptions of lubricants.

Newtonian behavior is very common and is the simplest type of viscous behavior of materials. This behavior is shown schematically in Figure 13(a) and is simply the case where the viscosity is not a function of the rate of shearing of the material but solely a function of temperature and/or pressure. The two most common ways of presenting the viscous behavior of lubricants is to plot the viscosity versus rate of shear or the shear stress versus the rate of shear, the latter being referred to as a flow curve. For a Newtonian fluid, these two plots appear as (A) and (B) respectively in Figure 13(a).

Non-Newtonian behavior simply refers to any shear behavior which does not have a viscosity which is constant with shear rate. In Figure 13(b), the most common type of non-Newtonian behavior in lubricants is shown. This behavior is the so-called pseudo-plastic behavior. For a non-Newtonian fluid the viscosity, sometimes referred to as apparent viscosity, is a function of shear rate being applied to the material. The non-Newtonian case shown in Figure 13(b) is the more common type of non-Newtonian flow found with lubricants, that is, a decreasing viscosity with increasing shear rate. However, many other types of non-Newtonian flow also exist.

The non-Newtonian behavior might be either time independent or time dependent. Time independent non-Newtonian flow is divided into_the categories of plastic or pseudo-plastic and dilatant depending on whether the apparent viscosity decreases or increases with increasing shear rate. The pseudo-plastic behavior, that is the decreasing apparent viscosity with increasing shear rate, is the most common type of non-Newtonian flow among lubricating materials. Figure 14 shows the typical flow curve and apparent viscosity curve for pseudo-plastic non-Newtonian lubricants. In this case it is common to speak of an apparent viscosity which is the ratio of the shear stress to the shear rate at a particular shear rate. As a general rule low molecular weight materials (< 1000 amu) tend to be Newtonian materials, while very high molecular weight materials (in excess of 10,000 or 20,000 amu) tend to be non-Newtonian materials. This means that unblended mineral oils could be expected to be Newtonian while those containing high polymers, particularly viscosity index improvers, would be expected to be non-Newtonian.

Non-Newtonian behavior introduces one of the major difficulties in the understanding of the relationship of the viscosity to lubricant performance. As shown in Figure 15, taken from the Tribology Handbook, the viscosity of a number of the typical lubricants as a function of shear rate might be Newtonian or non-Newtonian in the range of shear rates from those typically occurring in viscosity measurement apparatus

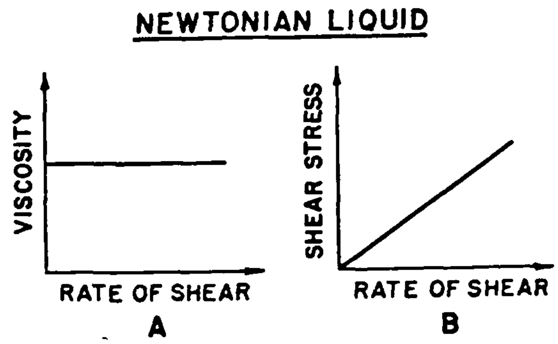


FIGURE 13(A) CHARACTERISTICS OF NEWTONIAN LIQUIDS. CURVE A: VISCOSITY IS INDEPENDENT OF RATE OF SHEAR. CURVE B: SHEAR STRESS IS DIRECTLY PROPORTIONAL TO RATE OF SHEAR

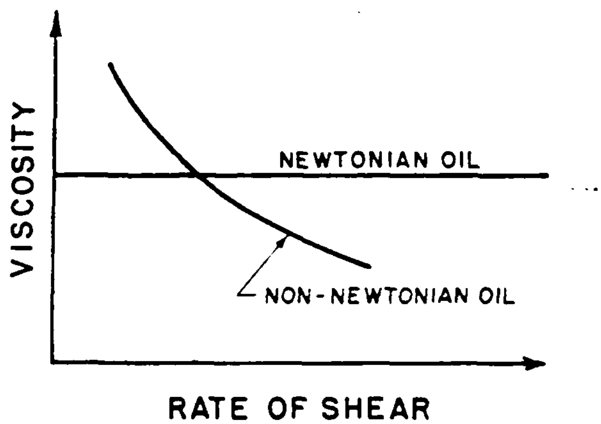


FIGURE 13(B) VISCOSITY VERSUS RATE OF SHEAR FOR NEWTONIAN AND NON-NEWTONIAN OILS

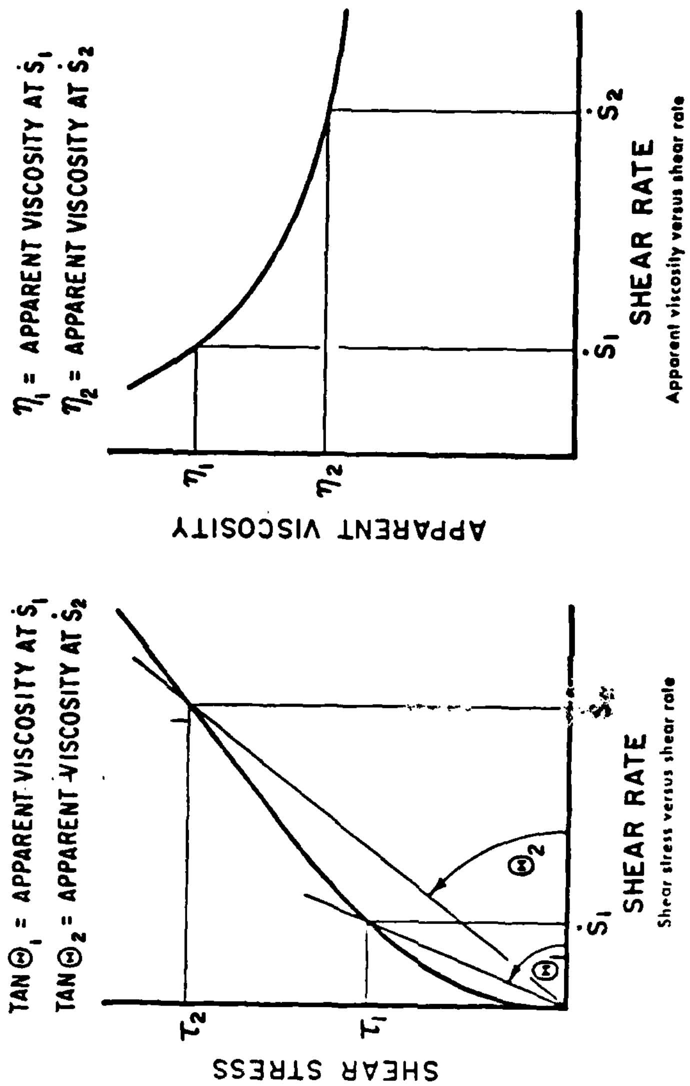


FIGURE 14 TYPICAL VISCOUS CHARACTERISTICS OF A NON-NEWTONIAN FLUID

to the very high values occurring in bearing applications[1]. Most viscosity measurements are made at very low shear rates, one reciprocal second or less. However, the shear rates occurring in most bearings are quite high, usually 10^5 and sometimes in excess of 10^6 reciprocal seconds. Depending on the polymeric material involved, this large difference in shear rate can cause the measured viscosity to be many times greater than the effective viscosity in a bearing.

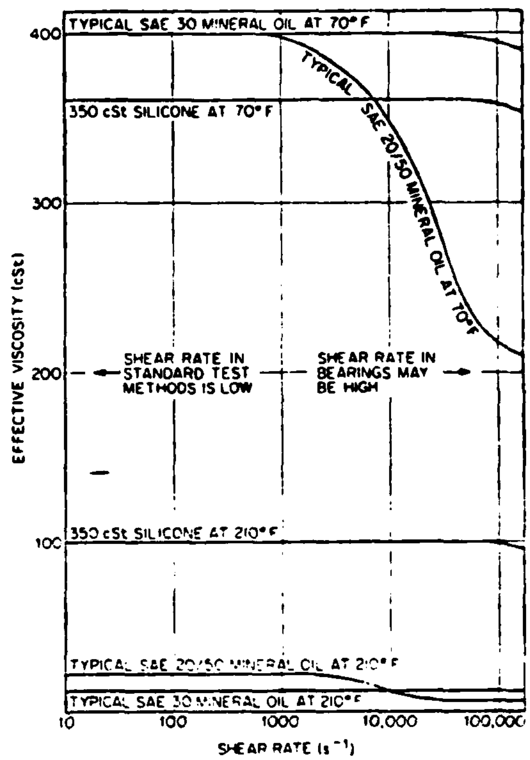


FIGURE 15 VARIATION OF VISCOSITY WITH SHEAR RATE (REF. 1)

In addition to time steady non-Newtonian behavior as a function of shear rate described above, it is also possible to have time dependent viscosity behavior. If the viscosity is a function of time at constant shear rate, the behavior is divided into two categories referred to as rheopectic and thixotropic depending on whether the viscosity is increasing or decreasing with time respectively. This time dependent behavior is shown schematically in Figure 16. Time dependent behavior is less common than time independent behavior among lubricants, but it does occur. For example, in the case of automotive engine lubricants, both thixotropic and rheopectic behavior can be observed over the drain interval for the engine. The thixotropic behavior is related to polymer degradation. In this situation the high molecular weight polymers will slowly be mechanically degraded into smaller molecules resulting in a lower viscosity with time. The thixotropic behavior in automotive engine lubricants can be observed as a result of combustion product blow-by and oxidation causing the oil to thicken with time over the drain interval. These two effects tend to counteract each other with the result that the viscosity will increase or decrease in a particular application depending on which effect predominates.

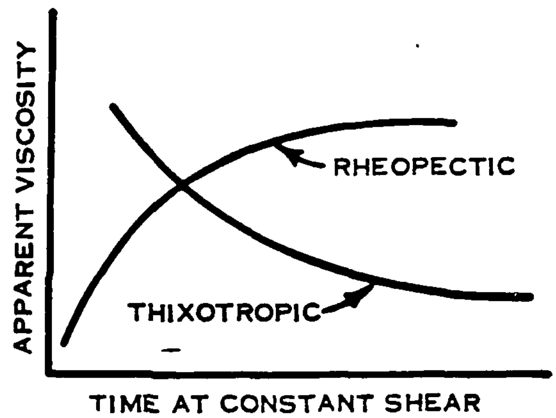


FIGURE 16 DIFFERENT TYPES OF NON-NEWTONIAN BEHAVIOR

There are many different types of apparatus for measuring viscosity of lubricants. Although the viscosity is defined as the ratio of shear stress to shear rate, it can also be viewed as a lubricant property which relates the way that mechanical energy is dissipated into thermal energy in the material. Therefore, measurements can be made by determining the time required for a given mass of material to be lowered from one height to another while passing through some flow constriction. Measurements can also be made by rotational devices in which a continuously rotating member, with a torque applied, is transmitting energy to the fluid film to be dissipated.

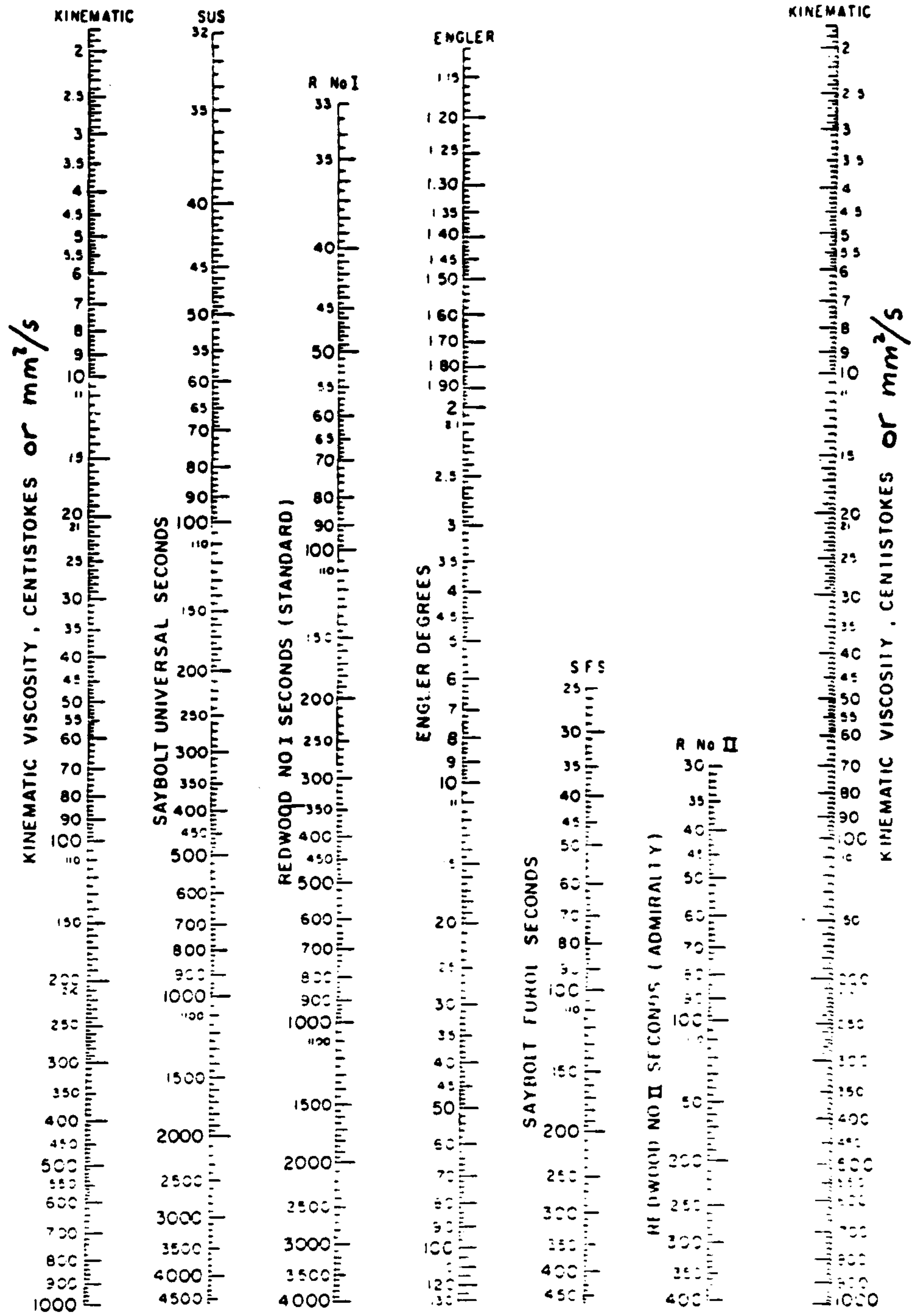
By far the most common method of measuring lubricant viscosity is by measuring the flow of a fixed quantity of material through a capillary constriction. If the volume of fluid, the distance to travel, and the geometry of the orifice are known, the measurement is simply a measurement of time for the flow of a given volume to occur. The Saybolt viscometer is a common device used for this type of viscosity measurement. In the Saybolt viscometer, a fixed quantity of oil (60 ml) is allowed to pass from a constant temperature bath through an orifice into a measuring flask. The measurment consists simply of the time required for the 60 ml to flow through the orifice. In this case, the density influences the potential to drive the fluid through the orifice and therefore the time is a function of the ratio of the absolute viscosity to the density of the material. The flow time is actually a measure of the kinematic viscosity of the oil. It is traditional to report viscosities measured in a Saybolt viscometer in terms of Saybolt's seconds, e.g., the time required for the flow to occur in that particular device. Unfortunately, the units of seconds bear no direct relation to kinematic viscosity or absolute viscosity in the terms most logically associated with the definition of the property as discussed above.

Other devices which are similar in principle and commonly used are the Redwood viscometer and the Engler viscometer. In each case the principle is the same as the Saybolt viscometer, but the geometry and volume of flow are different and therfore the generated numbers will be different. In the case of the Redwood viscometer, the viscosity is reported in Redwood seconds; while in the case of the Engler viscometer the viscosity, is reported in Engler degrees. In both the Saybolt viscometer and the Redwood viscometer different orifices are available for measuring viscosities in different ranges and therefore the times are usually qualified by which orifice has been employed.

Glass capillary viscometers are similar in function to the Saybolt viscometer described above. The glass capillary viscometers are immersed in constant temperature baths and permit the flow of a fixed quantity of fluid through a prescribed capillary in the glass. Again the measurement is the time required for the flow to occur. However, in the case of the glass capillaries, the results are usually reported in terms of traditional units of viscosity. The Saybolt viscometer and the glass capillary viscometers are both used in ASTM Standards for viscosity measurement .

Figure 17 is a nomograph which permits the conversation of kinematic viscosities from one set of units to another.

The viscosity of lubricants is a strong function of temperature, and whenever viscosity is quoted it is important that the temperature at which the measurement was made is also



LINE UP STRAIGHT EDGE SO CENTISTOKE VALUE ON BOTH KINEMATIC SCALES IS THE SAME. VISCOSITIES AT THE SAME TEMPERATURE ON ALL SCALES ARE THEN EQUIVALENT.

TO EXTEND RANGE OF ONLY THE KINEMATIC, SAYBOLT UNIVERSAL, REDWOOD NO I AND ENGLER SCALES, MULTIPLY BY 10 THE VISCOSITIES ON THESE SCALES BETWEEN 100 AND 1000 CENTISTOKES ON THE KINEMATIC SCALE AND THE CORRESPONDING VISCOSITIES ON THE OTHER 3 SCALES. FOR FURTHER EXTENSION, MULTIPLY THESE SCALES AS ABOVE BY 100 OR A HIGHER POWER OF 10.

(EXAMPLE: 1500 CENTISTOKES = 150 × 10 cSt ≅ 695 × 10 SUS = 6950 SUS)

FIGURE 17 VISCOSITY CONVERSION NOMOGRAPH
(COURTESY TEXACO LUBRICATION MAGAZINE)

quoted. The most common method of presenting viscosity as a function of temperature is done with the ASTM Viscosity Temperature Chart as shown in Figure 18. The ASTM Viscosity Temperature Chart is based on the Walther Equation for describing the viscosity change with temperature. The scales on this chart can be quite misleading if not viewed with care. The kinematic viscosity range is a doubled log scale while the temperature range is a logarithmic scale. As a result, subsequent plotting of the two variables results in the viscosity temperature behavior of most materials being a straight line on the graph. This permits the convenience of measuring the viscosity at only two temperatures, plotting them on the curve and making possible the extrapolation and interpolation over other temperatures.

The double logarithmic scale on viscosity results in a very distorted scale for the kinematic viscosity. The major increments at the lower portion of the kinematic viscosity scale are only from two to three centistokes while the major divisions at the top of the scale are from ten million to twenty million centistokes. Also shown in Figure 18 are relative temperature viscosity behavior of some typical lubricating materials.

Although very important, the temperature viscosity behavior of lubricants is difficult to conveniently quantify. The most common method of specifying the temperature viscosity characteristics is to use the ASTM Dean and Davis Viscosity Index Scale[9] This scale is a relative scale and shown schematically in Figure 19. It is based on two standard categories of mineral oil, one of which exhibits a relatively low change of viscosity with temperature which is arbitrarily set at 100 on the VI scale. Each of these groups of lubricants, the zero and the 100 VI standards, have a range of viscosities at 210°F. Viscosity index of an unknown fluid is determined by comparing its viscosity at 100°F with the 100°F viscosities of the standard fluids, which have the same viscosity at 210°F, as shown in Figure 19 and by using the Equation

$$VI_u = \frac{U - H}{L - H}$$ (variables as presented in Figure 19).

Tables for this purpose are in the Appendix to ASTM Method D-2270-77.

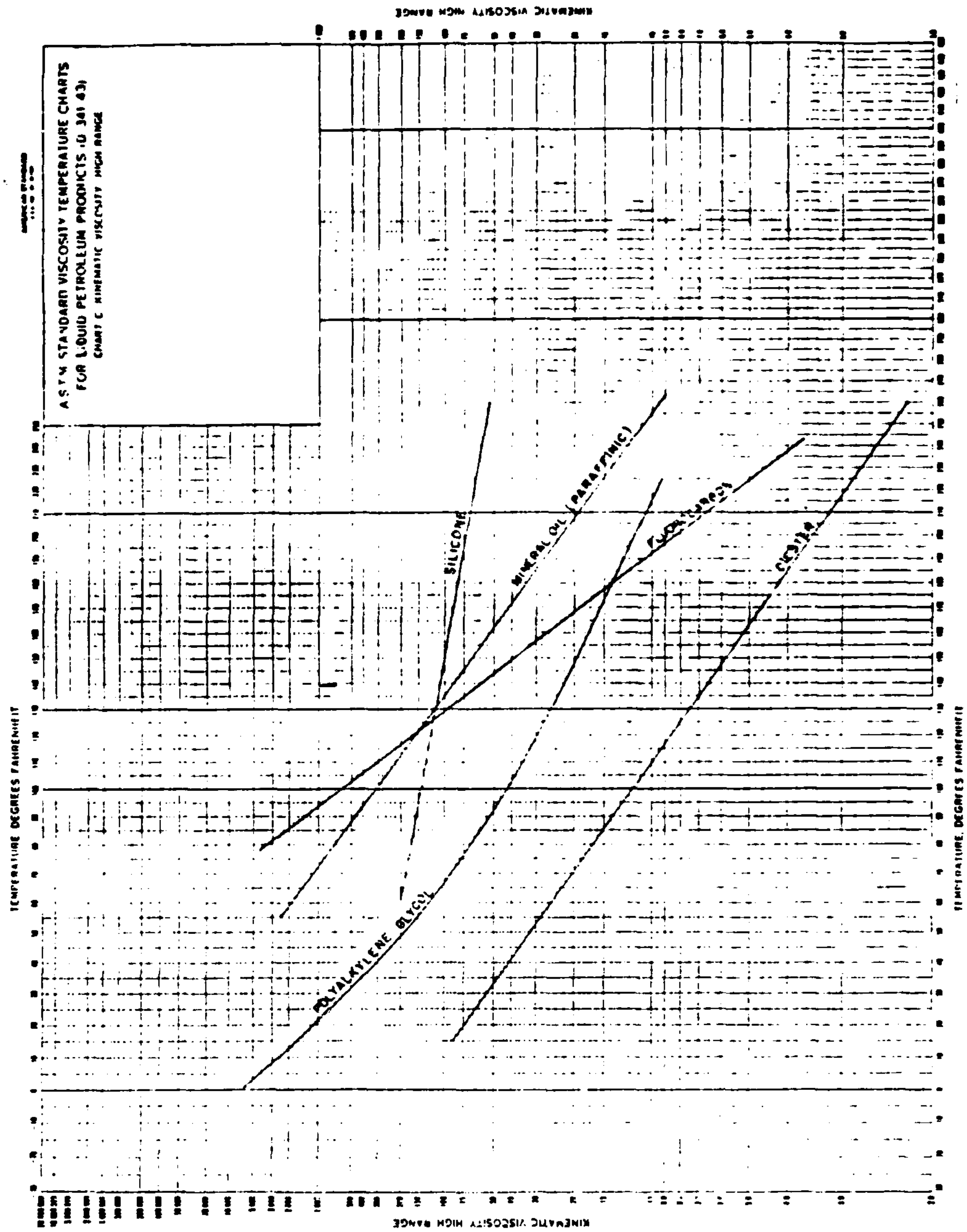


FIGURE 18 AN ASTM VISCOSITY-TEMPERATURE CHART WITH CURVES FOR SEVERAL FLUIDS

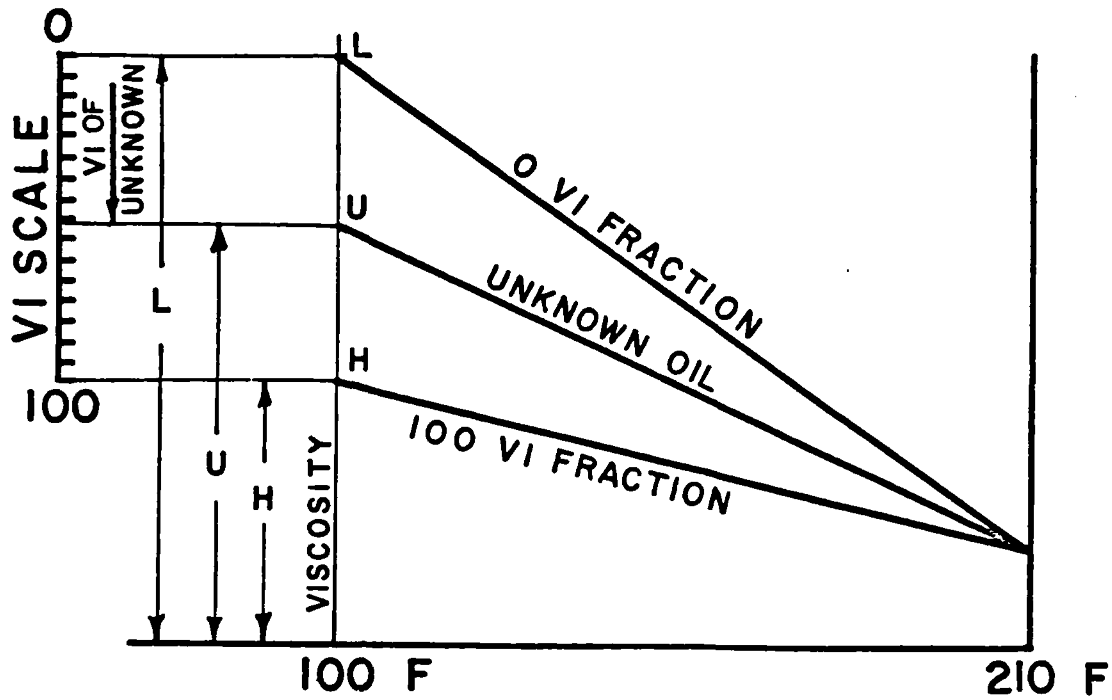


FIGURE 19 A SCHEMATIC ILLUSTRATION OF THE VISCOSITY INDEX SYSTEM

The viscosity index scale was established over forty years ago and has worked quite satisfactorily for some time. However, when considering today's lubricants, it is possible to have viscosity indices in excess of 100. The behavior of the viscosity index scale becomes somewhat different in the high VI range as compared to the lower 0 to 100 range for which it was originally established. Figure 20(a) represents a plot of kinematic viscosity at 100°F versus kinematic viscosity at 210°F with lines of constant viscosity index. Figure 20(b) shows the viscosity index of a hypothetical ideal material which has a constant viscosity versus the 210°F viscosity of the material. As can be seen from this figure, rather large changes in viscosity index occur for low viscosity materials. The viscosity index of a material can be obtained from extensive tables existing in the ASTM Standards .

Several other measures of viscosity temperature change can be found in the literature. These include simply reporting the ratio of the viscosity at 100°C to the viscosity at 40°C, the percent change in viscosity from 40°C to 100°C, or the logarithmic derivative of the viscosity at any particular temperature.

440

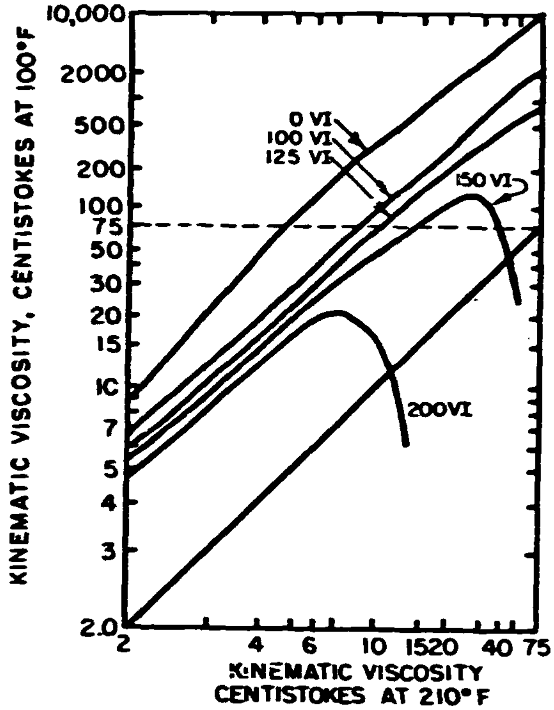


FIGURE 20(A) PLOT OF LINES OF CONSTANT ASTM VISCOSITY INDEX (COURTESY OF TEXACO LUBRICATION MAGAZINE)

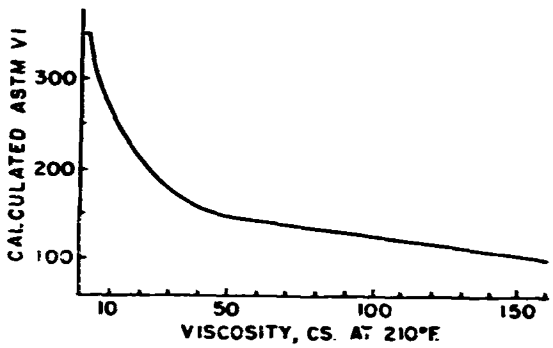


FIGURE 20(B) CALCULATED VISCOSITY INDEX VALUES FOR IDEAL OILS (WHOSE VISCOSITIES DO NOT VARY WITH TEMPERATURE) VERSUS THEIR VISCOSITIES AT 210°F (COURTESY TEXACO LUBRICATION MAGAZINE)

The viscosity of lubricants also changes with pressure. As a first approximation, the viscosity increases exponentially with pressure such that the viscosity pressure isotherm plotted on a semilog plot (log viscosity versus pressure) will be nearly a straight line. The viscosity change is among the largest of all physical properties to change with pressure. However, in lubrication applications this change does not normally become important enough to consider unless the pressures are in excess of 10,000 psi or approximately 100 MPas. Therefore, the variation of viscosity with pressure is generally not considered important in thick film hydrodynamic lubrication, but does become important in the relatively thin film elastohydrodynamic lubrication.

Some typical results of the pressure viscosity dependence of lubricants are shown in Figures 21(a) and (b) taken from the ASME Pressure Viscosity Report[10]. In the case of mineral oil shown in Figure 21(a), for the same base viscosity, the slope of the pressure viscosity curve tends to increase with increasing naphthenic content. Also for the same base viscosity, the temperature viscosity dependence also tends to increase with increasing naphthenic content of the oil. Figure 21(b) shows the relative behavior of a number of synthetic lubricants.

Figure 22 shows the pressure viscosity isotherms for a synthetic lubricant (Polyphenol Ether) and shows the rather large change of viscosity pressure dependence with temperature. The slope of the pressure viscosity curve is the pressure viscosity coefficient which is seen to decrease substantially with increasing temperature. As discussed in the previous chapter on lubrication, the pressure viscosity coefficient is the luricant property which, along with the viscosity, determines the film thickness in elastohydrodynamic lubrication. The pressure viscosity coefficient is frequently presented in two different forms. These forms are either the slope of the logarithmic plot of viscosity versus pressure for an isotherm at atmospheric pressure (αOT) or an integrated relation between the viscosity dependence on pressure which is referred to as α^*.

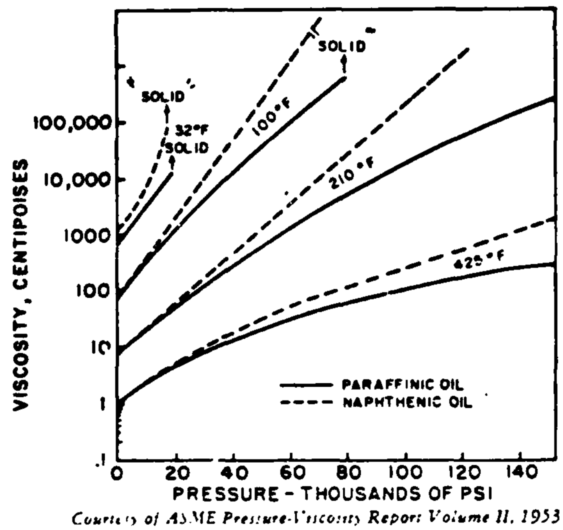


FIGURE 21(A) PRESSURE-VISCOSITY CURVES FOR PARAFFINIC AND NAPHTHENIC OILS AT FOUR TEMPERATURES

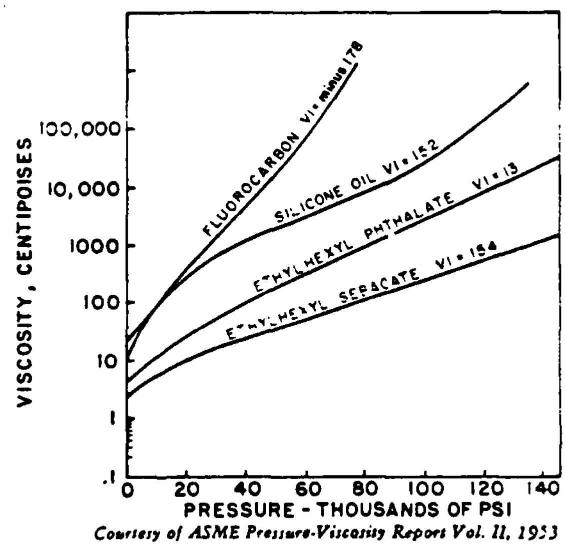


FIGURE 21(B) THE EFFECT OF PRESSURE ON THE VISCOSITY AT 210°F OF SEVERAL SYNTHETIC OILS

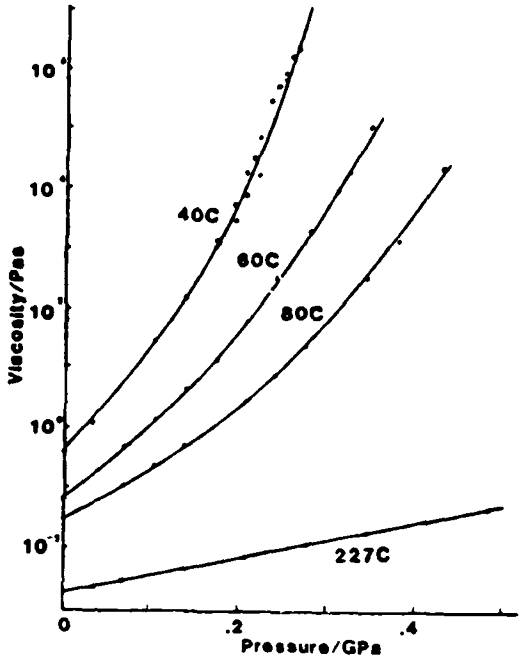


FIGURE 22 PRESSURE-VISCOSITY ISOTHERMS FOR 5PAE (POLYPHENOL ETHER), (REF. 11)

Table 7 presents the pressure viscosity coefficients, α_{OT} and α^* as a function of temperature for a number of typical lubricants. As decribed in the previous chapter, the lubricant property determining the film thickness and concentrated contacts or elastohydrodynamic lubrication is frequently presented as a product of the viscosity and the pressure viscosity, α^*, and referred to as a lubricant parameter. This lubricant parameter is presented as a function of temperature for a number of typical synthetic materials in Figure 23.

Fluid	T/C	α_{OT}	$\overset{*}{\alpha}_T$
R620-15	26	27.4	27.4
	40	21.9	21.9
	99	15.4	14.8
	149	10.7	11.0
	227	12.0	8.85
R620-16	26	35.6	35.8
	99	19.8	19.8
	227	10.8	10.6
R620-15 + 4523	26	25.5	25.7
	99	17.1	15.0
	227	16.8	10.3
R620-15 + 4521	26	24.2	24.9
	99	15.0	15.3
	227	13.8	9.8
5P4E	40	40.6	41.2
	60	27.6	29.3
	80	20.0	20.1
	227	6.8	6.8
LVI 260	30	31.9	34.8
VITREA 79	30	22.6	22.6
	40	22.6	22.6
Turbo 33	30	19.6	19.6
	40	16.7	16.7

$$\alpha_{OT} = \frac{d\ln\mu}{dp}\bigg|_{T,p=0} \qquad \overset{*}{\alpha}_T = \left[\int_0^{p=\infty} \frac{\mu(T,p=0)}{\mu(T,p)}\,dp\right]^{-1}\bigg|_T$$

Experimental fluids

Name	Type
R620-15	naphthenic mineral oil
R620-16	naphthenic mineral oil
R620-15 plus 4 wt. percent PL4520	Polymer blend
R620-15 plus 4 wt. percent PL4521	Polymer blend
R620-15 plus 4 wt. percent PL4522	Polymer blend
LVI 260	mineral oil
Vitrea 79	mineral oil
Turbo 33	mineral oil
5P4E	5-ring polyphenylether
Santotrac 50	Traction fluid
Krytox	Perfluorinated polyether
XRM 177F	Synthetic paraffinic hydrocarbon

TABLE 7 PRESSURE-VISCOSITY COEFFICIENTS (REF. 11)

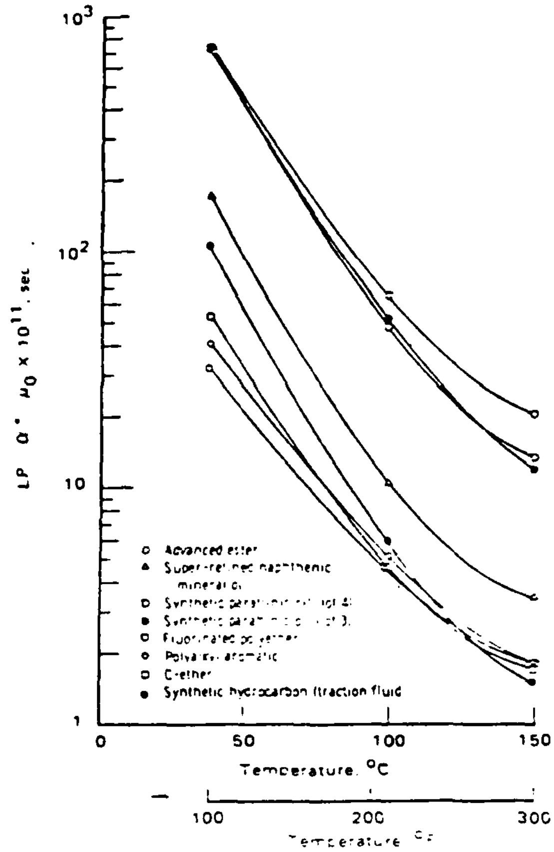


FIGURE 23 EHD FILM FORMING CAPABILITY AS FUNCTION OF TEMPERATURE FOR UNFORMULATED FLUIDS (REF. 12)

7. AMORPHOUS SOLID BEHAVIOR OF LUBRICANTS

In concentrated contacts with elastohydrodynamic lubrication, the pressures can vary from 600 MPa to 2 or 3 GPa. At these pressures, the viscosity of the lubricant becomes extremely high and the lubricant may no longer be able to respond by deforming viscously to the stress applied to it by the sliding surfaces. Therefore, the lubricant behavior may become similar to that of a relatively low shear strength amorphous solid. The liquid-amorphous transition in the lubricant is a function of temperature, pressure, and rate of stress application. This behavior is related to the glass transition of the material which is normally measured at atmospheric pressure. The glass transition is related to a very low rate of stress application and, for most lubricants at atmospheric pressure, occurs at temperatures below -20°C.

However, as the pressure is increased, the glass transition temperature also increases and at the very high pressures occurring in concentrated contacts, the glass transition temperature can be equal to or above the ambient temperature depending on the lubricant employed. Figure 24 is a temperature-pressure plot schematically showing a typical elastohydrodynamic contact and the glass transition behavior of three lubricants; 5P4E, a polyphenol ether which has the highest glass transition temperature of the lubricants shown; N1, a naphthenic mineral oil; and XRM 177, a low molecular weight synthetic material with very good low temperature viscosity. The curve for N1 is typical of many mineral oils.

Also shown schematically on Figure 24 are the approximate ranges of temperature and pressure in parts of an elastohydrodynamic contact. The inlet zone is at low pressure and the Hertzian zone is the higher pressure range. The temperatures in a rolling contact with little energy dissipation will be relatively low and near ambient temperature, while increased sliding in the contact will cause higher temperatures to be present. For a given lubricant whose curve is shown on Figure 24, the behavior in the concentrated contact would be expected to be liquid-like if the temperatures and pressures occurring are above and to the left of the curve shown. If the temperatures and pressures occurring in the contact for that lubricant are below and to the right of the curve, the material behavior would be expected to be that of an amorphous solid. If the material behavior is liquid-like, the lubricant viscosity should be the determining rheological property. If the behavior is amorphous solid in nature, properties such as elastic shear modulus and maximum shear stress should be the controlling rheological properties. To a first approximation, one would expect the inlet zones of virtually all elastohydrodynamic contacts to be in the liquid-like region. Therefore, because film thicknesses are determined by behavior in the inlet zone, film thicknesses in elastohydrodynamic contacts would be determined by the viscous behavior of the lubricant. This logic is consistent with what has been found experimentally. However, most of the Hertzian zones in concentrated contacts will be in the amorphous solid region for many lubricants. The friction in the contact is primarily determined by behavior in the Hertzian zone. Therefore, the friction or traction behavior will be determined by properties in the amorphous solid regime for the lubricant. Similar transition curves for a number of other lubricants can be found in the literature[13].

The transition curves shown in Figure 24 are for very low rates of stress application. If the rate of stress

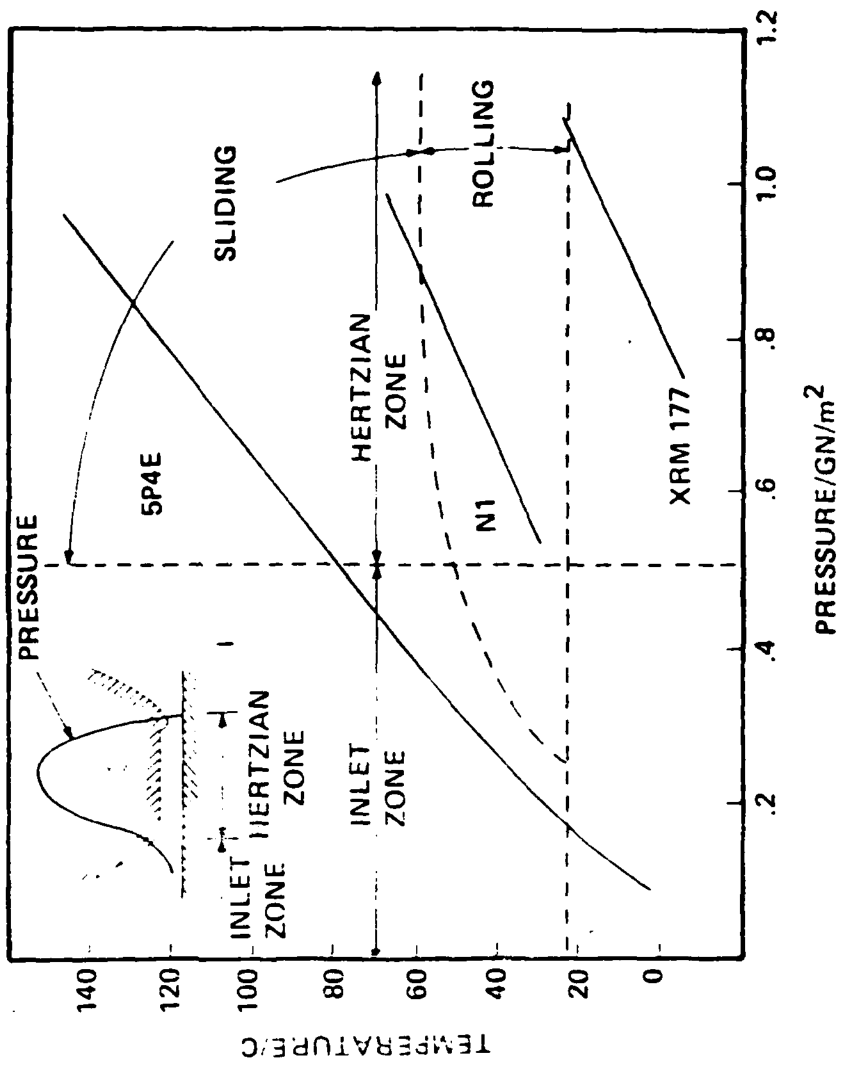


FIGURE 24 HEURISTIC ESTIMATES OF THE RELATIONSHIP BETWEEN CONDITIONS IN AN EHD CONTACT AND GLASS-LIQUID TRANSITION DIAGRAM OF SOME LUBRICANTS (LUBRICANT SUPPLY TEMPERATURE ABOUT 20°C) (REF. 13)

application is increased, these transition curves will shift up and to the left, e.g., to lower pressures for the same temperature or higher temperatures for the same pressure. This results in an expansion of the amorphous solid regime and a contraction of the liquid-like regime. Figure 25 shows the transition curves which result at a constant pressure, in this case atmospheric pressure, as the frequency of stress application, is changed. As the frequency of stress application increases, the temperature at which the transition occurs increases. The relative slopes of different materials is clear. For some materials, i.e., the polyphenol ether, the change of transition temperature with frequency is small as compared to the change that occurs with the mineral oil, N1. This change of transition with frequency can also be shown on the temperature pressure plot as shown in Figure 26.

It is reasonable to argue that the constant frequency relates to a constant relaxation time, which in turn is approximately equal to a constant viscosity for the material[14]. Therefore, lines of constant frequency of stress application and constant viscosity will plot as nearly parallel lines on the temperature-pressure transition curve. They should be parallel to the limiting low rate stress application performed by dilatometry. Such curves are shown in Figure 26 for polyphenol ether.

8. SHEAR STRESS STRAIN BEHAVIOR IN THE AMORPHOUS SOLID REGION

In the amorphous solid region the material will behave like an elastic-plastic solid. This elastic-plastic shear stress shear strain behavior will be the determining material property for the friction or traction in the concentrated contact. Figure 27 shows the stress strain behavior measured for a polyphenol ether at 40 kpsi pressure at the temperatures indicated. As the temperature is decreased, starting at 38°C and going down to -27°C, the elastic-plastic behavior of the material begins to become apparent as the temperature passes through the amorphous solid transition temperature as shown in Figure 26. The rate of stress application for the data in Figure 27 is very slow. As the temperature decreases the initial slope of the curve, which is the elastic shear modulus, slowly increases to a nearly constant value as one goes into the amorphous solid region. The maximum shear stress the material can withstand also increases as the temperature is lowered and the material is taken further into the amorphous solid region. The maximum values of the elastic shear modulus and the shear stress are referred to as the limiting values in each case. Similar behavior would be observed if the temperature and the strain rate were held

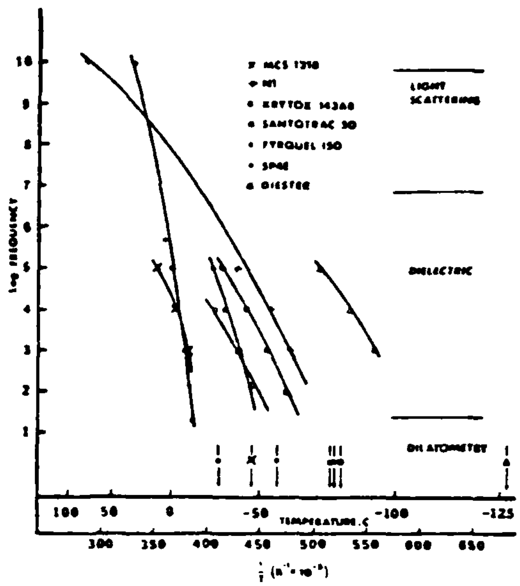


FIGURE 25 DIELECTRIC ($10 \leq Hz \leq 10^6$), LOW RATE DILATOMETRY (ARROWS) AND LIGHT SCATTERING ($Hz=10^{10}$) TRANSITION DATA AT ATMOSPHERIC PRESSURE, (REF. 14)

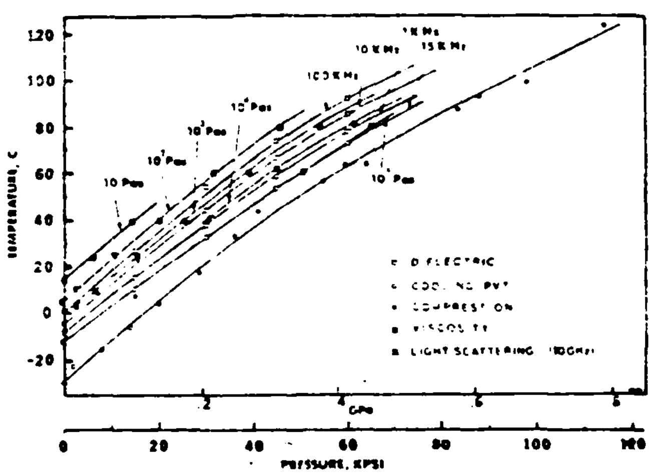


FIGURE 26 TRANSITION DIAGRAM FOR POLYPHENYL ETHER (5P4) AND SEVERAL METHODS (REF. 14)

constant while the pressure was increased. Similar behavior would also be observed if the temperature and pressure were held constant while the rate of stress application was increased.

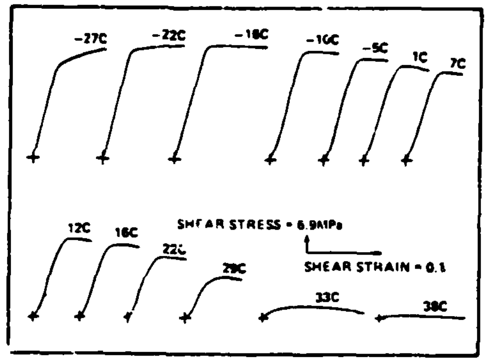


FIGURE 27 RECORDER PLOT OF SHEAR STRESS VERSUS SHEAR STRAIN FOR POLYPHENYL ETHER (5P4E) AT 0.275 GPa (40 kpsi) AND INDICATED TEMPERATURES (REF. 13)

Because in the concentrated contact the rate of stress application is very rapid, the limiting values of the elastic shear modulus and the maximum shear stress will be the controlling properties determining the traction in elastohydrodynamic contacts. Accompanying the maximum shear stress is a maximum recoverable elastic strain which can be seen from Figure 27 to be in the neighborhood of 0.02 to 0.03. If the strain in the concentrated contact at very low slide-roll ratios is less than this value of limiting recoverable strain, the traction will be determined by the elastic shear modulus of the material. However, at higher slide-roll ratios, the traction will be determined by the maximum shear stress the material can withstand.

Figure 28(a) and (b) contain the traditional format of viscosity versus shear rate or shear stress versus shear rate respectively for 5P4E lubricant. The entire range of behavior from liquid-like behavior on one side of the amorphous transition curve to limiting shear stress behavior on the other side of the transition curve is presented in a single flow curve. The data here presented represents measurements from three quite different types of measuring apparatus[15]. In Figure 28(b) the Newtonian behavior is represented by a line of slope 1 and the limiting shear stress is represented by a line of slope zero. As the shear rate is increased, the

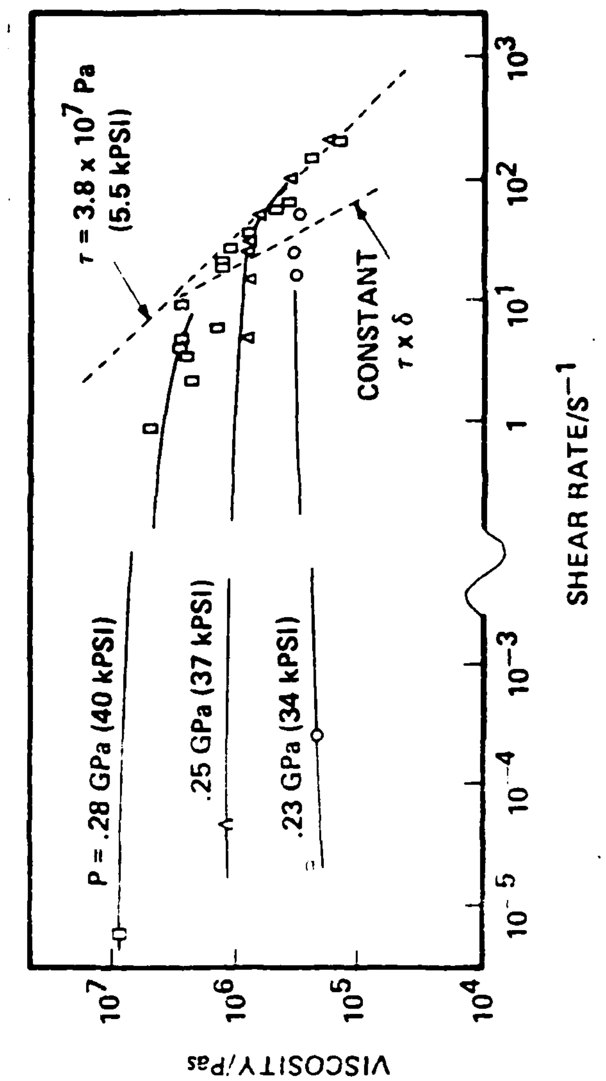


FIGURE 28(A) VISCOSITY OF 5P4E VERSUS SHEAR RATE SHOWING THE LIMITING SHEAR STRESS AT 40°C (REF. 15)

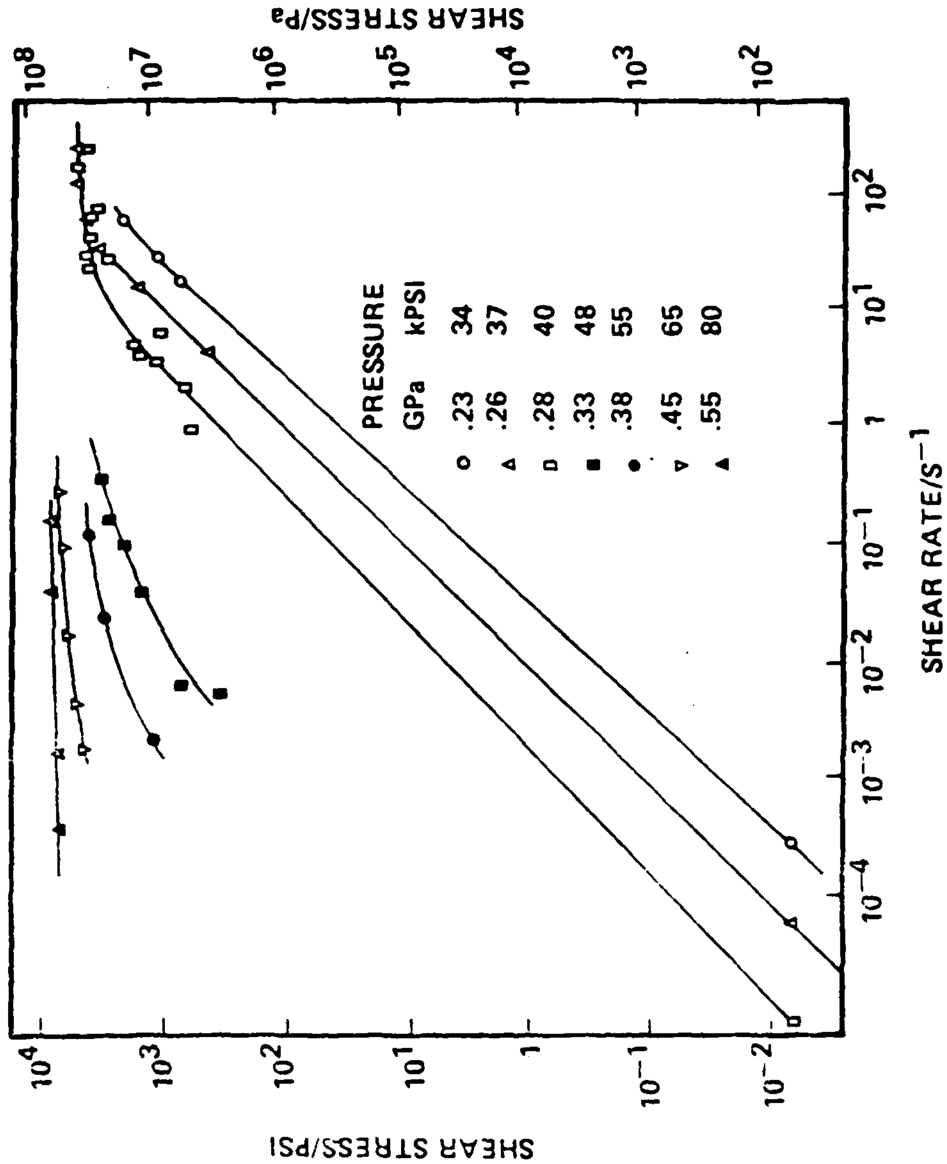


FIGURE 28(B) SHEAR STRESS-SHEAR STRAIN RATE FOR 5P4E AT 40°C AND INDICATED PRESSURE (THREE DIFFERENT METHODS) (REF.15) behavior is Newtonian up to some nearly maximum value where the shear stress reaches a maximum.

The limiting shear stress is the property primarily responsible for traction in highly loaded concentrated contacts. Limiting shear stress for a number of materials as a function of temperature is shown in Figure 29(a) and (b). The limiting shear stress decreases with increasing temperature and increases with increasing pressure. Also note on Figure 29(a), the influence of the presence of a polymer in a mineral oil by comparing N1, which is a naphthenic mineral oil, and N1 + 4% PAMA polymer. This influence has been found to be representative for PAMA polymers of several molecular weights. In all cases the limiting shear stress with the polymer present is approximately 15% less than that of the base oil alone[11]. This is somewhat unexpected in light of the fact that the presence of the polymer can increase the viscosity by a factor of five to eight times.

The limiting shear stress of some solid lubricating materials is very similar to that of liquid lubricants. In Figure 30 there are two solid plastic materials, PVC and Teflon, which are commonly used as solid lubricating materials. This similar behavior of limiting shear stress for both solid lubricating materials and typical liquid lubricants is consistent with the fact that in concentrated contacts with these materials, the coefficients of friction are similar. In the limiting case, the coefficient of friction should simply be the ratio of the limiting shear stress to the pressure on the contact.

Table 8 shows a comparison of the average shear stress based on traction measurements at low slide-roll ratios for three materials and the limiting shear stress measurement at the same pressure and temperature[11]. The agreement between the property measurement and the tribological traction measurement is extremely good.

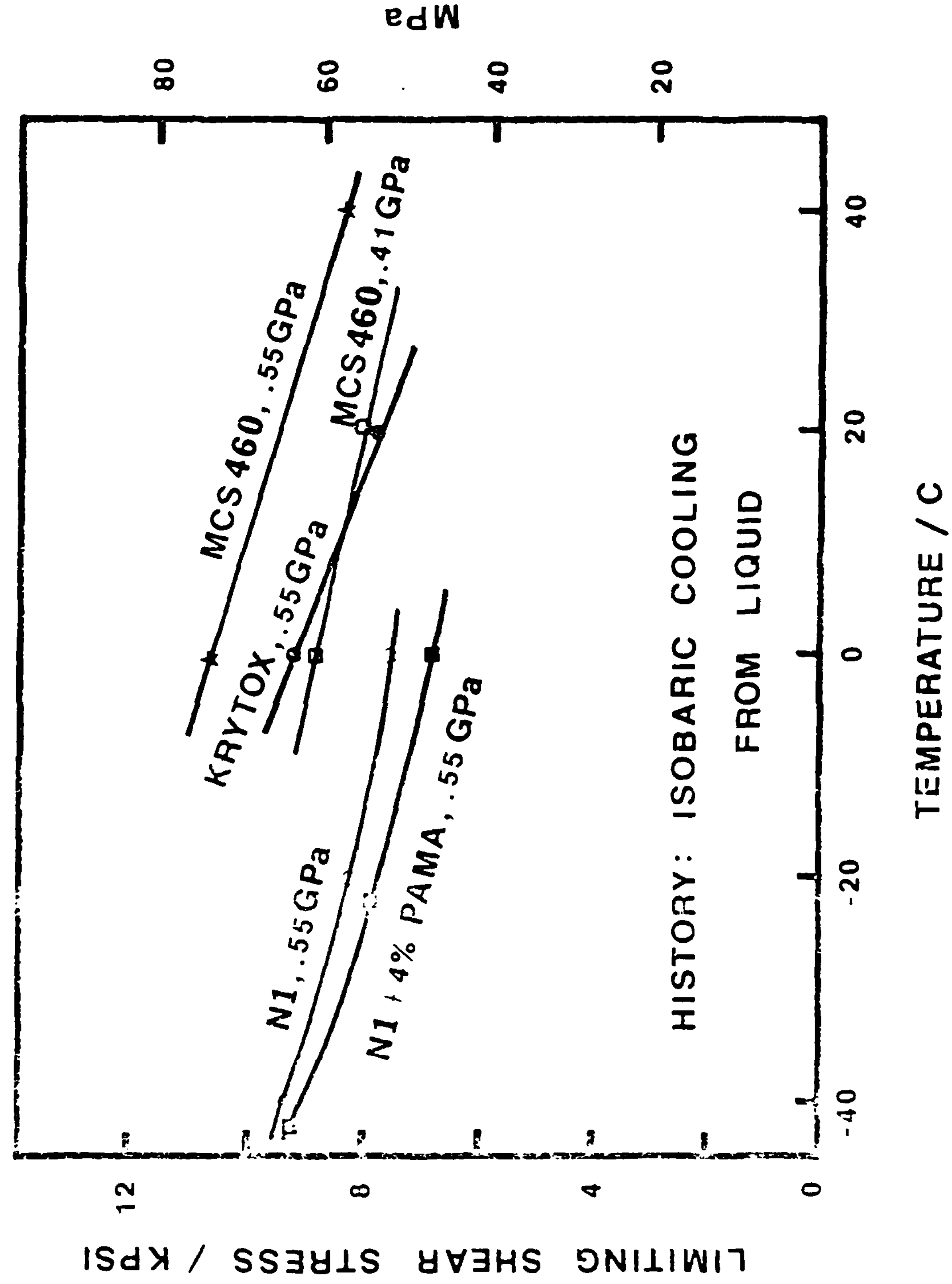


FIGURE 29(A) LIMITING SHEAR STRESS VERSUS TEMPERATURE FOR MCS 460, KRYTOX, N1, AND N1 + 4 PERCENT PAMA (COLLING HISTORY)

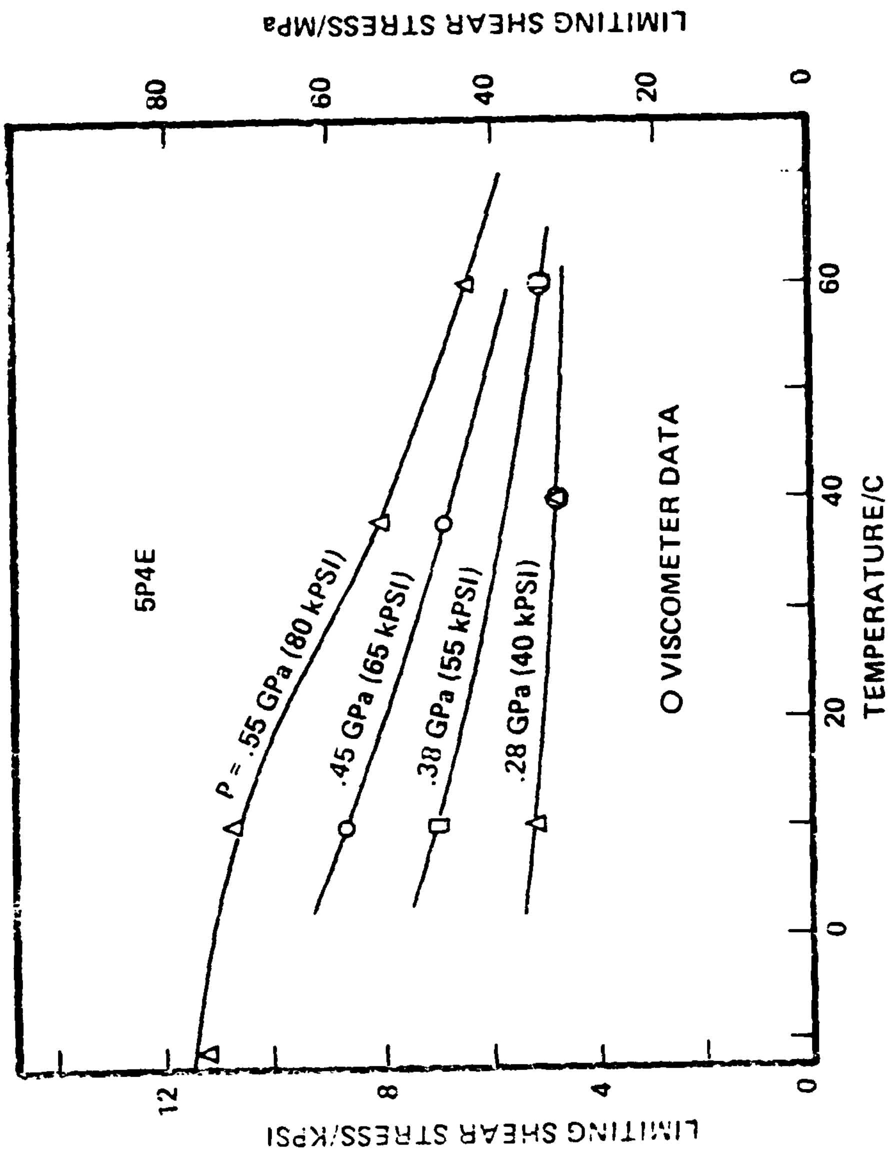


FIGURE 29(B) LIMITING SHEAR STRESS FOR 5P4E AS A FUNCTION OF TEMPERATURE AT INDICATED PRESSURES. CIRCLE AROUND DATA POINT INDICATES IT WAS OBTAINED WITH THE HIGH STRESS VISCOMETER (REF. 15)

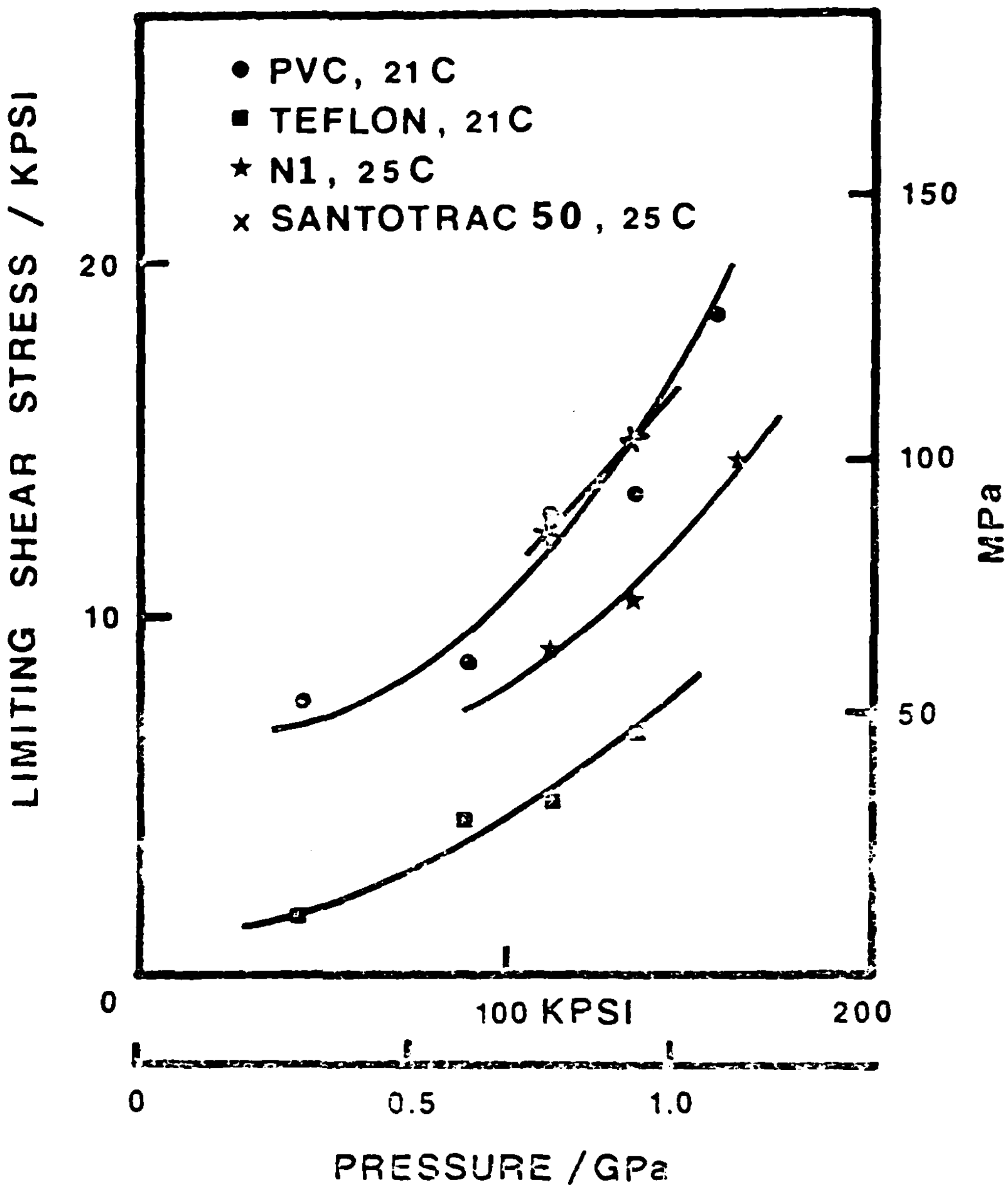


FIGURE 30 SHEAR STRENGTH OR LIMITING STRESS VERSUS PRESSURE FOR N1 (NAPHTHENIC BASE OIL), SANTOTRAC 50 (CYCLOALIPHATIC HYDROCARBONO, POLYVINYL CHLORIDE, AND TEFLON

Fluid	Limiting shear stress, τ_L, at P = 0.67 GPa	Average shear stress based on traction measurement at a slide-roll ratio of 10^{-1}, average pressure of 0.67 GPa [9]
5P4E	58 MPa (38°C)	55 MPa (35°C)
Vitrea 79	31 MPa (26°C)	33 MPa (30°C)
LVI 260	49 MPa (35°C)	51 MPa (35°C)

TABLE 8 COMPARISON OF LIMITING SHEAR STRESS WITH EHD TRACTION SHEAR STRESS, (REF. 11)

Figure 31 shows the typical log viscosity versus pressure isotherm for a material (5P4E) for different rates of stress application. At very low shear stress, the effective viscosity behavior is the traditional logarithmic dependence upon pressure. However, as the pressure increases the limiting shear stress is reached resulting in a nearly constant effective viscosity. A shear rheological model suitable for the entire range of behavior can be found in Reference 16.

9. SPECIFICATIONS OF LUBRICANTS

As indicated in the previous discussion on lubricants, lubricant behavior is dependent upon rheological properties and chemical properties. The specification of lubricants is a very complex consideration based on a number of criteria. The criteria include both readily measurable rheological properties, primarily viscosity at various temperatures, and performance criteria based on accepted application tests.

Probably the most common specification for lubricant selection is the viscosity of the material at the anticipated temperature of application. In this case there are ISO grades of lubricant viscosity, as indicated in Figure 32, which are approximately the kinematic viscosity in centistokes (mm^2/sec) at 40 °C[9]. The ISO specifications are commonly used in industrial lubricants. There are also the SAE viscosity grades for automotive engine lubricants and automotive transmission and axle lubricants[17].

The current SAE viscosity grades for engine oils are shown in Table 9(a) and (b). These viscosity grades are based on measurements at high temperature by kinematic viscosity (e.g., low shear stress viscosity) as well as low temperature viscosities based on a pumpability consideration and a non-Newtonian increase of viscosity at low temperature. Very

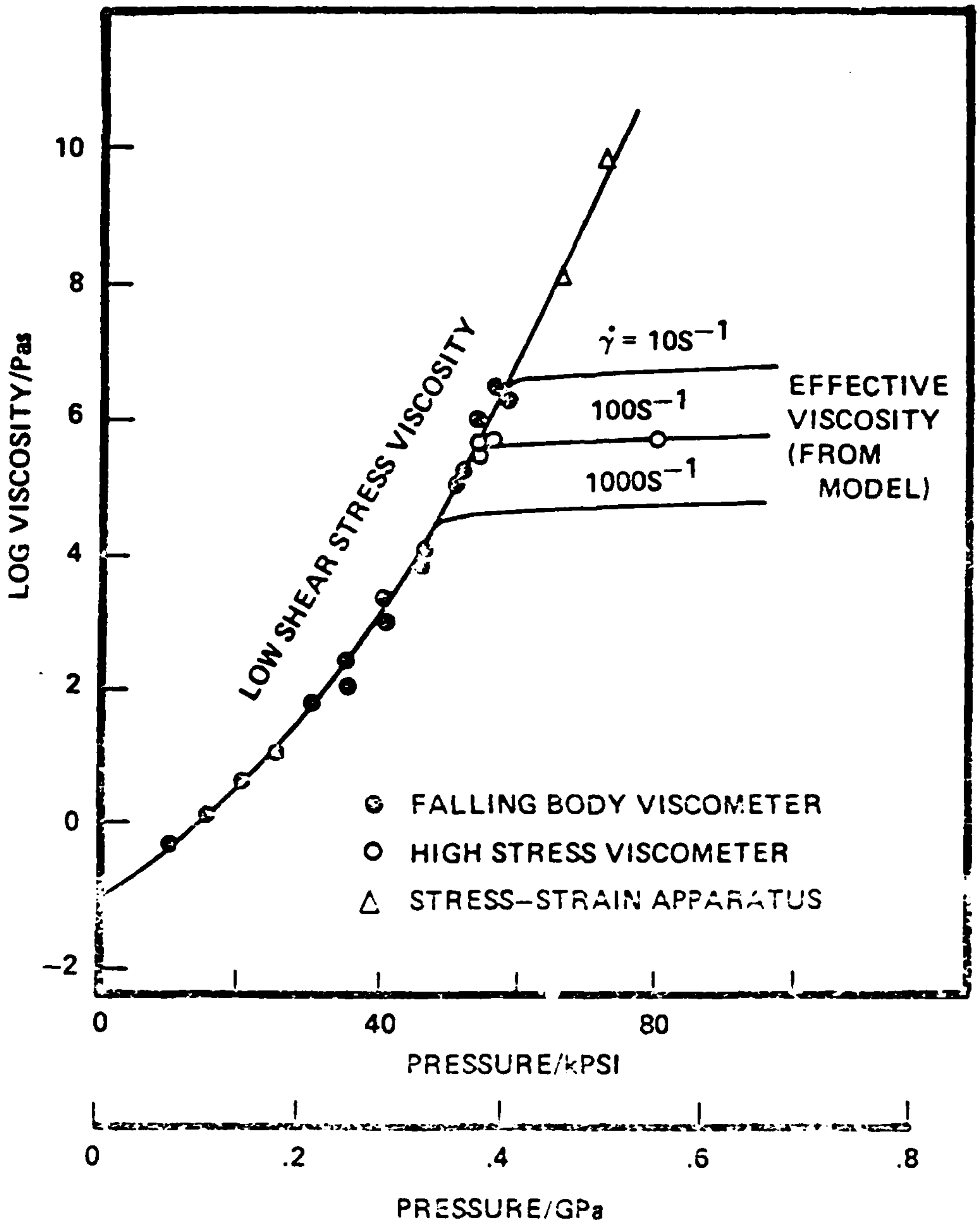


FIGURE 31 VISCOSITY PRESSURE ISOTHERM ($60^{\circ}C$) FOR 5P4E BY INDICATED METHODS OF MEASUREMENT. LINES OF CONSTANT SHEAR RATE PREDICTED FORM MODEL (REF. 16)

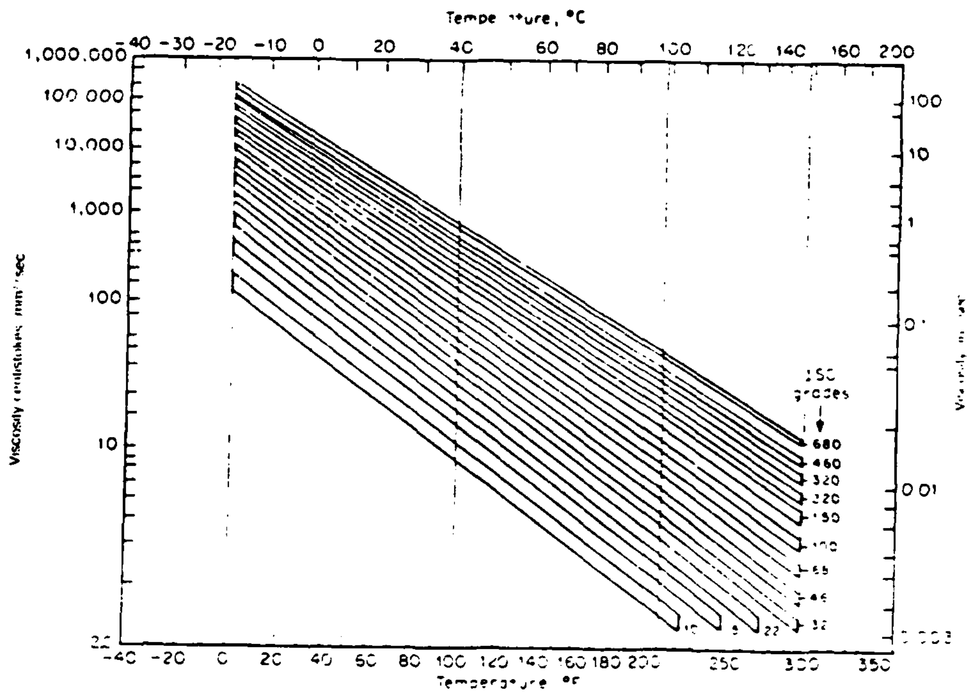


FIGURE 32 VISCOSITY-TEMPERATURE BEHAVIOR OF TYPICAL 100 VI LUBRICANT IN ISO VISCOSITY GRADES (REF. 8)

SAE Viscosity Grade	Viscosity Range		
	Centipoises (cP) at −18°C (ASTM D 2602)	Centistokes (cSt) at 100°C (ASTM D 445)	
	Max	Min	Max
5W	1 250	3.8	—
10W	2 500	4.1	—
20W*	10 000	5.6	—
20	—	5.6	less than 9.3
30	—	9.3	less than 12.5
40	—	12.5	less than 16.3
50	—	16.3	less than 21.9

Note: 1 cP = 1 mPa·s; 1 cSt = 1 mm²/s
*SAE 15W may be used to identify SAE 20W oils which have a maximum viscosity at −18°C of 5 000 cP.

TABLE 9(A) SAE VISCOSITY GRADES FOR ENGINE OILS (REF. 17)

SAE Viscosity Grade	Viscosity* (cP) at Temperature (°C) Max	Borderline Pumping Temperature**(°C) Max	Viscosity*** (cSt) at 100°C Min	Max
0W	3 250 at -30	-35	3.8	-
5W	3 500 at -25	-30	3.8	-
10W	3 500 at -20	-25	4.1	-
15W	3 500 at -15	-20	5.6	-
20W	4 500 at -10	-15	5.6	-
25W	6 000 at - 5	-10	9.3	-

* By proposed modification of ASTM D 2602,

** ASTM D 3829, *** ASTM D 445

SAE Information Report

TABLE 9(B) PROPOSED NEW DEFINITIONS FOR "W" GRADES likely in the near future, additional criteria will be added to the SAE engine oil viscosity specifications for high temperature (150°C) and high shear rate viscosity of the oil.

The SAE viscosity classifications for axle and transmission lubricants are shown in Table 10 and Figure 33, taken from the SAE Handbook[17]. Similarly in the case of greases, a rheological specification is provided by National Lubricating Grease Institute, referred to as NLGI consistency numbers and shown in Table 11. These classifications are also based on measurements according to an ASTM Standard.

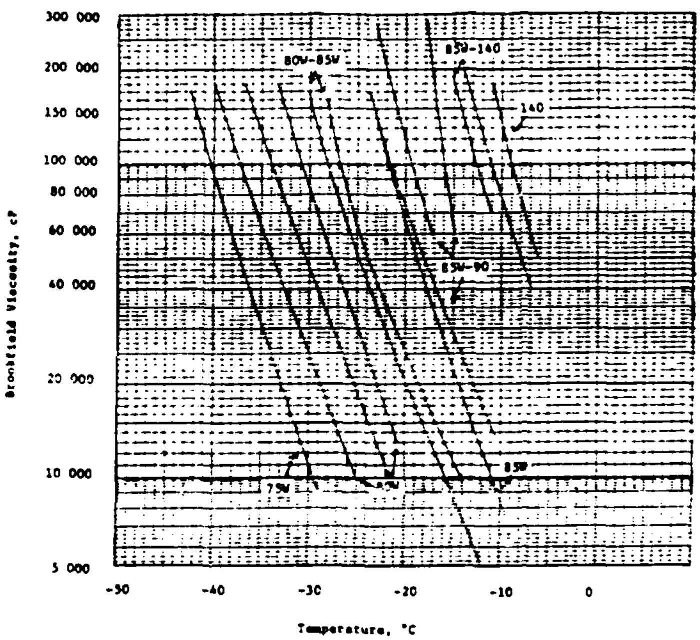


FIGURE 33 BROOKFIELD VISCOSITY VERSUS TEMPERATURE FOR TYPICAL GEAR LUBRICANTS (SAE VISCOSITY GRADES INDICATED) (REF. 17)

The performance specifications on lubricants vary with the particular application. Probably the most highly developed of these is the performance applications utilized by the American Petroleum Institute (API) concerned with automotive engine oil performance. A general description of the performance designations taken from the SAE Handbook is shown in Table 12[17]. More detailed discussion of the

SAE Viscosity Grade	Maximum Temperature for Viscosity of 150 000 cP* °C	Viscosity at 100°C** cSt	
		Minimum	Maximum
75W	-40	4.1	—
80W	-26	7.0	—
85W	-12	11.0	—
90	—	13.5	< 24.0
140	—	24.0	< 41.0
250	—	41.0	—

*Centipoise (cP) is the customary absolute viscosity unit and is numerically equal to the corresponding SI unit of millipascal-second (mPa·s)

**Centistokes (cSt) is the customary kinematic viscosity unit and is numerically equal to the corresponding SI unit of square millimetre per second (mm^2/s).

†The new viscosity classification represents a conversion to international SI units using degrees Celsius and with a minimum change in viscosity limits relative to prior practice. By early 1982, it is the aim to define the low temperature requirements at suitable multiples of 5°C while retaining 100°C for the high temperature range. The proposed revision will necessitate considering changes of the viscosity limits for the high and/or low temperatures used to define the new system.

TABLE 10 AXLE AND MANUAL TRANSMISSION LUBRICANT VISCOSITY CLASSIFICATION (REF. 17)

NLGI Consistency No	ASTM Worked (60 Strokes) Penetration at 25°C (77°F) tenths of a millimetre	NLGI Consistency No	ASTM Worked (60 Strokes) Penetration at 25°C (77°F) tenths of a millimetre
000	445 to 475	3	220 to 250
00	400 to 430	4	175 to 205
0	355 to 385	5	130 to 160
1	310 to 340	6	85 to 115
2	265 to 295		

*National Lubricating Grease Institute, 4635 Wyandotte St., Kansas City, Missouri 64112

TABLE 11 NLGI* CONSISTENCY NUMBERS

Letter Designation	API Engine Service Description	ASTM Engine Oil Description
SA	**Formerly for Utility Gasoline and Diesel Engine Service** Service typical of older engines operated under such mild conditions that the protection afforded by compounded oils is not required. This category has no performance requirements and oils in this category should not be used in any engine unless specifically recommended by the equipment manufacturer.	Oil without additive except that it may contain pour and or foam depressants.
SB	**Minimum Duty Gasoline Engine Service** Service typical of older gasoline engines operated under such mild conditions that only minimum protection afforded by compounding is desired. Oils designed for this service have been used since the 1930s and provide only antiscuff capability and resistance to oil oxidation and bearing corrosion. They should not be used in any engine unless specifically recommended by the equipment manufacturer.	Provides some antioxidant and antiscuff capabilities.
SC	**1964 Gasoline Engine Warranty Service** Service typical of gasoline engines in 1964–1967 models of passenger cars and trucks operating under engine manufacturers warranties in effect during these model years. Oils designed for this service provide control of high and low temperature deposits, wear, rust, and corrosion in gasoline engines.	Oil meeting the 1964–1967 requirements of the automobile manufacturers. Intended primarily for use in passenger cars. Provides low temperature antisludge and antirust performance.
SD	**1968 Gasoline Engine Warranty Maintenance Service** Service typical of gasoline engines in 1968 through 1970 models of passenger cars and some trucks operating under engine manufacturers warranties in effect during these model years. Also may apply to certain 1971 and or later models, as specified (or recommended) in the owners manuals. Oils designed for this service provide more protection against high and low temperature engine deposits, wear, rust, and corrosion in gasoline engines than oils which are satisfactory for API Engine Service Category SC and may be used when API Engine Service Category SC is recommended	Oil meeting the 1968–1971 requirements of the automobile manufacturers. Intended primarily for use in passenger cars. Provides low temperature antisludge and antirust performance
SE	**1972 Gasoline Engine Warranty Maintenance Service** Service typical of gasoline engines in passenger cars and some trucks beginning with 1972 and certain 1971 models operating under engine manufacturers warranties. Oils designed for this service provide more protection against oil oxidation, high temperature engine deposits, rust, and corrosion in gasoline engines than oils which are satisfactory for API Engine Service Categories SD or SC and may be used when either of these categories are recommended	Oil meeting the 1972–1979 requirements of the automobile manufacturers. Intended primarily for use in passenger cars. Provides high temperature antioxidation, low temperature antisludge and antirust performance

TABLE 12 DESIGNATION, IDENTIFICATION AND DESCRIPTIONS OF CATEGORIES (REF. 17)

TABLE 12
(Continued)

Letter Designation	API Engine Service Description	ASTM Engine Oil Description
SF	**1980 Gasoline Engine Warranty Maintenance Service** Service typical of gasoline engines in passenger cars and some trucks beginning with the 1980 model operating under engine manufacturers' recommended maintenance procedures. Oils developed for this service provide increased oxidation stability and improved anti-wear performance relative to oils which meet the minimum requirements for API Service Category SE. These oils also provide protection against engine deposits, rust, and corrosion. Oils meeting API Service Category SF may be used where API Service Categories SE, SD, or SC are recommended.	Oil meeting the 1980 warranty requirements of the automobile manufacturers. Intended primarily for use in gasoline engine passenger cars. Provides protection against sludge, varnish, rust, wear, and high-temperature thickening.
CA for Diesel Engine Service	**Light Duty Diesel Engine Service** Service typical of diesel engines operated in mild to moderate duty with high-quality fuels and occasionally has included gasoline engines in mild service. Oils designed for this service provide protection from bearing corrosion and from ring belt deposits in some naturally aspirated diesel engines when using fuels of such quality that they impose no unusual requirements for wear and deposit protection. They were widely used in the late 1940s and 1950s but should not be used in any engine unless specifically recommended by the equipment manufacturer.	Oil meeting the requirements of MIL-L-2104A. For use in gasoline and naturally aspirated diesel engines operated on low sulfur fuel. The MIL-L-2104A Specification was issued in 1954.
CB for Diesel Engine Service	**Moderate Duty Diesel Engine Service** Service typical of diesel engines operated in mild to moderate duty, but with lower quality fuels which necessitate more protection from wear and deposits. Occasionally has included gasoline engines in mild service. Oils designed for this service were introduced in 1949. Such oils provide necessary protection from bearing corrosion and from high temperature deposits in normally aspirated diesel engines with higher sulfur fuels	Oil for use in gasoline and naturally aspirated diesel engines. Includes MIL-L-2104A oils where the diesel engine test was run using high sulfur fuel.
CC for Diesel Engine Service	**Moderate Duty Diesel and Gasoline Engine Service** Service typical of lightly supercharged diesel engines operated in moderate to severe duty and has included certain heavy duty, gasoline engines. Oils designed for this service were introduced in 1961 and used in many trucks and in industrial and construction equipment and farm tractors. These oils provide protection from high temperature deposits in lightly supercharged diesels and also from rust, corrosion, and low temperature deposits in gasoline engines	Oil meeting requirements of MIL-L-2104B. Provides low temperature antisludge, antirust, and lightly supercharged diesel engine performance. The MIL-L-2104B specification was issued in 1964.
CD for Diesel Engine Service	**Severe Duty Diesel Engine Service** Service typical of supercharged diesel engines in high speed high output duty requiring highly effective control of wear and deposits. Oils designed for this service were introduced in 1955, and provide protection from bearing corrosion and from high temperature deposits in supercharged diesel engines when using fuels of a wide quality range.	Oil meeting Caterpillar Tractor Co certification requirements for Superior Lubricants (Series 3) for Caterpillar diesel engines. Provides moderately supercharged diesel engine performance. The certification of Series 3 oil was established by Caterpillar Tractor Co. in 1955. The related MIL-L-45199 specification was issued in 1958.

particular tests and the nature of the criteria of performance associated with the tests can be found in the SAE Handbook. The tests consist of several engine tests to simulate different types of automotive operating conditions. There are different tests and pass/fail criteria for spark ignited Otto engines and compression ignited diesel engines. The performance criteria change frequently and are determined by joint committee efforts among SAE, API, and ASTM lubricant activities.

There are also performance specifications in other areas of application, the most notable of which is probably the military specifications utilized by various military agencies.

10. CONCLUSIONS

The brief overview of lubricant properties and behavior presented in this chapter serves as an introduction to a rather complex field very dependent on both science and art. It should give the reader a feel for the complexity of the field and the state-of-the-art. It also provides an introduction to the broader literature that exists in the field.

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12. ACKNOWLEDGEMENTS

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CONTAMINATION IN FLUID SYSTEMS

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1. INTRODUCTION

Contaminant is the scourge of all fluid dependent systems - including lubrication, hydraulic, pneumatic, liquid/gaseous fuel and coolant types. Operational success of most modern machine systems depends on its control. Synonymous to wear control for fluid systems, contamination control is central to most modern technological systems. By utilizing this fundamental factor, the user can achieve efficiency, reliability, and service longevity. Productivity goals and customer loyalty can only be gained and maintained by achieving contamination control of the integrated fluid systems.

Contaminant, as universally defined today, is any foreign or unwanted energy or substance that can have deleterious effects on system operation, service life, or reliability. The presence of contaminant in the fluid of a system has a catalytic effect upon component performance deteriorating processes (such as tribological and fluid-to-surface type wear mechanisms).

Although the scope of the subject is too broad to present in any depth in this discussion, the practice of contamination control can be viewed which will help fluid systems engineers gain a working perspective of the subject. Details have invariably obscured the critical relationships between the controlling factors and hampered progress. This chapter will attempt to provide a practical introduction to the subject of fluid contamination. Material will be presented under the following headings: theory, types, analyses, sources, consequences, and control/prevention.

2. THEORY

Contaminants in fluid-dependent systems result from change that takes place during manufacture of the components and the system, change that occurs within the system during operation, and change that occurs in the system when it is exposed to the environment. The resulting contaminant must be accurately characterized and assessed before the user can appraise the seriousness of its manifestation.

The level of contaminant can vary from point to point in the system, from time to time, and unfortunately by the method of sampling. The contamination level is always highest just upstream of the contaminant separator and, of course, immediately downstream of the major ingression sources.

The ingression duty cycle corresponds to the severity of the actual work and respective work environment of the system and is the most important factor influencing contamination level on a time basis. Flow through the contaminant separator (or the residence time of the fluid in the system) also determines the magnitude of the contamination level. The only remaining factor influencing the contamination level of the fluid is the actual performance characteristics of the contaminant separator, i.e., the better it is, the lower the level.

A mathematical model can be written to express the contamination level of a circulating type system, as illustrated in Figure 1. This model would apply to all fluid system types, with the possible exception of a "single pass" fuel system. The theoretical relationship would be an accounting of the material ingressed per minute (R), the material removed by a filter having a filtration ratio of Beta, and the material still present in the system as reflected downstream of the filter (N_d).

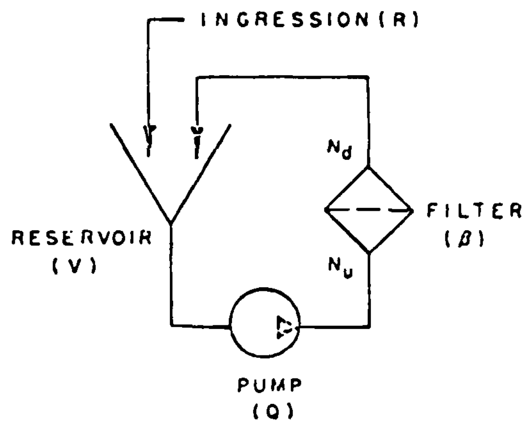

FIGURE 1 BASIC CIRCUIT

During the transient period, the ratio of the volume of the reservoir (V) to the flow rate (Q) of the pump can influence, along with Beta of the filter, the time constant of the contamination level curve of the system. However, since this transient period is generally very short in most modern systems, only the steady-state part of the differential equation is of real interest. For all practical purposes, the steady-state value of the downstream contamination level of the fluid can be described by the following equation;

$$\bar{N}_d = \frac{R}{(\beta - 1)\,Q} \tag{1}$$

The filtration ratio, Beta, is a function of both upstream and downstream particle concentration per unit volume of fluid as defined by

$$\beta = \frac{N_u}{N_d} \tag{2}$$

at the steady-state condition when the standard contaminant (AC Fine Test Dust) is the ingressed material. All particle-related terms in Eq. (1) are cumulative (i.e., includes all particles above a given size). For example, if the reference size is 10 micrometers, the units of "R" would be the number of particles 10 micrometers and larger which ingress the system per minute; while the filtration ratio would be identified as Beta Ten.

Cumulative efficiency of a filter is defined as

$$E_c = \frac{N_u - N_d}{N_u} \tag{3}$$

Using Eq. (2), the cumulative effeciency can be written as a function of only Beta, or

$$E_c = \frac{\beta - 1}{\beta} \tag{4}$$

This relationship is reflected in the efficiency graph shown in Figure 2.

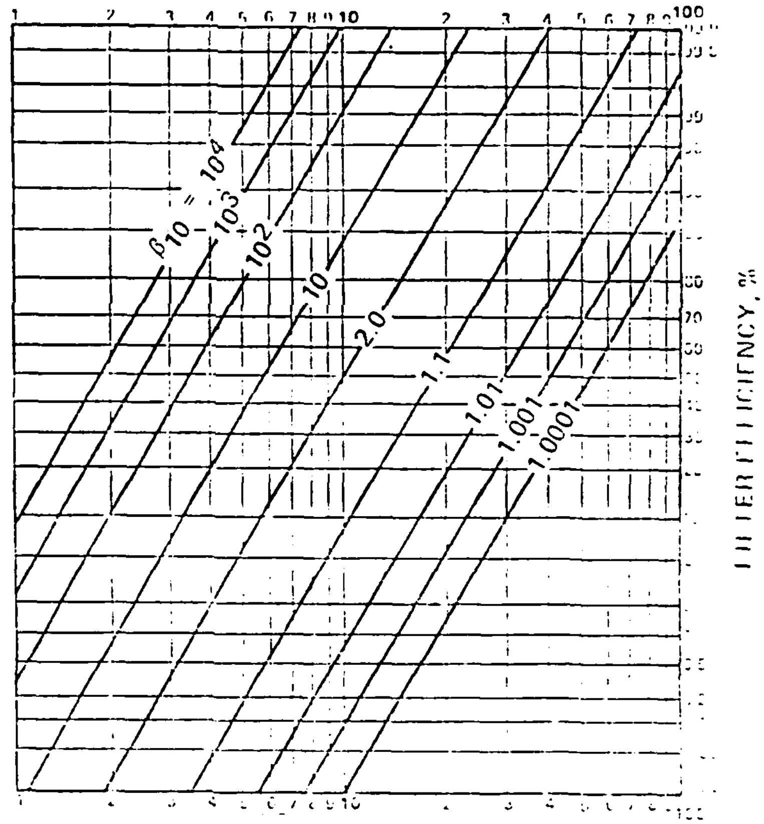

FIGURE 2 BETA VERSUS EFFICIENCY

Note from Eq. (1) that the contamination level of the system can be reduced an order of magnitude by simply changing either R, Q, or Beta appropriately. Control possibilities are evident and should be recognized.

Contaminant service life of a component or system depends on the balance existing between the two influencing factors - contamination level of the fluid and contaminant tolerance of the component or system in question. The contaminant tolerance level exhibited by a fluid component depends upon its contaminant sensitivity and operating conditions. The contaminant sensitivity of a fluid component, an inherent characteristic, depends upon three design aspects - basic mechanism, fabrication materials, and unit loading. Operating conditions which may be imposed on the fluid component include not only fluid properties but also the severity of the pressure, temperature and speed duty cycle.

All fluid components are sensitive in some degree to particulate contaminant entrained in the fluid. The term, "contaminant sensitivity," refers to degradation in performance occurring when a component is exposed to a specific contamination environment. For example, for a pump, rationalize that some finite fluid delivery volume is lost for every particle passing through its pumping chamber. This lost delivery volume (due to exposure to a given particle size) can be expressed as

$$Q' = \frac{\text{volume lost}}{\text{particle}} \times \frac{\text{particle exposure}}{\text{time}} \qquad (5)$$

Of course, the rate of particle exposure is simply the product of the particle concentration per unit volume and the flow rate.

Contaminant wear results from particle interaction with component surfaces. Contaminant wear can be reflected in two different ways; by actual wear debris generated from the critical surfaces and degradation in a responsive performance parameter of the component resulting from surface deterioration and clearance changes. The block diagram shown in Figure 3 relates these concepts for a hydraulic pump. Operating conditions and the exposed contaminant level cause the destruction of critical surfaces. The results are wear debris and clearance changes. Wear debris can be measured by a Ferrograph and reported in terms of the D54 density. The clearance change can be measured in terms of flow readings and reported in terms of flow degradation. The block diagram shows why an interrelation exists between flow degradation of a pump and its Ferrographic reading, a most important conclusion to the study of fluid contamination control.

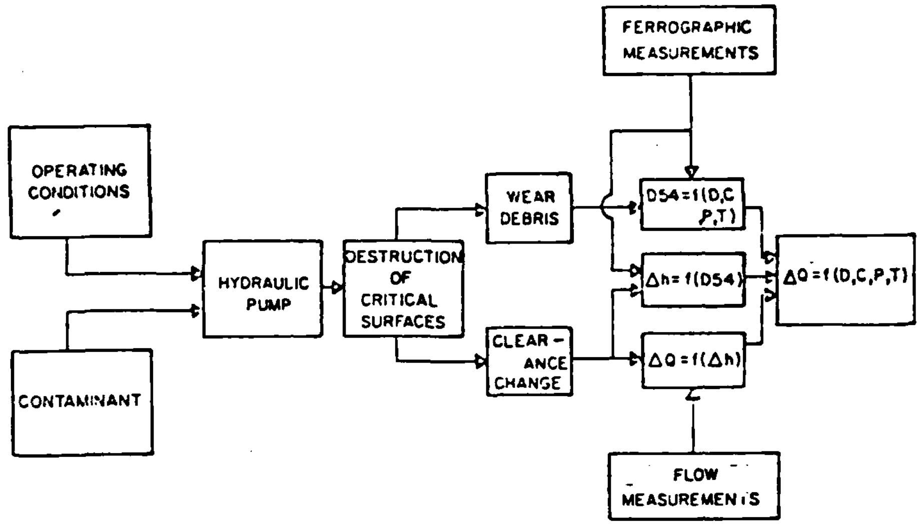

FIGURE 3 PERFORMANCE VERSUS WEAR

Research and application in fluid contamination control have been hampered by the complexity of the interacting parameters. Only recently have researchers gained sufficient insight in its practice, permitting recognition and understanding of the controlling parameters of the system and adequate description of the influence of each parameter on the system as a whole.

The Contamination Control Balance, illustrated in Figure 4, attempts to convey the nature of the interacting parameters in a pictorial form. The Balance should provide necessary insight for implementation of the theory and prove valuable in helping the reader understand and explain many principles on which contamination control is based.

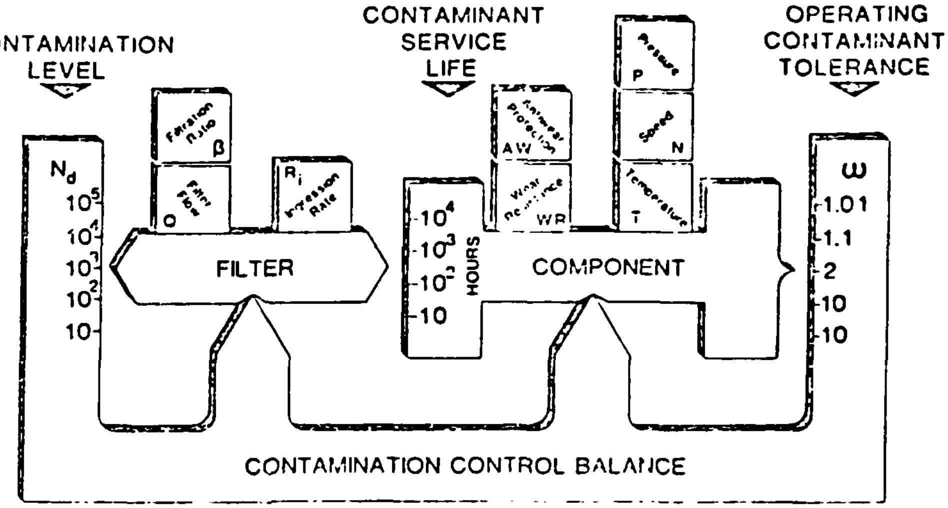

FIGURE 4 CONTAMINATION CONTROL BALANCE

3. CONTAMINANT TYPES

Contaminant, unwanted material or energy in the system, can exist as a gas, liquid, or solid. A gas can either be in the dissolved state or entrained in the liquid as bubbles or slugs. A liquid, depending on its volume and compatibility with the host fluid, can be free, dissolve, or emulsified. Since the term, liquid, encompasses all deformable, cohesive matter, precipitates (such as gums and biological debris) are included in this category. Solid matter found in the fluid is better known as particulates, or just particles. Material type contaminants of interest in fluid systems include particulate, aeration, tramp liquid (water), chemical and microbial.

Unwanted energy in fluid systems can deteriorate both the fluid and exposed system components and be the prime factor for failure, poor reliability and short service life. Forms of energy which can be highly detrimental to fluid-dependent systems are radiation, static electricity, corrosion, magnetism, and thermal energy.

Particulate contamination is characterized by specific descriptive, identifying terms which can be classified as follows:

- inherent physical and chemical properties
- environmental behavior
- geometrical shape
- size distribution
- concentration or population density.

Actual characterization factors include particle density, hardness, compaction, settling, dispersion, transport, agglomeration, limiting size, state, shape, size, size distribution, and concentration. Particle size distributions found in fluid-dependent systems are not normal but skewed in the direction of greater numbers of small particles. When plotted on the cumulative log-log^2 graph, these system distributions often exhibit a linear characteristic. If the distribution plot is not linear, problems usually exist.

Aeration is due to dissolved or suspended air in the system fluid. As long as air remains dissolved, no problems occur; but, in low pressure areas, it can become suspended air and the system will exhibit many undesirable characteristics such as the following:

- lower bulk modulus
- loss of power
- gaseous cavitation
- noisy operation
- accelerated fluid oxidation
- loss of lubricity
- higher temperatures
- loss of stiffness.

Water, a serious contaminant in most mineral-base fluids, is always present in operating systems to some degree, either in its free or dissolved form. Free water may be present as a precipitant or emulsified with the host fluid. Water is a major cause of rust and corrosion and the breakdown of additive packages in the fluid. When both water and dirt are present in the fluid, a synergistic effect results. Also, water in a system at subfreezing temperatures can transform into ice crystals that jam valves, clog filters, and cause controls to become erratic, the same problems as caused by sand particles and metal chips.

Chemical contaminants include solvents, fluid-breakdown residue, and surface-active agents. Literature abounds with horror stories of the contamination of complex fluid systems by chemicals. Incompatible chemicals can hydrolyze in the system, forming acids which in turn attack internal metallic surfaces. Surface-active chemicals, called surfactants, produce a variety of unfavorable conditions in fluid systems, particularly fuel systems.

Microbial contaminant is an ever present threat, particularly when water exists in the system. Once active growth has commenced in the aqueous phase of the fluid, it is generally self-sustaining due to the fact that water is one of the end products of an infestation. Microbial contamination results in the following:

- short fluid life
- degraded surface finish
- short filter life
- rapid corrosion
- obnoxious smells and fluid discolorations.

Radiation contamination is radiant energy in, on, or around the fluid system. It can change the state of a material, induce a chemical reaction, change a physical characteristic, or alter an electrical or magnetic characteristic of a material. In a fluid system, the fluid itself is most susceptible to damage by radiation. Changes have been noted in viscosity, acidity, volatility, foaming, coking, flash point, autogeneous ignition temperatures, and oxidation stability. Elastomers, plastics, and resins (which make up many of the seals, packings, and similar components of fluid systems) are also severely vulnerable to radiation contamination.

Static electricity or electrokinetic contamination in fluid systems result from fluid moving past surfaces within the system. At the solid-fluid interface, ions become detached, causing a streaming current, and the potential difference along the direction of flow is referred to as the streaming potential. This resulting charge depends on the flow velocity, properties of the fluid, and the surface area of the solid-liquid interface. This contaminant causes serious erosion in valves, and the charge generation has been responsible for many industrial fires.

Corrosion contamination includes electrochemical reactions that occur within the confines of a closed fluid system. This reaction, an oxidation and reduction process, can destroy critical surfaces and generate gross amounts of particulate contamination (rust). Types of corrosion identified include galvanic, pitting, crevice, intergranular, erosion, stress/cracking, and fretting.

Magnetism, a contaminant in a fluid dependent system when unshielded magnetic fields exist near control actuators, can force ferromagnetic particles in the fluid to move and collect in critical areas. Such capturing of particles can have serious consequences when they bridge, clog, or jam critical orifices, clearances, or invade load-bearing wedge-shaped openings.

Thermal energy (either excess or lack of it) can be just as devastating a contaminant to a fluid system as any other unwanted or foreign substance or energy. Temperatures above design conditions can cause a chain reaction leading to the total destruction of the system, early seal failure and fluid breakdown. The effect of subfreezing temperatures on the system can be highly detrimental to the proper operation and functioning of the system. Low temperatures result in high input power; pump cavitation; seals loose their memory, become distorted and crystallize; valves respond slowly; filters bypass or rupture, etc.

Regardless of its form, if the degree of manifestation of contaminant is intolerable to the components of the system, it must be reduced in level and controlled, limiting its ingression or removing it from the system.

4. ANALYSIS

The credibility of contaminant analysis depends just as much on having a representative sample of the contaminant as on the accuracy of the evaluation technique. To be representative of the contaminant in the system, the sample must be undefiled by exposed surfaces or by the environment with which it may come in contact prior to being appraised. Furthermore, for a sample to be representative, it must contain a full contaminant spectrum from a nondiscriminatory part of the system. Finally, the contamination assessment method has a direct influence on the potential value of the results.

Sample container cleanliness is the first prerequisite for achieving valid contaminant analysis. Furthermore, such control must be exercised over sampling appendages and any other exposed wetted surfaces within the analyzer process. Cleanliness control must also be maintained over the fluids used to flush, rinse, or dilute the sample. When particle evaluation methods do not restrict particle entry or consider the effect of tramp particles, the cleanliness of the environment surrounding the analysis station can be suspect.

Sampling method is the second prerequisite needed to ensure the validity of contaminant analysis of a fluid. Validity concerns the propriety of the sample with respect to the contaminant population it represents. Using an inappropriate sampling technique can compromise sample validity, as can withdrawing the fluid from a biased contaminant-level location or obtaining the sample during a nonrepresentative period of system activity. Selection of the sampling method and its proper application are critical aspects of the overall analysis effort. Obviously, if a sample is removed from a cyclonic or particle size stratified zone, the specimen is not representative. Finally, contamination profiles are dynamic in nature and continually change (depending on the environment, wear of exclusion devices, and the characteristics of the filter). Hence, the sampling period, with respect to the machine or to its system work and environmental cycle, is of paramount importance in obtaining a representative fluid sample.

Particle dispersion is a major requirement if the analysis method is to discriminate between discrete particles in the fluid. Of course, dilution helps minimize the interaction between particles and reduce flocculation and agglomeration; but, dispersing agents may be needed to stabilize the suspension. Poorly dispersed particles often represent the biggest single problem in size analysis of particulate contamination. When flocs form, particle size values are usually too large and the size distribution is too broad.

Particle settling is often a serious problem - particularly when particle density is great and fluid viscosity low. Particles must be maintained in suspension so that the full size distribution is presented representatively to the sensing zone of the analyzer, or a biased analysis results. The use of magnetic stirrers (for nonferrous particles) is an effective solution but inappropriate for most wear-type system samples. The settling rate of discrete particles in a given fluid can be assessed by Stokes Law or by conducting repeated analyses and establishing the rate at which the number of particles of a given size declines. Compensation for significant particle settling must be made or fluid viscosity increased (through dilution with a more viscous fluid or by refrigeration to hold heavier particles in suspension). The cumulative particle-size distribution curve steepens as the settling of particles continues unabated.

Aeration occurs when air bubbles are dispersed in a fluid. Particle counting instruments usually count them as if they are particulate matter. In fact, air entrained in the sample during agitation prior to analysis can totally mask the true size distribution of the particles. The size of the air bubbles depends on the surface tension of the fluid in which the air is dispersed. Normally most of the particles are small, producing an upward inflection of the cumulative particle size distribution curve at the small particle sizes. Researchers have developed effective techniques to give fast deaeration with minimal difficulty.

Undissolved liquid (e.g. water) in a sample can totally mask the true size distribution of the particles, just like air. Sometimes such a liquid is inadvertently added with dilutent, or it may have been contained originally in the sample of fluid. In either case, agitation of the sample just prior to anlysis effectively disperses the small droplets throughout the sample and results in an erroneous distribution of the particulate matter in the fluid. Undissolved and finely dispersed liquid droplets in a fluid cause an unrealistically large particle population and produce an effective flattening of the cumulative distribution curve.

Once a representative contaminant sample has been obtained and properly prepared (deaerated, dehydrated, dispersed and diluted); its assessment, reporting, and interpretation are all subjects of concern. Factors which deserve consideration are the following:

- effective analysis methods
- analysis method calibration
- cleanliness level reporting
- analysis interpretation.

Many methods are available for analyzing distribution of particle sizes in a fluid. Each method utilizes some particular property or combination of properties to distinguish one size particle from another. In some methods, the size dimension is measured directly; whereas, in others, the dimension is derived from measured physical behavior of the particles. Furthermore, some methods detect and measure each particle; whereas, in other methods, particles are evaluated as a unit.

Analysis methods can be classified as optical, electrical, geometric, gravitational, cylconic, magnetic, particle concentration, or ultrasonic. Optical methods include imaging (microscopes) and light extinction methods (such as employed by HIAC/ROYCO). Electrical types measure the change in fluid resistance when a particle passes through an aperature. Geometric methods include the ever popular sieving method and the modern version, the microsieve. Gravitational methods cover both sedimentation and elutration techniques. Inertial type separators perform cyclonic classification. Ferrography is the basic magnetic type analysis method. Particle concentration methods include gravimetric, silting index, turbidity, optical density, and the simple patch test. Ultrasonics is making a comeback, based on the work at Brown University.

Automatic particle counters are the most popular means of assessing the contamination level of fluids in industry today. These instruments would not be popular, however, if standards did not exist with which to calibrate them. For many years, users of particle counters were totally dependent on the manufacturer of the counter for calibration and achieving a commonality between laboratories. This dependency ended in the fluid power industry with the adoption of a calibration procedure that utilizes AC Fine Test Dust as the calibration medium.

A reproducibility survey was conducted prior to the adoption of the procedure by ISO and has become a standard (ISO 4402). This round-robin indicated that laboratories which had previously calibrated their counters using ACFTD procedure showed excellent reproducibility; whereas, those which depended upon intuition and nonstandard calibration techniques had unsatisfactory performance. Typical results of this survey are shown in Figure 5. The international survey demonstrated that, after each counter was calibrated per the ACFTD procedure, particles could be counted with an average standard deviation of $2/3\sqrt{N}$ (where N is the average number of particle counts). That is, if 10,000 particles are being counted, then the deviation would be 66.7 particles.

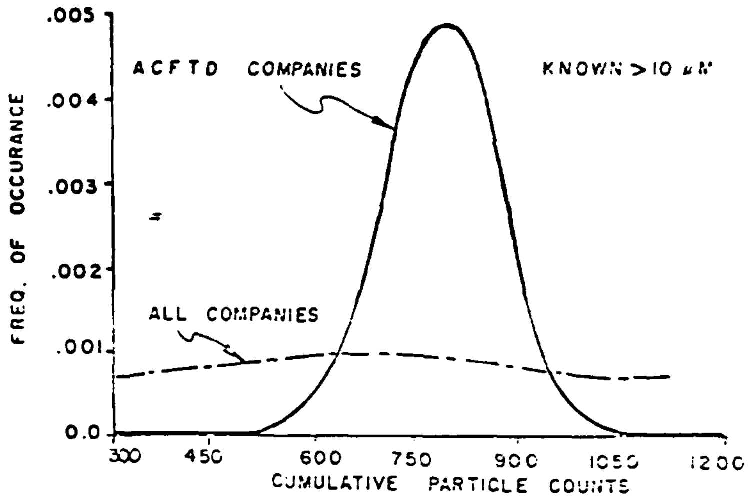

FIGURE 5 RESULTS OF INTERNATIONAL SURVEY

Categorizing particle counts to define cleanliness classes was instituted in the early 1960's by several organizations. Of these early attempts, only NAS 1638 is still in use within the aerospace field. All the early cleanliness codes had one common flaw - they were based on a fixed particle size distribution. Interestingly enough, this fixed distribution, selected as typical of field system ditributions of that period, corresponds roughly to that of AC Fine Test Dust. Undoubtedly, particle distributions from those early systems displayed the results of poor filter bypass control, which tended to flatten the distributions like AC Fine Test Dust.

The ISO Solid Contaminant Code (ISO 4406), approved in the late 1970's, represents a significant milestone in the area of fluid-contamination control. It provides a simple, unmistakable, meaningful, and consistent means of communication between suppliers and users. It applies to all types of fluid systems, and a theoretically infinite number of slopes is available to describe the contamination level of a fluid.

The ISO Code is assigned on the basis of the number of particles per unit volume greater than 5 and 15 micrometers in size. These two sizes were selected because of feeling that the concentration at the smaller size would give an accurate assessment of the silting condition of the fluid, while the population of the particles greater than 15 micrometers would reflect the prevalence of wear catalysts. Thus, particle size distribution by the ISO coding system, is described by a 5 micrometer range number and a 15 micrometer range number (with the two numbers separated by a solidus). When particle size distribution is plotted on the graph shown in Figure 6, the ISO Code can be read by noting the corresponding Range Numbers on the graph directly where the curve crosses the 5 and 15 micrometer particle size.

6. SOURCES

As far as functioning, service life, and reliability of a fluid system are concerned, the specific source of contaminant makes little difference. The fact that it exists, is entrained in the fluid, and gains exposure to critical surfaces is of fundamental importance. The term "ingressed contaminant" is reserved for contaminant from "wherever", the magnitude of which must be controlled.

Forms and types of contaminants present in the fluid encompass the complete range of materials and fluids used in

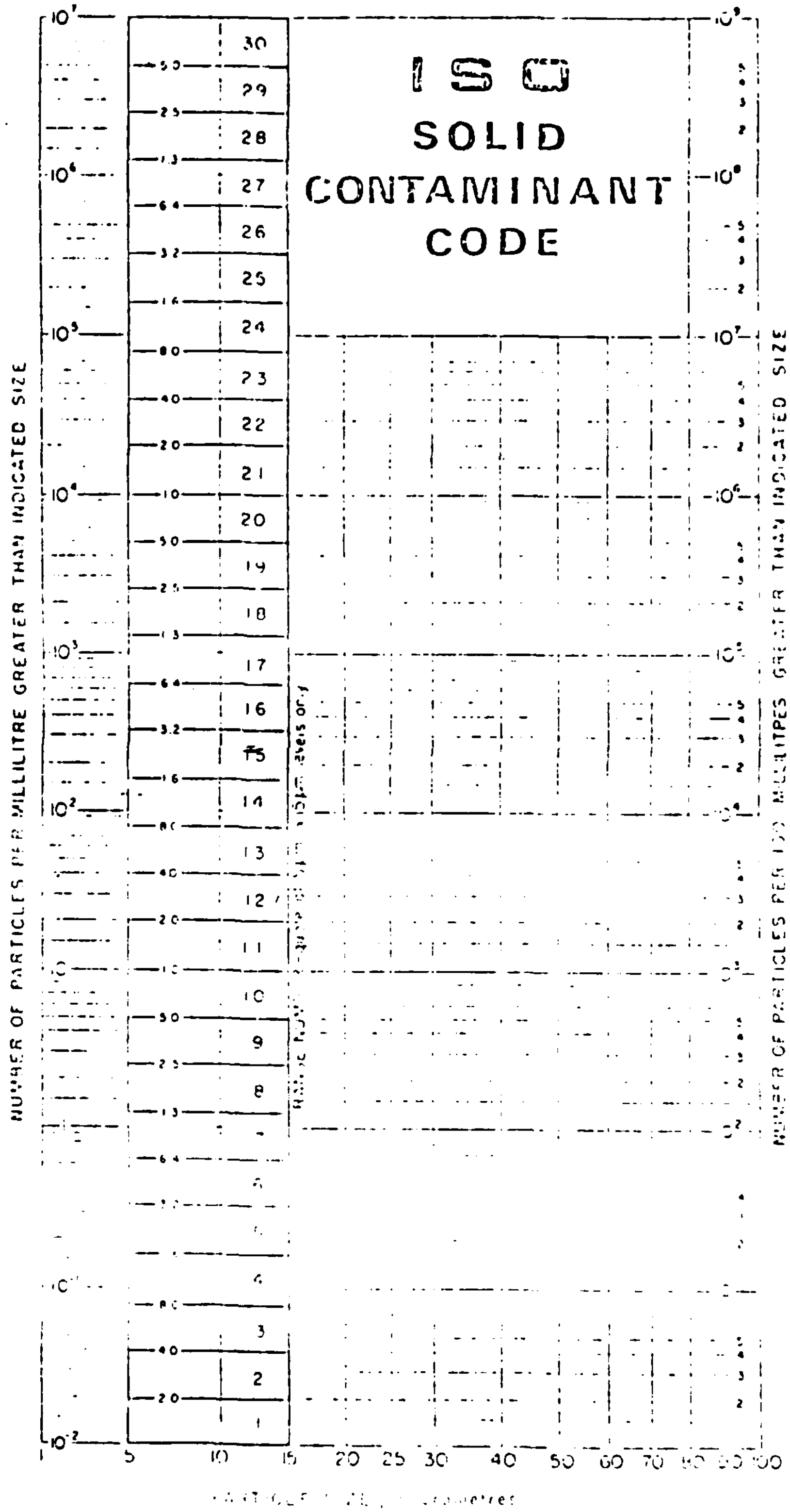

FIGURE 6 THE ISO SOLID CONTAMINANT CODE

producing and operating the system, as well as contaminant characteristics of the ambient environment in which the system has operated. Even the ingression mechanism affects the type system contaminants, such mechanisms as the wear modes, chemical reactions, and ingestion paths.

Specific origin of system contaminant is difficult to identify because sources of particulates are as numerous as the material constituents involved in the system. The dominant contaminant in a system is often far more important than the spectrum. Dominant contaminants are generally good indicators of system status; prevalent wear modes, internal state condition, severity of duty cycle, and degree of environmental hostility.

The sources of system contaminants can be classified as follows:

- Implanted contaminant
- Generated contaminant
- Ingested contaminant
- Escaped contaminant
- Induced contaminant
- Biological contaminant.

Implanted contaminant originates from manufacturing, handling, packaging, dispatching, and assembling process. This type of contaminant is the most critical to the system during new system break-in and commissioning exercises. Components such as lines and reservoirs with surface soil can be hazardous, particularly when the system is first placed in operation. But, if surfaces are rusty and corroded, they can cause damage throughout the life of the system, these are true perpetual ingression sources. The way to minimize implanted contaminant is to ensure that component surfaces are clean and protected prior to assembly. Once a system has been assembled, it should be flushed and checked by the Roll-off Cleanliness procedure (T2.9.8M-1979) recently approved by the National Fluid Power Association (NFPA). Filling the system with new "dirty oil" is another more subtle way of implanting contaminant in the system.

Generated contaminant constantly contributes ingressed contaminant. This contaminant represents the spoils of system inactivity, operation, and deployment. The magnitude of generated contaminant is normally controlled by respecting the severity limits of "rated" operation, environment, and fluid cleanliness. Barring the influence of human failings, only when the classical wearout mode occurs should the generated type of ingressed contaminant be alarming. Pump tests conducted by the Milwaukee School of Engineering and reported in SAE Paper 690866 showed that the average contaminant generated per hour is as shown in Figure 7. The Ferrograph offers an excellent method for monitoring generated contaminant from a fluid system, as illustrated in Figure 8 for a pump break-in test.

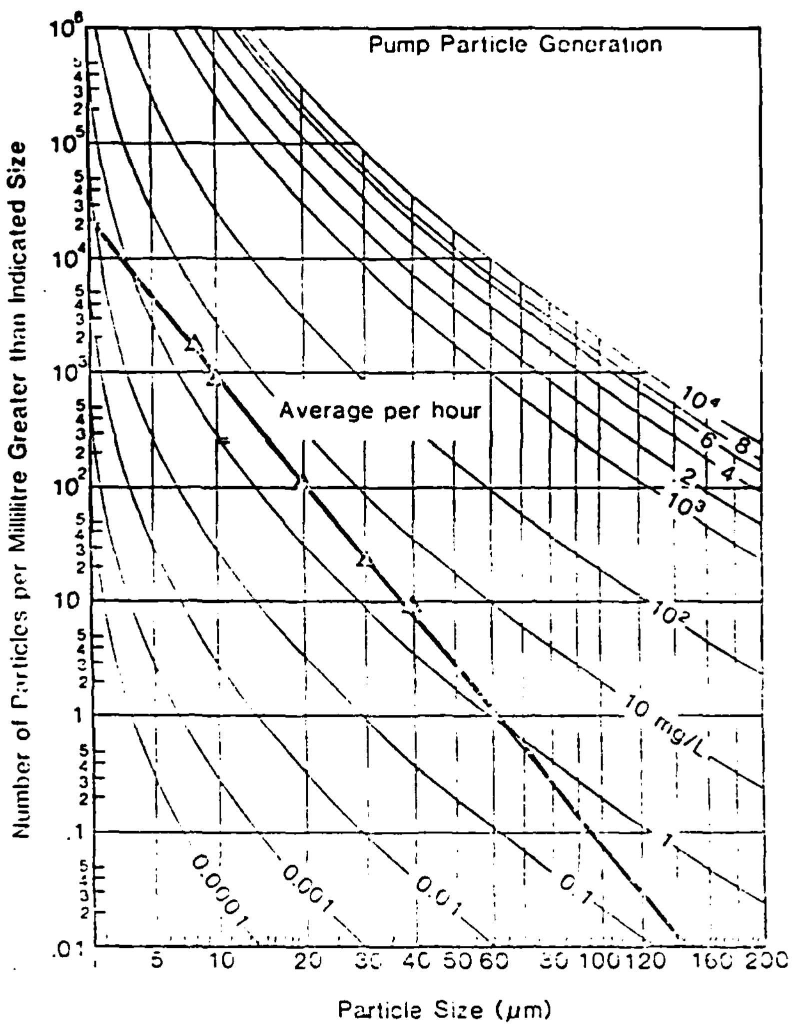

FIGURE 7 PUMP CONTAMINANT GENERATION RATE

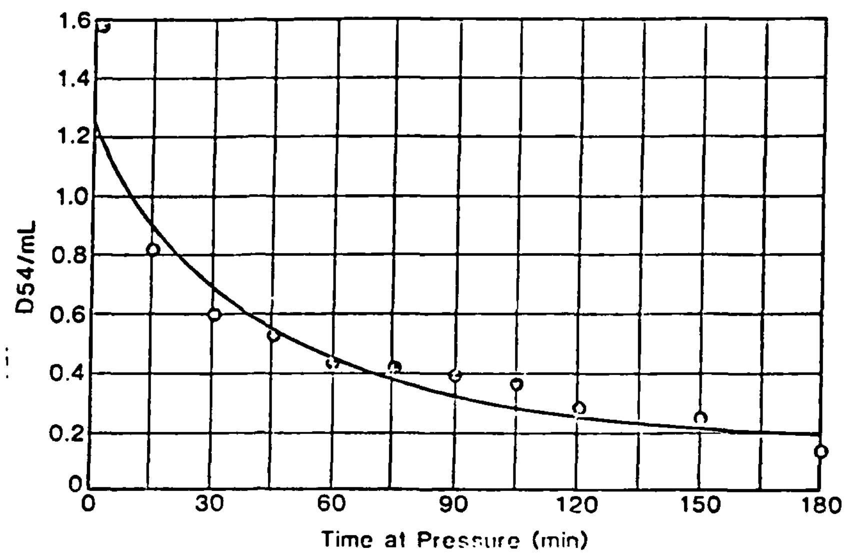

FIGURE 8 WEAR VERSUS PUMP BREAK-IN TIME

Ingested contaminant should be of utmost concern when the system must operate in hostile, dirty environments. Absolute exclusion of contaminant from the ambience is an idealistic thought. Field system studies have shown that ingested contaminant from wiper seals and reservoir breathers can totally dominate the contaminant ingression picture. When conditions warrant, this type of contaminant deserves almost "vigilante" attention.

Escaped contaminant is particles that have migrated through or around the filter. This type of contaminant must be identified quickly because the contamination level of the fluid is almost totally dependent upon the filter, not only for capturing particles but also for retaining them until they can be externally disposed. This contaminant results from a sneak flow path past the filter (an unseated relief or bypass valve or a cut in the filter medium) as well as from a flow surge that allows the viscous drag forces on the captured particles to exceed surface held forces of the fibers of the filter. Escaped contaminant is one of the major causes of early system failure and must be controlled in every conceivable way.

Induced contaminant is defilement from careless, improper, malicious, or hurried acts of system intrusions. This contaminant is often an operator's or user's nightmare. It exists because someone did the wrong thing - a farm worker adding water to the hydraulic system in order to finish a so-called perishable activity or mine personnel adding coal dust to the hydraulic reservoir to raise the level of the fluid to energize a float switch. Strict maintenance practices and operating policies are the only solution to this problem, rules that must be established by the one who must pay the consequences (the cost of equipment overhaul and the loss of production during downtime).

Microbial contaminant can only be arrested by the addition of a biocide to the fluid or by removal of one of the key physiological or nutritional requirements needed for microbes to flourish, that is, water, energy, or nutrients. Ingression of microbes may not be stopped, but the conditions for their survival and growth may be eliminated in the system. Unless arrested, microbial growth (regardless of the strain) results in a slime and usually an increase in the viscosity of the host fluid.

6. CONSEQUENCES

Every component has a limiting value for the contamination level which must be respected if the system goals for safety, performance, reliability, and service life are to be achieved. The consequences or penalty for not maintaining the contamination level below its limiting value is the loss of acceptable performace, a perceptible if not catastrophic failure. A perceptible failure includes loss of performance caused by abrasive, erosive, corrosive, electrostatic, and adhesive wear; wear types of which operator awareness generally exists. Catastrophic failures include such dreaded events as ruptures, collapses, lockups, obstructions, seizing, clogging, jamming, silting, and obliterations.

Degradation in performance can be related in most instances to the amount of contaminant wear (wear debris generation) occurring in the component. This relationship is a fundamental aspect of the contaminant sensitivity of components.

Sensitivity of a component to contaminant can be assessed by exposing it to ever-increasing sizes of contaminant, while measuring the influence of each size on the designated performance parameter. The performance parameter for a pump is flow; for a valve, it might be control pressure; for a hydraulic cylinder, the position hold capability; and, for a hydraulic pressure seal, static or dynamic leakage.

The test circuit needed to evaluate the contaminant sensitivity of a fluid compenent is schematically illustrated in Figure 9. With the test component operated at specified conditions, contaminants of various size ranges are exposed to the component for 30 minutes and then filtered. After each exposure period, the performance parameter is measured and recorded. The performance value after exposure is divided by the original performance value to obtain a performance degradation ratio. The performance degradation for both flow and pressure for a pressure-compensated pump is shown in Figure 10. In this example, the compensated pressure increases, which could lead to a dangerous safety situation. Flow degradation, as contaminant size ranges are exposed to the pump, continues to degrade in a normal fashion throughout pump life.

Another approach to contaminant sensitivity testing of fluid components is to monitor the amount of wear debris generated after each size exposure of contaminant. This can be done today using Ferrography. Standard contaminant sensitivity tests require test contaminant concentrations of 300 mg/l to yield measurable performance degradation values. Using Ferrography, only 10 - 20 mg/l concentrations have proved sufficiently sensitive and discriminatory on gear pumps to show even small changes in wear rates due to exposure to the standard size ranges of test contaminant. Debris generation rate has been correlated quite effectively with performance degradation in the case of gear pumps - Figure 11 illustrates both methods for two different pumps.

The assumptions that water in fluid systems simply vaporize, dissolve or emulsify and do no harm are entirely erroneous. The fact is that water can be extremely detrimental to the system long before it has a chance to "vaporize out," and the total elimination of dissolved water in oil is highly unlikely by the simple act of heating. Water-induced effects need investigation as much as any other contaminant of the system if high performance and reliability are to be achieved.

In most systems, there is very little chance of avoiding the entrance of moisture into the system. The U.S. Army studied water absorption tendencies of hydraulic fluids and found that fluids exposed for 30 days to an 80 percent relative humidity experienced an increase in water content as much as 804 percent.

The presence of water can grossly disrupt the "balance" within the chemical system of oil and interfere with the normal performance of additives through processes that currently are not fully understood. However, it is known that water reacts with zinc dialkyl dithiophosphate (ZDDP), an antiwear additive, and totally destroys its effectiveness. In fact, most investigators agree that ZDDP decomposes in the presence of water to form free sulfur or hydrogen sulfide. Thus, highly corrosive hydro-sulfurous acids can form that can destroy critical mating surfaces within the component when water is available.

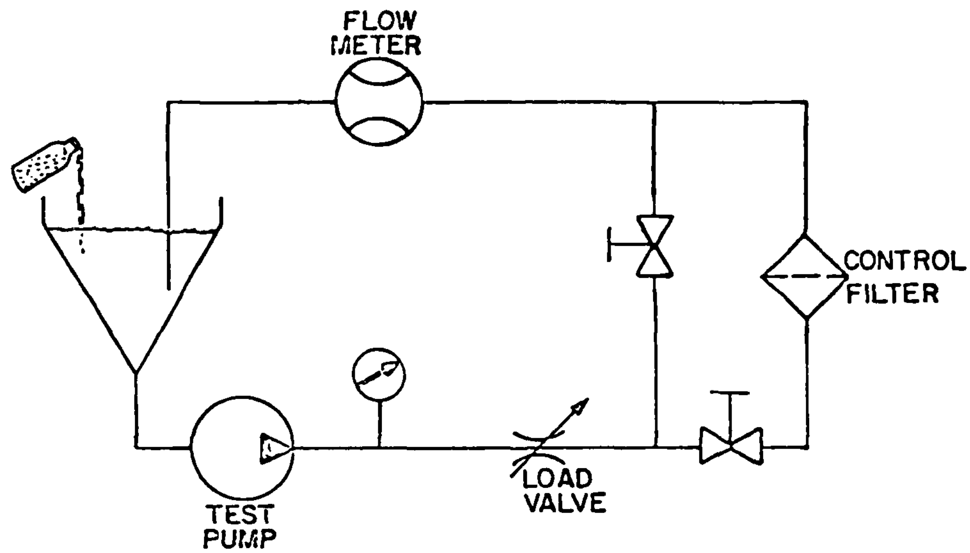

FIGURE 9 SCHEMATIC OF PUMP CONTAMINANT SENSITIVITY TEST CIRCUIT

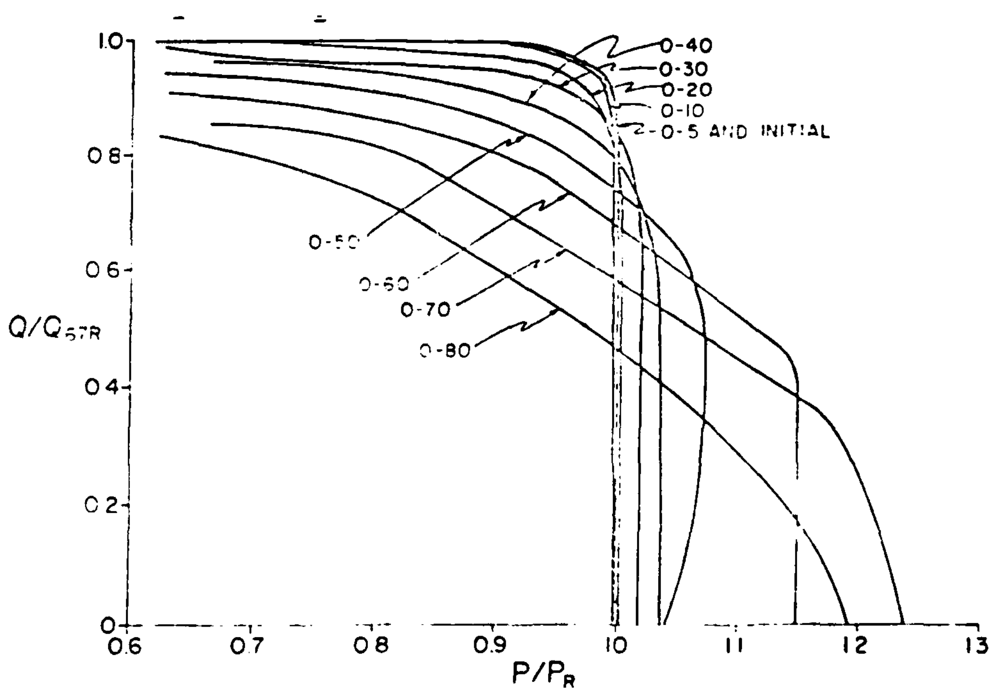

FIGURE 10 PRESSURE COMPENSATED PUMP CONTAMINANT TEST RESULTS

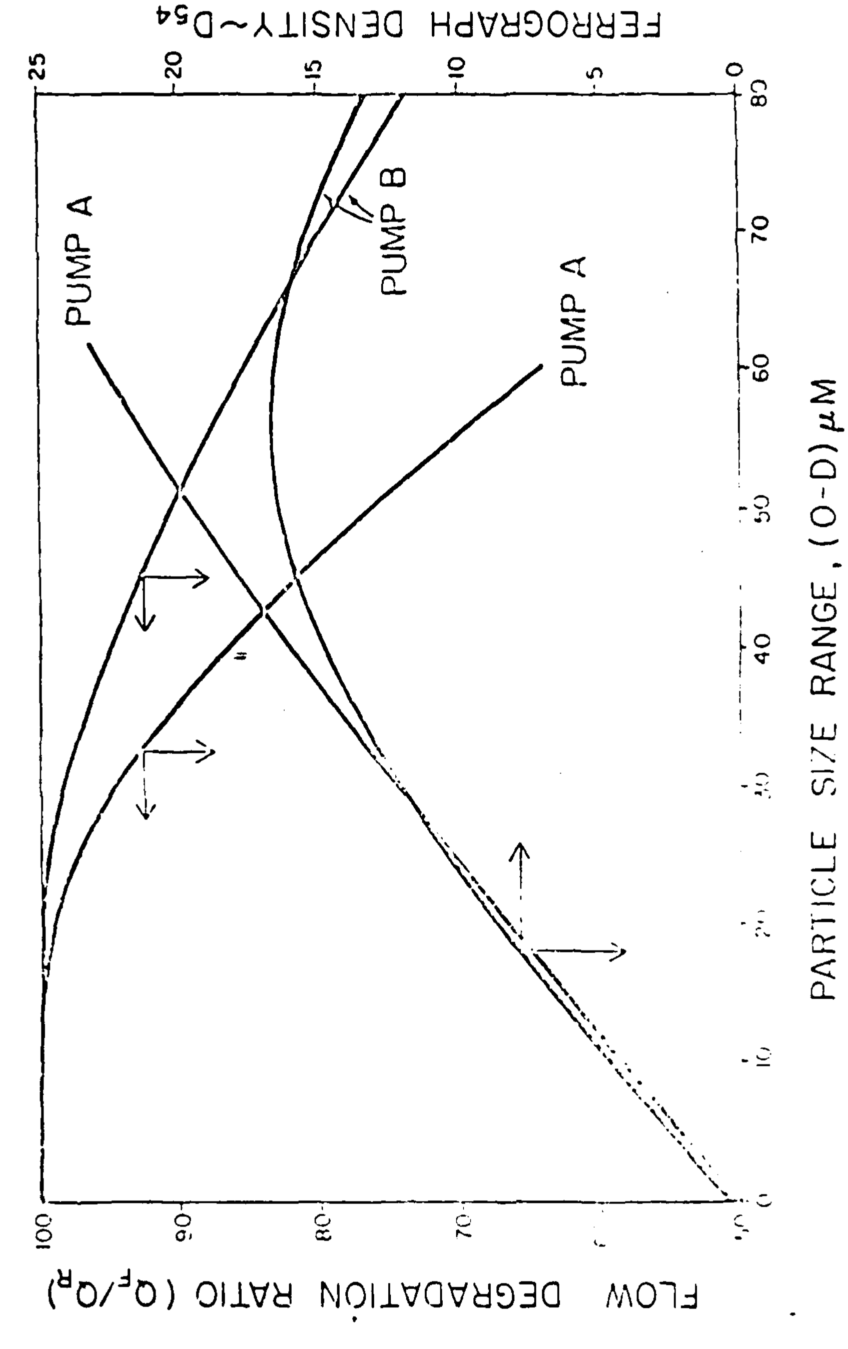

FIGURE 11 CONTAMINANT SENSITIVITY VERSUS PUMP WEAR DEBRIS

7. CONTROL/PREVENTION

In a fluid system, control over contamination is achieved and contaminant related failure is prevented simply by maintaining the contamination level of the fluid below the contaminant tolerance level of the components. If this balance is not possible, three basic options exist as follows:

- Increase the separation performance of the filter
- Decrease the contaminant ingression rate of the system
- Improve the contaminant tolerance of the components.

The separation performance of a filter depends upon three factors, namely;

- fluid residence time
- structural integrity of filter
- particle capture/retention capability of filter.

Fluid resistance time is equal to the circulating volume of the system divided by the flow rate through the filter. The lower the residence time, the higher the separation performance and the lower the contamination level. For any given filter, the lower the residence time, the shorter the time constant of the system - i.e., the faster the system responds in "cleaning-up" or reducing the fluid contamination level of the system. The more times that fluid is processed through the filter, the more opportunities that exist for the filter to remove the contaminating particles.

Structural integrity of a filter assembly is one of the most important factors in contamination control. If structural deficiencies exist in an assembly, all other "good" and desirable features that the assembly might possess can be totally overshadowed. Such deficiencies can result in unfiltered flow paths large enough to negate completely the intended function of the filter in the system.

To verify the absence of such deficiencies in the filter, five ISO approved standards exist to help assess the structural integrity of the filter elements:

- Fabrication Integrity (ISO 2942) - reveals defects in the element
- Collapse/Burst (ISO 2941) - shows ressitance of structure to differential pressure
- Material Compatibility (ISO 2943) - assesses deterioration tendency of element in hot system fluid
- End Load (ISO 3723) - shows axial compression resistance of element after Material Compatibility Test
- Flow Fatigue (ISO 3724) - reveals cyclic flow endurance of filter medium.

Other structural deficiencies in filters are common throughout the industry, and ISO procedures are not as yet available for their identification and assessment. These deficiencies include the following:

- Element to housing seal - sealing effectiveness under static and dynamic flow and mechanical vibrating conditions
- Bypass valve leakage - versus temperature, pressure differential, and mechanical vibration
- Silting susceptibility of bypass valve - under both open and closed conditions
- Cold soak bypass response - for filter element rupture prevention under cold startup conditions.

Filter media offer two types of capture/retention sites for particles entrained in the influent:

- Structural trap - where particles are held by mechanical means, like in a screen - called absorption
- Surface adhesion - where particles are held by surface forces, like electrostatic forces of van der Wall's attractive forces, called adsorption

Particles reach the capture site by action of one or more transport mechanisms, as illustrated in Figure 12.

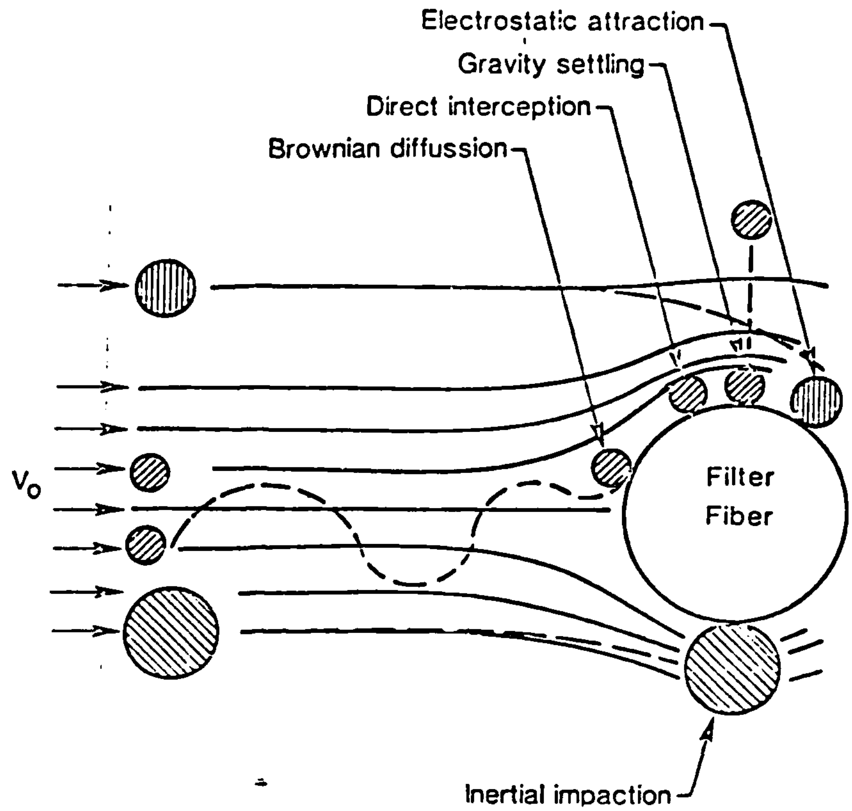

FIGURE 12 PARTICLE CAPTURE MECHANISMS

Evidence is mounting that particles retained by adsorption are mobile when perturbated flow or vibrating conditions occur. Hence, all particles below the sieve (pore) size of the media that are basically trapped by adsorption can become re-entrained by almost any dynamic action imposed on the filter, hydraulic or mechanical. For this reason, the trend toward silt control (small pore size) media is not only for the removal of such particles but to prevent "clouding" the fluid with silt size particles every time shock conditions occur. Clouding results in a highly visible concentration of sub-sieve size particles that were able to overcome the surface adhesion. This effect represents a dangerous situation to any fluid system.

The overall particle separation (absorption and adsorption) performance of a filter assembly can be assessed most accurately by conducting the ISO Multipass Filtration Test. This test (ISO 4572) is a static flow test so a discrepancy may exist between the rated performance (Beta Ten filtration ratio) and actual performance. An unsteady flow version of the standard multipass filter test has been developed and is ready for industrial perusal and promotion as a standard.

The ISO Multipass Filtration Test is restricted to the rating of filters which have Beta Ten values below 75 (because sample container cleanliness can start having a noticeable effect above the 75 value). Hence, a multipass test version for ultra-fine filters is being advanced and made ready for round-robin testing.

This test requires upstream and downstream in-line particle counting. Since AC Fine Test Dust will be used, the filtration ratio derived from the test can still be called "Beta." But, the rating size will probably be three instead of ten, and the upstream gravimetric level will probably be 3 mg/l rather than 10 mg/l. Instead of a Beta Ten rating system as ISO approved for "fine" filters, the "ultra-fine" filters will be rated by Beta Three values, if repeatability and reproducibility are both verified by the intra-lab testing.

The reader should realize that a Beta rating for a filter is tied to AC Fine Test Dust as the ingressed contaminant. If the particle size distribution of the ingressed contaminant in the field is not the same as that of AC Fine Test Dust, then a discrepancy in field performance prediction can exist. Fortunately, for those of us making predictions and unfortunately for those living with poor exclusion devices, field ingression distributions and the AC Fine Test Dust distribution are quite close, and the use of Beta in contamination control equations has worked very well up till now.

The future is not going to be as kind in this regard as the past has been. Contamination control managers are getting smart, controlling ingression sources much better, and altering the slope of field ingression. This means that, in order to make accurate performance predictions in the future, the Beta ratings (an accurate, well defined standard rating system which should continue) must be transformed mathematically into a performance oriented term which is independent of the ingressed contaminant.

The development of a unique size distribution rating for filter media has been the object of an ongoing research effort at the Fluid Power Research Center for several years. By ignoring surface forces as a particle retention mechanism, a unique rating can be derived. The accuracy of the new filtration model depends on the amount of adsorption which actually occurs in the system. Surge flow allows the model to predict particle separation performance quite accurately for any particle size distribution being ingressed. Work is continuing and an exchange of ideas between laboratories could be mutually advantageous.

One of the most perplexing and often depressing problems which commonly confronts machine system personnel is filter bypassing. Difficulty arises in pin-pointing the exact path of the sneak flow. It could result from a misalignment of the element in the housing, an unseated seal, a weak spring forcing the element against the sealing surface, a jostling action between the element and housing that repeatedly unseats the seal caused by the machine travelling over rough terrain, flow surges causing hydrostatic forces sufficiently great to drive the element off the seat, and "on-and-on." The effect of bypass is serious and, more often than not, will negate the otherwise good effects of the filter. Bypassing can be identified on the ISO Multipass Filtration Test and from fluid samples taken both upstream and downstream of the filter.

From what we have seen, it is important to locate filters where they are not subject to flow surges and severe mechanical vibrations, particularly low frequency, jerky movements. Hence, locations downstream from valves, motors, cylinders or anything else that contributes to perturbed flow should be avoided. Look for or create an ideal location where steady flow exists. Debris-catching filters downstream of pumps and "last chance" type filter screens ahead of critical components (servo-valves) make good engineering, maintenance, and economic sense.

All other factors remaining constant, the contamination level of the fluid is proportional to the contamination level of the environment. This is due to the fact that exclusion elements such as reservoir breathers and wiper seals for cylinder rods are proportional devices - that is, they let a certain proportion of each size particle pass through.

Environmental contaminant adhering to an exposed cylinder rod can be pumped into the confines of the cylinder and system when the rod retracts if the particles are not properly scraped off and rejected by effective wiper seal action. An effective wiper seal hugs a cylinder rod for an acceptable service life interval. Once a seal relaxes and allows environmental contaminant to ingress, the seal lip begins to wear, and ingestion continues at a an accelerating rate until the seal is worthless.

Wiper seal service life is a function of cycle distance traveled, environmental contamination level, wiper-seal sensitivity or ingressivity rating, and maximum level of acceptable system contaminant ingression. The SAE (J-1195) Wiper Seal Test establishes wiper seal ingressivity.

Fluid component contaminant tolerance can be improved when consideration is given to the type wear processes involved and contaminant properties contributing to the wear process. Contaminant entrained in the system fluid essentially goes wherever the fluid goes. If the fluid is used as a bearing lubricant, entrained contaminant has the opportunity to abrade or fatigue relative moving surfaces. Similarly, fluid exuding through annular leakage paths can become lodged and destroy the critical surfaces involved.

Another factor involved in the wear process is unit loading of mating parts. When particulate matter harder than the attendant surfaces separates them from each other, the severity of the abrasion depends on the normal force on the surface, the lateral movement of the surface, and the shear strength of the contaminant particles. Reduction in unit surface loading is an effective way to increase contaminant tolerances.

Whenever system debris lodges in valve clearances, the problem of stiction arises. Valves with high spool-positioning force levels are more tolerant of contaminant than those having low force capability. In fact, if valve spool forces are large enough, valve hysteresis reaches a limiting value above which additional contamination has little effect. Thus, valves with the smallest contact area between the spool land and its mating sleeve are least susceptible to contamination. By undercutting, or shortening valve lands not needed for metering and leakage control, more friction effects of contaminants are reduced.

Metering orifices are never intended to be used to shear particles. Therefore, when they become damaged, the valve operates erratically. Leakage of fluid resulting from such damage results in unequal pressures being exerted on the control valve and promotes actuator drifting. Particles intercepted by an orifice and collected within the flow passage result in a phenomenon called "silting," which again causes unequal pressures and actuator drift.

Particular attention should always be given to avoiding dead zones, where contamination can settle and cause direct interference damage and/or possibly become dislodged by flow surges or vibrations and slug critical downstream components. Contaminant tolerance of a valve can be improved by the following:

- Reducing spool/bore eccentricity
- Increasing the hardness of the spool/bore
- Reducing the asperity height of the mating surfaces.

8. SUMMARY

Contamination control is still not a science; but for those who have been following the progress of knowledge generation in this field, we are getting closer. Much research needs to be done. Hopefully, this discussion will spur some action in other parts of the world. Contamination control is a world problem, and we should all participate in formulating the science.

9. REFERENCE

Fitch, E.C., "An Encyclopedia of Fluid Contamination Control", Hemisphere Publishing Co., Washington, D.C., 1978, Revised 1980.

TRIBOLOGICAL FAILURES AND MECHANICAL DESIGN

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1. INTRODUCTION

Studies show that approximately 40 - 60 % of operating costs are maintenance related. This is illustrated in Figure 1 which shows the per flight hour costs of operating an A6E aircraft. It can be seen that almost 40% of the direct costs are maintenance related. A further analysis of the costs indicate that about 70% are unscheduled maintenance at the squadron level due to component failures or removal of components in anticipation of failure. These costs are only the direct cost of failures and do not include such items as stocking parts, purchase, down time, and loss of equipment. When these are added in, it can be seen that maintenance due to failures is expensive and serves no useful purpose if the failures can be avoided. In order to reduce these costs, a variety of different approaches to maintenance have been tried. These are listed in Table 1.

Table 1

ALTERNATIVE MAINTENANCE POLICIES TO REDUCE FAILURES

Preventive Maintenance	Condition Monitoring
Analytical Maintenance	Contract Maintenance
Failure-Free Design	Leasing
Modularization	Failure Warranties
On Condition Maintenance	

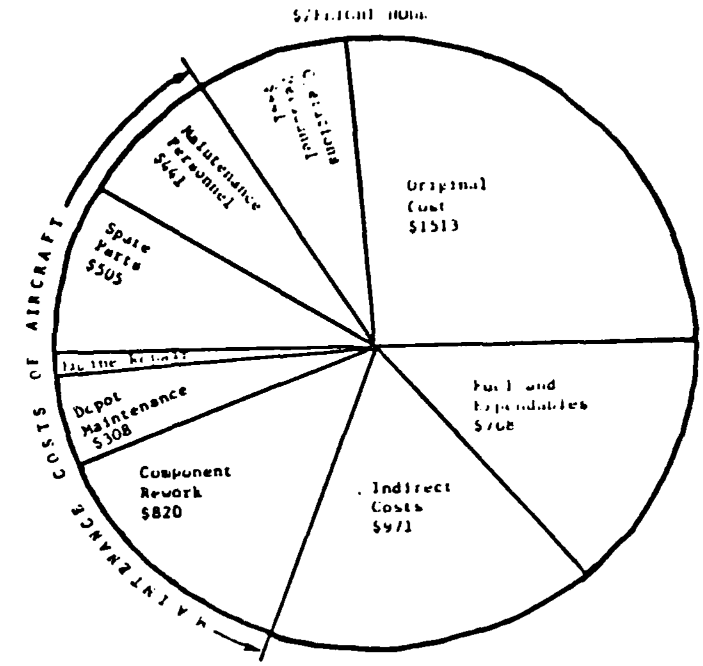


FIGURE 1 COST OF OWNERSHIP OF ONE A6E AIRCRAFT

Before the 1950's the preventive maintenance concept was promoted almost exclusively, the idea being that prevention of failures and equipment deterioration was always preferred to breakdowns and inadequate performance. However, as labor costs rose, and as technology produced better equipment, management began to question the need for certain maintenance actions. Furthermore, there was the impression that constant tinkering was in itself a significant cause for equipment malfunctions. The search for alternative maintenance policies has lead to a variety of approaches. These are described in the following paragraphs.

On-condition maintenance refers to performing maintenance only when required as dictated by machinery behavior. The difficulty with this approach was that a few catastrophic failures could eliminate any savings that resulted from elimination of preventive maintenance. Thus, attention was directed to methods of prediction of incipient failure. The expansion of this concept became "condition monitoring." By a variety of techniques (instrumentation, trend analysis, life projections, etc.) the condition of the equipment and its components was known and maintenance was performed only when the equipment deteriorated beyond prescribed limits.

During the same period of time there was a renewed interest in growth in contract maintenance. Essentially, the manufacturer of a given piece of equipment will perform maintenance on that equipment for a fixed fee. Such an approach fixes the costs of the user and provides certain benefits to the supplier such as service information, source of repeat sales, and for good equipment, an additional source of profit from a sale. The ultimate in contract maintenance is the lease where ownership is retained by the leasor who is responsible for the equipment. This approach does not reduce failures but rather stabilizes the cost.

Analytical maintenance refers to a maintenance program where service performance is recorded and the design modified to eliminate field failures. Such a program has as its basis a computerized maintenance actions system. By reviewing all maintenance actions and categorizing them, specific design, inspection, or repair problems become obvious. The essence of such a program is that the owner essentially accepts responsibility for the design. One of the results of the Analytical Maintenance Program has been the realization that much of the maintenance costs are not due to repair or replacement but rather due to removal and the need for general disassembly to reach the faulty component. As a result there has been increased use of modularization. Here, failures are accepted as inevitable and the technology was devoted to more rapid change. In some equipment like radar sets, it was estimated that this approach reduced costs by 50%. One problem with replaceable modules is that there is frequently an increase in "no defect" removals; any suggestion of a problem and the module is changed.

The understanding of service problems along with the growth in predicting reliability and life cycle costs has lead to an extension of the warranty idea. In this concept, the manufacturer guarantees a certain operating time between and/or a certain number of maintenance man-hours. The article could be as small as a valve or as large as a ship or aircraft. The advantage of this approach is that it forces the manufacturer to consider the durability as well as the performance of his design. To give such warranties it is, of course, necessary to be able to predict and control failures.

The conclusion which can be drawn from this chain of events is that there is great need to reduce service failures, and we are rapidly approaching an era where such failures will not be tolerated. Owners of equipment want maintenance-free, failure-free equipment and will exert their influence in the market place. As a result of this trend, component technologists must understand failures and learn how they can be prevented.

2. NATURE OF FAILURE PROCESS

If failures are to be prevented, it is necessary to understand the failure processes and how they occur in service. Some understanding may be gained from reference to Table 2. The designer starts with certain inputs: the requirements (function, cost, life); the conditions (operational and environmental); his past experience with similar components; and a knowledge of failure processes. From this he prepares several alternative designs to meet the requirements. Once a compatible design is chosen it is manufactured, assembled, put into service, maintained, and eventually removed from services. During this total life cycle, three different kinds of failures occur: inadequate design, poor quality control, or "conditional" changes.

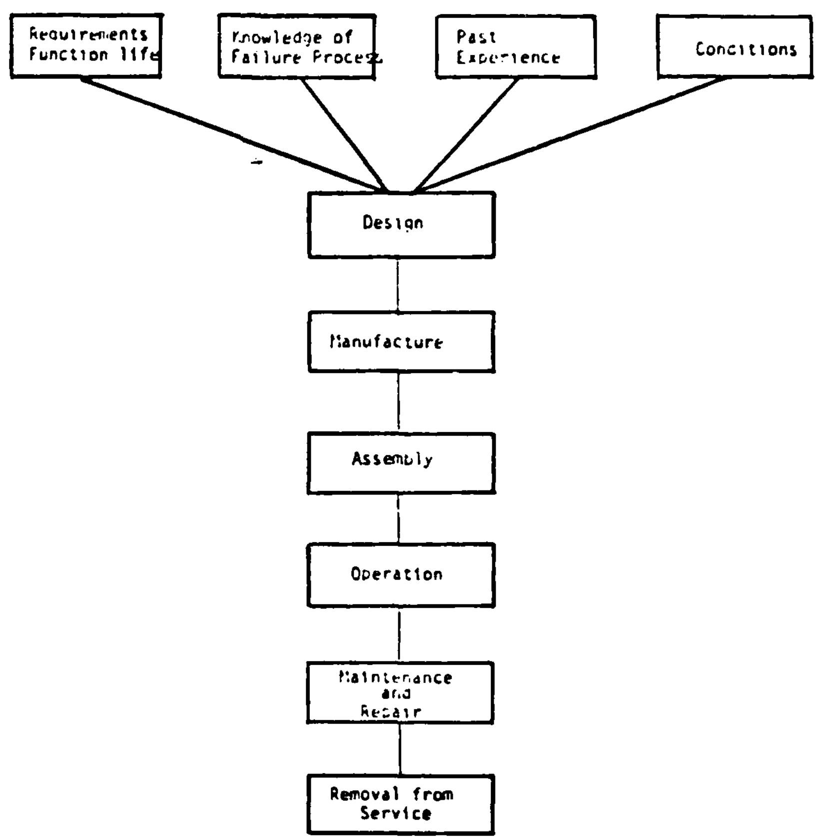
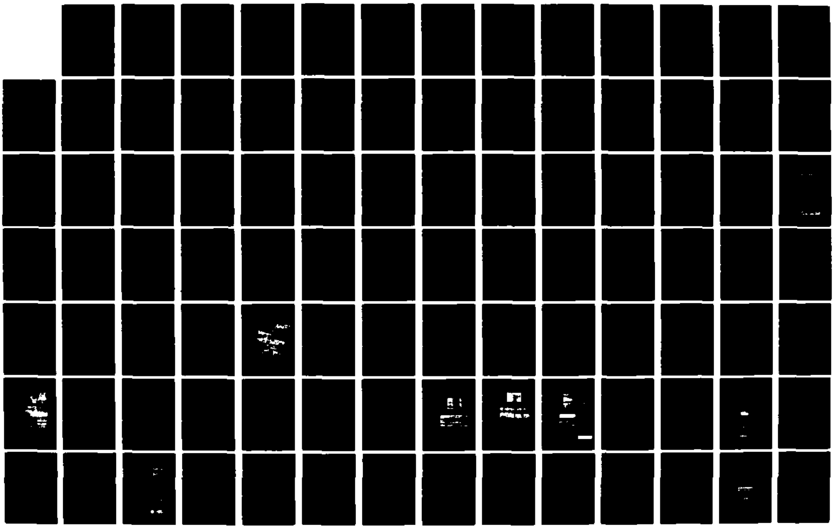
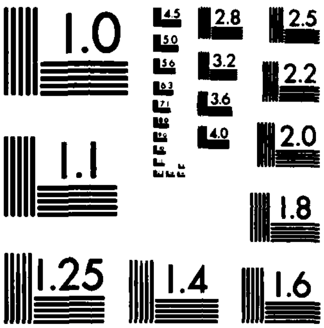


TABLE 2 EQUIPMENT DESIGN AND OPERATION

AD-A121 036 TRIBOLOGICAL TECHNOLOGY VOLUME II(U) MECHANICAL 2/4
TECHNOLOGY INC ANNAPOLIS MD P B SENHOLZI SEP 82
N00014-80-C-1736

UNCLASSIFIED F/G 13/8 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

Inadequate designs can be the result of mistakes in the design, however, they are usually the result of inadequate inputs; conditions are not exactly known, techniques are not available for life prediction, certain failure processes are not adequately understood and most important, specific resistance to failure of a given design cannot be estimated.

Once a design is selected which meets the requirements, is able to be manufacturedand is compatible with other components, it must be fabricated and assembled. This process can introduce a whole series of quality control problems. There are variations in materials, design tolerances, and assembly procedures which, in essence, change the design.

In operation, certain dissipative processes are initiated which will eventually determine the life of the component or part. At this stage of the life cycle, failures usually occur because the conditions have changed from those anticipated in the design. Some of the most significant kinds of changes are listed in Table 3. These changes, of course, may be due to operational mistakes or unanticipated changes like contamination.

Table 3

CONDITIONS RESPONSIBLE FOR FAILURES

Misalignment and Positioning	Contamination
Excessive Loads or Temperatures	Vibration
Material Degradation	Shock Loads
Inadequate Lubrication	Damage

This general view of the failure process is significant in that each different kind of failure can be prevented in a different manner as shown in Table 4. Design failures can be avoided by improved procedures or better information. In particular, the designer needs better information on the actual conditions encountered in service, how this affects component failure, and how designs can be modified to reduce these kinds of failures. If quality control is the problem, then tighter inspections and specifications are needed. If operation or environmental changes occur, then this situation must somehow be incorporated into the design; either the original design or by a design modification.

Table 4

APPROACHES TO FAILURE PREVENTION

Kind of Failure	Failure Prevention
Inadequate Designs	
Mistakes	Improved Design Procedures
Inadequate Inputs	Improved information on condition or failure prevention techniques.
Quality Control	Better inspections and specifications
Conditional Changes	
Mistakes	"Fool Proof" Designs
Unanticipated Conditions	Survey to define and modify the design to accept the condition.

If one is trying to reduce failures he could proceed on a broad front using all the listed approaches. However, it may be more effective to first determine the kinds of failure which most predominate in a given application and then to approach failure prevention in the most appropriate manner.

In order to determine which approach to apply to naval aircraft, a special investigation was conducted in order to identify the parts which fail most frequently and the respective type of malfunction[1]. These data from the Navy 3M system are shown in Table 5 for the A6 aircraft and Table 6 for the H46 aircraft. Each component on these lists was reviewed to determine the nature of its failure process. The results of this investigation are shown in Table 7. It can be seen that most of the aircraft failures are caused by changing conditions which are given secondary consideration in the design process. In other words, they are caused by one of the conditions listed in Table 3. These failure modes were, of course, known to the designer but he did not possess sufficient information for adequate control.

Work Unit Code	Identification	Total Hours	Total Incidences	Corroded	Broken	Cracked	Leaking	Stuck	Worn	Loose	Improper Lubrication
1121500	Wing Skin	12,778	1752	11,429	331	170	534	0	127	8	0
1113000	Misc Access Doors/Panels	11,122	2295	5,723	3,706	1,057	143	41	299	316	10
1113100	Forward Engine Access Door	9,491	1659	4,846	3,505	533	8	237	147	91	10
1113900	Access Panel	7,775	1646	4,080	2,071	762	72	6	290	162	18
1432100	Flaperon Actuator	5,724	139	2	19	7	5,622	60	10	7	4
148140C	Slat Assembly	4,895	533	3,559	595	346	7	54	103	4	3
1121200	Wing Outer Panel	4,591	296	2,880	159	703	591	0	100	1	0
1121100	Wing Inner Panel	4,558	378	3,991	27	87	434	2	0	0	3
1411200	Flaperon Skin	4,124	442	3,170	45	37	0	40	3	0	0
1118300	Boarding Ladder Assembly	3,972	879	2,331	774	501	6	27	783	19	6
1442100	Rudder Actuator	3,919	99	0	92	0	2,709	42	79	23	0
1452100	Stabilizer Actuator	3,496	83	16	62	0	2,272	1	20	0	0
1131700	Vert Stabilizer Skin	3,275	495	3,024	110	55	0	0	17	10	0
1116100	Ext Equipment Platform	2,494	530	1,127	872	180	15	43	123	79	4
1121400	Wing Leading Edge	2,578	407	1,778	79	60	70	2	744	21	0
1113500	Aft Engine Access Door	2,457	420	1,975	317	74	0	18	35	22	1
1113600	Tail Pipe Access Door	2,383	369	1,819	368	60	17	0	108	7	0
1311100	MLG Shock Strut	2,385	313	1,114	36	465	492	122	10	2	116
1311000	MLG Mechanical Components	2,365	291	1,144	217	77	47	162	99	12	578
1321100	NLG Shock Strut	2,559	167	1,464	9	3	897	4	16	4	147
1451100	Hor Stabilizer Cable	2,453	46	4	1,079	0	170	0	349	6	0

TABLE 5 A6 AIRCRAFT STRUCTURE 330/A/C/ 1 YEAR MANHOURS

Work Unit Code	Identification	Total Hours	Total Incidences	Corroded	Broken	Cracked	Leaking	Stuck	Worn	Loose	Improper Lubrication
1341800	Brake Assembly	4833	621	111	0	10	2260	0	2205	0	1
1521000	Fwd Rotary Wing Head	3253	214	333	171	0	2685	0	5	13	49
1321100	Nlg Shock Strut	2258	200	76	194	62	1523	89	122	105	0
1426700	Upper Boost Dual Act Cyl	2211	269	16	80	19	1547	47	213	119	0
1527000	Aft Rotary Wing Head	2008	181	253	102	1	1630	0	13	7	1
1421700	Cyl Stick Dual Boost Act	1790	205	5	10	132	1174	91	114	24	0
133110C	Wheel & Tire Assembly	1589	253	80	217	27	1523	0	174	0	5
1155000	Aft LH Clamshell	1468	207	49	309	231	0	6	658	113	0
2611000	Forward Transmission	1429	39	55	187	158	556	7	169	6	0
115540C	Fwd LH Clamshell	1252	176	103	398	277	0	1	427	51	0
1155800	Aft RH Clamshell	1189	288	55	588	172	0	1	249	115	0
1311100	MLG Shock Strut	1062	158	85	58	46	1051	1	113	75	0
115630C	Aft Fuselage Stec Assembly	1010	199	27	678	268	27	0	12	3	0
14288C	Dual Boost Actuator	1000	149	27	8	11	595	37	24	77	0

TABLE 6 H46 AIRCRAFT 300 A/C 1 YEAR MANHOURS

The same sort of analysis has been conducted for surface ships[2]. For corrective maintenance actions, 12,250 items were reviewed from 20 ships which had operated on the average of 3.7 years at a rate of about 2000 hours per year.

Table 7

COMMON AIRCRAFT FAILURES

Component	Failure Process
Wing Skin	Paint Damage due to Debris
Access Doors	Cracks and Paint Damage due to use of Door as Work Platform.
Panels	Structural Flexing Breaks Seal Between Panel and Fastener Causing Corrosion

Table 7
(cont.)

Component	Failure Process
Actuators and Struts	Dirt catches in seal -- Abrades rod causing leaking
Cables	Inadequate lubrication of cables
Wing Head	Seal leakage due to wear and distortion
Control Bearings	Corrosion due to water and salt in bearings

For illustrative purposes, the data for the propulsion system are presented. An analysis of these systems for the offending components is given in Table 8. It is interesting to note that of the 20 systems and hundreds of components, only a few dominate the maintenance expensives of the ship.

Table 8

PROPULSION SYSTEM REPAIR
17 SHIPS

	Reported Incidences
Air Start Motor (6)	25
Heads, Valves (96)	513
Injection Fuel Pumps	718
Injectors (96)	859
Fuel Booster Pump (6)	134
Salt Water Pump (6)	287
Governor (6)	66
Piston, Rings, Liners (96)	7
Turbocharger (6)	43

Table 8
(cont.)

	Reported Incidences
Fresh Water Pump (6)	17
Assessory Drive Gears (6)	7
Holset Coupling (6)	25
Lube Oil Pump (6)	2
Pedestal Bearing	15
Fawick Clutch	12
Reduction Gear (3)	16
Reduction Gear LO Pump (3)	15
Reduction Gear SW Pump (3)	9
Valves	8
Exhaust Leaks	37
Emergency Trip Wire	27

To be of assistance, the underlying cause of the malfunction must be stated, and costs and frequency attributed to those causes. This was accomplished by reviewing each maintenance item and attributing a cause to each, where possible. These resulting costs are listed in Table 9. The cost of the maintenance item was determined as the cost of material plus labor at a rate of $6.57/hour. This was the direct labor charge exclusive of overhead. If overhead is included, a figure about three times that amount should be used.

Table 9

COSTS OF CORRECTIVE MAINTENANCE

Wear	$1,420,513
Contamination giving corrosion	2,373,797
Leaks	505,590
Vibration	579,756
Corrosion	973,820
Broken	481,922
Contamination giving wear	3,674,622
Misalignment	282,482
Design faults giving wear	32,930
Vibration giving wear	33,549
Contamination control	565,939
Calibration	88,802
TOTAL	$11,013,722

In this table the phrase "contamination giving corrosion" means that system contamination leads to corrosion.

Preventative maintenance costs were obtained by reviewing each item in the planned maintenance schedule and again attributing the costs to independent categories. Here it is not a repair but usually a measurement to determine the need for repair. For example, a measurement of a piston ring to determine if it had to be replaced would be attributed to wear. The categories turn out to be somewhat different than that for corrective maintenance as shown in Table 10.

Table 10

PREVENTIVE MAINTENANCE COSTS

Wear	$2,125,420
Leaks	25,440
Vibration	71,800
Corrosion	214,900
Broken	4,080
Misalignment	40,840
Contamination control	1,511,380
Calibration	64,020
Check component operation	551,132
Lubricate	193,340
Record engine data	742,880
TOTAL	$5,545,232

The shipyard repair data was only available for 10 ships over a period of two years. This information had to be prorated to 20 ships over the 3.7 years in order to be compared with the previous data. These data are shown in Table 11.

Table 11

SHIPYARD REPAIR COSTS

Wear	$ 54,575
Leaks	4,440
Vibration	3,700
Broken	259,000

Table 11
(cont.)

Contamination giving wear		293,410
Contamination		11,100
Design fault giving wear		1,827,060
	TOTAL	$2,453,285

A summary of all costs is given in Table 12. From Table 12 it can be seen that a large amount of the total maintenance costs can be attributed to contamination, wear, and corrosion as shown in Table 13.

Table 12

SUMMARY OF COSTS

Wear	$3,600,508
Contamination giving corrosion	2,373,787
Leaks	535,470
Vibration	705,286
Corrosion	1,265,479
Broken	745,102
Contamination giving wear	3,968,032
Misalignment	323,322
Contamination control	2,088,419
Calibration	152,822
Design wear	1,859,990
Vibration giving wear	33,549
Testing component operation	551,132
Lubricate	193,340
Record Engine data	742,880

Table 13

COSTS PER OPERATING HOUR

	Cost	Cost/Ship Hour
Contamination	$8,430,238	$56.96
Wear	5,494,047	37.12
Corrosion	1,265,479	8.55
Broken	745,102	5.03
Vibration	705,286	4.77
Fuel Cost	----	75.00

The conclusions here are very much the same as that for the aircraft:

- A few components are responsible for the bulk of the failures.
- Most failures are caused by conditions which exist in these components which were not anticipated in design.
- Components are needed which are more resistant to conditions such as contamination, vibration, misalignmnet, etc.

Although the literature is not extensive these general conclusions also apply to other applications[3, 4, 5, 6].

3. FAILURE OF TRIBOLOGICAL COMPONENTS

3.1 Classification

If one attempts to classify failures, certain difficulties are encountered, as illustrated in Table 14 which lists causes of sliding bearing failures. The difficulty is that the list contains a variety of unequal elements which do not lend themselves to a rational approach to failure prevention. Some are causes, others are the result of the failure process. Some may cause failure in certain bearings but not in others. Wear is a process characteristic of all bearings and may or may not lead to a dimension change of significant magnitude to interfere with the function of the part. In order to better understand failure processes, it is sometimes convenient to break the failure process into the categories shown in Table 15. There are sources, conditions, dissipative processes, characteristic component failure modes, observable results, and reasons for replacement and repair. These categories are illustrated using the example of the bushing. The failure process begins with severe misalignment due to structural deformation of a base plate and eventually leads to seizure. It should be noted that all categories need not be different. The observed result is an inoperable bearing and that is the reason for replacement. However, under the same circumstances, siezure or welding may not have occurred, only a large increase in friction. The observable result may have been "noise" and the reason for replacement might be "anticipation of failure" if high friction was tolerable.

Table 14

CAUSES OF BEARING FAILURE

Oil passages plugged	Misalignment
System low on oil	Shock leads
Poor surface finish	Poor materials
Lubricant Degradation	Corrosion
Lubricant Deterioration	Fatigue
Excessive temperature	Wear
Differential thermal expansion	Improper crush
Excessive sliding	Scoring
Poor thermal conductivity	Dirt
Improper dimensions	Improper viscosity
Dimensional instability	

Table 15

FAILURE PROCESS CATEGORIES

Category	Bushing Example
Source	Misaligned Shaft due to Structural Distortion
Condition	Restricted Area of Contact High Pressure/High Temperature
Dissipative Process	Thermal Softening
Characteristic Failure Mode	Seizure (welding)
Observable Result	Inoperable
Reason for Removal	Inoperable

The important point here is not the uniqueness of the categories but approaches to prevention. Causes may be important in legal situations, but they are too varied to be of much use in failure prevention. In the bushing example cited, the original cause may have been the settling of the earth but this is of little help in prevention. For prevention of failures, it is important to know the conditions and dissipative processes which can lead to failure and the characteristic failure modes of the various components. If these are known then these conditions may be avoided or the components may be modified to prevent the characteristic failure modes.

A large amount of literature was reviewed on the tribological components in order to understand the failure processes and to suggest means of failure prevention. A collation of data leads to the results shown in Table 16, which now refers only to the tribological components. For each component a given condition will aggrevate one of the dissipative processes which will lead to one of the failure modes (DIRT ———> CUTTING ———> WORN), (WATER ———> CORROSION ———> SURFACE DAMAGE), etc.

Table 16

TRIBOLOGICAL FAILURES

Conditions Leading to Failure	Dissipative Processes	Component Failure Modes
Dirt/Contamination	Stress Cycling	Inadequate Friction
Inadequate Lube Supply	Heating & Heat Cycling	Instability
Shock Loads	Plastic Flow	Distortion/ Deformation

Table 16
(cont.)

Conditions Leading to Failure	Dissipative Processes	Component Failure Modes
Water in Lubricant	Fatigue (Crack propagation)	Fractured/ Broken
High Temperatures	Adhesion and Transfer	Surface Damage
Inadequate Film Thickness	Cutting/Tearing	Worn
Overload/Speed	Material Diffusion	Seizure
Vibration	Corrosion	
Improper fit		
Inadequate Materials		
Inadequate Lubricant		
Misalignment		
Improper Mounting		

The different characteristic failure modes are related to specific components in Table 17. The terms used in a given category are customary names which are used to describe a given kind of failure. For example, "fade" is used to describe inadequate friction characteristics of brake materials. Thus it can be seen that in the tribology area there are seven major types of failure which occur. The real question is whether these can be appropriately linked to specific conditions and whether appropriate prevention techniques can be identified. This is given more detailed consideration in the following sections where specific components are considered.

	Inadequate Friction	Instabilities	Distortion Deformation	Fracture	Surface Damage	Seizure	Worn
Bushing	Excessive Torque	Stick Slip	Plastic Flow	-		Galling Sliding Seizure	Shaft or Bushing Dimension Chnge
Seals	High Friction	Whirl	Creep Coining Distortion	-	Tearing Deposits	-	Seal Wear or Shaft Grooving
Rolling Contact Bearings		-	Ball Indentation	Cage or Race Fracture	Spalling Pitting	Smearing	Cage Wear Roller Wear Race Wear Land Wear
Brakes	Fade	Squeal	-	-	Heat Check		Wear
Gears	-	-	Surface Rippling	Tooth Breakage Case Crush		Scoring	Tooth Wear
Splines	-	-	-	-	-	Galling	Tooth Wear
Cables	-	-		Strand Breakage	-	-	-
Fits			-			Galling	Fretting
Fasteners	Over Torque	-		FRACTURE	-		WEAR
O-Rings			COLD FLOW				WEAR

TABLE 17 COMPONENT FAILURE MODES

3.2 Dry Bushings[6 - 15]

Dry bushings refer to a shaft and a journal where fluid lubricants are not used. There are several kinds of bushings, the most common being polymers, solid lubricant filled polymers, and static metal bushings. Bushings are widely used in small devices (pumps, fans, valves, etc.) where speeds and loads are low and a large number of cycles are not accumulated during the lifetime of the part.

A failure chart for the polymers and filled polymers is given in Table 18. The most common failure modes are distortion, seizure, and wear. Since these materials are much softer than the shaft material they are very sensitive to load and temperature effects. Although Table 18 appears to be quite complicated, it merely says that the common failure processes are:

- Bearing deformation due to high ambient temperatures or shock loads.
- Seizure due to inadequate clearances in design.
- Wear due to contamination, operation at high load and speeds, or at high pressures (low contact areas); wear due to high temperature operation, or use of a poor shaft finish.

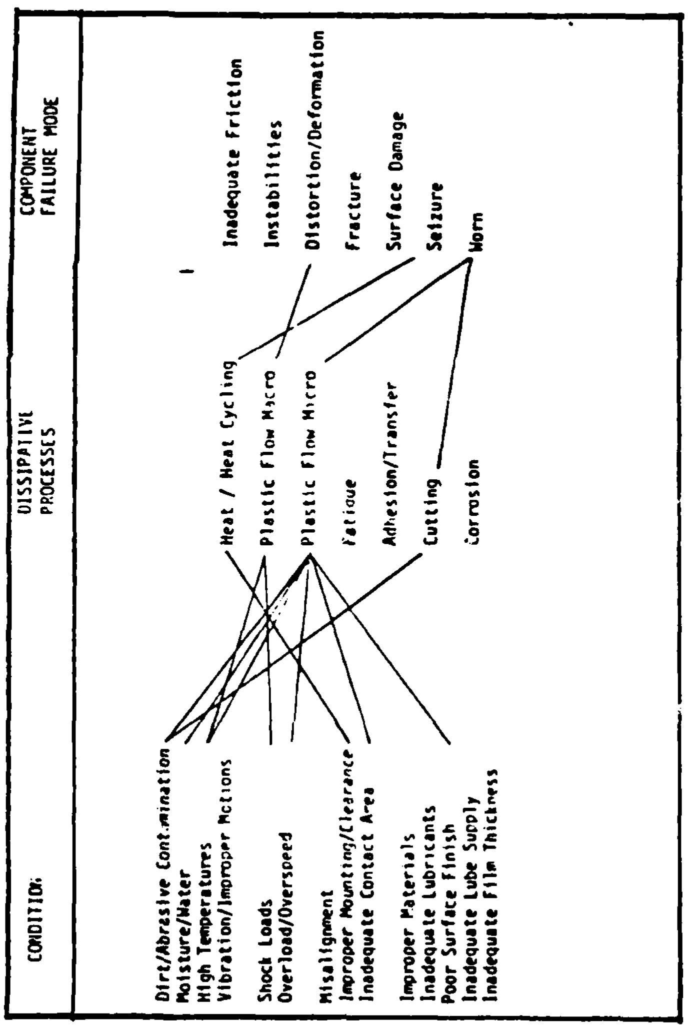


TABLE 18 DRY BUSHING FAILURE MODES (POLYMERS & FILLED POLYMERS)

The most common approach to failure prevention is to change materials. A wide variety of materials are available which operate under a variety of conditions and stress levels[16]. Even with a given material like teflon, a variety of fillers are available for different stress and temperature levels. Usually, it is not the design loads and ambient temperatures that are at fault. Rather, it is the shock loads not anticipated in the design and the unexpected temperatures of operation which create a problem. Materials are available for temperatures to 600°F and pressures to 1000 psi (depending upon the desired life).

Seizure is strictly a design problem. Due to unequal thermal expansion, the metal shaft may grow at a faster rate than the bearing material so at a certain temperature the clearance is reduced to zero and seizure may occur. Prevention of this kind of failure is simple; clearances should be adjusted to allow for thermal expansion.

Excessive wear can be attributed to a variety of conditions. Wear is known to increase rapidly at a given interface temperature[17]. The interface temperature is a function of operating conditions (load, speed, contact area) and the bearing ambient temperature. To prevent excessive wear, the interface temperature must be lowered. This, of course, can be accomplished by reducing the load or speed and/or increasing the contact area. However, these adjustments are often not possible, in which case the materials must be changed. Almost all manufacturers provide data on the operating limits (pressure and velocity) for their respective materials. Although these data are obtained under limited conditions, they do provide a good guideline for selecting bushing materials. Excessive wear based upon this condition is a design fault. If changing materials is difficult, cooling may be appropriate. The results presented in Reference 18, show that considerable convective cooling will result with small amounts of air flow across a bearing. This is not difficult to accomplish since it usually means redirecting the natural air flow about the moving shaft. Other means of cooling also exist such as hollow shafts and materials with better thermal conductivities.

Since bushing materials are soft, they are sensitive to shaft roughness. Designers usually tend to use rough surfaces since it is less expensive, however, in order to minimize wear, finishes should be in the range of 10 to 20 RMS. Improving the surface finish is a simple design modification and should be investigated first.

Wear due to contamination, either abrasive particles or water, is very frequent. Elimination of the contamination (by sealing) is difficult since the purpose of using dry bushings is design simplification. However, some approaches are available based upon the literature which was accumulated. Grooves cut in the surface of the bearing collect the abrasive particles and reduce the wear[19]. Reducing the clearance or extending the bushing length prevents abrasives from entering the bearing[20]. Putting a teflon coating on the surface of a hinge pin also acted as a seal[21]. This can often be accomplished relatively easily.

Water or other fluid contaminants are known to increase wear. The low wear of many plastic and filled plastic materials is contingent upon the transfer of a thin film to the shaft surface. This transfer film reduces the effective surface roughness and thus reduces wear. If water or other contaminants prevents this film from forming then wear will be higher in magnitude, being a function of the surface roughness. Wear can also be reduced by coating the shaft with the bushing material prior to operation.

Thus, the literature provides a variety of techniques to reduce failures depending upon the nature of the failure process. These are summarized in Table 19.

Table 19

DRY BUSHINGS
APPROACHES TO FAILURE PREVENTION

Component Failure Mode	Failure Prevention
Distortion/ Deformation	Change to higher Temperature or higher strength materials.
Seizure	Check bushing clearance Increase clearance

Table 19
(cont.)

Component Failure Mode	Failure Prevention
Wear	Change to higher temperature material
	Cool Surfaces
	Improve surface roughness
	Use 10 - 20 RMS
	Coat shaft
	Reduce Clearance
	Increase Bearing Length

3.3 Fluid Lubricated Bearings/Bushings[22 - 27]

Fluid lubricated journal bearings are of two varieties: boundary lubricated and fluid film lubricated. Although there is no clear distinction, the fluid film bearings are designed to operate with a given lubricant film thickness while boundary lubricated bearings may or may not, depending upon the operating conditions and the materials. Boundary lubricated bearings are usually made from bronzes, copper alloys, steels, cast iron, plastics, or ceramics, although any material may be used.

The failure chart for boundary bearings/bushings is shown in Table 20. Basically, the same failure processses apply as for dry bushings except that this bearing type depends upon the presence of a lubricant at the interface. If this lubricant is not present, failure is almost inevitable. The lubricant provides the protection against adhesive processes, thus compromises have been made in that property of the materials. Consequently, any condition which tends to remove the lubricant will tend to increase the probability of failure. These conditions include: high temperatures, excess loads and speeds, severe misalignment, inadequate contact area, and insufficient or interrupted lubricant supply. This increased adhesion can either lead to seizure or to inadequate friction, depending upon the requirements of the bearing.

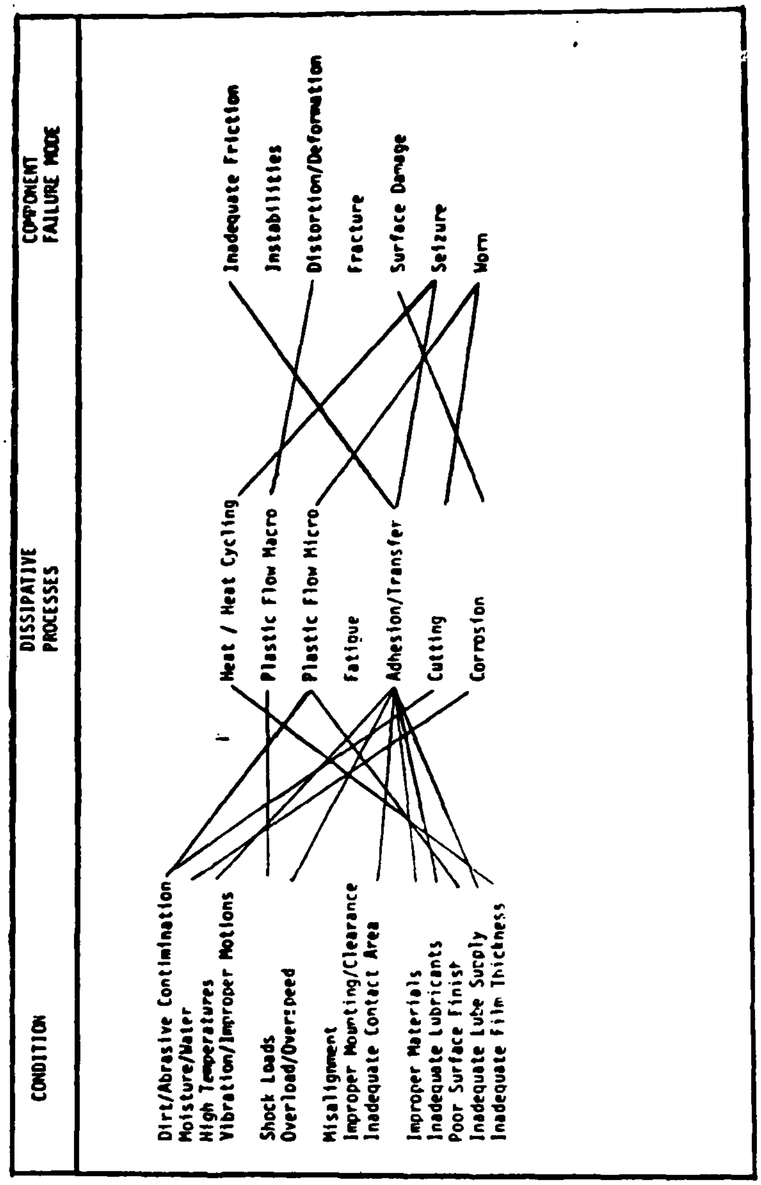


TABLE 20 FLUID LUBRICATED BUSHINGS (BOUNDARY LUBRICATED)

Failure prevention techniques can concentrate on removing the condition or reducing adhesion as shown in Table 21. A detailed investigation of bearing failures shows that most of the problems arise from contamination which increases the wear and reduces the lubricant flow to the bearing. A second factor is neglect. Many of the lubricant points are almost inaccessible and thus do not receive relubrication. The problem in this case is not adhesion but lubricant supply and distribution, debris removal, and sealing. The minimization of this factor will take a special bearing design with grooving and an internal reservoir. Simple designs of such bearings are available.

Table 21

APPROACHES TO PREVENTING SEIZURES

Reducing Adhesion

Use Nonsoluble Materials
Increase Shaft or Bearing Material Hardness
Add Low Adhesion Coatings
Use Lubricant with Improved Additives
Use Less Ductile Materials

Eliminate Condition

Increased Cooling or Heat Removal
Grooving for Better Lubricant Distribution
Use Positive Lubricant Supply System
Increase Contact Area

3.4 Seals[23 - 45]

Seals may be divided into two main categories: static seals such as gaskets and O-rings, and dynamic seals which have a sliding interface. Dynamic seals may be face seals, - circumferential seals, or lip seals. Lip seals are given primary consideration because of their preponderance in aircraft structures and because an anlysis indicates that they are one of the main causes of tribological maintenance. The technology of lip seals has advanced rapidly in the past 20 years. There is now a much better understanding of their mechanism of sealing and the factors which control their life. This new understanding has produced new designs and design guides[46, 47]. This new technology provides a variety of means to prevent and correct service failures. In addition, a variety of new elastomeric materials have been adapted to seal usage which greatly expands their range of operating usefulness. A seal has become a precise engineered component, a fact not yet recognized by maintenance and manufacturing personnel. As a result, seals are frequently damaged before they are used. They should be treated with the same respect as is given rolling element bearings. Such care would reduce failures dramatically. The seal interface consists of a thin lubricant film approximately .0001" in thickness. This film thickness must be maintained if a satisfactory life is to be expected. If the film becomes too thick, leakage will occur. If it becomes too thin, the interface will be inadequately lubricated and surface damage and increased wear will occur. This, too, will result in increased leakage. The factors which control the film thickness are: load, velocity, temperature, alignment, shaft roughness, and the supply of lubricant. Factors which affect its change are: misalignments, wear or surface damage, and changes in the material properties due to environmental effects. These factors are illustrated in the failure chart shown in Table 22. Leakage is not, of course, the only technical reason why a seal is replaced. Some removals result from noise, vibration, and convenience. Seals are often replaced whether they are defective or not, in order to assure a failure-free duty cycle.

Leakage is caused by a variety of conditions as illustrated in Table 22. Many of these conditions (high ambient temperatures, overloads or overspeeds, inadequate lubricants, poor lubricant supply and inadequate film thickness) lead to plastic deformation damage to the seal interface which results in leakage. Many of these same conditions lead to overheating at the interface which causes the seal to distort, again causing leakage.

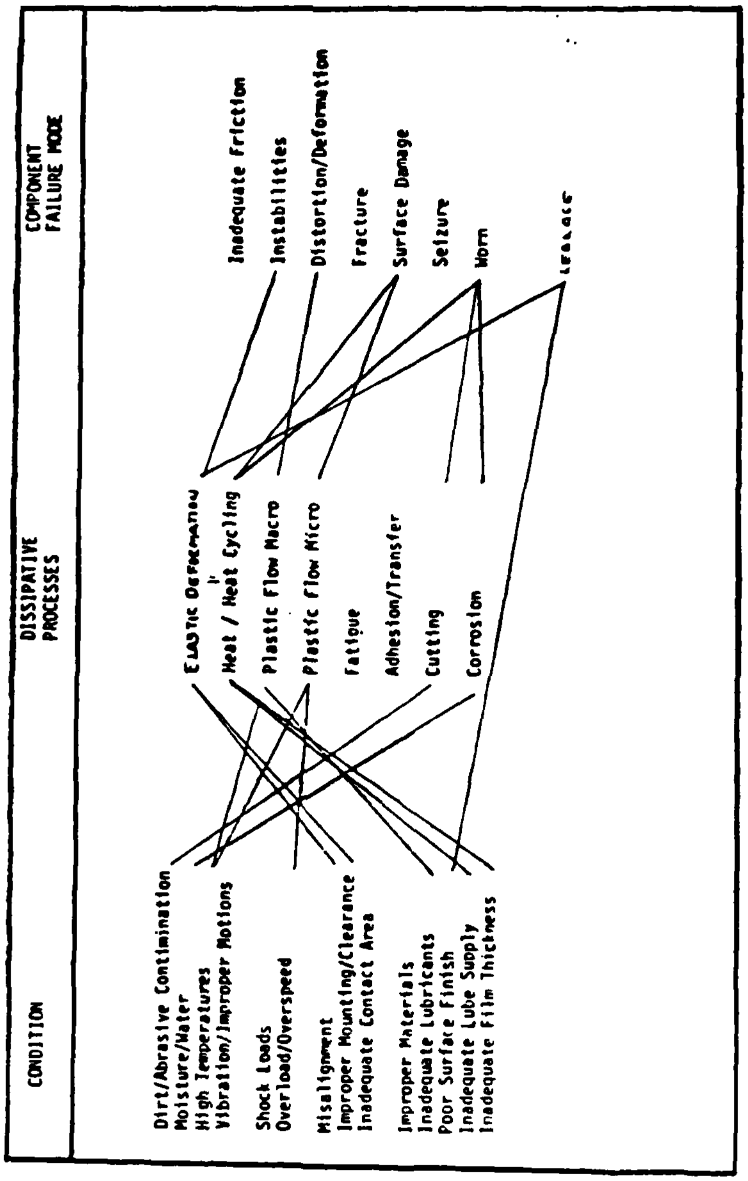


TABLE 22 SEAL FAILURE MODES

A second major area of failure is leakage caused by seal elastic deformations which exceed the material compliance and present a direct leakage path. This most frequently happens at high speeds when the seal is unable to follow shaft movements. Misalignments may be caused by shaft "out of roundness," nonparallel center lines of shaft and seal, or eccentricity.

Another major failure area is caused by contamination, either abrasive dirt particles, moisture, or oxidized or decomposed lubricant. These conditions lead to increased seal wear. Certain conditions (misalignment, finish) can in themselves produce leakage if they become excessive.

Primary failure modes can then be classified as wear and surface damage, distortions of the required seal geometry, and dynamic instabilities which produce leakage. Based upon this, some approaches to failure prevention are shown in Table 23.

Table 23

APPROACHES TO PREVENTION OF SEAL FAILURES

Wear and Surface Damage	Add wear sleeves or coatings of more wear or corrosion resistant materials. Improve seal/shaft alignments. Improve lubricant supply to interface cooling. Hydrodynamic lip seals. Use fluoroelastometer seal.
Seal Deformation	Improve seal/shaft alignments.
Instabilities	Improve seal/shaft alignments. Improve lubricant distribution. Clean interface.

3.5 Rolling Contact Bearings[48 - 69]

A properly applied, well lubricated, properly installed rolling contact bearing operating in a contamination free environment, can have a life of greater than 10 years in continuous operation. Yet, in service it is one of the most frequently replaced parts. This is usually not a fault of the design but of environmental factors such as dirt, corrosion, lack of adequate lubrication, and excess temperatures. Although these can be avoided in the design, they are often overlooked or unspecified. Designs are based upon static or dynamic load capacity. The static capacity is the bearing load necessary to yield a given indentation in the race. The dynamic load capacity is the operating life at a given load based upon bearing fatigue. This life is determined by running a large number of bearings to failure at a given load and then extrapolating through a range of loads. It is found that these loads are conservative for other failure modes such as wear or fracture. Unfortunately, most bearings do not operate under the ideal conditions of laboratory tests and much shorter lives are found in the service environment.

The failure modes for rolling contact bearings are listed in Table 24. Most of the common failure modes are exhibited by rolling contact bearings. In contamination free bearings, failure is most common in bearings which have appreciable sliding or with those which have very thin lubricant films. Thin films are characteristic of low speed, high load bearings.

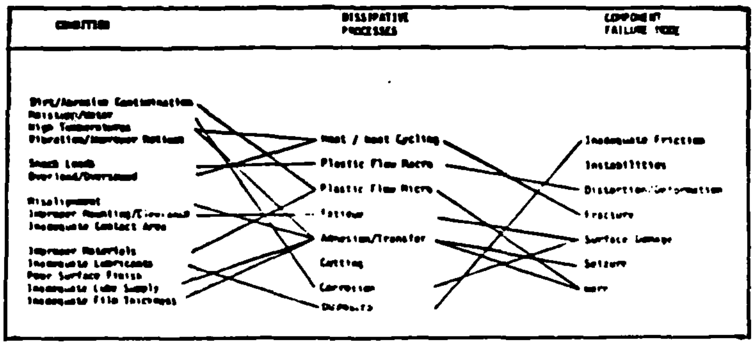


TABLE 24 FAILURE MODES - ROLLING CONTACT BEARINGS

Inadequate friction is a special case for instrument bearings which require constant torque. The bearings removed from service would be considered satisfactory for most other applications. The friction variations are usually due to small amounts of deposits in the contact area or small variations in concentricity. Bearing deformations are usually the result of shock loads which cause indentations. Fracture of the race can also result from shock loads or from thermal expansion of the bearing or the housing due to high temperatures. Actually, fracture of the cage is more common than fracture of the race and usually takes place after considerable cage wear. Cage fractures result from centrifugal stresses, misalignment, or accelerating forces on the cage due to uneven ball loads. There are very few ways to prevent these stress failures except to over design the bearing, that is, to use a much larger bearing than the application requires. Such failures are often isolated cases attributable to quality control in the bearing. If they are the fault of the machine, then the condition should be eliminated.

Seizure failures are most often due to insufficient lubrication. They are usually referred to as "smearing," a term which describes the surface appearance of the bearing. "Smearing" failures can also be the result of high temperatures in the contact area. At high temperatures the lubricating capabilities of the lubricant are destroyed and the effect is the same as having insufficient lubricant. In the failure process, high friction first occurs; there is a transition from fluid film friction ($f < .01$) to boundary friction ($f <= .10$) and then to dry friction ($f => .40$). This increase in friction brings about a forty fold temperature rise in a small contact area, causing the failure area to enlarge. This process continues until a definite event occurs which reduces the friction or the surfaces are damaged. This damage can result in seizure with certain materials and wear with others. The cage is often more sensitive to this type of failure and is the first to be influenced. Coatings are often used to prevent seizure failures. In most cases they provide auxiliary lubrication to compensate for the depleted fluid. In other cases, the coatings will extend the operating temperature range before lubricant failures occur. This is particularly true for soft metal platings (silver, gold, etc.) which provide improved heat conduction and flow, reduced surface roughness, and thinner effective lubricant films. Coatings could also be used for another purpose. If the coating melts it could provide the friction reducing event required to reverse the failure process. This approach has not been used in practice but is worthy of consideration.

Surface damage is the most common bearing failure mode. The specific kinds of surface damage are spalling due to fatigue, and pitting due to localized surface fatigue, corrosion, or debris indentation. If the bearing is not removed because of the pitting, continued operation will eventually lead to its removal because of either spalling or excessive wear. A survey of the literature indicates that a large number of studies have been directed at the understanding of the variables influencing fatigue damage.

Changing these variables appropriately will increase the fatigue life as shown in Table 25. Several important prevention techniques can be noted. For example, more frequent lubrication insures a cleaner lubricant and increased fatigue life. Longer fatigue life can also be obtained by use of improved materials. Precise correlations with specific material properties are difficult because different materials are processed differently. However, small increases in material hardness are known to greatly increase fatigue life. Certain materials like the tool steels (M50) are usually substituted for increased life, however, the increased life may be more related to their processing than to the material. Recent evidence has shown that all surface distress can be drastically reduced if the effective film thickness is increased. Variables which increase film thickness are reduced surface roughness, increased viscosity, reduced temperatures, lower loads, increased speeds, and larger contact areas.

Table 25

FACTORS INFLUENCING FATIGUE LIFE

Variable	Effect on Fatigue Life
Load	Varies $\frac{1}{L3}$
Velocity	Depends upon velocity effect on temperature, film thickness and centrifugal loading
Temperature	Reduce life by decreasing viscosity
Water in Lubricant	Reduces fatigue life
Lubricant film thickness-	Thicker film increases life
Viscosity	Higher viscosity increase life
Abrasives in lubricant	Reduces fatigue life
Reactive lubricants or or additives	Reduces fatigue life
Solid lubricant additives	Increases fatigue life

Table 25
(cont.)

Variable	Effect on Fatigue Life
Materials	Specific for material and processes
Hardness	Higher hardness increases life
Roughness	Increased roughness decreases life
Surface damage	Reduces fatigue life
Stress Concentrations	Reduces fatigue life
Residual compressive stresses	Increases life

Wear is a common failure mode only in those bearings which experience appreciable sliding. This wear is usually associated with the cage since it rubs against the ball and the race land. Improved materials and appropriate application of solid lubricants can eliminate many cage wear problems[70, 71]. Coatings are also used. Race wear is almost always due to dirt or corrosive materials in the lubricant. If the conditions are not appropriate to accelerate fatigue damage, then excessive wear will result.

4. SUMMARY OF RESULTS

In the past 10 years there has been considerable reevaluation of maintenance policies with emphasis on reduced maintenance while still requiring high levels of reliability, availability, and durability. To achieve this, several different approaches to maintenance are being considered. Condition monitoring avoids preventive maintenance and many inspections and performs maintenance only after some indication is given that it is required. This "indication" may be a sensor which monitors a given condition, service data which indicates that a particular maintenance task is required, or some analytical life prediction scheme. Each of these techniques are currently in use to some degree with instrumentated receiving the greatest emphasis. Contract maintenance, more characteristic of commmercial markets, transfers maintenance responsibilities to another who presumably can achieve economies of scale and specialize in a particular technology or system. Very often the maintenance is performed by the equipment manufacturer. This is a particularly useful arrangement since it provides an immediate source of information for the manufacturer and provides economic incentives to reduce maintenance. Adequate engineering is also available to make the required design changes. Failure warranties are similar to contract maintenance except the actual maintenance is performed by the equipment user. However, the manufacturer is responsible for maintenance costs. In these cases, the manufacturer and the user agree on what is to be warranted, how failures are defined, and the cost.

Whatever the approach, reduced failures are desirable since the user ultimately pays the service costs. The critical element of a maintenance policy is to completely understand the nature of the service problems encountered. Once such information is available the most cost effective appraoch to meet maintenance objectives can be devised. This may be the introduction of new technology (maintenanace technology) or the use of more conventional approaches. With condition monitoring it is only necessary to monitor these critical problem areas rather than all possible failure modes. The literature suggests that condition monitoring has been successful in these instances. Where it has been unsuccessful is where it has been too broadly based. An understanding of the failure modes also allows a better understanding of where warranties are required and where they will be cost effective. Thus it can be concluded that new approaches to failure control should be investigated.

This chapter classifies failures using a variety of categories. The source of failures were classified as inadequate designs, poor quality control, or changes in operating conditions. Most failures were found to result from unexpected changes in operating conditions; for example, a bearing designed for 200°F actually operates at 300°F. The changing conditions most responsible for failure were factors such as contamination, misalignments, vibration, and inadequate lubrication supply techniques. Alternative component designs are required which will remove these conditions or provide tribological components which are more tolerant to these conditions. A review of the literature indicates that sufficient technology is available to provide failure-free components, however, data are not available to predict the life benefits so gained. Such data are needed to justify the costs of change. For example, is a higher cost, longer life bearing more cost effective than the more frequent replacement of a lower cost one.

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TRIBO-TESTING

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1. INTRODUCTION

Tribo-testing is a critical facet of tribological technology. A key element in the machinery wear integrity optimization process is the existence of viable wear test techniques and approaches. This tribo-testing element will be discussed in this chapter with respect to extrapolation and simulation.

2. EXTRAPOLATION IN TRIBOLOGY

The difficulties encountered in attempting to extrapolate friction and wear data, obtained on laboratory rigs, to industrial problems is discussed under this section in the light of the Three Body model. Third body (or intermediate film) rheology along with explicit transverse and longitudinal boundary conditions are shown to be necessary before extrapolation can be expected. Various domains of tribology (thick film lubrication, solid lubricants, dry bearings, etc...) are explored to see how much of the information required for extrapolation is available in each domain. Simulative testing must be undertaken when extrapolation requirements are not met.

2.1 Generalities

Three types of experimental programs are performed today in tribology. They differ from each other both in purpose and method. They can be grouped under the headings: friction mechanisms, material testing, and simulation.

Research in friction mechanism is most often carried out on fairly simple tribometers. Experiments are performed under carefully controlled conditions and working surfaces can be examined with sophisticated equipment. The purpose here is to analyze the basic mechanisms of friction, independently of any specific application. This type of work is reported in References 1 - 3.

This section will not be concerned with experiments run to improve the basic understanding of friction but with those run for practical purposes as outlined later. Let us therefore attempt to define briefly what is meant by material testing in tribology and by simulation.

2.1.2 Material Testing in Tribology

The purpose of material testing in general is the determination of intrinsic material constants or limits which characterize material behavior over a large range of conditions. Once determined, these constants or limits can serve to predict the response of the material in a given environment. One classical example is the determination of Young's modulus, E, Poisson's ratio, ν, and the proportional limit, σ_0, in solid mechanics. The intention in material testing is to go from the particular, which is the test environment, to the general, which is the field of application, Table 1. Tests are few and standardized. The key word is "extrapolation."

Materials testing is not necessarily viewed in this fashion in tribology. It is now clear that there is no such thing as an intrinsic friction property of a given or even of a pair of materials[5]. Extrapolation of values of friction or wear, from one test condition to another, is condemed and even material ranking is known to be dangerous. Tests are unlimited in number and are rarely standardized[6, 7, 8]. Indeed there are at least as many tests as there are applications, and existing tests serve to qualify a material for a given usage (i.e. for specific loads, P, velocities, U, temperatures, T, and configurations, G) but do not yield parameters which can be related to the "intrinsic friction property" mentioned above.

The only true satisfactory work which can be listed under the heading of material testing in tribology is the study of

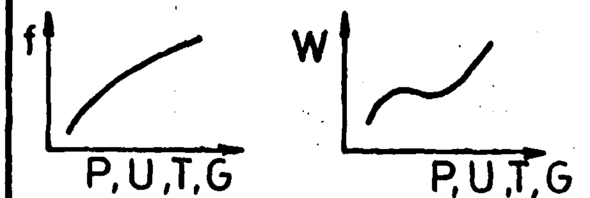
	Solid mechanics	Tribology
Types of tests	1 (traction)	∞
Results	E , ν , σ_0	
Consequence	extrapolation	no extrapolation

TABLE 1 MATERIAL TESTING

lubricant behavior under different pressures and shear rates[9, 10, 11]. Constitutive laws which apply over a fairly wide range of conditions are developed and performances can be predicted[12 - 16]. The approach here is very similar to the one referred to earlier in solid mechanics. It can be transposed to tribology, as the material which governs the friction process (i.e. the lubricant) is identified, it can be isolated and tested separately.

The situation is different in "dry friction". As, in most instances, the material which governs friction is neither identified nor isolated and therefore cannot be tested[17]. This situation will be considered later in this discussion.

2.1.3 Simulation

A detailed discussion of simulation will be presented later in this chapter[18]. Simulation is a global process, directed to the determination "in vitro" of the performance of a given mechanical component, isolated from a complex engineering environment[19]. This process is carried out independently of both basic friction phenomena and material evaluation. It is intended to solve a particular technological problem[20 - 23]. The procedure is therefore not standard and has to be redefined for each application. Many examples of material assessment through simulation are found in the literature[24]. By definition, no extrapolation is expected from simulation. The simulation can be compared to a black box, entry conditions are fed to it and results are transposed to the application with a minimum amount of interpretation. The difficulty resides in carefully designing the black box.

2.2 Third Bodies

Before getting into the details of material testing in tribology, it is necessary to try and set the pattern which exists and see what is known today and what should be known later to extrapolate laboratory results to full scale industrial applications as is done in structural design for instance.

Let us therefore recall a few well established facts around which we will attempt to construct our reasoning.

- Solid bodies adhere to each other in the absence of surface films [25]. Surface fimls are found where repeated sliding between solids is encountered. These films can be formed artificially, generated "in situ" by chemical action or wear, and entrained kinematically in the contact[26]. These films will be grouped under the heading of third bodies[27]. Oil and oxide films form effective third bodies.

- A contact is therefore made up of two first bodies (i.e. machine elements such as gears) and an intermediate film or third body.

- Third bodies can be defined in a general or "material sense" as a zone situated between the two first bodies enclosed by frontiers close to the surface which mark a change in composition or structure from the bulk material, Figure 1.

- Third bodies can also be defined in a more restrictive or "kinematic sense" (when the first bodies undergo relative motion) as the thickness across which the difference in velocity is accommodated.

The first definition is preferred by material scientists, while the second is more useful when the problem is analyzed in terms of continuum mechanics.

- Third bodies can be continuous or discontinuous, thus generating respectively "full" or "empty" contacts, Figure 2[17]. Continuous third bodies are found for instance in classical EHD, Figure 3. Different third body patterns are obtained with empty contacts. Solid lubricants exhibit essentially longitudinal patterns, Figure 4. Elastomers exhibit lateral patterns, Figure 5[30]. Further random patterns can also be expected to exist.

- Third body origin is diverse. The contact can be supplied in third body by either tangential or normal feed[31]. In normal feed, the third body is formed "in situ" through wear. Resulting wear particles are compacted during motion and separate the first bodies thus serving as load carrying agents. This situation is well illustrated during the first pass of experiments run with chalk, plastics, carbon brushes, etc[32]. Normal feed implies wear, with the notable exceptions of hydrostatic and porous bearings.

In tangential feed, the third body is drawn through the contact by the moving bodies causing separation of these bodies and thus acting as load-carrying agents. This situation is found with both liquid and solid lubricants and also when wear debris originally formed through normal feed is recirculated by the first bodies. Tangential feed prevents wear. Table 2 presents a few representative cases.

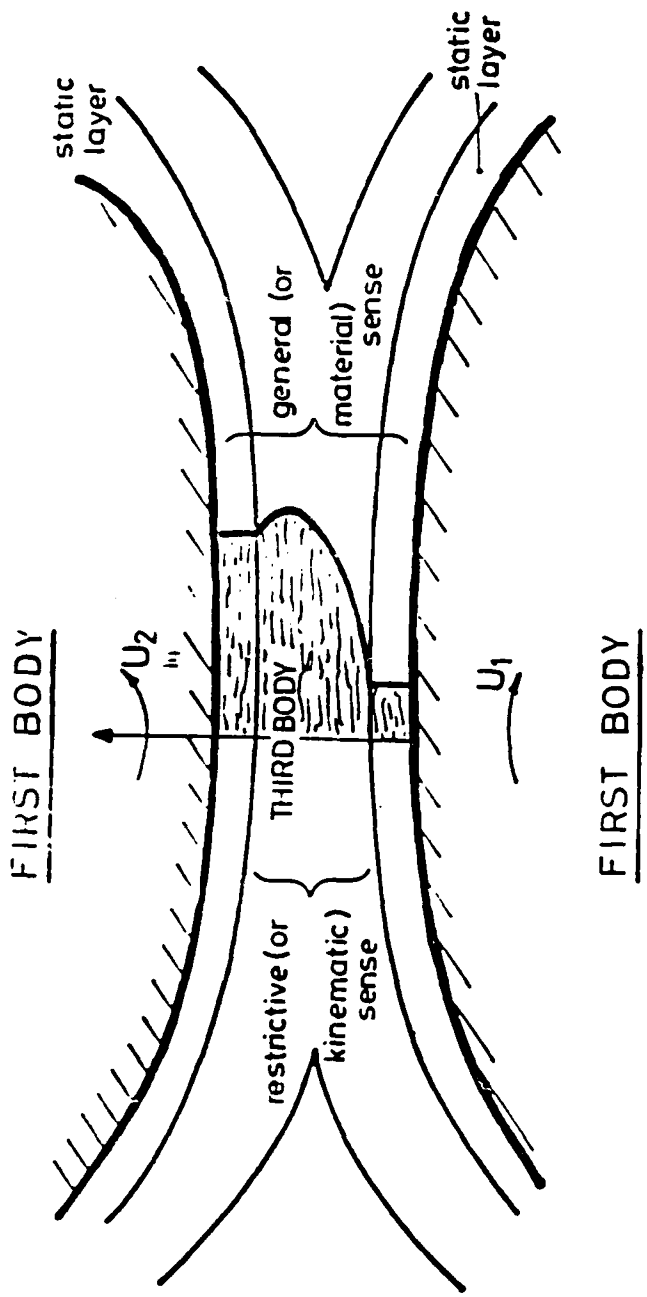


FIGURE 1 THIRD BODY DEFINITIONS

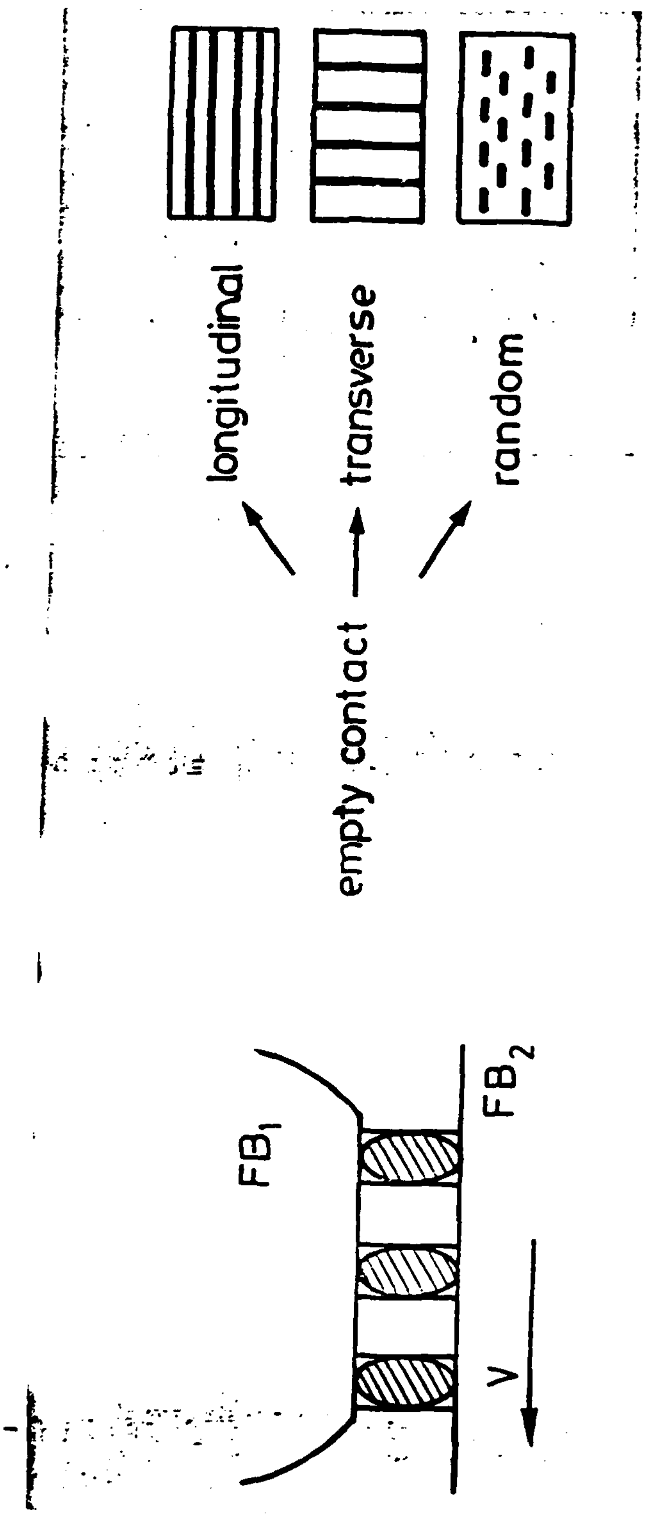


FIGURE 2 EMPTY CONTACTS

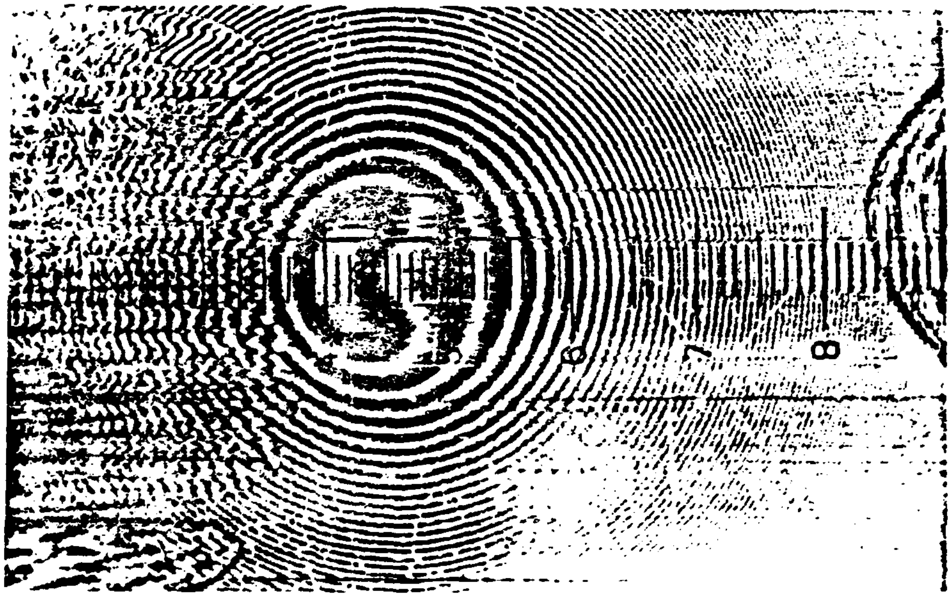


FIGURE 3 FULL CONTACT (EHD REF. 28)

FIGURE 4 EMPTY CONTACT-LONGITUDINAL (REF. 29)

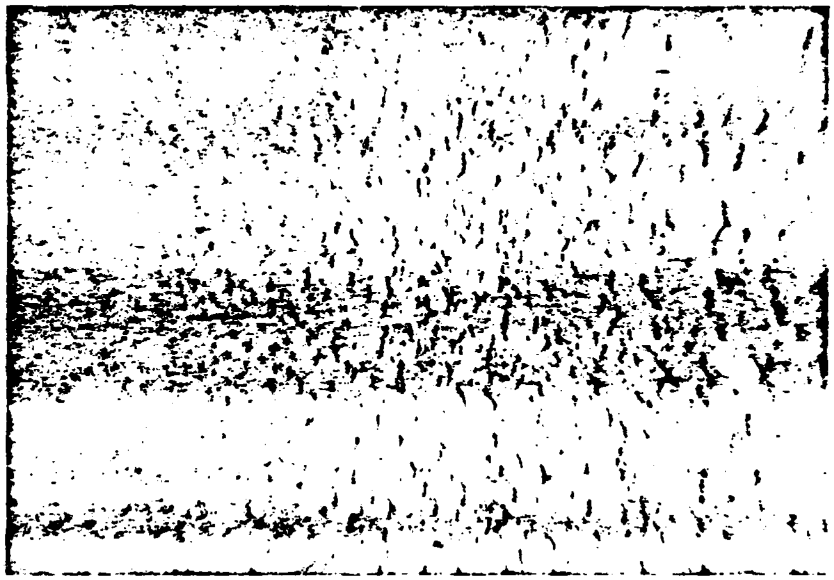


FIGURE 5 EMPTY CONTACT - TRANSVERSE (REF. 17)

cases studied		feed		
		pure normal	pure tangential	mixed
I	hydrodynamic lubrication		0	
II	thick film : solid lubricant		0	
III	chalk	0 ①		0
IV	boundary lubrication		0	0
V	rubber	0		0
VI	plastics	0 ①		0
VII	pencils	0		

①: first pass

TABLE 2 NORMAL AND TANGENTIAL FEED

In both hydrodynamic and solid lubrication, feed is exclusively tangential. Visualization shows that when chalk is rubbed against glass, feed is first normal and wear particles are formed, which are deposited on the track and recirculated in the contact during the following passes thus initiating tangential feed[32]. Similar behavior was noted with some plastics. Boundary films are reaction products formed on the surface by active components of the lubricant[33]. Feed is therefore mixed as the third body is partially formed from first body material (normal feed) and is also drawn kinematically (tangential feed) into the contact. Rubber rolls are first formed through normal feed and depending on configuration, are recirculated tangentially. Pencils which are rarely called upon to write over a "used" track, present an interesting, if not very common case of pure normal feed[34 - 35].

In conclusion, the above remarks suggest that third bodies:

1) separate partially or totally first bodies,
2) are fed either normally or tangentially to the contact,
3) are transported along the contact,
4) are load-carrying agents which permit relative motion.

Accordingly, from an industrial point of view, a tribology problem is solved when for a given set of dynamic, thermal, and geometric conditions, a material combination is found which is capable of:

a) sustaining heavy loads (of the order of tons) transmitted by the first bodies,

b) accommodating differences in velocity (of the order of meters per second) between the first bodies.

That combination must therefore:

a) withstand large normal pressures ($1 < p_{max} < 5 \times 10^3$ MPa) i.e. possess good volume properties, or more specifically, a <u>high shear resistance</u>,

b) accept high shear rates ($10^3 < u/h < 10^7$ sec^{-1}) i.e. posssess good surface properties or more specifically a <u>low shear resistance</u>.

Hence right from the start, it is clear that no one homogeneous material can satisfy both requirements simultaneously and that more complex combinations have to be sought. The simplest combination is the "Three Body Model" which has been described above. Clearly first bodies withstand the high loads while third bodies accommodate the high shears and transfer the loads.

2.3 Material Testing in Tribology

2.3.1 Idealization of the Problem

If it is true that a minimum working combination in tribology is characterized by the three body model, two classical material tests are required if both first bodies are identical and three are necessary if they are different. Further the conditions at the interface between first and third bodies must be specified. This argument sets the pattern of what follows. To simplify this discussion, it will be assumed that:

1) first bodies are identified and characterized and that no further testing is required,
2) contacts are full, and thick films are considered,
3) surfaces are smooth,
4) stationary conditions prevail.

Three important factors are thus left out of the discussion:

1) roughness effects on load-carrying capacity,
2) short and long term transients; the first being concerned with changes in dynamic or thermal conditions, the second being concerned with contact modifications during tests generally grouped under the heading of running-in[36].
3) empty contacts.

Finally, wear will not be discussed but it will be understood that, all other things being equal, a decrease in load-carrying capacity causes an increase in wear. Even within these important limitations, extrapolation from material tests results in tribology, other than in lubrication, will be shown to be practically impossible. This point will be developed later.

2.3.2 Levels in Extrapolation

Let us therefore try to define what, in this context, is meant by extrapolation. Three arbitrary levels of extrapolation with three different levels of efficiency can be defined, Table 3.

In the <u>first level</u>, one standardized and repeatable test yields "coefficients" and "limits" (such as Young's modulus, heat conductivity etc..) which are introduced in a "theory" that presents "absolute" results for a given engineering application. This procedure has been applied, at least in principle, for close to a century in classical strength of materials problems. The number of problems that can be solved in unlimited. From an engineering point of view, Level I efficiency is remarkable, and material testing can only reach that efficiency when all governing parameters or coefficients, within known limits, are identified and clearly <u>expressed</u> in terms of <u>mechanics</u>. Hence, theory predominates. Level I is used in tribology to solve straight medium pressure hydrodynamic problems.

In the second level, the mathematical theory is incomplete or more often non-existant because the governing parameters are either not identified physically or not expressed in terms of mechanics because of mathematical difficulties. However, definite trends can be drawn either from material and surface science studies or from experimental programs which can guide the engineer in the choice of materials for the conditions considered. The number of practical problems that can be solved largely justifies the approach. The answers furnished in Level II are comparative or relative, and the number of practical problems that can be solved is large. Level II extrapolation efficiency is adaquate and this approach is commonly used in metallurgy, corrosion problems, etc.

In the third level, little is known of either the mechanics or the pyhsics of the problem. Neither theory nor established trends exist. One test gives one point for a particular condition and nothing can be inferred concerning the system performance if that condition is varied only slightly. The answer funished in Level III is discrete or punctual. Extrapolation efficiency is nil as there is no basis for extrapolation. The description of Level III can appear to be somewhat extreme, however it is fairly close to reality when one considers, as noted in Table 3, the number of tests necessary to come forth with a workable industrial solution. The expense of this approach is of course formidable.

2.4 First Level Extrapolation in Tribology

2.4.1 First Level Requirements

An essay on testing in tribology should attempt to identify the level of extrapolation reached in various branches of this discipline and see what can be done, if anything, to move up to the level above. The most efficient way is to list the requirements imposed by Level I extrapolation and see how close representative branches come to it.

In order to reach Level I efficiency in tribology, it is necessary to predict load-carrying capacity or third body thickness and friction in a system working under given operating

EXTRA-POLATION LEVEL	TESTS number	TESTS results	UTILISATION formal	UTILISATION phenom.	UTILISATION empiric.	type of information	number of cases treated	efficiency	examples
I	1	coeff. limits	x			absolute	∞	10	strength of materials
II	10	trends		x		relative	>>1	7	metallurgy
III	1000	points			x	discrete	1	1	fretting wear

TABLE 3 LEVELS IN EXTRAPOLATION

conditions. These predictions require a theoretical analysis. Let us therefore attempt to identify the information needed by theory in order to solve the elementary three body case presented earlier (i.e. full contacts and smooth surfaces). The approach used in thick film lubrication which, as noted earlier, is the only case solved in a satisfactory manner will serve as guide. By contrast, lubrication will be compared to dry friction. It proves easier to start from the solution and work backwards thus meeting each aspect required to get that solution to work, and concentrating on third bodies as (section 2.3) all information and theories concerning first bodies are assumed to be available. This is usually the case in practice.

Load-carrying capacity and friction are forces. They result from the vectorial sum of normal and tangential stresses acting at contact. Independently of the origin of these stresses, both components are expressed in terms of mechanics. It is necessary therefore, to identify the information needed to develop a third body mechanics theory. Third body mechanics, in turn, is nothing more than thin film mechanics of continua (TFMC). TFMC equations, both dynamic and thermal, are formed around constitutive equations and require boundary conditions for their solution as they are partial differential equations. Constitutive equations can be written when the behavior (or rheology) of the third body considered is known. Kinematic boundary conditions, Figure 6, are divided in transverse and longitudinal conditions. The transverse conditions express the interaction between first and third bodies in terms of either displacement or stress. The longitudinal conditions take in the factors which govern the extent of the third body. As such they govern "entry" and "exit" conditions. Thermal boundary conditions specify heat fluxes or temperatures in both the transverse and longitudinal directions.

2.4.2 Third Body Rheology

As mentioned in Section 2.1, a considerable amount of work has been done recently in order to determine the rheological properties of lubricants under different pressures and strain rates. Results have been applied with success, in thick film lubrication. Constitutive equations have been developed.

In dry friction, however, the situation is radically different as the third body is rarely identified. An intermediate situation exists in polymer work where third bodies, both transfer films and polymer surfaces films, have been detected and their composition determined through surface analysis[37]. In these cases, third bodies are found not to be necessarily homogeneous as composition gradients and structure variations are noted across their thickness. These variations

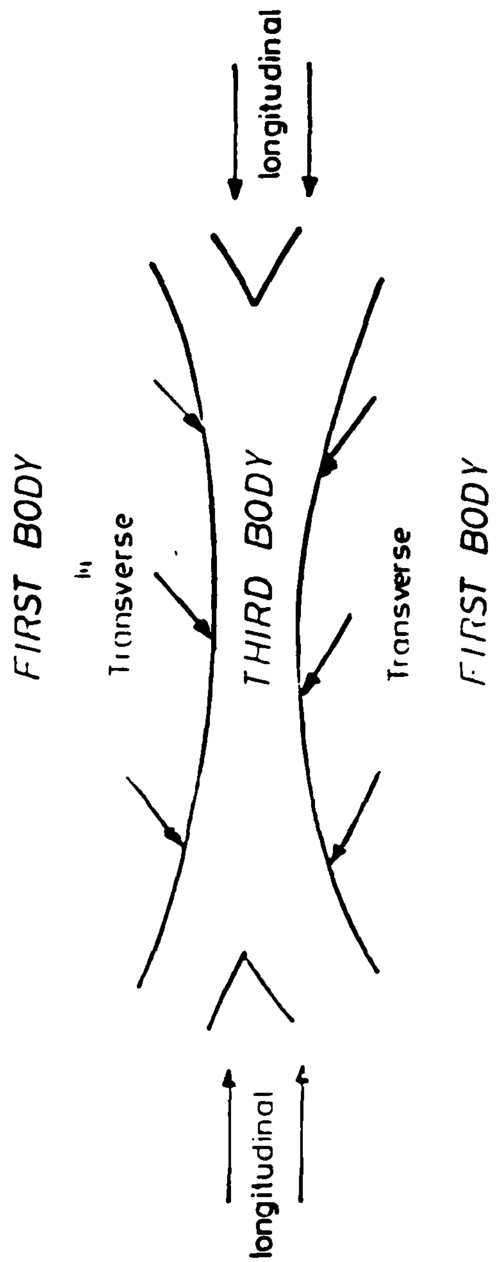


FIGURE 6 BOUNDARY CONDITIONS

underline the difference noted earlier between third bodies defined in the general sense and the strict sense. However, in a purely formal basis, this composition gradient would not constitute an insurmountable obstacle for analysis if the corresponding variations in rheology were known. The real difficulties lie elsewhere, and result from the fact that:

- The third body approach has not been completely worked through and all of its consequences analyzed and acted upon. This obstacle is one of attitude.

- Very little third body data are available particularly in cases where normal feed prevails. This obstacle results from difficulties encountered in the development of microrheometers capable of analyzing minute volumes of material.

Note, however, that as far as Level I extrapolation is concerned, precise information concerning third body composition is not needed. The only information demanded is that required to write the constitutive equations.

2.4.3 Boundary Conditions

In thick film lubrication, transverse boundary conditions are clearly expressed in mechanical terms by the "no slip at the wall" condition irregardless of the shearing stress. This condition seems to be quite satisfactory in most instances even though:

- It has been questioned recently under particular running conditions[39].

- The physics of the problem are not completely worked out[40].

From an engineering point of view, the second point is interesting as it suggests that a simple global mechanical expression of complex liquid solid interface phenomena is enough to reach Level I efficiency in a great many applications. However, concerning both entry and exit conditions, flow continuity conditions are in most instances sufficient to express longitudinal boundary conditions[42]. Thermal boundary conditions are often untractable mathematically, but basic concepts are clearly outlined for most cases and will not be discussed further here[43 - 46].

In "dry friction", the problem is not even approached in these terms and progess in the expression of boundary conditions appears remote as the respective "scientific partners" do not talk the same language. Statement of the type "beyond temperature, T_o, material A is seen to deposit transfer films when rubbed against material B under such and such condition," which conclude the work done by physicists are not yet exploitable in terms of mechanics to reach Level I efficiency[48]. Such statements are of course of a great help in Level II work. Clearly some effort must be expended, if third body analysis is to progress, and thus insure that the underlying physical concepts of material transfer incorporate stress and displacement notions[26].

Little is known about longitudinal conditions in "dry friction".

2.5 Extrapolation Levels in Different Areas of Tribology

The preceeding paragraphs attempted to identify the information required in thin film mechanics studies to reach Level I extrapolation. It appears useful now to see how close typical tribology systems come to that ideal, Table 4. As stated earlier, in order to calculate load-carrying capacity and friction in full smooth contacts, it is necessary to:

1) determine the values of third body rheological parameters (i.e. coefficients and limits),
2) express both transverse and longitudinal boundary conditions in both kinematic and thermal cases,
3) have available the corresponding thin film mechanics of continuum theory and the necessary programs to apply that theory to the particular problem under study.

In order to give a full account of the situation, and understand how far along in third body analysis the particular subject has advanced, it is necessary to specify whether or not the third body has been identified both in composition and as a load-carrying capacity agent. Further, an attempt will be made to predict whether efficiency or extrapolation level changes can be expected shortly. Finally, priority areas of work and application are noted in each instance. Table 4 lists seven cases chosen arbitrarily. The first three are taken from the thick film lubrication domain and obviously belong to the Level I class of solution. This does not mean that all aspects are either known or expressed in usable terms from an applied mechanics point of view, but only that most problems which belong to these cases can be solved theoretically:

- Some basic problems concerning feed and recirculation exist in bearings which are representative of low pressure (L.P.) hydrodynamic problems[49].

E.P.	steels 1000°C	plastics	sol lub.	hydrodynamics				CONTACT
reaction prod	debris	debris	ex Mo[illegible]	oils				T. B.
				VLP	HP	LP		SPEC.
F	E	F.E	E	?	F	F	F: Full ; E: Empty	TYPE
N.T	N.T	N.T	T	T	T	T	N: Normal T: Tangentiel	FEED
P	N	P	[illegible]	Y	Y	Y	Y: Yes N: No	THIRD BODY ID.
P	N	N	P	Y	Y	Y	P: Partly	THIRD BODY Rh.
U	U	U	U	K	K	K	K: Known	THIRD BODY Coef.
U	U	P	U	P	P	K	U: Unkn. P: Partl.	THIRD BODY Lim.
N	N	N	[illegible]	N	N	N	E: Expressed N: Not exp.	Boundary conditions TRANS PH
N	N	N	[illegible]	T	T	T	T: Transcribed N: Not trans.	Boundary conditions TRANS M
N	N	N	[illegible]	P	P	P	E: Expressed N: Not exp. P: Part.	Boundary conditions LONG. PH
N	N	N	[illegible]	P	T	T	T: Transcribed N: Not tran. P: Partl.	Boundary conditions LONG. M
N	N	N	[illegible]	Ex	Ex	Ex	Ex: Existant P Partial N: No	SOLUTIONS FORM.
Y	P	Y	[illegible]				Y: Yes N: No P: Part	SOLUTIONS PHEN.
Y	Y	Y	[illegible]					SOLUTIONS EMPI.
III	III	III	[illegible]	I	I	I	LEVEL	EXTRA. 1981
M	L	M	M	S	S	S	S: Short M: Med L: Long term	EXTRA. Prog
id 3rd B theol [illegible]	id 3rd B	debris thro BC [illegible]	[illegible]	lim BC	BC (entr.) lim. (HP) ph of fluids	B C* [illegible]	priority	FUTURE WORK
gears, cams, etc	oven steel works	dry bearings	[illegible] bearings	seals	ball bearings gears	bearings	examples	APPLICATIONS

*B.C. Boundary Conditions

TABLE 4 EXTRAPOLATION LEVELS IN DIFFERENT AREAS OF TRIBOLOGY

- Longitudinal boundary conditions are not completely understood in ball bearings which are representative of high pressure (H.P.) lubrication[50].

- The conditions which define cavitation zones in seals representative of very low pressure (VLP) lubrication, are not always clearly defined[51]. Note that a significant advance in third body high pressure, high strain rate rheology has been accomplished in the last ten years as a result of the work undertaken on traction of different oils.

The classification of the last four cases in Level II or III is somewhat arbitrary. Indeed trends are found in the four areas, but tests must be run to validate each solution. There is a significant difference however, between the four when expressed in terms of closeness to Level I extrapolation. The substantial difference between Case IV (solid lubrication) and the other cases is that:

1) the third body is identified,
2) it exists in sufficiently large quantities to insure that satisfactory rheological determinations can be conducted.

However, many problems remain concerning:

1) contact fullness,

2) boundary conditions,

3) corresponding TFMC theory.

Contact fullness and longitudinal boundary conditions, particularly entry conditions, are related as feed is controlled by conditions at inlet. Transverse boundary conditions must also be looked into with great detail as indications here are scarce. Solid lubrication however, appears to be a likely area for progress. Attempts should be made in this case, even if in very limited applications, to reach Level I efficiency. The situation is clearly much less promising in plastic wear studies. This skepticism as mentioned earlier, results from the complexity of the third body composition[37]. The whole process of load-carrying capacity in high temperature refractory steel friction, has proven to be of such complexity that there is very little hope of reaching any form of theoretical development in the near future[52]. However, visualization studies have given new lines of approach which can help breach the gap between Level III and Level II extrapolation. The very same ideas can be developed concerning E.P. studies.

Table 4 is fairly eloquent as areas of ignorance are shaded while areas of competence are left plain. One sees at a glance, the distance that separates the well coded disciplines such as lubrication from those such as fretting, which from a mechanical point of view, currently offer little hope.

2.6 Conclusion

In conclusion, this section has attempted to show that the notion of testing in tribology is as varied as the discipline itself. Tribology does indeed include elements of all three levels of extrapolation which command these radically different approaches in testing.

Level I requires material tests in the most classical sense. Material constants are determined and included in well established computer programs. Experiments are conducted occassionally to "verify" a result. "Indicative" tests can also be run to choose the best method among different levels of complexity in theoretical approaches. Thick film lubrication is the only example of Level I extrapolation in tribology. Progress in lubrication will come from the application of known methods to more amd more complex technological situations and from transfer of new mathematical methods from older and therefore more developed sciences such as fluid mechanics.

A significant effort is needed in micro-rheometry and in the understanding of both transverse and longitudinal boundary conditions. Further, developments of thin film mechanics of continuum solutions with varied constitutive equations and different longitudinal and transverse boundary conditions should be encouraged. Application of these solutions to specific cases should be followed with extreme care and with a well developed critical sense in order to detect unfounded claims and unrealistic mathematical developments. Precise answers to industrial problems come from simulation in Level II and III cases.

3. SIMULATIVE TESTING

Straight material testing has been shown to be not applicable to tribology as it does not yield "intrinsic" parameters which can later be used to extrapolate test results to real applications. The design engineer must thus rely on simulative testing.

The purpose of this section of the chapter is to present summaries of three cases of simulative testing which have been worked on in the laboratory over the last few years. Detailed accounts of the methods and the relevant bibliographies are found in the original published papers. The first case considers rib/roller contacts in tapered roller bearings; the second, gear damage simulation; and the third, simulation of helicopter oscillating dry bearings.

In simple terms, it can be said that engineers call upon data tables, extrapolation techniques, and of course experience to produce new designs. It was suggested in the first part of this chapter that three levels of extrapolation are found in engineering, depending upon the subject considered. Examples of each level are found in tribology. In Level I, all governing parameters are identified and taken into account in fully developed theories mostly through efficient computer programs. Straight fluid bearing design is classified in Level I.

The distinction between Level II and Level III is not as clear and easy to express. In tribology, Level II would include:

a) Cases are included in which the lubricant (intermediate film or third body) is identified and available in quantities sufficient for analysis, but whose rheology and also feed (longitudinal boundary conditions) and adhesion (transverse boundary conditions) aspects are unknown. Solid lubricants can be classified in Level II.

b) Cases are included in which the lubricant is identified and characterized rheologically but present particular feed conditions. Special bearing configurations belong to Level II.

c) Cases are included in which the lubricant is identified and characterized rheologically with classical feed on longitudinal boundary conditions, but for which thick films cannot be formed. Here, Level I techniques can predict partial separation but cannot foretell the damage which results from that condition. Partial elastohydrodynamic cases including highly loaded gears, belong under Level II.

Each case lacks some information which prevents it from being listed in the Level I table. However, trends exist which can be used in an engineering sense as guides to help solve the problem.

Level III would include all cases in which the third body is not identified. Dry bearings and most of the dry friction applications are recorded under that heading.

In the last few years through "Systems Analysis", a much needed and determined effort to tidy up simulative testing has been conducted[53]. This approach which is presented elsewhere, will not be discussed here. Its aims, however, are similar to those pursued in this treatment. The emphasis in this discussion is in the careful identification of the governing parameters and their simulation.

In order to avoid being side-tracted, the logic behind simulative testing, and its relation to testing in tribology is as follows;

1) All mechanical rubbing components are made out of at least three elements; two first and one third body.
2) Satisfactory material testing requires characterization of all three elements along with the definition of conditions at both first and third body interfaces. Only under these conditions can Level I efficiency be reached.
3) When these conditions cannot be satisfied, the engineer who seeks an answer to a specific industrial problem, as opposed to the scientist who attempts to understand a phenomenon, has to turn to simulation.
4) Consequently, test devices can be, or rather should be, radically different in fundamental and industrial laboratories.

3.1 Simulation of Rib-Roller End Contact in Tapered Roller Bearings

This discussion is part of a collaborative program between the Société Nouvelle de Roulement (SNR) in Annecy (FRANCE) and the Laboratoire de Mécaniques des Contacts of INSA. Detailed information is presented in References 54 and 55.

Figure 7 shows a section of a tapered roller bearing in which three contacts are found:

- rolling elements and cage (not shown)
- cone 3 and roller 2 line contact
- rib 1 and roller 2 point contact

The problem under consideration is the last contact. Without entering the details of the problems outlined elsewhere, the rib and roller end contact presents a certain number of original points which must be understood;

a) The hydrodynamic domain is very limited with the pressure build up entry abscissa, Figure 7, being very small which corresponds to starved conditions.

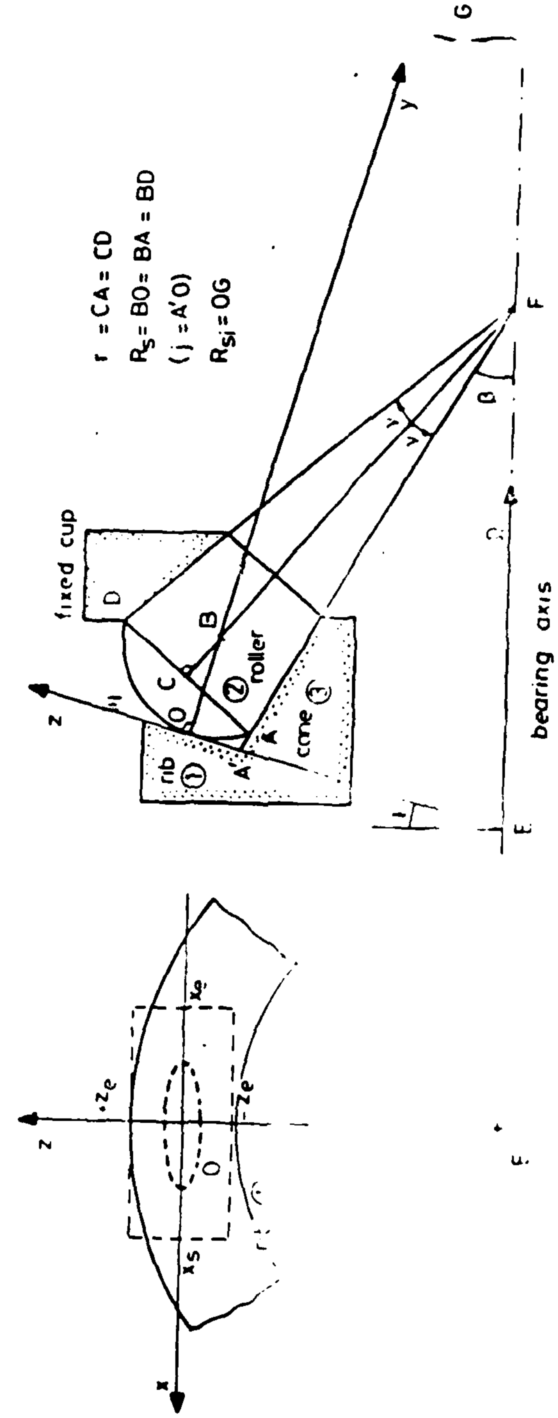


FIGURE 7 RIB ROLLER END CONTACT GEOMETRY

b) The large axis of the contact ellipse is parallel to the rolling direction, thus lateral or transverse flow is very important.
c) The kinematics of the problem are complex as rolling, sliding and spinning coexist.
d) Loads are comparatively small, thus the lubrication regime is either piezo viscous or lightly loaded elastohydrodynamic.

Classical theoretical methods can be used for the piezoviscous situation however, existing point contact elastohydrodynamic solutions cannot be applied because of the peculiar geometry. Consequently, under this case the isoviscous and piezoviscous problem will be solved but experimental simulation will be instituted for the entire range of conditions.

Respective simulation will be divided in two parts:

- Simulation and investigation will be initiated concerning the particular contact condition of only one contact on the INSA interferometric apparatus.

- Simulation will also be initiated concerning the entire bearing on the SNR special device in which rib and cone contacts are isolated.

In the original study, optimal geometry, film thickness and friction were studied. The optimal geometry could be determined by choosing the spherical roller end radius which gave the largest film thickness for given set of conditions.

3.1.1 The Two Simulators

Figure 8, a and b, show a schematic view and an actual photograph of the INSA one contact simulator. The apparatus is made of a flat glass disc and a steel toric specimen. The disc is driven by a variable speed motor which is fixed on a hydrostatic bearing possessing one degree of freedom in order to measure the lateral friction force. The toric specimen is fastened to a shaft connected to a second variable speed motor which itself is held by a cylindrical hydrostatic bearing. This bearing allows both axial and angular displacements. The axial displacement is used to measure the traction force in the rolling direction and the angular displacement is used to apply and measure the load.

The maximum height variation at contact location on the disc and on the toric specimen, is less than 0.5 μm. Surface roughnesses are respectively 0.005 μm and 0.01 μm CLA. The lower

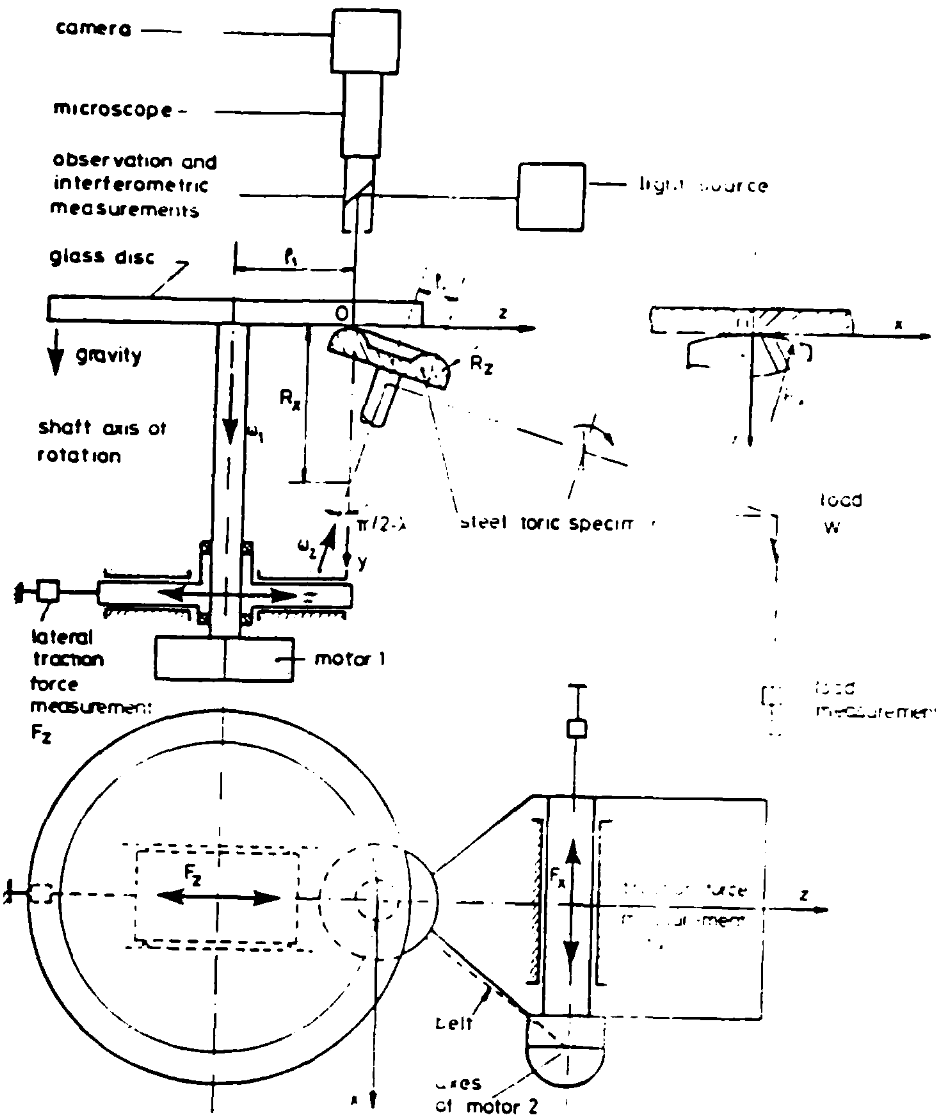


FIGURE 8(A) SCHEMATIC VIEW OF THE APPARATUS

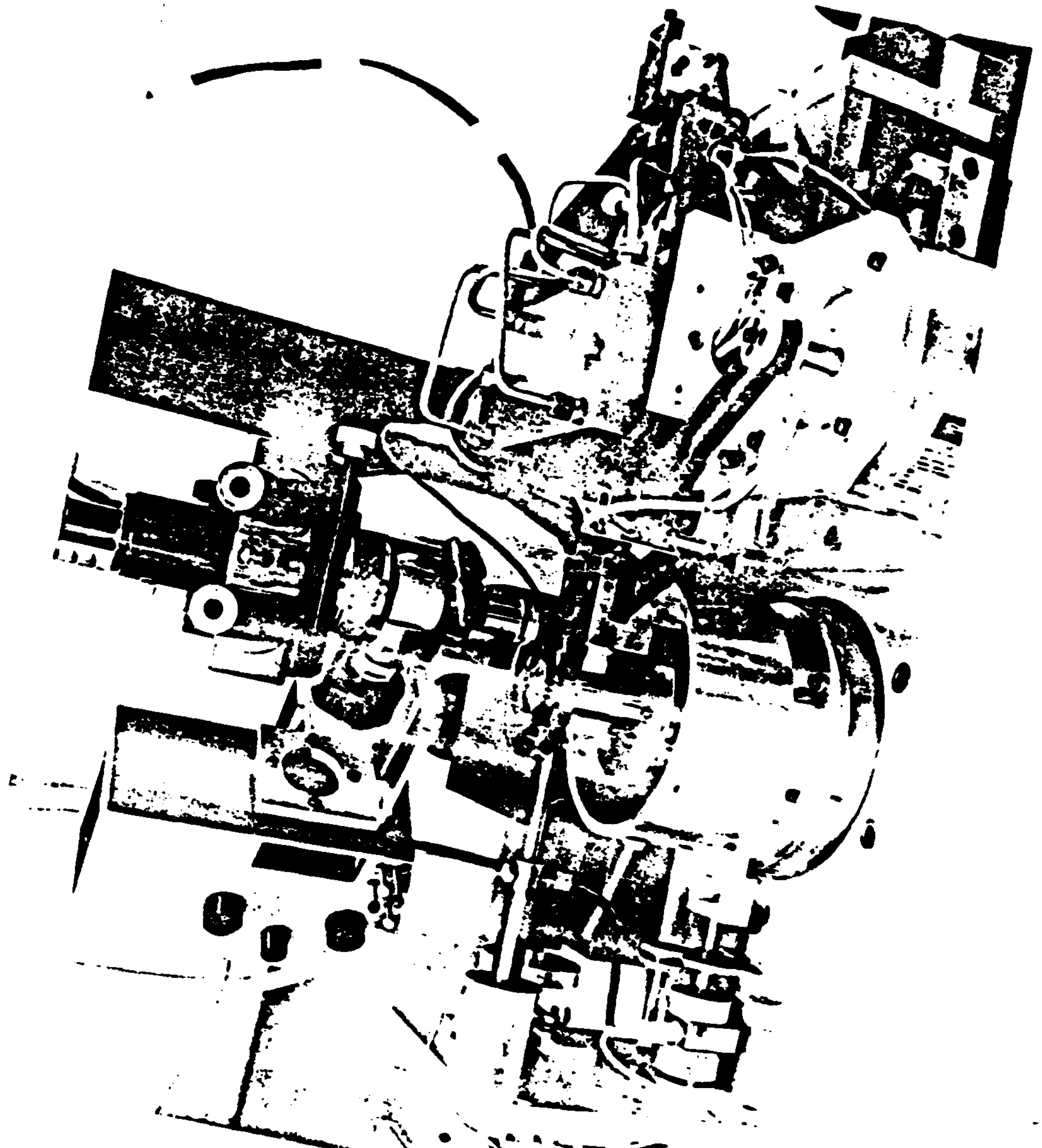


FIGURE 8(B) ONE CONTACT SIMULATOR

face of the disc is coated with a chromium layer and the upper face with an antireflecting coating. A classical optical system composed of a microscope, light source and camera creates an interferometric pattern.

Simultaneous measurements of load, angular speeds of both specimens, inlet oil temperature, traction force on the toric specimen in the rolling direction, lateral traction force on the disc, and film thickness are performed for this peculiar contact under hydrodynamic and elastohydrodynamic conditions.

A test rig was built by SNR to study the friction of the rib-roller end contact in order to determine an optimum geometric configuration as well as to study influence of the surface roughness on friction of the SNR tapered roller bearing 32314 BA, used primarily in truck axle applications.

Figure 9 shows a schematic view of the SNR apparatus in which rib and cone are separated in order to measure individually rib and cone friction. Hydrostatic pads are used to minimize friction generated between the bearing rib and cone and intermediate parts which are necessary for load application. These pads allow friction torque measurements. Axial load is applied on the main shaft by a hydraulic jack and is measured with a strain gauge transducer. Load is transmitted through the rollers to the cone and rib which are supported by the hydrostatic pads. Friction torques are also measured with strain gage transducers. Load and speed and their variations with time are regulated electronically. The rib-roller end contact temperature is measured by a thermocouple embedded 1mm below the surface of the rib. Inlet oil temperature is also monitored.

3.1.2 Specific Conditions To Be Simulated in the Rib-Roller End Contact

The mechanical parameters required to define the running conditions in the axis Oxyz at the point of contact 0 are:

- the two principal radii of curvature of the rib (R_{1x} and $R_{1z} = \infty$) and of the roller end which is spherical ($R_{2x} = R_{2z} = R_s$).
- the three linear and angular velocity components of the rib ($U_{1x}(0)$, $U_{1y}(0)$, $U_{1z}(0)$, and Ω_{1x}, Ω_{1y}, Ω_{1z}) and those of the roller end ($U_{2x}(0)$, $U_{2y}(0)$, $U_{2z}(0)$ and Ω_{2x}, Ω_{2y}, Ω_{2z}),
    - the extent of the contact region,
    - the surface roughness of surfaces 1 and 2,

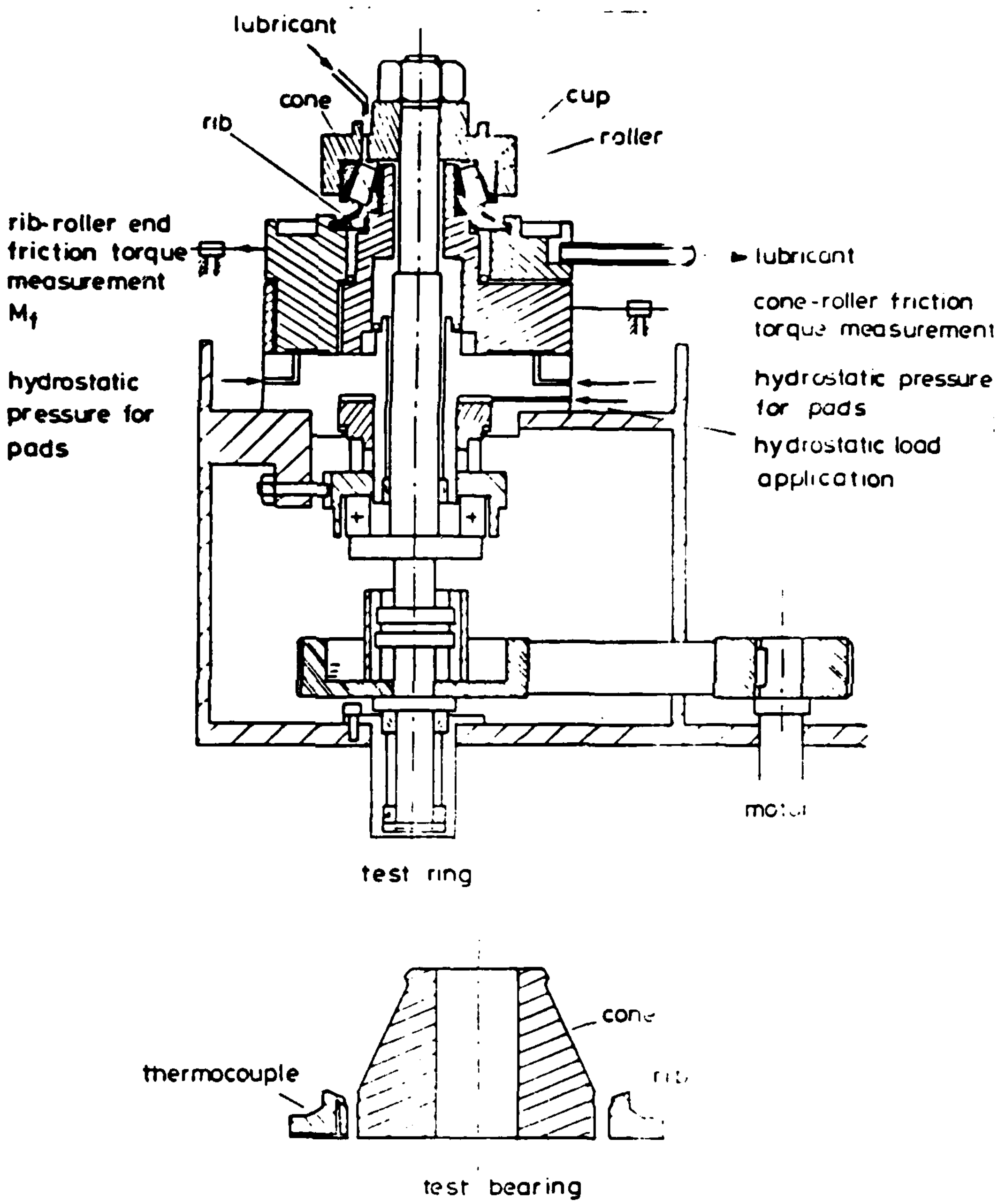


FIGURE 9 S.N.R. APPARATUS

- the applied load along the y axis,
- lubricant and material characteristics.

For the SNR bearing n° 32314 studied here, the kinematic and geometric parameters are given in Table 5 where Ω is the angular speed of the bearing in radians per second.

Table 5

Interference radius $R_{si} = OG$		0.21040 m
Equivalent radius $R_x = \dfrac{R_s\, OG}{OG - R_s}$		0.72874 m
Equivalent radius $R_z = R_s$		0.163262 m
	$U_{1x}(0)$	0.03124 Ω m/s
	$U_{1y}(0)$	0
	$U_{1z}(0)$	0
Rib (1)		
	Ω_{1x}	0
	Ω_{1y}	0.5614 Ω rd/s
	Ω_{1z}	0.1485 Ω rd/s
	$U_{2x}(0)$	0.01917 Ω m/s
	$U_{2y}(0)$	0
	$U_{2z}(0)$	0
Roller end (2)		
	Ω_{2x}	0
	Ω_{2y}	-2.8821 Ω rd/s
	Ω_{2z}	0.11743 Ω rd/s

As mentioned earlier the rib-roller end contact is an unusual lubrication problem. More specifically:

- The contact is lightly loaded (0 to 10^3 N) with the maximum Hertz pressure usually lower than 0.3 GPa.

- The velocities which govern the performance of this epicyclic contact are characterized by the rolling speed

  $U_{1x}(0) + U_{2x}(0)$, the sliding speed $U_{2x}(0) - U_{1x}(0)$ and the spinnning of both bodies Ω_{1x} and Ω_{2y}. The value of the rolling speed is similar to the one of the roller-cone contact but the sliding speed and spinning are independent.

- The contact geometry in which R_x is large (R_x ~ 1 to 10 m) has a radius ratio, $R = R_z/R_x$, less than unity (~0.001 < R <~0.5).

- The domain size in which the pressure can build up is very small thus the contact is severely starved (xe -~ 2.4 mm and z_e = 0.66 mm).

- Thermal effects must be taken into account as sliding speed and spinning are important.

3.1.3 Results

This discussion will not list all the results obtained in this study. The discussion will, however, present the method used to simulate the contact chosen and show points of agreement between the three approaches.

3.1.3.1 Theory

An optimal roller end spherical radius was determined by hydrodynamic theory. This was achieved by simultaneously changing γ and R_s to ensure that the contact point, "o", is situated half way along the rib and choosing the situation which yields the largest film thickness. For Ω equal to 1000 rpm, the film thickness is on the order if 1µm for an oil of viscosity 0.03 P1. The prime parameter is R_s, and the optimum values for load and traction forces are found for

$$\frac{R_s}{R_{si}} \sim 0.85 \text{ to } 0.90,$$

where R_{si} is the interference radius value of the spherical roller end. Thermal effects were shown to be much less important than geometric and domain size effects.

3.1.3.2. Experimental Results (INSA rig)

Tests were run for the particular geometric and kinematic conditions that prevail in this contact. More specifically; film thicknesses and traction forces were obtained at ambient temperature with low viscosity mineral oils under the following running conditions;

- The contact is formed by a toric steel specimen whose principal radii of curvature are Rx $\simeq$ 1.374 m and Rz $\simeq$ 0.663 m and a flat glass disc.

- The applied loads varied from 10 to 320 N giving maximum hertz pressures up to ~ 0.1 GPa.

- Rolling, sliding and spinning speeds are those found in the rib-roller end contacts.

Good agreement between the previous hydrodynamic theory and these experiments is obtained in the hydrodynamic regime for both load and traction forces. Departure from fully flooded elastohydrodynamic theory is due to domain size and starvation effects. These results show that hydrodynamic and the elastohydrodynamic lubrication conditions can be maintained at the rib-roller end contact for high loads and high speeds.

3.1.3.3 Experimental Results for the SNR Rig

Test rig speed and speed variation can be programmed between 0 and 1000 rpm. Oil inlet temperature is regulated at 63° C. Axial loads, ω_a, vary by steps up to 10^5N and generate maximum Hertz pressures between 20 and 300 M Pa. Representative values for contact ellipse dimensions are a $\simeq$ 2.48 mm and c $\simeq$ 0.78 mm for $R_s/R_{si} \simeq 0.85$ and a contact load $\omega \simeq 10^3$N. Rib and roller end surface approximate roughnesses measured along a direction perpendicular to the grinding direction are respectively 0.18 µm CLA and 0.08 µm CLA. Hence, the composite surface roughness is $\varepsilon \simeq 0.20$ µm CLA.

Sixteen different geometries suggested by the previous theoretical analysis, were tested. For a given geometry γ and R_s are specified. The SNR criterion through friction torque measurement, is based on the minimal speed, Ω, which gives thick film operating conditions. The optimum contact geometry configuration which can be deduced is

$$\frac{R_s}{R_{si}} = 0.88$$

which is very close to the theoreticl hydrodynamic prediction given in the preceeding paragraph.

A typical friction coefficient versus speed curve is given in Figure 10 for a given load.

An example of the good correlation between the SNR rig and the following is presented in Figure 10.

1) the INSA experiment in the EHD zone.
2) theory in the hydrodynamic regime.

3.1.4 Discussion

The hydrodynamic approach performed for rigid contact surfaces is useful in the understanding of the behavior of this lightly loaded contact and clearly shows that the geometry, the domain size and starvation are the main factors which govern the respective lubrication. Obviously, the surface deformation and the surface roughness along with thermal effects are also important factors which must be taken into account.

The INSA simulator gives a confirmation as to the importance of the geometry, the kinematic conditions, and the domain size in the rib-roller end lubrication process. These factors remain important when the experiments are extended to the elastohydrodynamic regime.

Good theoretical correlation was obtained with the SNR experimental results for a tapered roller bearing operating in the hydrodynamic or elastohydrodynamic regime. Clearly, low speed conditions which might bring about glass/metal contact cannot be simulated in the INSA rig. This constitutes one of the limitations of the system. However, the complexity of the INSA simulator is justified due to the fact that the inlet and kinematic conditions along with the lateral flow conditions could only be simulated in this manner.

3.2 Gear Simulation

The following nomenclature is utilized in this section:

E_1, E_2 = Young's modulus W = normal load

E' = reduced modulus

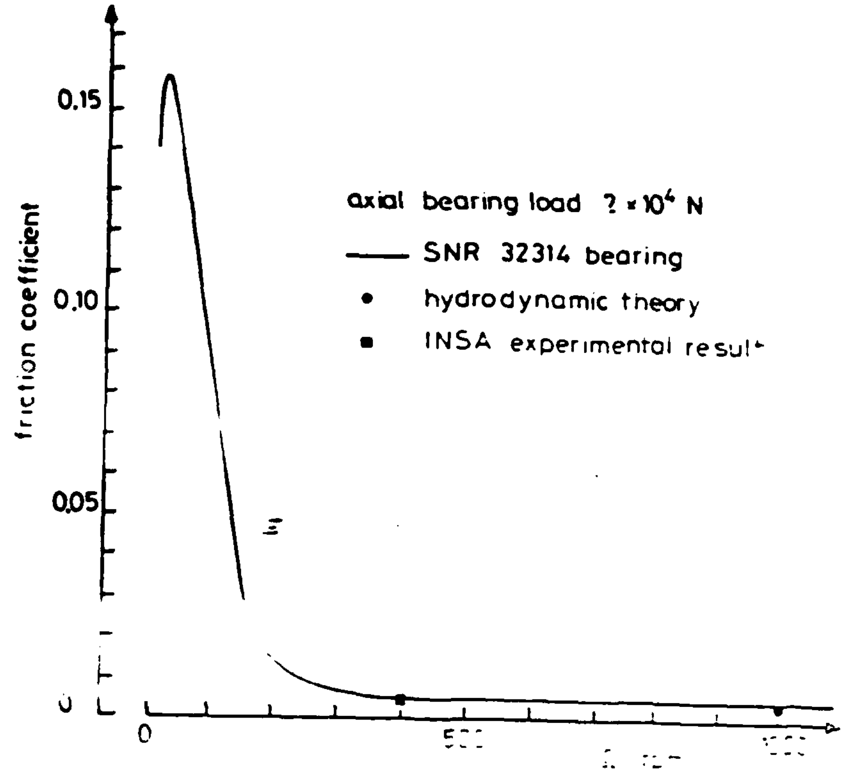


FIGURE 10 FRICTION COEFFICIENT VERSUS ROTATING SPEED FOR A GIVEN LOAD. COMPARISON OF THE INSA AND SNR RESULTS.

H_v = Vickers' hardness

h = EHD film thickness (Dowson and Higginson)

L = track width

p = pressure

R = reduced radius of curvature

R_1, R_2 = radius of curvature

T = temperature

U_1, U_2 = rolling speeds

α = piezoviscosity coefficient

β = thermosviscosity coefficient

ε_1, ε_2 = asperities heights

λ = slide/roll ratio

μ = viscosity

ν_1, ν_2 = Poisson's ratio

ρ = density

$\sigma = \sigma_1{}^2 + \sigma_2{}^2$ = composite roughness

σ_1, σ_2 = RMS roughness

This case study is discussed in detail in Reference 56 and 57.

The purpose of this effort is to compare the results furnished by a gear machine and a disc machine. Both are run under closely controlled conditions (defined below) in order to establish whether, under what appears to be the most favorable conditons, correlation between gear and disc surface distress is possible. The damage considered is surface durability commonly known as "pitting." Gear tests were run on the modified FZG machine of the Societe Nationale Industrielle Aerospatiale of Suresnes (France). Disc tests were performed on the variable center disc machine of the Institut National des Sciences Appliquees de Lyon (France).

Figure 11, a and b, give respectively a schematic view, and an actual photograph of the disc machine. Table 6 lists the machine characteristics.

3.2.1 Simulation

Loads, speeds, oil film thickness (h), surface roughness (T), roughness ratio (σ/h), gear materials, surface treatments, and lubricants are known to affect surface durability. They are

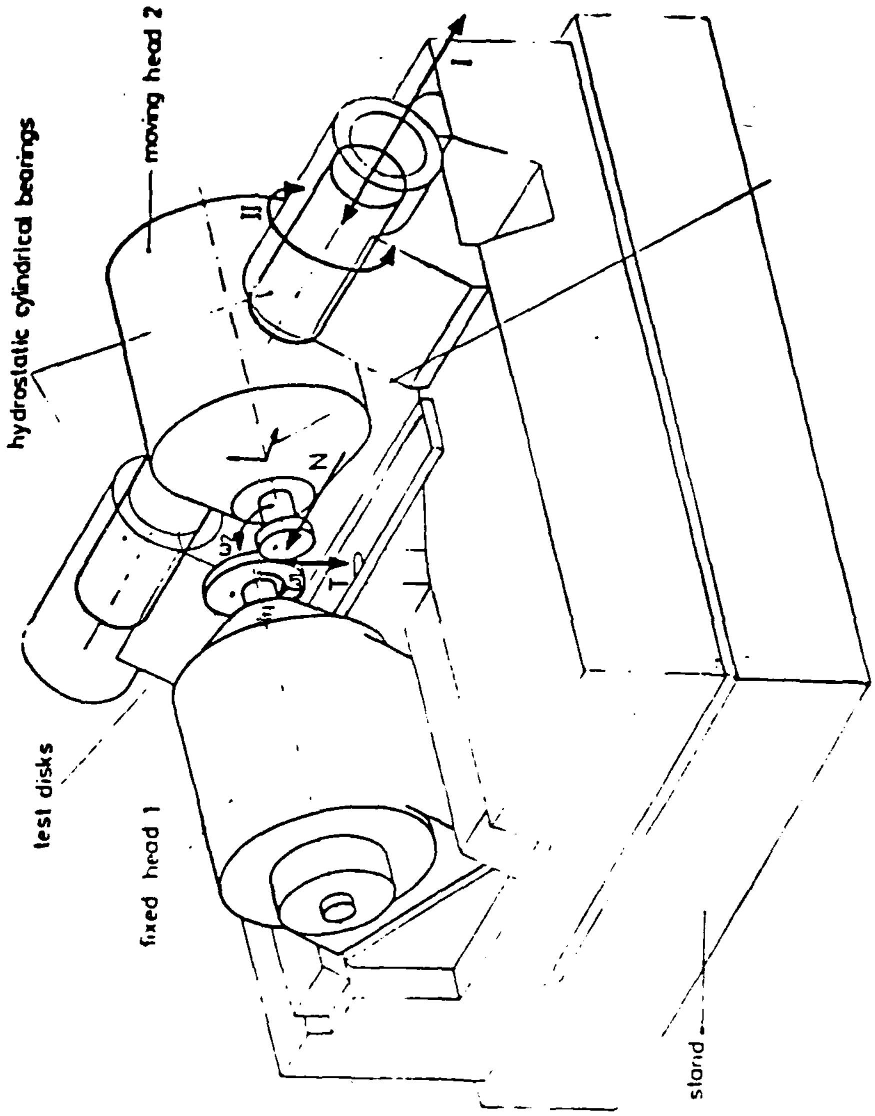


FIGURE 11(A) DISC MACHINE

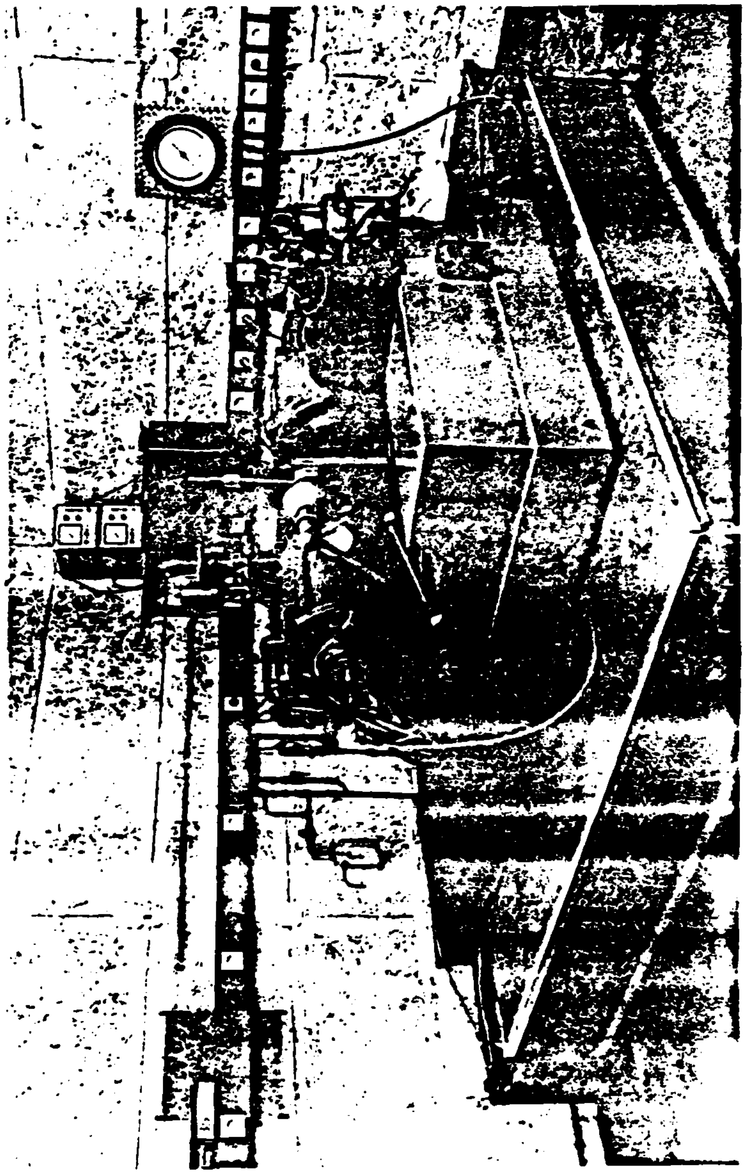
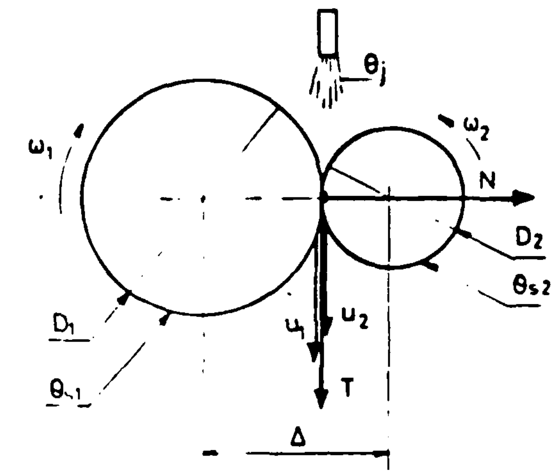


FIGURE 11(B) GENERAL VIEW OF DISC MACHINE



center to center distance Δ (mm)	-10 ⟶ 100
disk diameters D_1, D_2 (mm)	15 ⟶ 100
maximum disk out of roundness (µm)	0.25 ⟶ 1.6
disk widths L_1, L_2 (mm)	2 ⟶ 12
rotating speeds ω_1, ω_2 (rpm)	3 000 ⟶ 15 000 ⟶ 30 000
peripheral speeds u_1, u_2 (m/s)	3 ⟶ 80 ⟶ 160
sliding speed $u_1 - u_2$ (m/s)	0 ⟶ ± 157
normal load W (N)	50 ⟶ 15 000
maximal Hertz pressure p_0 (hbar or 10^5 Pa)	20 ⟶ 500
friction force T(N)	5 ⟶ 3 000
available power at contact kW	0 ⟶ 37
lubricant	any
oil jet temperature θ_j °C	20 ⟶ 250 ⟶ 500
surface disk temperature θ_s (°C)	80 ⟶ 400
vibrations of moving head (2) mS^{-2}	0.5 ⟶ 1000

TABLE 6 DISC MACHINE CHARACTERISTICS

not independent of each other and have to be considered singularly or grouped in the simulation.

In this discussion, simulation is based on the faithful reproduction, in a disc machine, of the Contact Mechanical Definition (CMD) prevalent at a particular point along the gear profile. CMD includes the mechanical parameters which are taken into account in the recent elastohydrodynamic theories of rough surfaces (EHDR) and the material and environmental parameters. The mechanical parameters of CMD are listed in Table 7a. Material and environmental aspects cannot obviously be defined as clearly, and identical conditions must therefore be created. Consequently both discs and gears were taken from the same blank and treated in the same oven. Both types of specimens were ground with approximately the same finish expressed in terms of both CLA and R_t. The lubricants tested are taken from the same batch. Both gears and discs were run in the air.

Simulation of gears by discs performed in this effort is, however, incapable of reproducing faithfully:

- surface roughness direction,

- hydrodynamic transients introduced by the variations in CMD which occur along the gear profile,

- the thermal effects,

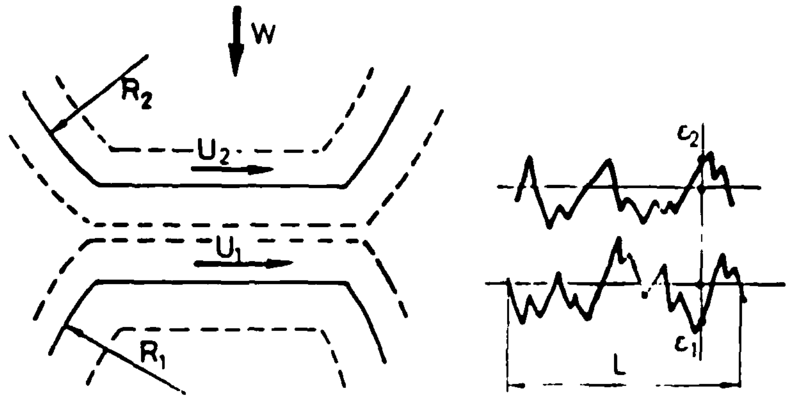
- any eventual interaction between two neighboring but different points along the profile.

Surface roughness direction is different in the gears and discs tested. Gear surfaces are generated on a Maag machine which yields a cross-striated pattern, while the discs are finished on a cylindrical grinder which give longitudinal roughness. This difference is known to produce differences in scuffing. The effect on pitting is less clear as the film thickness differences introduced by roughness is small at the low slide/roll ratios at which pitting is produced.

Film thickness is known to be modified by transients, however the change noted under the EHD conditions found in gears is negligible.

Oil viscosity at the contact entry which is known to control film thickness, cannot be measured in gears. Oil entry temperature can only be approximated with trailing thermocouples on the discs. Simulation of thermal effects is therefore very difficult. No significant discrepancies in the simulation results are however, expected from these effects, as all tests

CONTACT MECHANICAL DEFINITION (CMD)



geometry	R, $1/R = 1/R_1 + 1/R_2$
	L
roughness	$f(\varepsilon_1)$; $f(\varepsilon_2)$
kinematics	U_1 , U_2
normal load	W
material and lubricant characteristics	μ_o
	α
	E', $\frac{1}{E'} = \frac{1}{2}\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)$

TABLE 7(A) MECHANICAL PARAMETERS OF CMD

were run with fairly fluid oils (7.1 10^{-3} Pa.s) at a relatively high temperature (80° C). At this high temperture the viscosity temperture gradient is very low, therefore any variation in film thickness can be expected to be small. Interaction effects on simulation results will be shown to be of importance and will be discussed later.

Note that correct simulation of a given point along the gear profile can only be achieved on a fairly sophisticated machine with a variable center to center distance and independently controlled drives on each disc.

3.2.2 Test Procedure

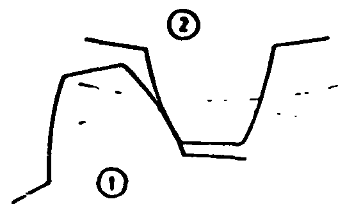
Gear characteristics and corresponding EHDR parameters at the profile point chosen for simulation are listed in Table 7b. The load program for both machines is given in Figure 12. The test procedures are slightly different as the load is applied at rest in the gear machine and at speed in the disc tester. A temperature controlled oil jet (80°C) lubricates the discs at contact entry while the gears are splash lubricated. The gear rig bath temperature is also controlled at 80° C. Lubricant filtration is not present in either the gear or the disc rig. Track observations are performed during the tests in both machines according to the schedule given in Figure 12. Oil changes are also indicated in this figure. Disc and gear roughness are measured before and after each completed test. Gear profiles are recorded at the end of the test. Friction force is constantly monitored on the disc machine. Vibration level is also monitored in both cases and tests are stopped when the vibration reaches a given level, usually resulting from a classical spall or by any form of surface damage which gives a signal of equal magnitude. Tooth breakage obviously causes the gear test to stop.

3.2.3 Surface Damage

Various forms of surface damage were obtained during these tests as follows:

1) micropits: craters roughly 20 µm deep and equal in diameter. They come either isolated or grouped, Figure 13. According to theory they should occur above a roughness ratio of 1.25.

2) sponges and flats: an area which can extend over several mm^2 and in which a high concentration of micropits is observed. Average depth of sponges can be slightly higher than that of micropits. The sponge areas can exhibit flats, Figure 14.

GENERAL GEAR PARAMETERS		
		
CHARACTERISTICS	PINION 1	GEAR 2
external diameter (mm)	88.5	105.5
pitch line diameter (mm)	81	99
number of teeth	27	33
modulus (mm)	3	
operating pressure angle	22° 27'	
addendum modification factor	0.33	0.20
base circle diameter (mm)	70.14	93.2
speed (r p m)	3666	3000
linear normal load (N/mm)	445 → [illegible]	
face width (mm)	15	

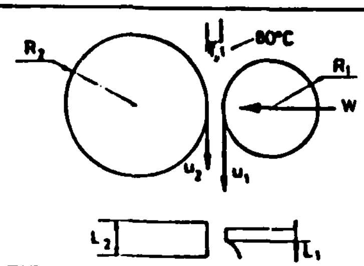
CONTACT MECHANICAL DEFINITION RETAINED		
		
geometry	R_i	$2R_1 = 31.4$ mm $2R_2 = 38.5$ mm
	L_i	$L_1 = 4$ mm $L_2 = 10$ mm
roughness	Ra_1 Ra_2	0.5 → 1 μm
	Rt_1 Rt_2	3.6 → 8 μm
kinematics	U_1	6.627 m/s
	U_2	6.047 m/s
	λ	0.045
linear normal load N/mm	W	455 → 832
material and lubricant characteristics	E	2.3×10^{11} Pa
	μ	synthetic oil 7.5 cSt (80°C)
	α	$\simeq 1.3 \times 10^{-8}$ Pa^{-1}

TABLE 7(B) GEAR CHARACTERISTICS & CORRESPONDING EHDR AT THE PROFILE POINT

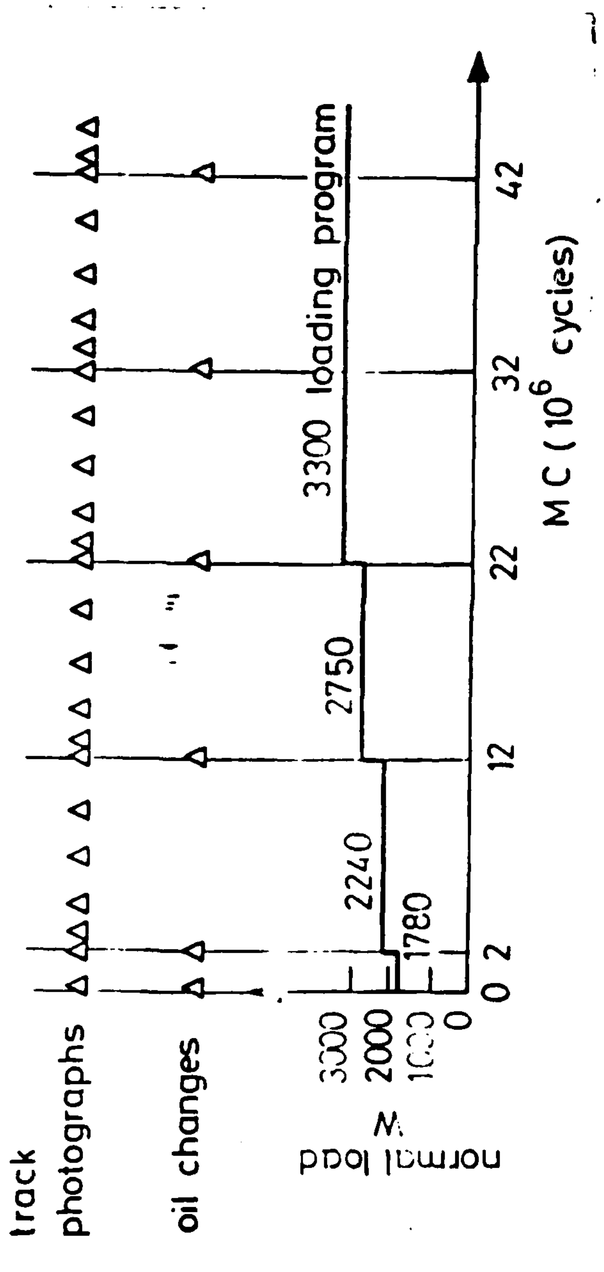


FIGURE 12 TEST PROCEDURE

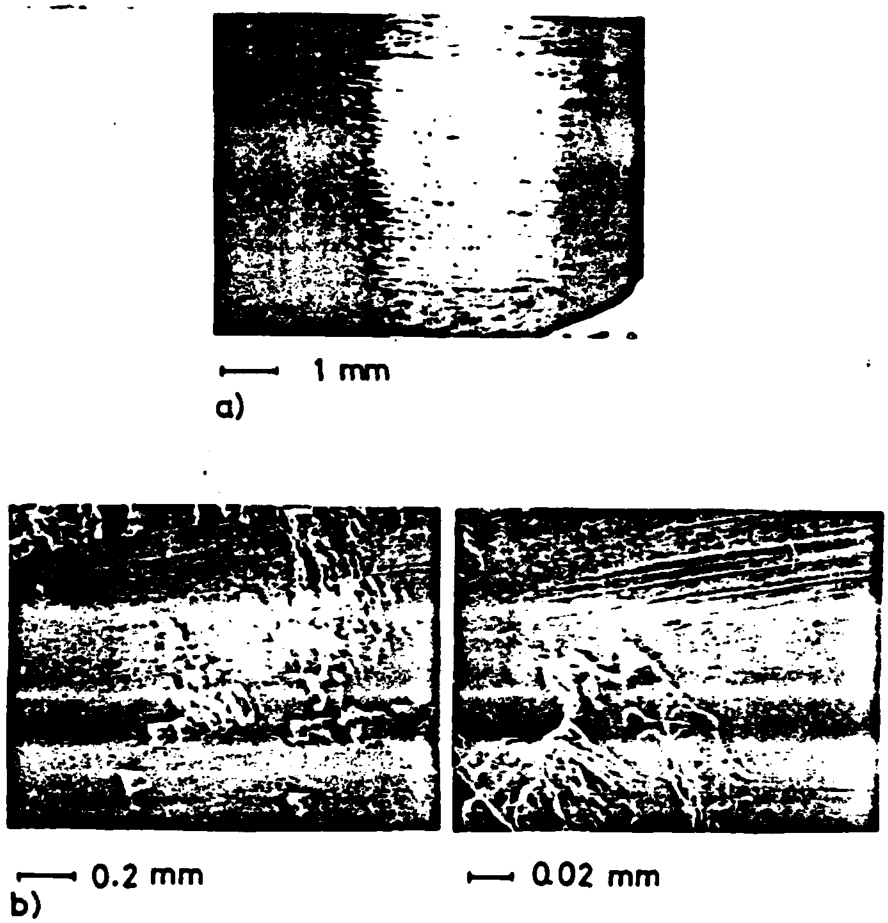


FIGURE 13 MICROPITS (A) OPTICAL, (B) S.E.M.

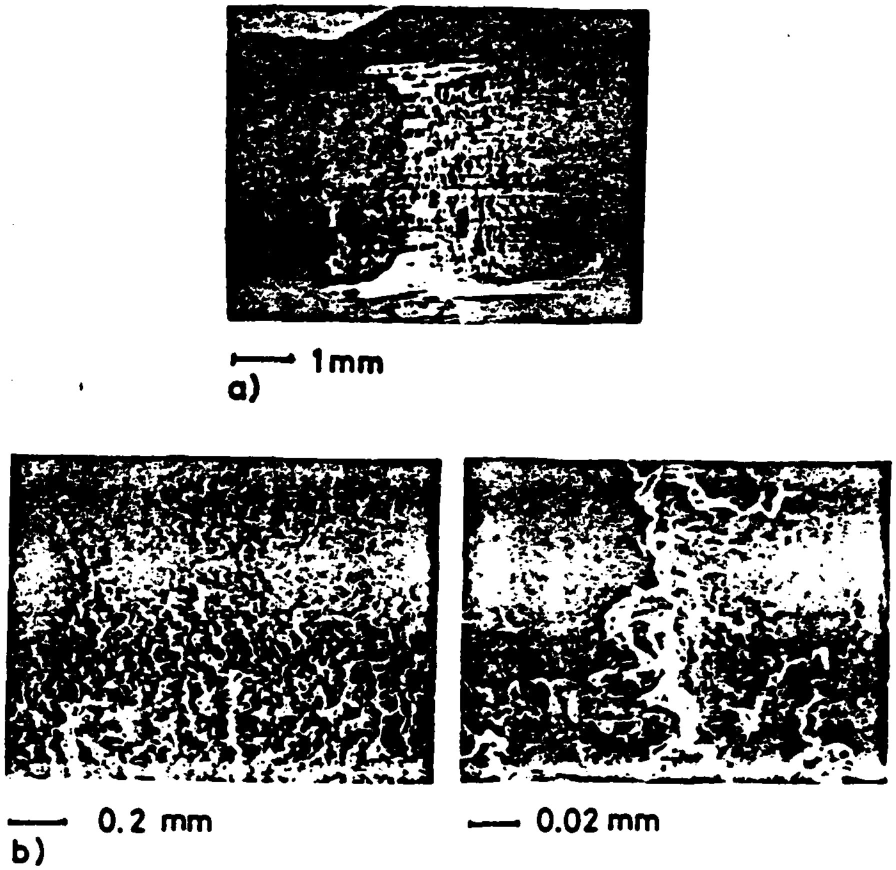


FIGURE 14 SPONGE AND FLATS

3) macropits: a shallow zone in which micropits have joined to form a void of several mm^2 in area. Macropits must not be mistaken for spalls which are 200 µm deep, Figure 15.

4) surface wear: a fairly evenly worn track from which the sponge zones were swept away. This condition is enhanced by sliding, Figure 16.

5) spalls: subsurface initiated fatigue pits abundantly described in the literature, Figure 17.

Clearly, macropits and sponges only characterize different densities of micropits but their individual appearance is sufficiently different to be noted. Running conditions are not affected by micropits and sponges. Flats, however, increase the

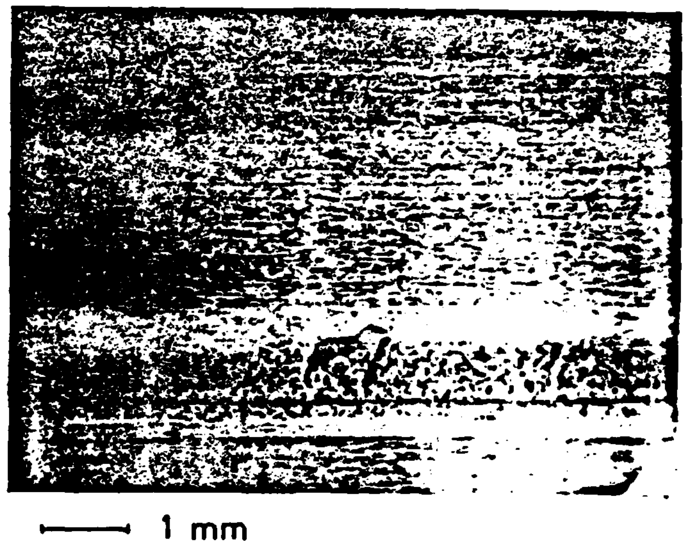


FIGURE 15 MICROPITS AND MACROPITS

FIGURE 16 SLOW DISC SURFACE WEAR

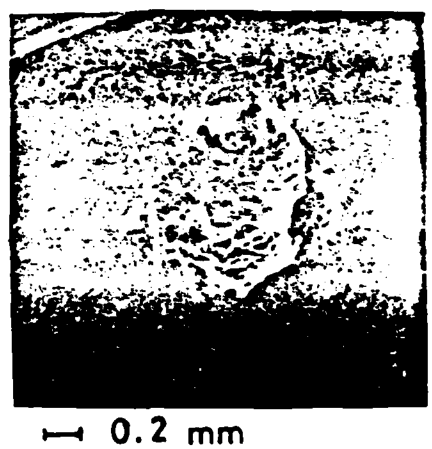


FIGURE 17 SPALL SURROUNDED BY MICROPITS

vibration level. Macropits grow in time and only become dangerous when they reach spall size which increase the vibration level significantly. In general terms, it appears that if material is removed by one cause or another, over an area equivalent in size to the hertzian band, the test cannot be pursued.

Spalls do not systematically develop in densely pitted areas. One spall however, was found below a macropit. In accordance with results found in the literature, the slow track suffers greater damage than the fast one and unless marked otherwise, results presented below deal with the slow track.

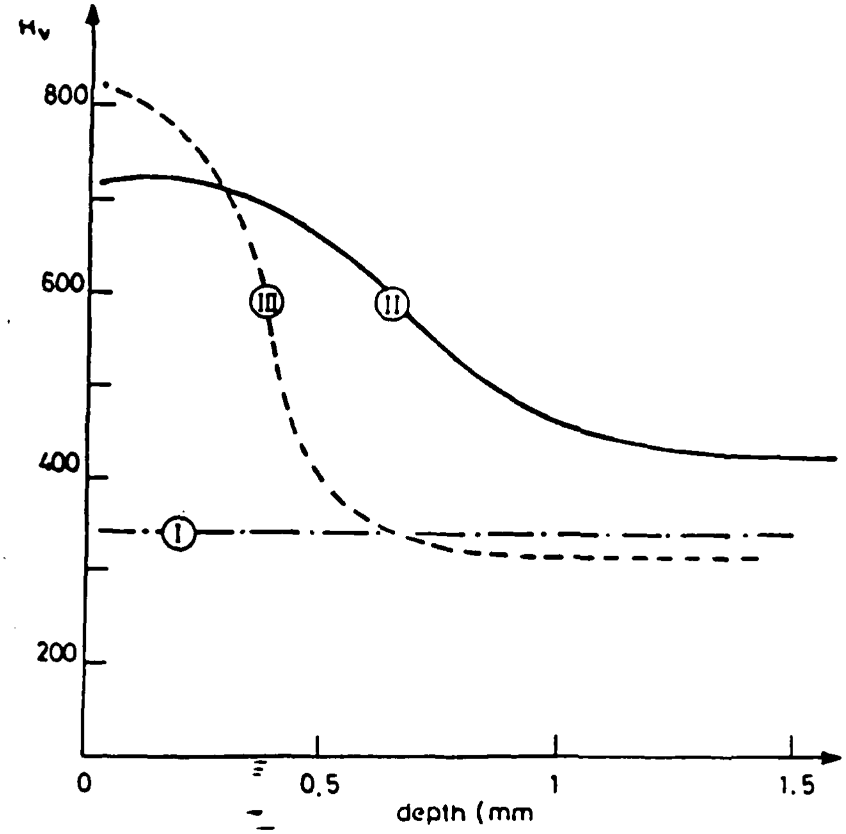
3.2.4 Results

Correlation studies were performed for 3 gear material combinations, Table 8, and one synthetic lubricant of the diester type. A minimum and maximum of respectively 3 to 5 runs were made for each test on both maachines. The final number of runs was a function of data scatter. Life reported is the arithmetic average of values measured in each test. Maximum dispersion with respect to the average value is approximately $\pm$ 60%, which is relatively low for fatigue tests. The number of runs is limited by time as each, including all measurements and controls, takes an average of two weeks. Test results are presented in Table 9.

In Table 9 the columns list the different forms of damage encountered in both mechanisms. The table also shows the roughness ratio, σ/h, at the beginning and at the end of the tests for both gear and disc. Correlation observed between discs and gears will not be judged strictly on life before final damage but on the existence and progression of similar forms of surface distress which might lead to different failures in the two systems.

Table 9 clearly shows that the correlation on micropit formation is very good between discs and gears for all three materials. Micropits appear between one or two megacycles (MC). They grow in number and reach a high density on material combination II. Their number is limited in material combination I and II. No macropits were observed on material I and III for both gears and dics, but extensive formation is observed with II in both mechanisms. Correlation therfore is also satisfactory for macropits.

Spalls were generated early on gears and discs with material combination I. The final damage was thus the same in both cases. Disc life was however, on the average 6 times longer than gear life which is better than the differences noted elsewhere in the literature. Correlation is adequate in this case.



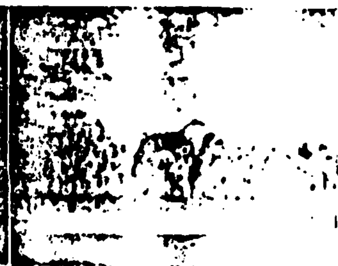
material combination	composition (%)					thermal treatment	grinding after treatment
	C	Ni	Cr	Mo	V		
I	.35	3.50	1.7	.30		through hardened	yes
II	.16	3.25				case hardened	yes
III	.32		3	1	.2	nitrided	no

lubricant (diester)
_ viscosity variations $\mu = \mu_o \exp\left[\alpha p + \beta\left(\frac{1}{T} - \frac{1}{T_o}\right)\right]$
_ thermoviscosity coef $\beta = 4\,230\,°K$
_ piezoviscosity coef $\alpha = 1.32 \; 10^{-8} \; Pa^{-1}$
_ viscosity at $T_o = 313\,°K$: $2.3 \; 10^{-2}$ Pa s
_ density : $\rho = 0.94$

TABLE 8 MATERIALS AND LUBRICANT

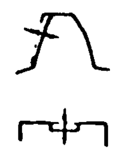
micropits

macropits



spalls

surface wear



test material	micropits gears	micropits disks	macropits gears	macropits disks	spalls gears	spalls disks	surface wear gears	surface wear disks	variation of roughness ratio a/h, gears, initial	gears, final	disks, initial	disks, final
I	scattered	scattered	none	none	some at 0.27 MC	some at 1.8 MC	none	none	1.37	1.10	1.70	0.38
II	dense > 1.2 MC	dense > 2.0 MC	yes at 22 MC	yes at 22 MC	none at 32 MC	some at 40 MC below macropits	50 μm at 32 MC ↓ tooth breakage	20 μm at 40 MC	0.59	1.15	0.81	1.49
III	scattered > 2.0 MC	scattered > 2.0 MC	none	none	none, nitrided case crushing ↓ tooth breakage at 60 MC	none, nitrided case crushing at 20 MC	40 μm at 60 MC	5 μm at 20 MC	1.46	0.78	1.45	0.90

TABLE 9 RESULTS

The situation found with material combinations II and III is more complex. With material combination II in gears, macropits were noted at 22 MC and accompanied by a relatively high wear rate. This situation led to tooth breakage, occuring at 32 MC. In discs, macropits were observed also at 22 MC with spalls forming below the macropits at 40 MC. Wear was significant but less than in gears. Clearly the micropit density, which increased in discs subjected to low slide/roll ratios, were swept away in gears. This situation is the result of the sliding which occurred in the neighboring zone and which, because of proximity, affected the zone simulated. If true, this hypothesis highlights the possibile interaction noted earlier, which can exist between points along the gear profile that see different CMD. Clearly this interaction does not exist in discs. To substantiate this conclusion, a heavily micropitted disc of material combination II, was tested under a higher slide/roll ratio (0.08). The test disc pits were rapidly worn away. However, final disc surface finish was not as good as that found in gears which is not surprising if one considers the initial roughness in that particular test. Two hypotheses can be advanced to explain the absence of spalls in the gears of material combination II. The first is that high wear retards spall formation. The second is that tooth rupture may have occurred before spall intitiation. The correlation which is clearly unsatisfactory if final damage is the only criteria considered, becomes more acceptable if the causes which lead to the two different forms of final damage are compared.

With materials combination III, case crushing was noted in both discs and gears at 20 and 60 MC respectively. Early disc failure was attributed to edge effects. Disc tests were stopped because of high vibration, while gear tests were terminated by tooth breakage. The final damage appeared to be different in both mechanisms but the cause was the same. Here again correlation is acceptable if, as above, the causes which lead to these two different forms of final damage are compared.

Finally, the variation in surface roughness during the tests follow the same trend in both disc and gear for a given material. Running-in was noted for material combination I and III. Surface deterioration was measured for material combination II. This leads to an increase of roughness ratio for material II and to a decrease for materials I and III for both disc and gear testing.

3.2.5 Conclusion

Simulation of gears with discs of different material combinations was performed on a high performance disc machine. The point simulated was chosen from previous experiments on gears where damage was observed to the the most important. Results show the following:

- The first forms of damage, i.e. micropits observed during the first few megacycles, are identical for the three materials in both gears and discs.

- The progression from micropits to macropits also followed the same trends in both mechanisms.

- Final damage in the two mechanisms however, is not always the same. Indeed if spalls are noted in both gears and discs for material I, the situation is different for material II and III where gear tooth rupture was noted. However, gear tooth rupture observed with materials III followed case-crushing which was also noted on discs and the correlation can therefore be considered to be acceptable. The tooth rupture observed with material II was due to high wear, which was not noted on the corresponding discs although they were severly pitted.

All results therefore clearly show that as long as gear geometry at the point simulated is not affected by wear, good agreement is found between discs and gears. Leaving roughness variation effects aside, the tests show that if the gear profiles are modified by wear, the gear CMD is altered while if disc wear occurs, the disc CMD is unchanged. This results from the fact that the variation in radii of curvature due to wear is negligible. Hence as long as the CMD of both mechanisms is identical, surface damage is the same. As soon as the CMD of gear differs from that of discs, mechanism damage is necessarily different and the correlation fails.

Concerning interaction, sliding causes densely pitted surfaces to wear uniformly. Interaction is imposed by the necessity of maintaining profile continuity during the wearing process. A given surface element cannot wear independently of its neighbor and a given point on the gear surface does not necessarily show the damage which corresponds intrinsically to that of the CMD calculated at that point. It can show however, a form of damage which is modified due to the interaction of neighboring points having different CMD and thus different forms of damage.

This study, which was conduced from 1973 to 1975, has been followed by quite considerable work on different materials and for different running conditions. The new work confirms what was observed originally and allows the following points to be highlighted:

a) Micropit density is very sensitive to material heat treatment. Materials with identical surface hardness but which have been produced in two separate batches can yield different micropit densities. These densities in turn, due to durface/volume interaction, can significantly modify life.

b) Slight changes in disc ratio do not appear to significantly alter surface damage when:

   1) the slide/roll ratio is low
   2) the hertz pressure is simulated independently of load.

c) Attention must be paid to synchronization effects. In the type of disc machine used here speed is controlled very accurately. However, a point of one of the discs does not come into contact repeatedly with a corresponding point of the other disc. With gears, by definition, synchronization is guaranteed. Synchronization effects have been noted at various times by different authors.

3.3 Simulation of Helicopter Oscillating Dry Bearings

Design of dry bearings is known to present problems in particular applications as guide lines are often missing in practice. Dry bearings are commonly used in aircraft, in transport, and in general also in industrial machinery. Classical criteria of the PV type approach (Pressure x Velocity = Constant) have been proven to be unsatisfactory. Significant improvements have been brought following the introduction of dimensional considerations. However, these improvements are unable to optimize high performance bearings. This section will deal briefly with these design problems. Further information can be found in References 58 and 59.

3.3.1 Governing Parameters in Dry Bearing Simulation

It is well extablished that in all contact conditions which perform satisfactorily, a thin film or third body separates both machine elements. The third-body composition varies from one type of contact to another and is rarely homogenous. In contacts formed by a moving steel ring, Figure 18, or shaft and a fixed bearing made out of friction material such as a plastic liner, a thin transfer film or third-body one (TB_1) covers the steel ring (first-body on FB_1) and under some conditions a layer of amorphous material known as third-body two (TB_2) is packed on the surface of the liner (first-body two, FB_2). As third-body formation, elimination and endurance govern both friction and

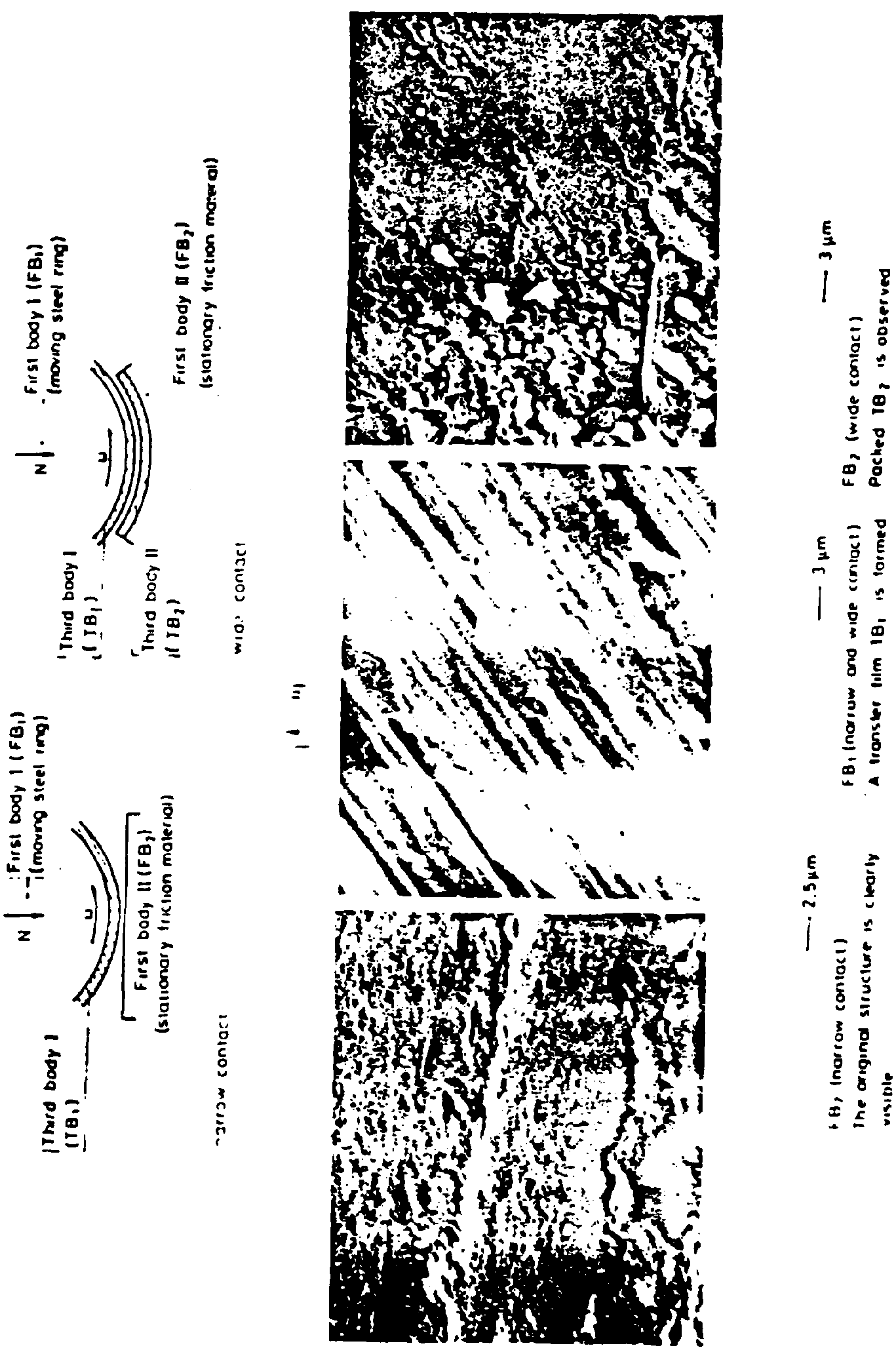


FIGURE 18 THIRD BODIES OBSERVED IN NARROW & WIDE CONTACTS

wear, the parameters which control these phenomena must be identified before a simulation can be attempted.

3.3.1.1 Contact Geometry

The effect of contact geometry on TB formation is significant. As an illustration, Figure 18 shows SEM pictures of both TB_1 and TB_2 formed in narrow (or Hertzian) and in wide (or distributed) contacts when a steel ring rubs against a carbon-reinforced resin under identical average pressures and linear velocities. TB_1 is formed in both cases, although TB_2 was only observed in the wide contact. Clearly, narrow and wide contacts are not satisfactory definitions but the observations noted are sufficient to indicate that bearing simulation cannot be performed with either point or line contact machines and that distributed loads have to be considered. Further, contact microgeometry cannot be ignored in simulation as wear or TB elimination has been shown to depend on surface roughness.

3.3.1.2 Contact Temperatures -- Effects of Boundary Conditions

Contact temperatures are believed to govern friction and wear in plastics. The effect of the heat path or the actual design of the machine on contact temperatures for a given energy input is first considered for a given energy input and for given external cooling conditions.

The experimental data presented in Figure 19(a) shows the variation of temperature with frictional energy measured under the same external conditions with the same test specimens but for two different heat paths. The thermocouple used is situated close to the surface of the stationary rubbing specimen. Curves I and II are characteristic of heat paths I and II respectively, which were imposed by two different cooling conditions. Further evidence is given in Figure 19(b) in which both the variation of friction and temperature with time are recorded. Conditions are identical in all four tests with only the heat paths varied as indicated. Heat path I led to liner destruction although satisfactory operation was obtained in the other three cases.

Figure 19 clearly shows that relatively minor differences in heat paths, determine the performance of a given frictional material for the same P.V. value. The differences achieved artifically in Figure 19, exist naturally between two industrial designs which use the same friction material. They must therefore be taken into account in any simulation attempt. Quite obviously, at the design stage, it is impossible to determine experimentally the heat path of a new mechanism. Therefore, this determination has to be done theoretically.

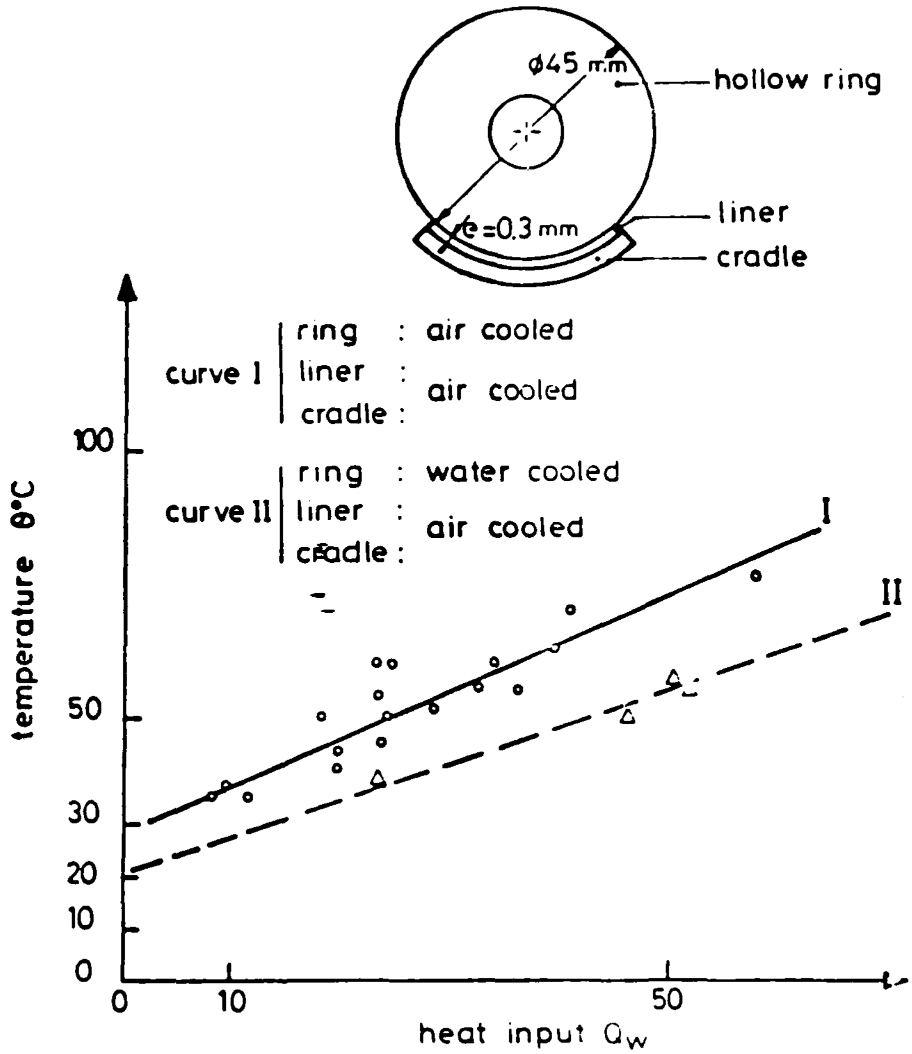


FIGURE 19(A) TWO CHARACTERISTIC HEAT PATHS OF THE SAME MACHINE

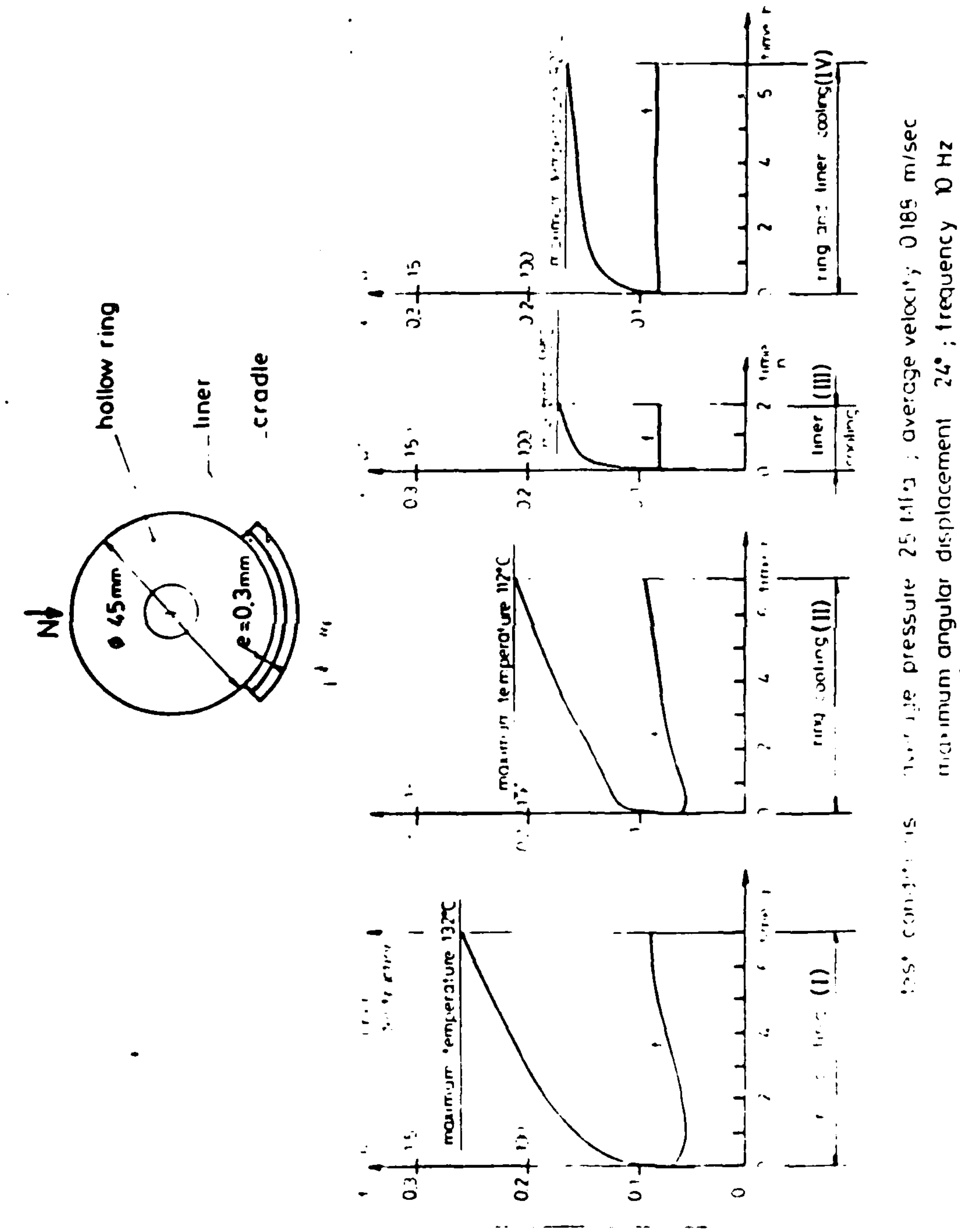


FIGURE 19(B) EFFECT OF FOUR DIFFERENT COOLING CONDITIONS ON TEMPERATURE AND FRICTION OF A STEEL PLASTIC LINER COMBINATION OPERATING UNDER OSCILLATORY MOTION

3.3.1.3 Contact Temperatures - Effect of Kinematics

Figure 20 shows the effect of the type of motion on contact temperature. The analytical techniques used to obtain these results are presented elsewhere and will not be discussed here. It is sufficient for our purpose to note that, for the same energy input, the temperature calculated for a steel ring rubbing against a plastic sector to be 126° C if the ring motion is limited to small angular displacements and only 80° C if this ring rotates continuously at 25 rev min^{-1}. Thus, the heat path of the machine clearly depends on the nature of the movement of its parts which therefore has to be taken into account in the simulation.

3.3.1.4 Minimum Sliding Distances

It was stated earlier that third bodies governed friction and wear. It was also shown that the nature of the motion determines the contact temperatures which inturn influence TB formation. However, independently of contact temperature, there is no evidence concerning the effect of the nature of the motion on TB formation. Third bodies are created during sliding but the minimum sliding distance necessary to generate them for a given material combination is not known.

3.3.1.5 Load, Pressures and Velocities

All test machines have fixed specimen geometries and information concerning the variation of both friction and wear with load is available. However, no data on the effect of pressure or pressure distribution for a given load are given, as this requires changes in specimen geometries. Thus, in simulation attempts, assumptions have to be made to relate the load and pressure conditions which exist in the real system to those created in the simulator.

Instantaneous contact velocities can be calculated accurately in all applications. In reciprocating systems the velocity variation can be complex and is therefore practically impossible to reproduce exactly on a simulator. Hence, equivalent velocity conditions must be defined for the simulator.

3.3.1.6 Surface Treatment and Environment

Figure 21 shows the difference observed in both friction, temperature, and wear when a plain and a phosphated steel ring rub against a carbonfiber reinforced resin. Clearly, third-body stability is enhanced by surface treatment and wear is considerably decreased. Environmental effects on the friction and wear of plastics are also well known and it is therfore

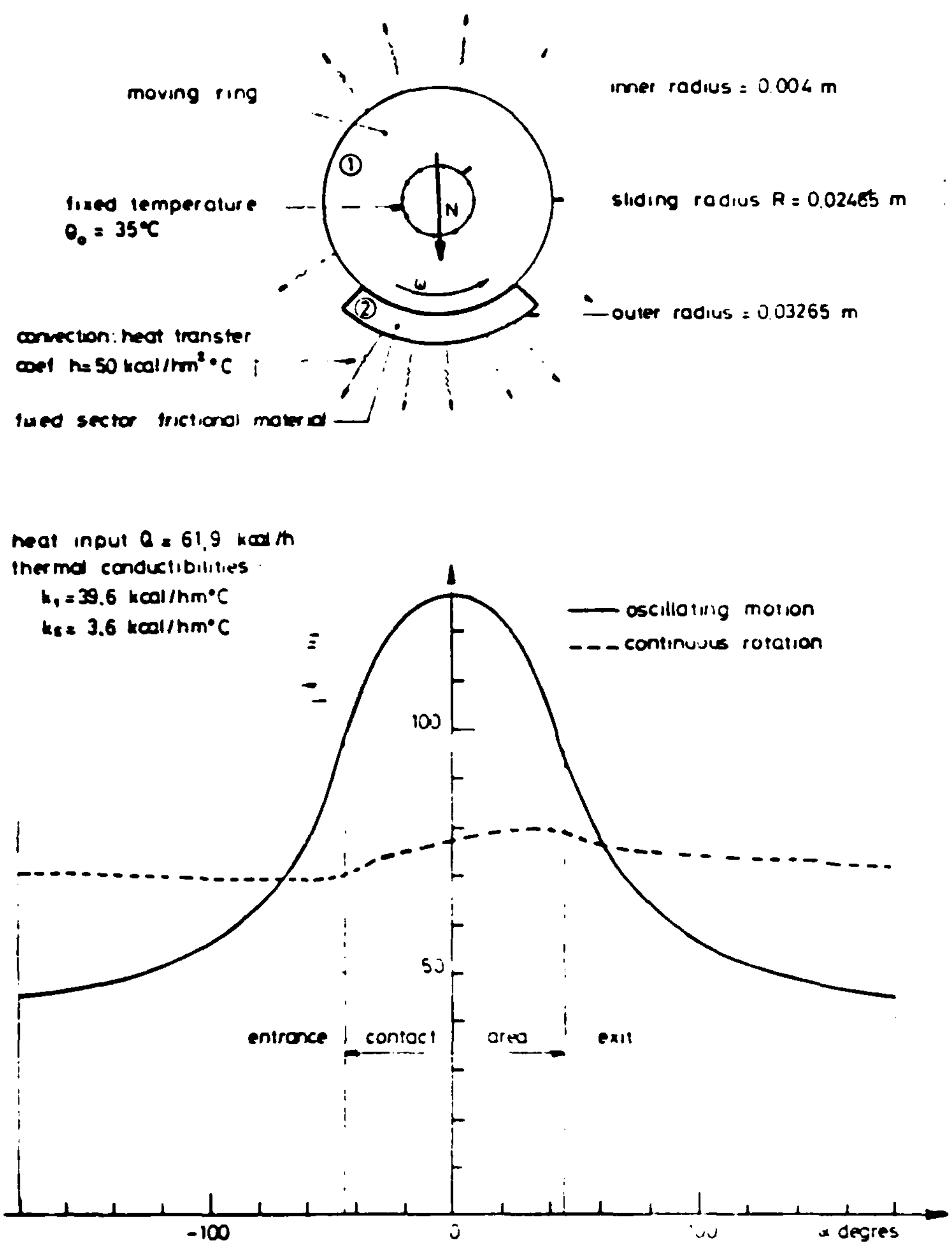


FIGURE 20 VARIATION OF CONTACT TEMPERATURE WITH POSITION FOR OSCILLATORY MOTION AND CONTINUOUS ROTATION

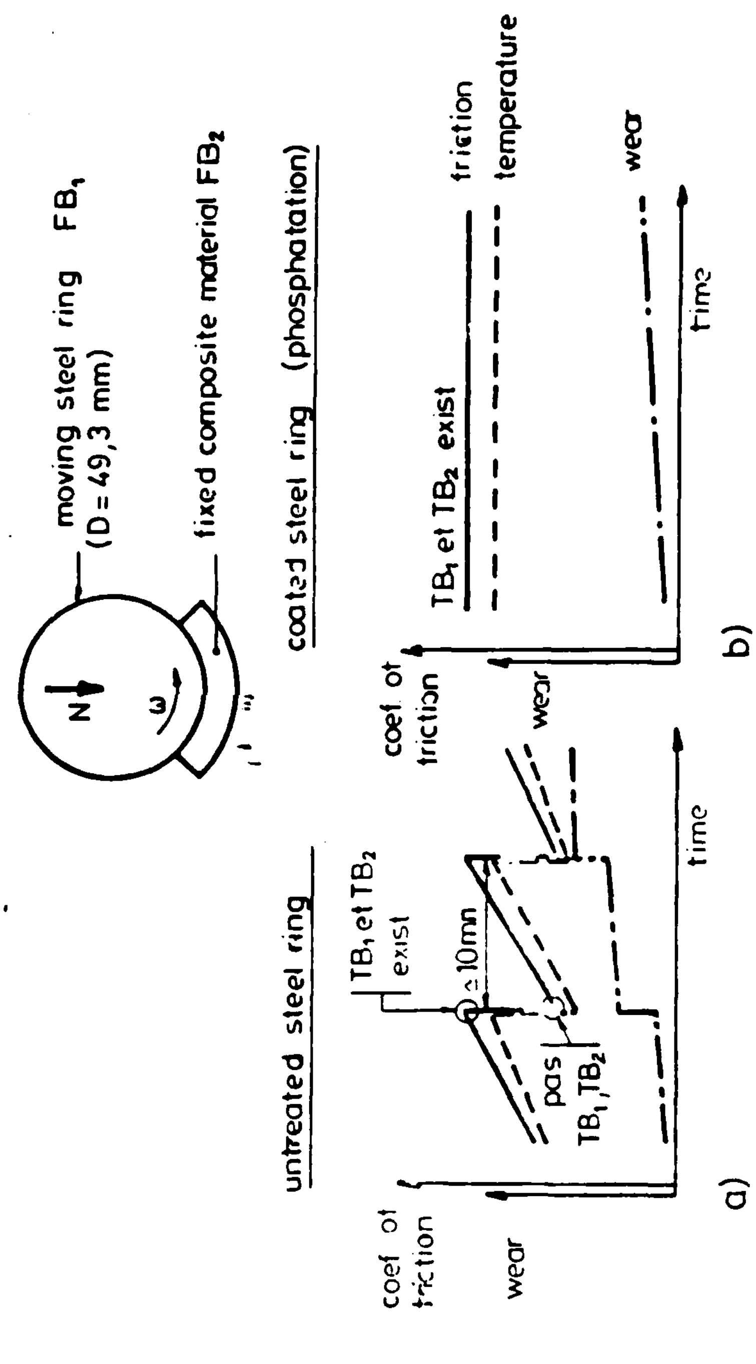


FIGURE 21 EFFECT OF SURFACE TREATMENT ON THIRD-BODY STABILITY AND THUS OF FRICTION TEMPERATURE AND WEAR

necessary for simulation to take into account both surface treatment and environment.

3.3.1.7 Wear Stimulation

Wear controls bearing life. However, the problem of wear simulation is very difficult and extrapolation of data from simulator results to the real application is still uncertain. This uncertainty arises from the fact that the basic phenomena are unquantified. A first step, however, is to consider that, for a given material combination, wear rates differ significantly with third-body stability, Figure 21. If the parameters which control TB stability are identical in both the simulator and the real bearing, TB behavior will be identical in both cases. Thus, a low wear condition on the simulator will lead to a low wear situation in the bearing even if the scale factors cannot be defined. The limiting values for the low wear rates imposed for the simulator studies were based on earlier experience obtained on materials tested both on the simulator and on the full-scale rig during a short preliminary investigation. A wear rate of 0.8 $\times 10^{-3}$ μm m^{-1}, determined on the simulator, was chosen.

3.3.2 Simulation Parameters

It is clear that contact geometry, contact temperature, contact kinematics, pressures, load, and velocities must be considered individually if adequate simulation is to be achieved. Environmental and surface treatment requirements must also be met. The full-scale simulation of the dry bearing described in Table 10 leads to the construction of a test machine that is well beyond the scope of this effort and defeats the very purpose of simulation which, as much as possible, must be sophisticated but which must be simple and economic to run.

It was therefore decided to build a simulator capable of reproducing:

a) the contact microgeometry and macrogeometry;

b) the maximum and average contact pressures and their variation in time due to cyclic loading;

c) the type of motion and the average velocity;

d) the contact temperature (this must be met independently of a, b, and c);

e) the environment.

Bearing dimensions and operating conditions	
Dimensions :	
bearing shaft diameter	0.125 m
bearing width	0.019 m
Load and pressures :	
cyclic loading	$N = N_o (1 + \sin \omega t)$
frequency $f = \omega/2\pi$	4.4 Hz
maximum amplitude 2No	0 < 2No < 22 000 Newtons
maximum pressure p_{max}	$0 < p_{max} < 12$ MPa
average pressure	$0 < p_a < 9.3$ MPa
Displacement :	
cyclic motion	$\alpha = \alpha_1 \sin \omega t$
frequency $f = \omega/2\pi$	4.4 Hz
maximum angular displacement	$0 < 2\alpha_4 < 20°$
Material and environment :	
shaft material	440 C steel (untreated)
surface roughness	0.4 μm CLA
friction material	to be defined
atmosphere	air
Limits :	
maximum coefficient of friction for the highest load 0.15	
maximum wear rate 0.8×10^{-3} μm/m	

TABLE 10 BEARING DIMENSIONS AND OPERATING CONDITIONS

3.3.2.1 The Simulator, Figure 22

The contact used for the simulator was that of a ring rubbing against a 90^0 sector. Motion can be continuous or oscillating and load is either constant or cyclic. The heat path can be varied by different cooling techniques. The simulator characteristics are given in Table 11 along with the contact parameters found in dry bearings. The conditions that can be simulated are cross-hatched.

The test conditions for the bearing described in Table 10 and established from simulation principles are given in Table 12. Four different woven friction liner materials which were retained after an exhaustive preliminary study formed the list of test materials. The simulator tests were then performed. As an example, the results obtained with the optimal test liner are presented and the performed calculations are described below.

Table 13 gives the results obtained on the simulator with a Dacron weave liner filled with Teflon. Five cyclic loads and four angular displacements were tested with the load and displacement frequency being 10 Hz.

3.3.2.2 Experimental Procedure

The test specimen ring is cleaned ultrasonically in a dimethylethyl ketone bath. The sector is wiped with a clean cloth and the load applied after the ring is set in motion. Each test lasts 30 hours. Friction and temperature are recorded continuously and wear is measured at predetermined times. The wear rate is calculated after running-in and is expressed in terms of wear thickness in micrometers per length of travel in meters.

3.3.3 Simulation

The different steps in the simulation procedures required in the design of high performance bearings are outlined in Table 14.

(1) The bearing running conditions together with the limiting requirements of the friction force, wear rate and maximum contact temperatures are clearly defined. A thermal model for the bearing and its housing is developed.
(2) The load and speed to be applied to the simulation are determined from simulation principle.
(3) A short list of promising liner materials is established after either a literature survey or a preliminary gross ranking campaign.

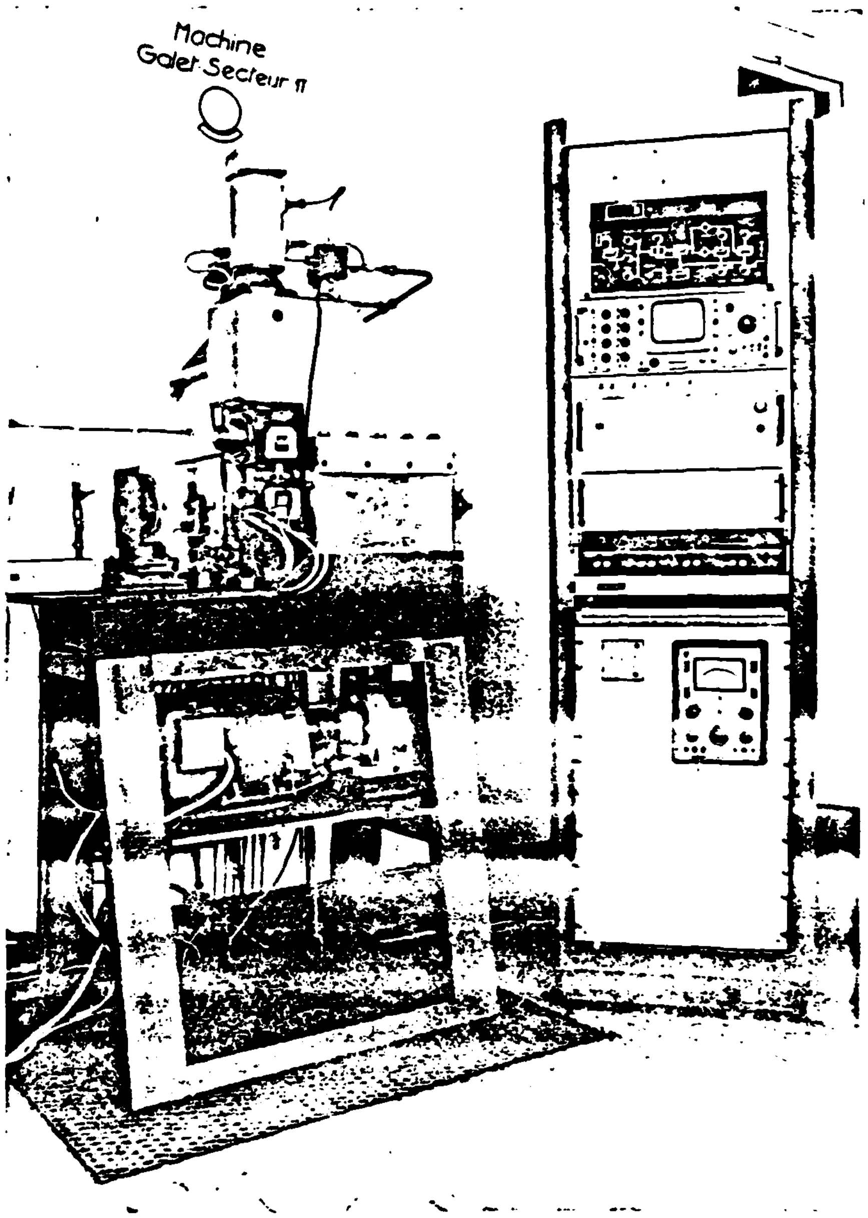


FIGURE 22 GENERAL VIEW OF THE APPARATUS

PARAMETERS		RANGE OF VARIATIONS
load	constant	N or $2N_o$ in Newtons; 0.1 1 10 10^2 10^3 10^4
	pulsed $N=N_o(1+\sin\omega t)$	ω rd/s; 0.1 1 10
contact pressure	constant pulsed	$P_{av} \times 10^7$ Pa; 0.01 0.1 1 5 10
movement	continuous	V m/s; 0.001 0.01 0.1 1 10
	oscillation $\alpha=\alpha_1\sin(\omega t+\varphi)$	$\alpha°$; 0 90 180 360; $\varphi°$; -180 0 180
geometry	diameter D	D: m; 10^{-3} 10^{-2} 10^{-1} 1
	L/D ratio	L/D; 0.2 0.6 1
	sector arc	$\zeta°$; 45 180 360
contact temperature	constant cyclic	T °C; -200 0 200
environment	air or various gases	oxydant air reductor; RH %; 0 50 100

TABLE 11 CONTACT PARAMETERS IN PLASTIC BEARINGS

AD-A121 036 TRIBOLOGICAL TECHNOLOGY VOLUME II(U) MECHANICAL TECHNOLOGY INC ANNAPOLIS MD P B SENHOLZI SEP 82 N00014-80-C-1736 3/4

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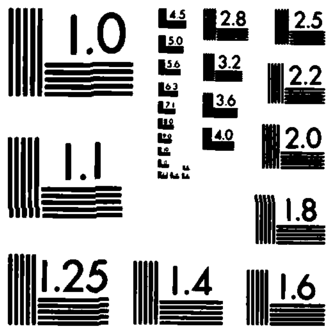
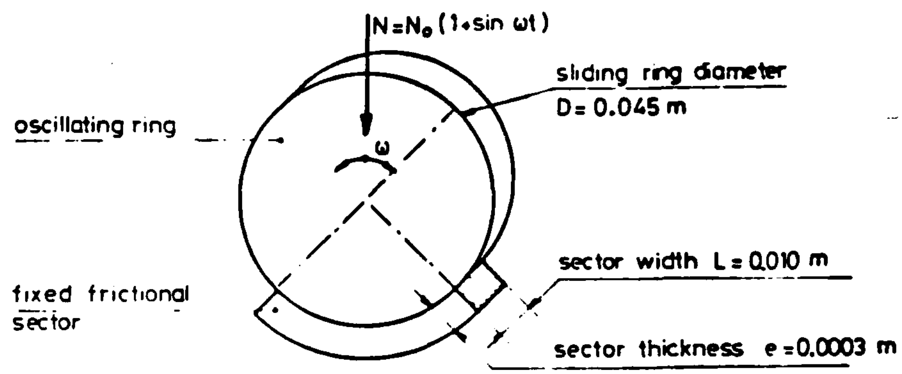

1.0
1.1
1.25
1.4
1.6
1.8
2.0
2.2
2.5
2.8
3.2
3.6
4.0
4.5
5.0
5.6
6.3



TEST CONDITIONS						
cyclic loading at 10 Hz	N_o (N)	660	1000	1330	1760	1930
	p_{moy} for $2N_o$ $\times 10^7$ Pa	0.41	0.62	0.82	1.09	1.2
maximum angular displacement α at 10 Hz	α (°)	6	12	18	24	
	V_{moy} (m/s)	0.050	0.090	0.140	0.190	
material for ring	440 C steel ± 60 HRc, 0.4 μm CLA					
thermal conditions	air cooled shaft, air cooled support					

TABLE 12 SIMULATION TEST CONDITIONS

cyclic load $N = N_0(1 + \sin\omega t)$ at 10 Hz		maximum angular displacement at 10 Hz: 6° $V_{moy} = 0.05$ m/s	12° $V_{moy} = 0.09$ m/s	18° $V_{moy} = 0.14$ m/s	24° $V_{moy} = 0.19$ m/s	
$660(1 + \sin\omega t)$	f	0.28	0.32	0.25	0.19	0.33
	W*	8.7	19.9	23.3	23.5	40.9
	θ °C	35	43	60	54	70
	$\dot{u}$**	1.18	0.74	0.41	0.665	0.79
$1000(1 + \sin\omega t)$	f	0.21	0.25	0.21	0.18	
	W*	9.9	23.5	29.6	33.8	
	θ °C	37	45	60	60	
	$\dot{u}$**	1.9	0.74	0.4	0.6	
$1330(1 + \sin\omega t)$	f	0.185	0.16	0.13	0.13	
	W*	11.6	20	24.4	32.51	
	θ °C	35	40	50	56	
	$\dot{u}$**	3.55	1.6	0.4	0.4	
$1750(1 + \sin\omega t)$	f		0.11	0.115		
	W*		18.1	28.4		
	θ °C		50	52		
	$\dot{u}$**		0.6	0.59		
$1930(1 + \sin\omega t)$	f				0.1	0.15
	W*				36.3	58.05
	θ °C				55	76
	$\dot{u}$**				0.82	[illegible]

material : teflon filled dacron weave cloth
*heat input (watt) - **rate of wear (x 10^3 $\mu m/m$)

TABLE 13 EXPERIMENTAL RESULTS

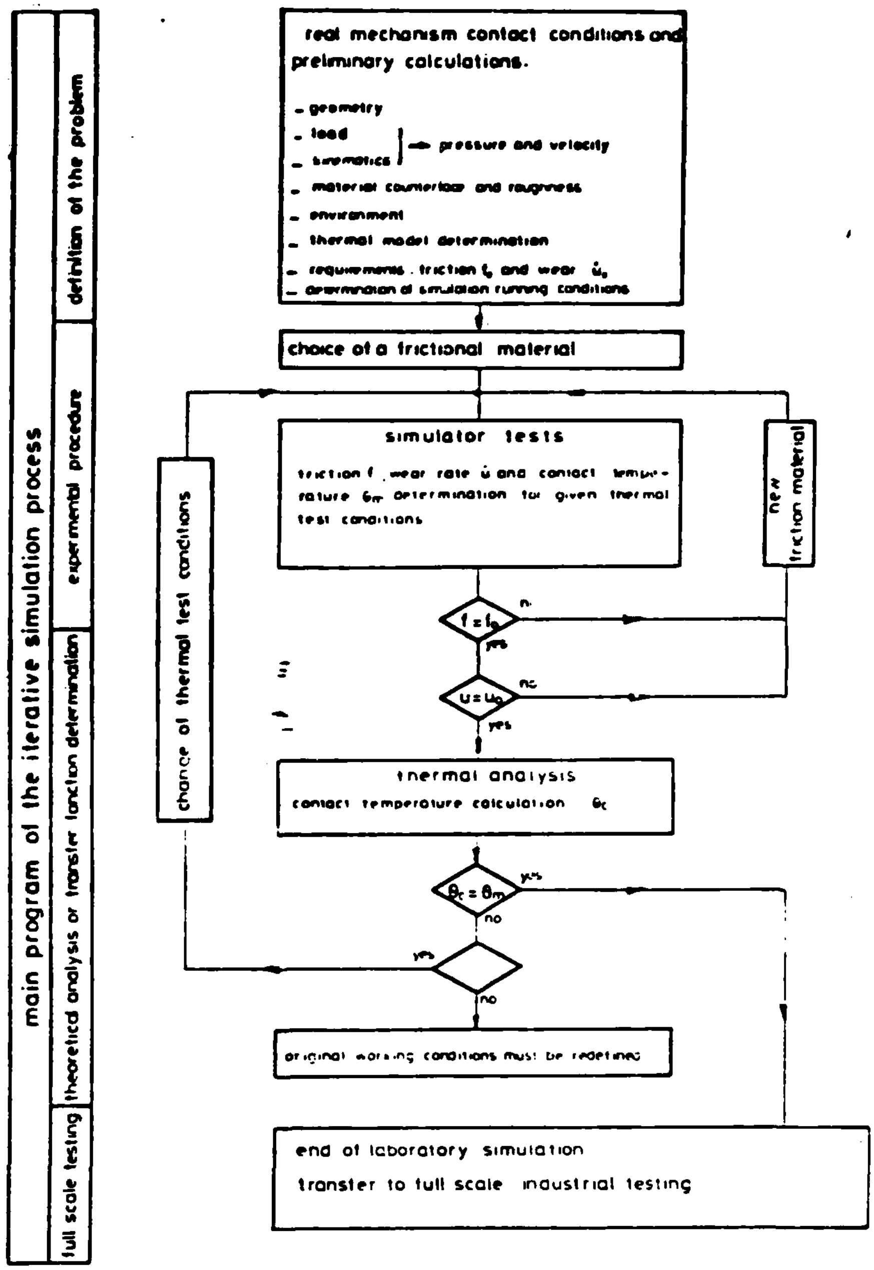


TABLE 14 SIMULATION FLOW CHART

(4) Simulator tests are run and the coefficient of friction, f, the wear rate, $\dot{u}$ and the contact temperature, θ_m, are recorded or estimated. The performances noted are compared with the requirements listed in step 1. If the requirements are not satisfied, a new material is chosen and step 4 is started again.

(5) If the requirements are met, the contact temperature, θ_c, in the real application is calculated for a representative value of the coefficient of friction measured in step 4. The values of θ_c and θ_m are then compared.

(6) If the calculated and the measured temperatures θ_c and θ_m differ significantly, the thermal condition in the simulator are changed and step 4 is started again with the new conditions. This process is repeated until both values agree. Experience shows that industrial applications have larger thermal masses than laboratory benches and as a result θ_m is usually larger than θ_c. Convergence however, can be obtained by cooling the test bench.

(7) If θ_m and θ_c cannot be made equal, the original working conditions must be redefined. If θ_m equals θ_c, the laboratory simulation is terminated and full scale industrial testing can begin.

3.3.4 Thermal Analysis of the Real Mechanism

Step 5 of the simulation flow chart, Table 14, requires that contact temperatures be calculated in the actual mechanism. The two-dimensional thermal model studies by finite elements, Figure 23, includes three elements: the inner shaft, the liner, and the outer bearing. In most applications, the liner is glued to the outer bearing and this disposition is introduced in the model for the calculation performed for simulation purposes. As such, the maximum temperature of 82° was calculated.

Returning to the simulation analysis, Table 8 shows that a maximum temperature of 76° was measured one millimeter below the contact during the tests. The value calculated in the real application at that distance is 75°. As the agreement between the theoretical and experimental values outlined in the simulator flow chart Table 14 is satisfactory, the laboratory simulation is therefore ended.

3.3.5 Full Scale Tests

Full scale tests were conducted under this regime. As expected, no detailed observations concerning friction and wear rate values were available. However, good agreement between

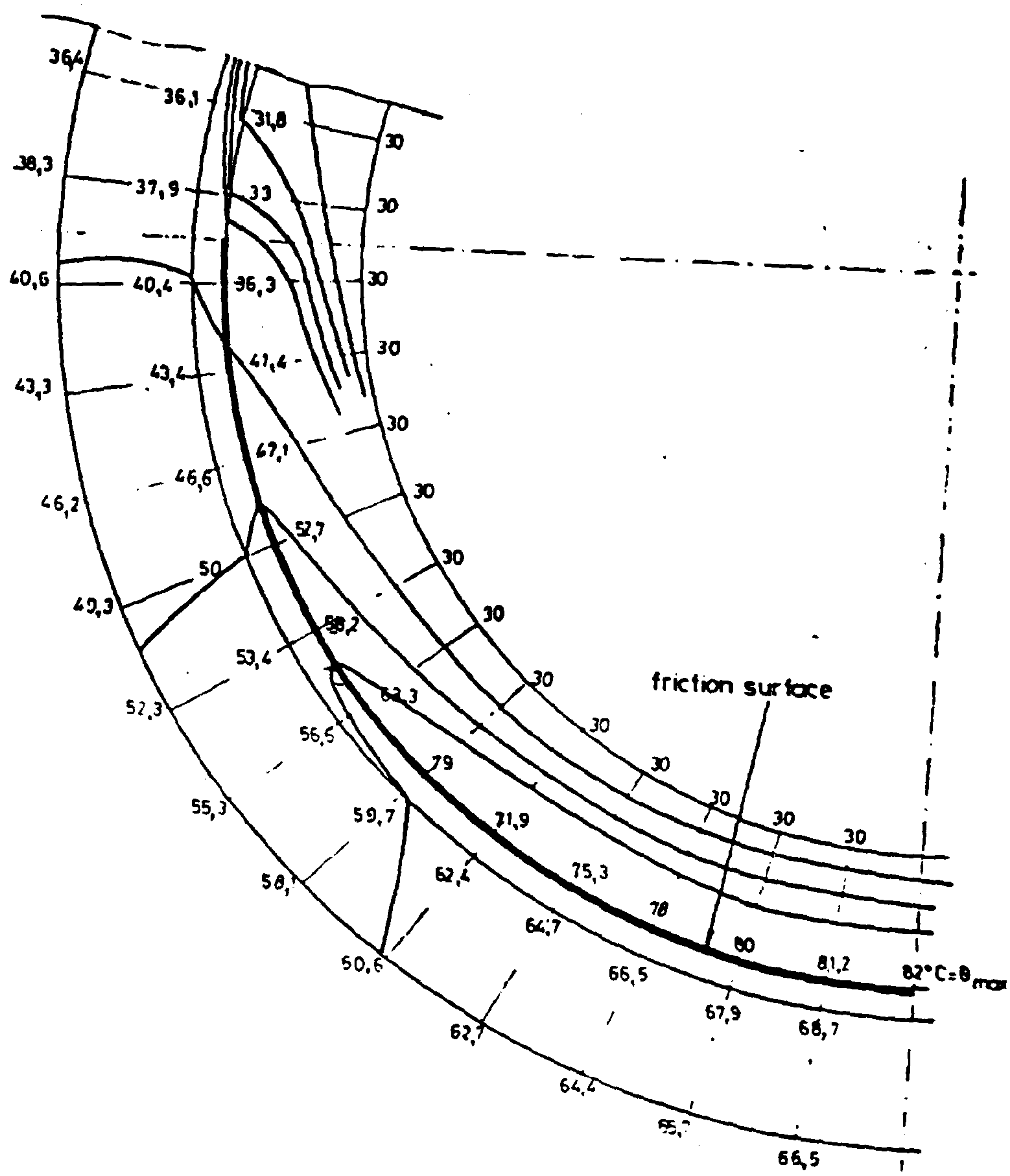


FIGURE 23 TEMPERATURE DISTRIBUTION WITH LINER GLUED ON OUTER BEARING (CONFIGURATION I)

measured and calculated temperatures was observed and the performance qualitatively confirms simulation test results.

3.3.6 Conclusions

Simulation based on the reproduction of contact pressures and temperatures, average velocity, type of motion, and contact geometry appears to give valuable results. As such, it constitutes a first step in the actual design of high performance bearings whose life performance could not be predicted earlier with any measure of certainty. The method suggested here is still cumbersome and can be improved. It is hoped that, as general calculation programs aand specific test data are accumulated, many steps in the iterative procedure outlined can be eliminated for each application. Further, computer aided design techniques are being tried in order to enhance the design process.

3.4 General Conclusion

Simulative testing has been shown to be capable of producing useful answers to industrial tribological questions when intrinsic coefficients, boundary conditions, and theories are not available. As such, it represents a very important link between fundamentals and application. The examples considered here have been chosen among the more complicated and sophisticated simulation attempts. This was the case because the information sought tends towards absolute performance prediction rather than comparative behavior. Clearly accurate absolute prediction can only be expected when Level I requirements are met. However, the case studies presented here can give an indication as to how well the governing parameters have been identified over a given range of mechanical conditions and thus serve to help isolate the factors which must be introduced eventually in a more general theory. To be efficient, from an industrial point of view, simulative testing must follow closely the advances in basic sciences in order to capitalize on their contribution. As such, tribotesting is a dynamic subject when progresses along with basic understanding.

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MONITORING

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1. INTRODUCTION

Many advances in engineering have been made by turning *failure into success. Thus, the history of* engineering in the past century and a half is in part a story of failure in service followed by improved materials and design incorporating greater reliability and less maintenance as a result of successful failure investigation[1]. However, in the present and foreseeable future world economic situation, prevention of failure in service is more beneficial than any lessons which can be learned from the failure investigation.

Most modern machines are complex and expensive with their vital parts totally enclosed. The practice of withdrawing such machines from service at periodic intervals for examination and maintenance to avoid failure, involves expensive dismantling. Means have thus been developed to assess machinery condition while in operation and allow predictive maintenance to be carried out when most convenient before the breakdown point is reached[2].

Condition monitoring is thus concerned with extracting information from machines to indicate their condition[3, 4]. This can be done by various techniques and the resulting information used for planning machine operation and maintenance in order to improve reliability, safety and economy of operation.

This discussion outlines the philosophy of condition monitoring, describes the various techniques available, reviews their application, compares their effectiveness and discusses economical aspects of their use.

2. APPROACHES

Machines can be run until they fail and then repaired. This is feasible if failure does not involve personnel risk and production losses and if spare machines are readily available and repair is relatively inexpensive. However, in many instances breakdown maintenance can be expensive in terms of lost output and machine destruction. In some situations it may be dangerous.

A better method is to stop machines at regular intervals based on past experience, for planned preventative maintenance in order to reduce the chance of unplanned stoppages through breakdown. A compromise however, is required between too frequent maintenance which reduces productivity and increases expense and too long an interval which involves the risk of some unacceptable failures in service.

Economically, a more acceptable method is to carry out preventative or on-condition maintenance at irregular intervals determined by the actual condition of the machine. Thus the main function of condition monitoring is to provide knowledge of machine condition and its rate of change which is essential for the successful operation of on-condition maintenance. The knowledge may be obtained by selecting a suitable parameter for measuring deterioration and recording its value continuously or at suitable intervals.

Assessing the trend of this measurement can provide a useful lead time in warning of incipient machine failure. The overall level of the condition monitoring activity appropriate to a particular plant or industry is usually decided by the potential economic savings.

Monitoring a complete plant necessitates a large number of measurements of numerous parameters and as such the cost of such monitoring systems is high. Complete condition monitoring is only applicable to very few major plants of strategic national importance. Only key machines in a plant may be monitored but this may also involve fairly expensive equipment and facilities. This method is generally applicable to most larger industrial establishments. Generally, monitoring a few critical components of key machines based on past failure experience may be adequate to avoid expensive unplanned maintenance and loss of production. This method is widely applicable in industry and can be readily used on existing equipment to show economic advantages.

3. TECHNIQUES

Although there are numerous techniques and a large amount of instrumentation available there are only four basic methods of machinery condition monitoring. These are visual, performance, vibration, and wear debris monitoring. There are two levels of assessment. The first determines that some wear or deterioration is occurring in a system or machine and the second determines the specific component of the system or machine which is wearing or deteriorating.

In visual monitoring, machine components are inspected for the direct observation of wear and change using microscopes, stroboscopes, thermography, dye penetrants, and X-rays. In performance monitoring, changes in machine performance such as output and operating parameters are measured to establish how well the machines are performing their intended duties. Parameters such as operating temperature and power losses are important. Vibration, noise, and shock measuring techniques serve to indicate wear, misalignment, and changes in component tolerances. A considerable selection of transducers, recorders, and signal analysis procedures are available. Wear debris monitoring involves examination of machine lubricants for the presence of entrained wear debris, contaminants, and lubricant degradation products. Wear debris can be detected directly, assessed by collection methods, or assessed by lubricant analysis using spectroscopy or Ferrography.

Monitoring systems may be classified into three categories; manual, semiautomatic, and fully automated. In a manual system, the data are collected manually, processed manually, and displayed manually by reports or graphs. Manual systems are simple to implement and are relatively inexpensive to design and operate. For a small operation seeking limited performance monitoring objectives, a manual system can be a cost effective solution. One objection to a manual system is the added work load it imposes on the observer who normally has a primary responsibility other than monitoring which can result in missing or inaccurate readings. Some degradation in the quality of the data can occur due to inaccuarcies inherent in the processing method. Tedious smoothing or statistical analysis techniques which can improve data analysis, are usually avoided. Manual data systems must deal with essentially steady state phenomena since a high probability exists that transient or rapidly varying data will not be observed or will be missed due to human limitations or to the simple instrumentation which may not be capable or accurately following such data.

Semiautomated systems combine manual and automated capabilities. Usually the manual portion is data collection, transcription and conversion to a form compatible with the automated portion of the system which handles data storage, processing, and output or display of the processed information. There is usually a time-lag inherent in semiautomated systems between data collection and data output by a central location. Monitoring is therefore aimed at trending performance degradation and/or identifying anomalous behavior. A semiautomated system is easy to design and operate but exhibits a problem, in common with the manual system, of the inaccuracy inherent in manual reading and recording of instruments. A benefit of computer processing in a semiautomated system is the ease with which the data can be organized for comparative purposes.

A full automated system from data acquisition through data display is capable of gathering and processing high amounts of data, handling signals of all types, maintaining a high level of accuracy, and of processing and displaying data in real time. The computer can be used to perform selective recording using software programs to accomplish data compression. Automated monitoring represent the ultimate in capability but the cost of such systems is high and the analysis software requires considerable development.

4. HARDWARE

Machinery condition monitoring requires hardware that falls into four categories; transducers, electronic equipment, display or output devices, and interface hardware[5].

Transducers perform the task of measuring machine parameters and converting the information to an electrical signal proportional to the value measured. They are thus a basic building block in any machinery condition monitoring system. Electronic hardware conditions transducer output signals to a common format, processes the signals according to a predetermined decision sequence, and selects information for recording or output. Display/output devices provide information in the form of flags, lights, instruments, recordings, or print outs for machine safety, diagnostics, or performance trending. Interface hardware consists of equipment for troubleshooting or for remote data processing on a more extensive basis for trending or storage.

Transducers used for monitoring, convert measured parameters such a speed, pressure, and temperature into proportional electrical signals. Selection of the parameters necessary to accomplish the established objectives of the condition monitoring system determines the basic transducer types required. Transducer selection is based on several variables. Transducer accuracy is critical to the accuracy of the condition monitoring system, as system accuracy will always be less than transducer accuracy. Thus, system requirements determine the transducer accuracy required. Systems reliability is only as reliable as transducer reliability, so that selection of transducers of the required reliability is of prime importance. The environment in which the transducer is required to operate also determines the required design specification. Transducer cost is a direct function of the required environmental, accuracy, and reliability specifications.

4.1 Temperature Measurement

The operating temperature of a machine is a useful indication of the effectiveness of its operation. For example, the exhaust gas temperature (EGT) of a gas turbine engine is an important control parameter.

There are several methods generally used in measuring temperatures in a condition monitoring system. These include thermocouples, thermal resistance temperature sensors, bi-metallic switches, and radiation pyrometers. The most commonly used sensor is the thermocouple which operates on the basis of the electro motive force (EMF) developed in a circuit of two dissimilar metals. The EMF generated is proportional to the temperature measured. Thermocouples are standard equipment, are reasonably linear within a given range, are stable, and are capable of measuring wide temperature ranges. Thermocouples may be connected in parallel so that if one fails, the average temperature can still be measured with little loss of accuracy. The disadvantages of thermocouples are a small signal level in the order of millivolts and the necessity for cold junction compensation. Thermocouples are usually supplied in configurations with distinct temperature ranges for each configuration, Table 1.

Table 1

Temperature Ranges for Some Typical Thermocouples

Thermocouple Material	Temperature Range
Copper - Constantan	-300° to 700°F
Iron - Constantan	0° to 1400°F
Chromel - Alumel	0° to 2300°F
Platinum - Rhodium	0° to 2700°F

In the thermal resistance sensor, the resistance element produces a change in electrical resistance with respect to temperature which is measured with some form of Wheatstone bridge. Resistance temperature sensors may be either metallic or semiconductors. Some typical temperature ranges of thermal resistance sensors are given in Table 2.

Table 2

Temperature Ranges for Some Typical Thermal Resistance Sensors

Resistor Material	Temperature Range
Platinum	-450° to 1200°F
Copper	-325° to 300°F
Nickel	-100° to 300°F
Thermistors	-100° to 500°F

Metallic resistance sensors are linear, have a high signal level, and are very stable. However, maximum temperature range is the limiting factor. A semiconductior resistance sensor or thermistor has advantages of a high signal level and low cost. The disadvantages are a narrow measurement range, nonlinear output, and lower accuracy than a metallic resistance sensor. Thermistors also suffer some of the inherent problems of semiconductors such as difficulty in control of operating characteristics in batch manufacture. They are not used where a high degree of accuracy is essential.

Bi-metallic switches may be used where a discrete signal is required. They have the advantage of accurately switching within a very limited temperature range, require no signal conditioning and are usually inexpensive. They are used mainly for specific applications.

Radiation pyrometers utilize the intensity of emitted radiation as a measure of temperature. They consist of an optical focusing sensor head for collecting the radiant from a selected area typically .1 inch diameter, steel sheathed flexible fiber optical cables, a silicon detector and necessary signal conditioning. The advantages of radiation pyrometers include high accuracy of temperature measurement, rapid response to temperature changes, and a high temperature range. The disadvantages are susceptibility to lens coking in certain environments and complicated signal conditioning requirements.

They are potentially attractive for gas turbine blade temperature measurement[6, 7].

Other methods of temperature indication range from simple, inexpensive temperature indicating labels and paints, low melting point metals, hardness changes in selected alloys to thermography, or thermal imaging[8, 9, 10]. The latter technique converts the infrared radiation picked up by a scan of the surface with an infrared camera into a proportional electrical signal.

The signal is amplified and displayed on a cathode ray tube or recorded on photographic film.

4.2 Pressure Measurement

Performance monitoring usually requires accurate pressure measurement so that pressure transducers are common to many condition monitoring systems. Pressure transducers may be passive or active. A passive pressure transducer, such as a metal foil or semiconductor strain gauge requires excitation for output whereas an active pressure sensor generates an output voltage without excitation.

In a pressure transducer, the strain gauge is connected into a Wheatstone bridge circuit and the strain on the strain gauge produces a resistance proportional to the pressure displacing it. The most commonly used pressure transducer, the metal foil strain gauge, provides very accurate measurement and has low sensitivity to thermal effects, shock, and vibration. It is capable of either static or dynamic measurement with A.C. or D.C. excitation and has continuous resolution. The only disadvantage is a relatively high cost when designed to meet aircraft specifications. A compromise is often required between high resolution, durability, and capital cost. Semiconductor strain gauge pressure transducers possess high sensitivity and are smaller and cheaper than metal foil strain gauges but their use is limited owing to their poor reproducibility, reliability, and accuracy due to temperature effects.

The only active pressure transducer used, piezoelectric, is based on the principle that asymmetrical crystalline materials produce an electrical potential on the application of strain or stress. The most widely used crystals are quartz, tourmaline, rochelle salt, and barium titanate. Piezoelectric crystals are most widely used in accelerometers. When used in pressure transducers, they can only be used for dynamic measurement as current is generated only under dynamic loading of the crystal. They are mainly used where accuracy in dynamic response is only required. They are usable to high temperatures and are very rugged. Their static response is poor and requires a change amplifier to render the signal usable.

4.3 Vibration Measurement

For vibration measurement it is important to consider not only the vibration pick ups but the monitoring system as a whole. The overall vibration system objectives must be finalized prior to transducer selection. Software capabilities should indicate the final system requirements. Capability may range up to a full diagnostic program able to fault isolate to component or plant level. This capability dictates the complexity of the hardware required to monitor vibration.

In any system, other than a gross overall displacement check, some frequency filtering of transducer outputs is required in the form of band pass filters in the signal conditioning or software capability to provide such filtering. Overall objectives dictate filtering requirements. It may be desirable to concentrate on one or two octave bands in the frequency spectrum associated with machine rotational speeds or to filter many bands to diagnose the complete operational spectrum of machine vibrations. Filtering also eliminates bands not associated with the machine frequency spectrum. The transducers used in vibration monitoring are velocity pick ups and accelerometers, depending upon system requirements.

Although the degree of accuracy available is relatively high the inherent design of velocity pick ups presents disadvantages. Vibration is measured by the relative motion of a coil with respect to a magnetic field and as it is a mechanical device it is subject to wear and reliability problems. Velocity pick ups are unsuitable for the higher frequency ranges (4000 + Hz) due to a relatively low natural frequency. It is usually necessary to integrate the output to reduce the effects of signal noise and provide the displacement. This limits the use of velocity pick ups to an overall gross system check rather than a more sophisticated analysis.

Accelerometers are constructed of a stack of piezoelectric crystals in compression with a mass mounted on the stack. An output is generated when the accelerometer is subjected to vibration, but the low level output requires substantial amplification to obtain a usable signal. Advantages of the accelerometer include no moving parts, high reliability, and a high natural frequency. Integration of the accelerometer output provides a velocity measurement for more sophisticated diagnostic work. A disadvantage is the low level output which necessitates shielded cabling to reduce interference from extraneous noise on the vibration signal. Higher output accelerometers are being developed to increase signal to noise ratio and to reduce the effects of noise. Some very sophisticated condition monitoring systems have been developed based on gas turbine engine parameter interrelationships[11].

4.4 Position Transducers

A potentiometer is a variable resistance transducer used to measure displacement. In aircraft it may be used to measure fuel control settings, engine and airframe displacements. Potentiometric transducers which have a large electrical output and are available for any desired displacement range, are inexpensive and require simple signal conditioning. A major disadvantage is that mechanical contact renders the potentiometer subject to wear and vibratory stresses which curtail the useful life.

A synchro is basically a variable transformer which provides an electrical output proportional to the angular displacement of its shaft. An alternating current excitation is provided to the synchro and the phase relation changes as the shaft is rotated. An advantage of a synchro over a potentiometer is that brushless construction eliminates mechanical contact; however, signal conditioning requirements are more complicated than for a potentiometer.

A linear variable differential transformer (LVDT) used to measure linear displacements is an inductance transducer which produces an electrical output proportional to the displacement of a moveable magnetic core. A primary coil and two secondary coils surround the core. These secondary coils symmetrically spaced on either side of the primary coil are connected in a series-opposing circuit and motion of the magnetic core varies the mutual conductance of each secondary coil to the primary coil which determines the voltage induced in the secondary coils. If the core is central, the voltage induced is identical and 180° out of phase so there is no net output. If the core moves off center, one secondary coil has a greater voltage induced. The output of an LVDT is linear within its range of displacement and there is no physical contact between the coil and core so that there is no deterioration by wear. An LVDT is not affected by mechanical overload and has excellent dynamic response due to a low core mass and the absence of friction but it requires complex signal conditioning.

4.5 Lubricant Monitoring

Lubricant monitoring provides a means of detecting abnormal conditions of oil wetted components. Magnetic chip detectors, screens and oil filters with pop out buttons to indicate an excessive oil pressure drop are inexpensive methods. Chip detectors and screens are checked periodically to determine particulate concentration. Some magnetic chip detectors contain an electrical circuit which is completed when a metal particle makes contact with the magnet to provide a signal that there is an abnormal lubricant condition. The disadvantage of these methods is that they do not provide continuous monitoring. They must be properly located in the lubrication system and may not provide a warning in sufficient time to allow remedial action to prevent secondary damage.

An on-line lubricant system monitoring method has been developed based on the principle of light scattering for particulate debris detection and light attenuation for chemical or thermal lubricant degradation. As the lubricant passes through the transducer it causes a rotor to turn. The rotor contains fluid passages and optical references which are alternatively inserted in a light measuring system as the rotor revolves. The optical paths utilize sealed fiber optics to conduct light into and out of the fluid and to change its direction. One photo sensor is mounted radially to view the light beam at 90° to provide the scattering output. The attenuation sensor views the axial component of transmitted light. The output of each sensor is a series of pulses alternating between reference and signal. The output requires extensive signal conditioning which is expensive.

5. WEAR DEBRIS ANALYSIS

As the history of a wear process is recorded in the debris produced, an attractive method of monitoring the condition of machinery is by the careful analysis of wear debris and contaminants in the lubricant used[12]. By measuring the quantity and observing the nature of the wear debris, it is possible to obtain an indication of the condition of the various machine components which are lubricant washed. As the failure of load-carrying lubricated surfaces is usually a slow progressive process, wear debris analysis allows advanced warning of surface deterioration towards failure.

Wear debris monitoring may be carried out by direct detection using inductive or capacitance detectors and filters which detect the presence of conducting debris. Monitoring may also be carried out by debris collection using removable magnetic plugs, filters, and centrifuges. However, wear debris analysis is usually applied to lubricant samples. Elemental analysis of lubricant samples may be carried out using spectrographic oil analysis procedures (SOAP). Wear particle analysis is carried out by particle counters and Ferrography. Great care is necessary with oil sampling. A sample of lubricant should be taken, if possible, from the machine or system while it is in operation or in less than two minutes after stopping in order to ensure that most particles are still in suspension.

5.1 Spectrographic Oil Analysis Procedure

SOAP may be carried out using either atomic absorption or atomic emission spectroscopy[13, 14]. The atomic absorption spectrophotometer which costs less than other spectrometric methods operates on the principle that atoms absorb only light of their own specific wavelength. The lubricant sample suitably diluted is vaporized in a flame and each element required, determined separately using a source lamp which emits light of wavelength characteristic of the element. Light is absorbed on passage through the flame and comparison with a reference beam allows a direct measurement of element concentration. Standards of known element concentration are used for instrument calibration.

The atomic emission spectrometer measures the characteristic wavelength of light emitted when elements are excited by an electrical discharge. Usually a small quantity of lubricant picked up on the periphery of a rotating graphite disc is vaporized in an electric discharge to promote emission of spectra by metallic elements present. In a direct reading instrument, photomultipliers aligned to spectral lines of interest are used to measure element concentrations simultaneously. A print out of element concentration in parts per million by weight reduces test time. Complex calculations are necessary and spectral interference may require compensation. However, the method is very useful for large numbers of specimens handled by a central control unit. A big disadvantage of SOAP is that it is blind to large particles which can be most dangerous in precision machinery[18].

X-ray fluoresence by exposure of a medium to a source of irradiation can detect the same elements as atomic absorption plus phosphorous, sulphur and chlorine. Although expensive it has considerable potential as an on-line continuous elemental concentration detector for very critical systems.

5.2 Particle Counting

Particle counting can be carried out using atomic counters such as the HIAC, Coulter, and Quantimet. The HIAC and Coulter counters provide a detailed particle size distribution in a fluid by counting particles individually and then automatically totalling. The HIAC counter works on a light blockage system and the Coulter counter by electric pulses caused by the passage of particles through an electrolyte.

The Coulter counter is more involved than the HIAC as the latter was specifically designed to analyze hydraulic fluids whereas the former was developed for blood cell counts[16].

Quantitative study of particles collected on filters or ferrograms can be carried out with the Quantimet image analyzing computer which can be used to discriminate between metal, oxides and polymeric material by light contrast relative to background. Numerical outputs include numerical concentration, particle size, ratio of length to width, and statistical distribution of particle size. Simple statistical parameters provide a clear indication of wear transitions occurring in multi-component systems[17].

A new area of particle detection in condition monitoring form the gas stream of a gas turbine is by electrostatic discharge[18, 19]. The electrostatic discharge is due to metal debris oxidation (burning) and metal rubbing or pitting. Metal rub produces a positive charge and metal burning (over temperature) a negative charge. Monitoring was first accomplished with a metal rod in the gas turbine tail pipe, but it has been found that a combination of an electrostatic wire grid and metal ring provided a much better indication of turbine engine distress due to monitoring the entire exhaust instead of just a portion by a rod.

A short review of methods of examination of debris and lubricant contaminants has been published[20].

5.3 Ferrography

Ferrography is a method of recovering particles from a fluid and depositing them on a substrate according to size and magnetic susceptibility for analysis[21, 22, 23]. The direct reading (DR) Ferrograph, a simple instrument for use at plant or depot level, is used to determine the amount and size distribution of wear particles in a sample of lubricant from which significant data can be derived. Use of a simple equation provides a single figure for the severity of wear or other index and experience can determine ranges to allow a code of monitoring practice to be established. Full Ferrographic analysis using the bichromatic microscope, electron microscopy, heating techniques, and Quantimet can be used to supplement the information[22, 24, 25]. Trend analysis has proved to be potentially attractive[26].

Particles generated by different wear mechanisms have characteristics which can be identified with the different wear mechanisms[21, 24]. Rubbing or adhesive wear particles found in the lubricant of most machines have the form of platelets and are indicative of normal permissable wear, Figure 1. Cutting or abrasive wear particles take the form of miniature spirals and loops, Figure 2. An accumulation of such particles is indicative of a serious abrasive wear process. Particles consisting of compounds can result from an oxidizing or corrosive environment, Figure 3. Steel spherical particles, Figure 4, are a characteristic feature associated fatigue crack propagation in rolling contacts[27]. Specific regimes of wear have been classified by the nature of the particles produced by surfaces in sliding contact[28]. Different mechanisms such as rolling bearings, gears, and sliding bearings produce distinctive particles. An atlas of such particles is available[29]. Atlases of characteristic nonmetallic and nonferrous particles are in preparation.

The condition monitoring of nonlubricant washed, inaccessible components by the extraction of particles from exhaust or gas streams has been investigated[30]. Lubricants, lubricant additives, lubricant contaminants, and friction polymer and their influence on machinery conditions can be monitored[21]. Solvents are available to fluidize grease to prepare ferrograms. An on-line Ferrograph can be incorporated in a machine or system to allow continuous condition monitoring. Wear debris in a lubricated systems if isolated by a high gradient magnetic field and quantitative measurement of the debris is achieved by a surface effect capacative sensor.

Ferrography has been successfully applied to the conditions monitoring of natural and prosthetic joints. Particles retrieved from the synovial fluid or saline washings of joints by centrifuging are treated with a magnetizing solution containing erbium ions and the resulting suspension used to make ferrograms[31]. Scanning electron microscopy, in conjunction with X-ray energy analysis, is a powerful tool for particle identification[32]. The ability to monitor the condition of prosthetic joints ferrographically should aid the development of implant materials and the design of artificial joints. Ferrographic synovial fluid analysis should augment the understanding or the etiology and pathogenesis of degenerative arthritis and provide a method for the diagnosis, documentation and prognostication and treatment of the disease.

Monitoring of the content of nonmagnetic or nonmetallic materials in fluids can be effected by the use of special fluids which contain both lubricant solvating solvents and water miscible solvents to dissolve the lubricant and allow magnetic salts to become absorbed on the nonmetallic or nonmagnetic wear particles so that they can be precipitated by the Ferrograph.

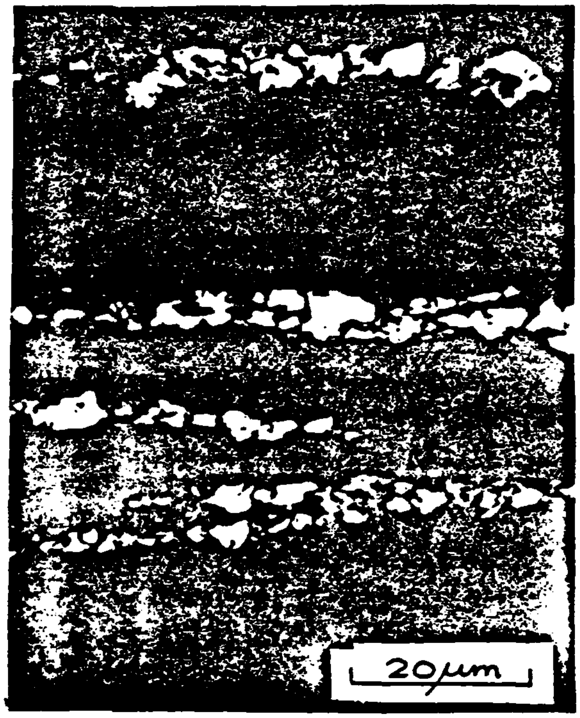

FIGURE 1 RUBBING WEAR PARTICLES

FIGURE 2 CUTTING WEAR PARTICLES

FIGURE 3 SPHERICAL PARTICLES

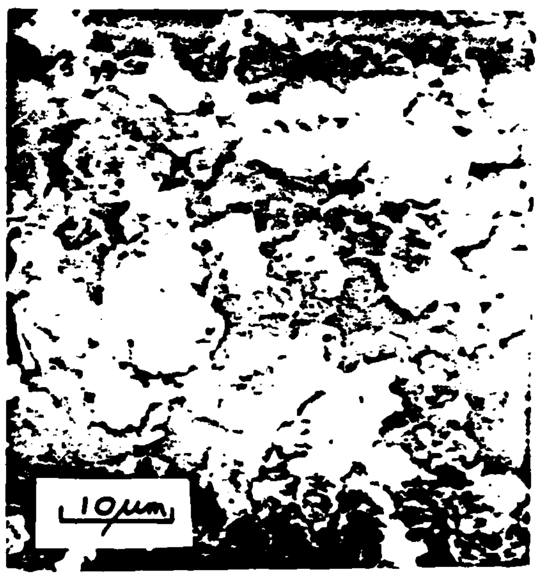

FIGURE 4 OXIDIZED PARTICLES

6. THE SIGNIFICANCE OF MONITORING

Overall economic benefit is usually the most important consideration affecting the decision to apply machinery condition monitoring and in particular the benefits arising from the reduction of output losses and maintenance costs. Energy and fuel conservation are also an important consideration as a machine such as a gas turbine will consume considerably less fuel to perform its required duty than a machine which has suffered deterioration. Prevention of failure also conserves material required for new components and the energy required to extract, refine, and fabricate the materials of construction.

The application of condition monitoring is essential where a safety risk is likely to arise, such as in plant handling dangerous materials and machines for the transportation of people. Monitoring is also desirable where accurate and advanced planning of maintenance is essential for normally inaccessible continuously operated equipment.

Condition monitoring enables the early detection of faults while damage is still slight in plants of recent design which may have development problems. It can provide useful information to guide design improvements[33]. Where operators cannot be expected to detect faults in expensive or sophisticated vital equipment, condition monitoring enables incipient faults to be detected to allow remedial measures before expensive failure or expensive consequential damage occurs. Condition monitoring may be significant when a plant manufacturer or lubricant supplier can offer a service to users of equipment thus reducing the cost to each and allowing a useful feedback to prove product design and development and the formation of improved lubricants and additive packages.

Machinery condition monitoring may also be affected economically by the use of equipment already available for quality control, process control, or servicing requirements.

7. EFFECTIVENESS OF MONITORING

Benefits which can be derived from the successful monitoring of the condition of plant and machinery include increased availability resulting in increased output from the capital invested and reduced maintenance costs. Machine running time can be increased by maximizing the time between overhauls by safely switching from regular periodic maintenance to on-condition maintenance. Overhaul time can also be reduced as the nature of the problem or the specific module in trouble for complex modular constructed machine is known so that spares required and essential personnel are available. If damage consequential to initial damage is avoided, considerable savings in time and capital can be effected. The storage of spares and the expensive tying up of capital can be eliminated if advanced warning of requirements is known to enable procurement of the spares in time.

Machinery condition monitoring can allow more efficient plant operation by controlling output and operating life to the best compromise. Recorded experience of the operation of most machinery aids the specification of improved design of future machines.

8. THE ECONOMICS OF MONITORING

If safety is the primary factor, then machinery condition monitoring is essential no matter what the cost. However, machinery conditions should generally only be considered where savings are considerably in excess of the cost of monitoring. Condition monitoring is most cost effective with capital intensive plants utilizing continuous operation of high cost machinery. It is not likely to be effective for lots of small cheap machines where ample spare capacity is readily available.

The main source of financial savings which can be made from machinery condition monitoring in an industrial establishment, are a reduction in the loss of production due to breakdown and the cost of maintaining plant and machinery. As a rough initial guide, industries are likely to achieve savings of the order of 1.2% of their added value output calculated from total sales revenue minus the cost of raw materials and energy brought in[3]. With expensive machines, which are expensive to maintain, it is essential to delay overhaul as long as possible but to avoid expensive consequential damage from simple intial wear.

To illustrate potential cost savings by effective machinery condition monitoring and the safe switch from periodic routine to on-condition maintenance, the cost of overhauling a modern large jet engine is approximately 12½% of its initial capital cost of £½M. It has been reported that a simple bearing failure in a fully integrated steel mill can lead to a total shutdown which at full output rate may cost up to £300 per minute[34, 35]. A similar bearing failure in a modern generator set could involve Central Electricity Generating Board in a loss of up to £20 per minute till the set was again operational. This being the difference in the cost of generating electricity with a large modern generator and smaller stand by equipment. A similar bearing failure in the U.S.A. has been quoted to cost $28,000 per day[36]. It has been reported that the total cost of wear for a U.S. Naval aircraft amounted to $243 per flight hour and of this total $140 was due to unscheduled maintenance[37]. Although SOAP has often been criticized, the U.S. Defense Department spends £40M on oil analysis to predict only certain types of failure in one power system, the aircraft gas turbine, but this expenditure effects savings of twice this amount in terms of direct repair costs[36]. A high safety factor is also achieved.

Owing to environmental problems, it has not been possible to replace some older paper mills and so the aging plants are being worked continuously at full or overload to maintain the required output. In one mill failure of a roller bearing on a main roll results in total shut down involving a loss in overall profit of £100 per hour. As it can take up to 24 hours to effect repairs and experience has shown that in other mills, of the 75 vital bearings, some 25 failures are experienced per year. Prevention of even one failure by monitoring would be cost effective, but an efficient monitoring system may be expected to prevent most of this failure and unscheduled maintenance.

9. SELECTION OF A MONITORING SYSTEM

The selection of an appropriate method for monitoring the condition of a machine must be based on a consideration of which of its many components are most likely to fail and in what way. Possible methods for monitoring these components can then be considered. It may be possible to choose a single method by which all likely failures of components in a machine may be detected with acceptable efficiency. In many instances however, it is necessary to use many techniques as specific information from each technique allows comprehensive data to be accumulated from which a suitable lead time to failure can be established before any sequential damage has occurred. For instance in Germany, concern had been expressed regarding the use of a single method of aircraft engine condition monitoring particularly in the light of experience[38]. Deterioration of vital components had occurred and sometimes not been detected so that expensive failures in service had resulted. Equally expensive had been engine removal on the basis of SOAP results when no deterioration had been found. As in many cases, it is difficult to make a correct decision regarding engine removal on the basis of SOAP alone. Ferrography has been used to advantage to supplement SOAP. Experience with the use of both techniques has indicated that ferrographic analysis can detect the incidence and build up of wear debris too large to be detected by SOAP and still too small to be detected conveniently by MDP. The use of three techniques cover the usual size range of wear debris, as shown in Figure 5, to prevent failure and false alarms by SOAP and to prevent secondary expensive damage by particles large enough to be picked up by MDP[39].

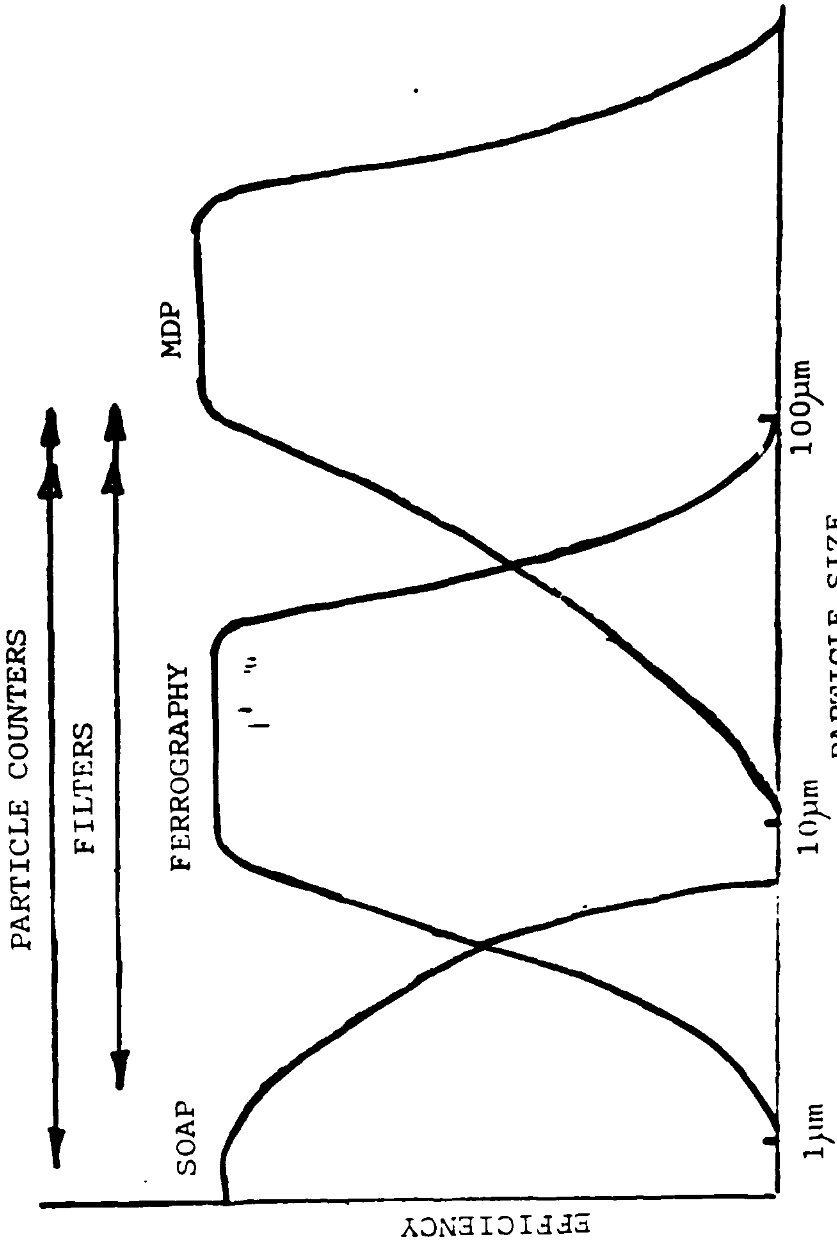

FIGURE 5 PARTICLE SIZE RANGE EFFICIENCY OF DIFFERENT TECHNIQUES OF WEAR DEBRIS ANALYSIS

The monitoring of nonlubricated vital components, such as turbine blades and discs of a gas turbine, has proven to be a difficult problem. Although Boroscope examination is currently used, it is difficult to carry out effectively. Over the past few years there has been a gradual acceptance of vibration analysis, although this has proved to be neither the simplest nor the most effective method to use. There is the problem of interpreting the data caused in part by the fact that the vibration signal tells as much about the general dynamics of the machine as its damaged state. Modern software, to effectively separate the information, requires extensive investigation and development and thus is expensive.

10. CONCLUSIONS

For machinery condition monitoring, the simplest method providing the accuracy of prediction required should always be selected. The simplest method is usually the cheapest to purchase and the easiest to install and operate. The more sophisticated the equipment used, the less reliable is it likely to be and generally the more prone to misuse and misinterpretation. Sophisticated equipment also requires specialist operators and maintenance.

The time taken to interpret the result of monitoring after measurement has been completed, must be less than the failure propagation rate. Where possible, the monitoring should be controlled by the person responsible for making the maintenance decision and thus monitoring on the flight deck of an aircraft, on board a ship, or in the control room of an industrial plant is preferable to control by a remote central laboratory. Continuous monitoring is preferable to periodic measurement and alarm signals can be beneficial even with fully automated systems.

The best method of monitoring is usually derived from long experience arising from the monitoring of the particular machine. An optimum economic service life may be determined for many expensive to operate machines. For example, with the present escalating trend in fuel price, keeping a jet engine in service too long may involve extra expensive fuel consumption with a worn engine together with eventual expensive overhaul due to extensive replacements caused by overall general long term wear. Withdrawal at the optimum service life may result in a less expensive overhaul with a return to improved fuel consumption. The optimum period will probably shorten with increase in fuel cost compared with increase in maintenance costs.

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TRIBOLOGY: THE MULTIDISCIPLINARY APPROACH

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1. THE ROLE OF THE DISCIPLINES

1.1 The Importance of Tribological History

As with many other aspects of technology, the rate of expansion of research,—innovation, analysis, synthesis, and product development in Tribology has probably followed some form of exponential curve if such expansion is to be quantified, however recondite the connection, in terms of the rate of production of technical papers in the field.

On this basis, it is highly improbable that any single individual could be conversant either with the scope of the whole spectrum in depth or offer a prognosis on the detailed ramifications resulting from the dissemination of such information on current and future tribological evolution.

Bearing this in mind, however, it is clearly important that the individual tribologist does not extend this proposition to the point where he will eschew any attempt to view the subject as a unified entity, merely because of the extent and variety of the available material. Indeed, it is the opinion of the author that there is a growing tendency in technology in general towards the cultivation of the individual and specific tree, unfortunately at the expense of the wood, and that this philosophy, especially in the field of Tribology, could have consequences which might be litte short of disasterous if the results of the manifest efforts in the field of research are not widely implemented at the grass roots of industrial utilization.

In the present world climate of limited natural resources and rapidly shrinking energy reserves, it is the prime requisite, if not the moral duty, of the industrial designer to employ as far as possible, both the cropus of information which has been placed at his disposal and the specific resources available to him in his particular milieu.

The question, however, remains as to how such a goal is to be accomplished, bearing in mind not only the multiplicity of the applications of the subject, but also the interdisciplinary nature of the complete tribological spectrum in the context of its current utilization.

The human mind is, however, a flexible system, highly developed in a role of adaptation to ever-changing situations. Fortuitously, also, it relies on its experience and especially on precedential information in framing both its present problems and establishing its future solutions. In this, like the development of language itself, it relies heavily on the chronological experience of other minds, for in the knowledge of what has gone before lie the seeds of what will ultimately follow.

It is to this hierarchy of past experience, therefore, that we must initially turn to establish the foundation of our study of the interaction of the various disciplines, in order to place them both in historical time sequence and to present specific developments in what, with the benefit of hindsight, we may justifiably call a 'temporal perspective', remembering always that we stand as a judiciary with this perrogative of time denied, both to ourselves in our current problems and to our tribological antecedents when they were coping with theirs.

Indeed, the author considers it of paramount importance if any fundamental understanding of the interdisciplinary nature of the tribological spectrum is to be achieved, that the seeker after such insight turns to the process of tribology history itself.

By this process, he will firmly establish a broad foundation on which any conceptualized integration of the various disciplines within the tribological framework must necessarily rest. For this reason, the historical aspect will be treated in some depth.

1.2 Development of Tribology From 1850 to 1925

Historically, the 'Age of Steam Power' is considered by many text-books to begin around the year 1850 and to continue unabated until the end of the century[1]. Undoubtedly, the period saw a great upsurge of interest in the training and education of engineers in all the industrialized countries of the world. Technical education flourished and academic chairs were established in many branches of engineering, particularly in the U.S.A., U.K., and the European continent and Russia. In the light of this, it is to a certain extent understandable why prior to 1850, lubrication, per se, had received little or no attention - the demand for bearings capable of coping with heavy duty while producing low frictional loss, simply did not exist.

As steam power became more efficient and bearing loads and speed escalated, problems inevitably arose with bearing failures, either through rapidly diminished life or of a more catastrophic nature.

The coming of the railways introduced bearings for locomotives and rolling stock in large quantities with the concomitant problem of their efficient lubrication. Indeed, it was the 'father of the railway', George Stephenson, who became the first president of the Institution of Mechanical Engineers when it was first formed in England in 1847.

It was not, therefore, entirely fortuitous that around the 1850's several of its members were publishing papers on the efficient lubrication of railway axle boxes. Mineral oil was already being produced for use in the Lancashire cotton mills and its use as a rival to the more conventional vegetable and animal greases in lubrication was a subject of some contention, the basic problem then, as it is today, being its efficient containment in the vicinity of the bearing.

1.3 Journal Bearing Friction - The Early Workers

Experimental work with loaded half-bearings with and without lubricants had been published by Hirn in 1854, who noted that the coefficient of friction was directly proportional to speed at constant temperature and was also directly related to the lubricant's viscosity[2]. Although this work revealed the essential prerequisites of efficient lubrication, it received little or no recognition since it contravened the established laws of dry friction proposed by Coulomb and confirmed by Morin[3, 4]. Hirn's work, which included studies on air as a lubricant in addition to a range of animal, vegetable, and mineral oils, was later to recieve due recognition from Robert Thurston, the first President of the American Society of Mechanical Engineers[5].

Thurston, himself a university graduate, was essentially desirous of conserving energy through efficient use of materials and lubricants and sought diligently to convince his contemporaries of this necessity. He published much experimental data on friction coefficients for a wide spectrum of lubricants and recognized the importance of mechanical test machines for performance evaluation of bearings. Arising from his work in this area he established that, for a lubricated bearing, a point of minimum friction coefficient existed within a range of increasing load.

1.4 Petrov's Analysis and 'Mediate Friction'

It was a Russian, Petrov, however, who, motivated both by the inevitable problems with railway axle boxes and the desire to find a market for Caucasian mineral oil, (which had been first produced in 1876) undertook an extended program of friction measurements on these units. Using oils of differing viscosities, he was the first to establish the relationship between working oil viscosity and bearing power loss. More important however, was his discovery of 'mediate friction' i.e., that power loss arose from an intermediary, the lubricant, which physically separated the bearing surface, thereby resolving the hitherto contentious problem of surface contact friction into one of simple viscous drag. Employing the established viscous friction law of Newton in a simple analytical model he produced, in a paper of (1833), basic equations for viscous power losses[6, 7]. Through further extensive experimental work and an analysis of Hirn's results, Petrov established the validity of his concept of 'mediate friction' and laid the foundation for the classical hydrodynamic analysis that was to follow shortly.

1.5 Beauchamp Tower - 'The Gentleman' and the 'Wooden Plug'

Meanwhile, in England, the Institution of Mechanical Engineers had not been idle. A Research Committee had been formed to encourage studies into prevalent engineering problems and at a Council Meeting in 1879, one of the subjects felt worthy of investigation 'should time and money be found to be sufficient' was 'Friction between solid bodies at high velocities'.

The need for this, it would appear, arose from a conglomeration of contradictory evidence in various kinetic friction experiments and a 'gentleman' was sought who would be in 'a position to take the subject in hand'. Happily both the 'time' and 'money' and the 'gentleman' finally appeared for, on April 20th, a Mr. Beauchamp Tower was duly appointed. His test machine, a simulated axle box bearing, was quickly assembled and commissioned and experiments were commenced.

Tower produced his 'First Report' to the Institution in 1883 and its findings, though he may not have appreciated it at the time, were to have momentous results in the history of Tribology. Initially, Tower found that erratic friction results arose with the common methods of lubrication prevailing at the time and resorted to an oil bath system which gave better consistency; thus establishing the importance of adequate lubrication[8].

Towards the end of the initial experiments the bearing, having seized, was stripped. While the brass was out, a ½" diameter hole was drilled into the half brass for a lubricator. On running the machine on reassembly, oil flowed out of this hole and to prevent its egress the hole was blocked with a wooden plug. On restarting the machine, the plug was observed to be slowly forced out by the oil, indicating a 'considerable pressure'. A pressure gauge, being screwed in, registered a pressure over twice that of the average on the bearing cross-section and in Tower's words 'showed conclusively that the brass was actually floating on a film of oil, subject to a pressure due to the load'.

This discovery, that a pressure sufficient to separate the surface, was generated within the lubricant marked a fundamental experimental breakthrough, confirming Petrov's concept of mediate friction and initiating a hydrodynamic analysis which was to form the bedrock of fluid bearing design.

Within two years, in 1885, Tower's second paper, devoted to mapping the complete pressure envelope over the area of the bearing surface had been produced. The accuracy of his measurements can be judged from the fact that the integrated pressure gave a load of 7,988 lb f, the applied load being 8,008 lb f[9].

1.6 Reynold's Equation

It has often been said that good experimentation produced good theory and Tower's work was no exception. In 1886, Professor Osborne Reynolds submitted his classical paper 'On the Theory of Lubrication and its application to Mr. Beauchamp Tower's Experiments' to the Royal Society[10]. Indeed, there is evidence that he was already at work on this in 1884, for he delivered a paper 'On the Function of Journals' in that year to the British Association for the Advancement of Science in Montreal. It was also clear at this meeting that both Lord

Rayleigh and Professor Stokes had discussed Tower's work in relation to the concept of the hydrodynamic taper wedge from remarks made by Lord Rayleigh during his presidential address. However, it was to be Reynolds who launced the Hydrodynamic Theory which is the cornerstone of fluid-film lubrication as we know it today.

The basic differential equation formulated by Reynolds

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\eta}\frac{\partial \rho}{\partial x}\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{\eta}\frac{\partial \rho}{\partial z}\right] = 6 \quad (u_o - u_1)\frac{\partial h}{\partial x} + 6h\frac{\partial}{\partial x}(u_o + u_1) + 12V$$

expresses differential functions of pressure in two coordinates in terms of the relative movements of the bearing surfaces and requires a sequential process of integration to produce equations characterizing the bearing's performance. The conceptual intuition behind its formulation implies a highly erudite mind and an outstanding capacity for logical deduction and imagination.

From this basic differential equation, Reynold's derived analytical expressions for pressure distribution and load-carrying capacity per unit width of bearing (i.e., neglecting the effect of side leakage), but found difficulty in expressing these for bearings operating at high duty because of the lack of convergence in certain trigonometrical expressions in a series solution.

1.7 The Sommerfeld Solution

It was, in fact, a nuclear physicist, Arnold Sommerfeld, whose analytical solution in 1904 to the problem posed by the Reynold's equation, paved the way to the establishment of a concrete foundation for future fluid bearing analysis and design[11]. Neglecting the effect of side leakage reduces the Reynold's equation to the form:

$$\frac{dp}{dx} = 6\ \eta U\left[\frac{h - h_o}{h^3}\right]$$

which expresses the slope of the pressure curve in the direction of motion without restriction on the geometry of the fluid film. The equation could therefore be applied to bearing systems other than circular, the problem of determining the pressure distribution being one of overcoming the erected integrals resulting from the substitution of the film thickness 'h' in terms of the (x, θ) coordinate, Figure 1.

In overcoming three erected integrals for a circular bearing (with 'h' expressed in terms of the variable coordinate 'θ) Sommerfeld employed the standard substitution $\delta = \tan(\theta/2)$ to solve the first integral on the assumption that the oil film was continuous around the bearing. Having obtained a solution to this first integral, Sommerfeld used reduction formulae to obtain solutions to the remaining two.

It is of interest in passing to discover that the so-called 'Sommerfeld Transformation' expression:

$$\cos\psi = \frac{\varepsilon + \cos}{1 + \varepsilon\cos\theta}$$ (where 'ε' is the eccentricity ratio, Figure 1)

widely attributed to Sommerfeld was probably never actually used by him at all. Professor Duncan Dowson, in his magnificent book 'A History of Tribology', a most comprehensive and exhaustive piece of historical scholarship, writes: "Perhaps the most amazing finding which emerges from a careful study of this classical paper is that the mathematical transformation attributed to Sommerfeld was apparently never used by him. He simply used the standard substitution $\delta = \tan(\theta/2)$ and reduction formulae to solve the integrals which yielded expressions for pressure, load-carrying capacity, and viscous traction in his solution of the full 360° journal bearing problem[12].

Professor Dowson goes on to say that, although Sommerfeld used a function which can be rearranged to form a similar expression to the 'Transformation', it was probably Boswall in 1928, who first wrote it in its now familiar form[13].

1.8 Slider Bearing Studies - The Michell Analysis

As already indicated, both Reynolds and Sommerfeld obtained solutions to the full fluid bearing by negelecting side leakage effects. It was left to an Australian, Michell, in 1905, to obtain a solution in which side leakage effects were included, in this case for the problem of the slider bearing[15].

Expressing the fluid pressure as a series of terms, whose solution involved Bessel functions with specified boundary conditions, Michell employed numerical procedures to obtain

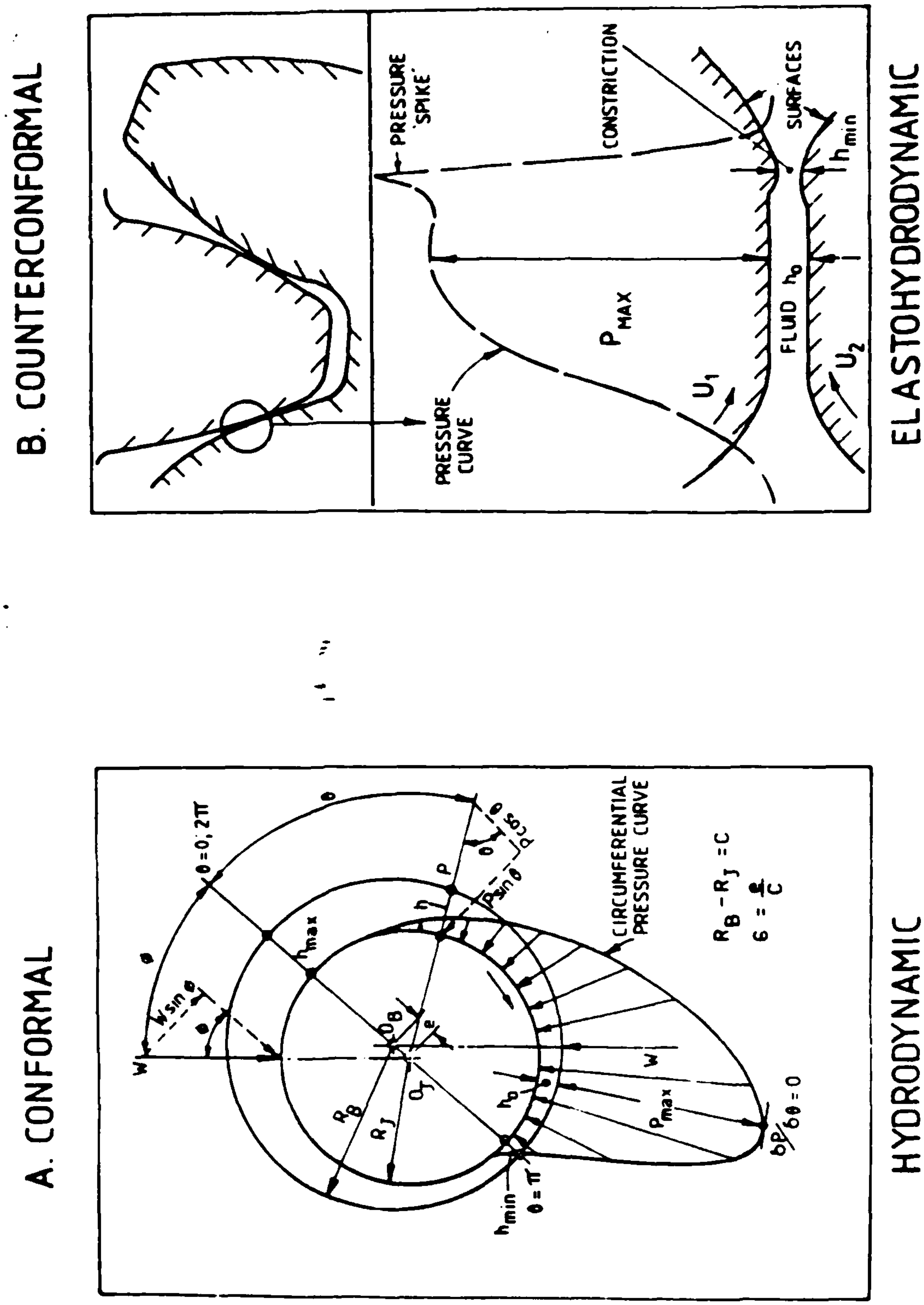


FIGURE 1 FLUID FILM PRESSURE GRADIENT

the first analytical three-dimensional solution of the Reynold's equation for planar surfaces, producing pressure envelopes for four rectangular pad geometries.

Here was a mathematical exposition of the highest order, the first unrestricted solution of the Reynold's equation, worthy of the great master himself. It showed, as may be seen from Figure 2, the dramatic effect of the fluid side leakage on the axial pressure gradient directly corroborating the experimental findings of Tower, some twenty years earlier.

It is noteworthy that, in addition to being a brilliant analyst, Michell was a practicing engineer, his interest in thrust bearings arising from problems with these units in centrifugal pumps and water turbines in Australian hydroelectric plant. This culminated in the invention and patenting of the tilting pad thrust bearing, Figure 2, in 1905, the year of his analytical publication.

1.9 Viscometry

In this period towards the end of the century, experimentalists and inventors in the lubricant field had been particularly active. Reynolds, himself, when presenting his classical paper in 1886, included empirical temperature viscosity relationships for olive oil based on his own findings and for water and air based on work by Poiseuille in 1846, and Maxwell in 1860[16, 17].

The viscometer itself had been introduced by Slotte in 1881 and in 1884 Engler's viscometer had been officially adopted by a section of the German railways for the comparison of lubricants[18, 19]. In England, in the same year, Redwood had produced his own instrument describing it in a paper in 1886[20]. During this meeting, he produced an instrument developed for the same purpose by a Mr. G. M. Saybolt from the Standard Oil Company U.S.A. All the instruments measured oil viscosity as an efflux in seconds at a standard temperature and expressed their results, either as a time ratio with a standard fluid, or in absolute time. These instruments were, with development, related to each other by empirical equations and became the foundation for standard industrial viscometric systems.

We are today apt to treat viscosity with a certain sang-froid as a basic concept easily cognized when considering viscous energy dissipation. It must be appreciated, however, that, prior to about 1880, viscosity was an exceedingly nebulous quality, although Dolfus as far back as 1831, had demonstrated an efflux instrument which he called 'viscometre'

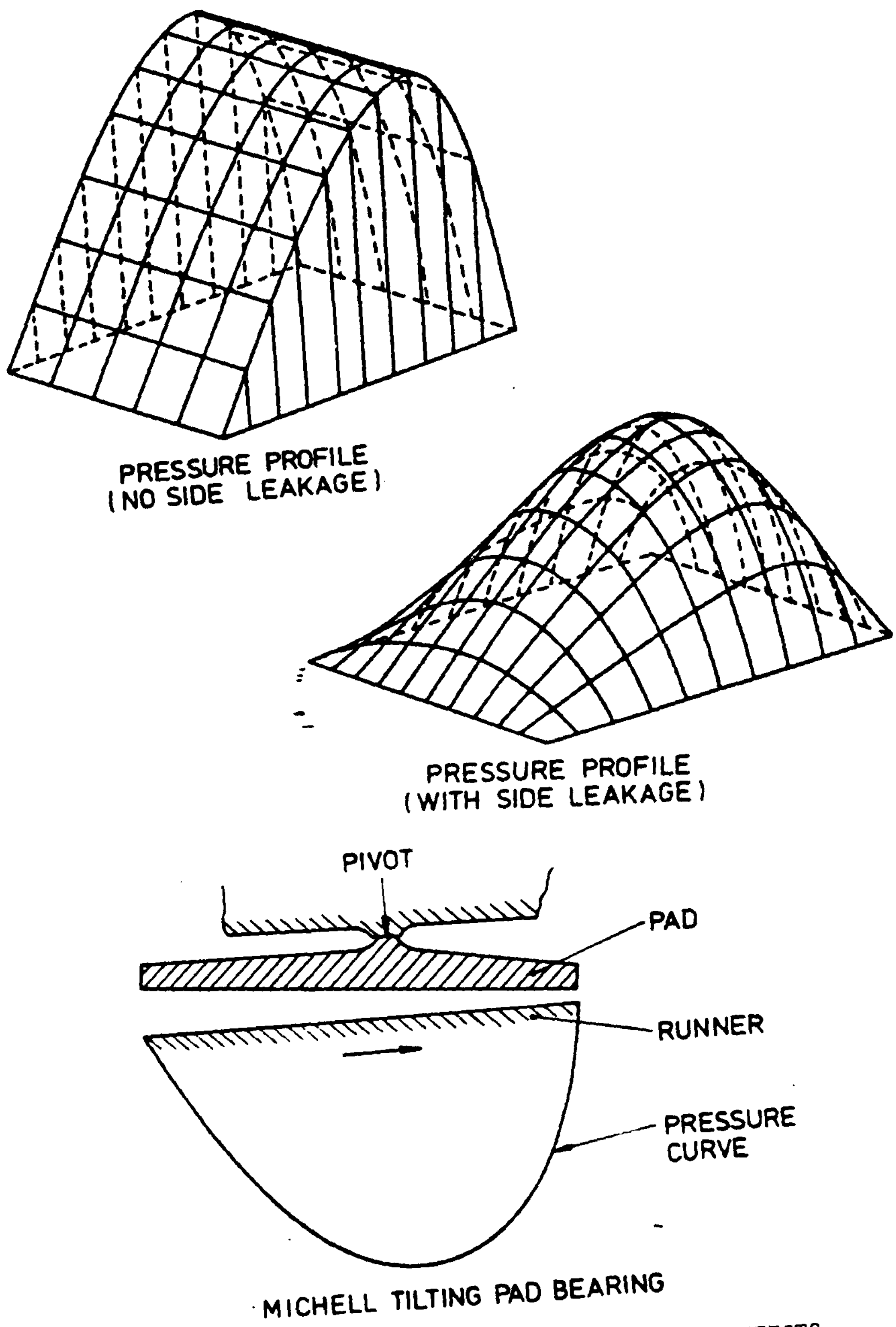


FIGURE 2 FLUID SIDE LEAKAGE PRESSURE GRADIENT EFFECTS

deriving, on the basis of the liquid's efflux time, an 'index of liquidity'[21]. 'Fluidity' of liquids had certainly been appreciated but the more obvious properties of density or specific gravity were generally considered by engineers to be a lubricant's cardinal property in bearing applications prior to the last quarter Century*. The escalation of bearing duty concomitant with increasing engine power in this period, however, rapidly exposed the limitations of the established animal and vegetable lubricants and this, together with their relatively high cost, inititiated their eventual demise to be supplemented by the, by then, ubiquitous mineral oil. The genre of this age of lubrication investigation is reflected in books by Thurston in 1885 and by Archbutt and Deeley in 1900 and provides most illuminating reading[5, 22].

1.10 Hydrodynamics - The Aftermath

Bearing testing, per se, had not stagnated since Beauchamp Tower's monumental breakthrough in 1883. Indeed, in some respects his results may well have had a catalytic reaction on other experimentalists. In 1886, Goodman read a student paper to the Institution of Civil Engineers in which he reported the first direct measurement of fluid film thickness between bearing surfaces using a micrometer device and a simple electrical circuit, thus confirming Petrov's concepts three years earlier and lending experimental credence to Reynold's paper which was published shortly after it in the same years[23, 10].

In the year 1892, a most intriguing phenomenon was observed by Albert Kingsbury in the United States. Kingsbury was investigating screw thread friction and in order to apply loads to his test machine, employed a 6" diameter piston within a cylinder. With the cylinder in the vertical position, he found that the piston could be rapidly spun without frictional resistance from the cylinder wall. The experiment was repeated with the cylinder in a horizontal position with the same result. From these simple experiments, Kingsbury postulated that the piston was rotating on a film of air and building an air journal bearing test rig, measured circumferential and axial pressures at specific points on the bearing surface. Kingsbury was unaware of Reynold's work when he first demonstrated the test rig in Washington in 1896, but was shown the paper at a later demonstration before the Bureau of Steam Engineering at the Navy Department. Subsequent to reading Reynold's paper, he published his paper 'Experiments with an air lubricated bearing' in 1897, a classical paper, in

* Maxwell in 1860 had already discussed viscosity and 'viscosity coefficient'[17].

many ways years before its time[24]. Reynold's paper also encouraged Kingsbury to employ the new hydrodynamic concepts to the hitherto troublesome problem of thrust bearing lubrication and in 1898, instigated the design and construction of a pivoted-pad thrust bearing, the bearing pads pivoting on spherical seatings. The bearing was first used by the Westinghouse Co. in 1904, by whom Kingsbury was employed; certain teething troubles arose however, and it was replaced by a ball bearing. By this time Michell had completed his analysis and obtained an English patent for his bearing in 1905. Parallel development of both bearings continuing from this period onward, Kingsbury being awarded a patent in the U.S.A. in 1910. Unlike Leibniz and Newton, however, and the wrangle over the development of the infinitesimal calculus relations between Kinsbury and Michell were amicable, each respecting the other's independent development[25].

1.11 Rolling Contact Bearings

In Germany Friherr Drais with his invention of the 'Draisine' in 1818, inititated the velocipede, later to be known as the bicycle. This technological development spread throughout Europe and in the process attention naturally became focused, both objectively and subjectively, on the problem of its tendency to resist locomotion.

Frenetic activity arose in an 'effort to reduce effort' following the first patent in 1962, granted to A. L. Thirion for a ball bearing device for such a machine. This culminated towards the end of the nineteenth century, not only in a new industry, but in a new field of study in metallurgy, materials science, and manufacturing techniques arising from the high contact stress engendered in ball contacts; case hardening, developed at the turn of the century, together with better precision in manufacture, providing extended utilization.

1.12 Contact Mechanics

As a corollary to these developments, the science of contact mechanics became greatly extended, yet, perversely, one of its foremost contributors, Heinrich Hertz, became involved in the subject through an interest in optics and the contact between glass lenses, rather than steel balls.

In 1881, Hertz presented his now classical paper on the subject to the Physical Society of Berlin the expression derived enabling deformations and stresses to be calculated for generalized elastic surfaces in counter-conformal contact[26]. This work, as we shall see, was destined to play a very significant role in tribological studies some sixty-five years later.

Experimental support for Hertzian contact theory was provided around the turn of the century by Professor Richard Stribeck, working in Berlin.

Stribeck both considered the mechanical properties of bearing steels arising from heat treatment and tested the Hertzian predictions by mechanical testing ball assemblies in a variety of contact situations. He established, for a radial ball bearing of 'n' balls under load, that a simple expression 4.37 W/n (Stribeck's formula) expresses the greatest load taken by any ball but suggested the number be approximated to '5' to allow a factor of safety[27].

1.13 Stribeck & Gümbel

Stribeck also studied friction in hydrodynamic bearings in addition to rolling contact bearings and in 1902 confirmed, with a series of carefully controlled experiments, the point of minimum friction previously reported by Thurston, today recognized as the transition point between incipient asperity contact and a full fluid film[28]. In doing so he ended the controversy which had existed during the fifty years, prior to this time, regarding the mechanism of friction in journal bearings. Many engineers today speak of the characteristic form of the journal bearing friction curve as the 'Stribeck Curve', on the basis of this early pioneering work.

Stribeck's results were analyzed in 1914 by a compatriot of his Dr. Ludwig Gumbel[29]. Gumbel demonstrated that the family of curves produced by Stribeck could be unified into one single curve, a dimensionless parameter

$$\frac{\eta\omega}{\rho}$$

being plotted against the friction coefficient. This grouping was later to be known as the 'Gumbel number'. Gumbel also considered the problem of cavitation in oil films and suggested possible boundary conditions.

1.14 Dimensionless Groups - The 'Hersey Number'

Although dimensionless groups had been utilized sporadically by workers prior to 1914, it was the Pi Theorems of Buckingham of the National Bureau of Standards, U.S.A., who placed such approaches on a firm foundation[30]. The technique proved attractive to engineers and Mayo D. Hersey (1914) was to become the first worker to apply such an approach to journal bearing analysis showing that hydrodynamic friction could be expressed as a function of

$$\frac{\eta n}{p}$$

a restatement of the Gümbel number in terms of 'n' the speed of rotation[31]. The term appears today in the upper case form

$$\frac{ZN}{P}$$

where 'Z' is the abbreviation for the German word 'Zahigkeit' (viscosity).

1.15 Fluid Analysis - The Consolidation

Starting with Reynolds and extending through Sommerfeld and Michell, fluid analysis was continued for gaseous fluid by Harrison in 1913[32]. Although unaware of Sommerfeld's solution, he proceeded to solve the Reynold's equation for a full journal bearing, drawing attention in the process to discrepancy between the journal and bush reactive torque arising from the journal's eccentricity, Figure 1. He extended this work to compressible fluid flow between parallel plates and on the basis of mass flow continuity and neglecting side leakage, numerically integrated the Reynold's equation. He then compared results for incompressible and compressible flows against those obtained experimentally by Kingsbury; excellent agreement being obtained. Between 1913 and 1919, Harrison treated inclined surfaces under varying load and speed and in 1919 produced a paper on dynamically loaded journal bearings which stands today as a pioneering work years before the practical problems of bearing dynamics had presented themselves to the designer[33]. In the same paper, he also produced an analytical study of surface profile effects in pivot bearings.

1.16 The Problem of Counter-Conformal Contacts

The success attained by analysis on conformal surface contact problems led to studies on the more complex problems of counter-conformal surfaces.

In 1916, in the journal 'Engineering', Martin produced an analysis of the contact conditions between two lubricated discs loaded together and rotating in nominal line contact, recognizing that such a configuration closely modelled the conjunction conditions between straight spur gears[34]. Treating the system as isoelastic and the lubricant as isoviscous he obtained an analytical expression for the minimum film thickness in the contact area, i.e.

$$h_m = 2.44748 \cdot \left[\frac{\eta URL}{W.}\right]$$

Although this expression yielded a magnitude of film thickness considerably less than the measured surface finish of gears at that time, the implication of the approach is of considerable significance since it represented an adherence to concepts of classical hydrodynamic lubrication initiated by the Reynold's paper of 1886. However, around this time, a feeling was growing that there were other factors which influenced the lubrication between sliding surfaces in close proximity.

1.17 Boundary Lubrication

Young, as far back as 1805, had considered the equilibrium of a liquid droplet on the surface of a solid in terms of the interfacial surface tensions and Tomlinson in 1867 and 1875 had discussed the influence of surface contamination upon gas release at interfacial contacts between solids and liquids in terms of adhesive forces[35, 36, 37]. Lord Rayleigh in 1918, only two years after Martin's analysis, in a paper entitiled 'On the lubricating and other properties of thin oily films' referred to his earlier experiments on surface tension and wrote: "We see that the phenomena here in question probably lie outside the usual theory of lubrication, where the layer of lubricant is assumed to be at least many molecules thick[30]."

Already Langmuir in 1917 was involved with work on thin surface films, principally on the idea of oriented monomolecular layers as were Hawkins, Davis, and Clark in the U.S.A., in a paper of the same year[39, 40].

In 1919, Hardy and Hardy spoke of the nature of an 'adsorbed' layer of lubricant onto a surface and considered that the lubrication depended wholly on the chemical composition of the fluid[41]. In the following year, a report from the Department of Scientific and Industrial Research (D.S.I.R) recognized three distinct stages of lubrication governed by three laws: 1. Dry friction. 2. 'Greasy' friction. 3. Viscous friction; while Hardy, again in 1920 used the term 'boundary conditions'[42, 43].

However, it was both Hardy and Doubleday in joint papers to the Royal Society in 1922 who laid the foundation of the school of boundary lubrication and molecular chain theory, emphasising the idea of multilayer orientated films, rather than a mono-layer and stating what, in retrospect, was to prove to be a portentous conclusion: " ... Boundary Lubrication differs so greatly from complete lubrication (i.e., full fluid) as to suggest that there is a *discontinuity between the two states.*" (Author's italics)[44, 45].

Thus, Hardy and Doubleday papers not only provide a starting point from which a whole new world of surface studies evolved, but provided the intitation of a whisper which, however unintentional, was later to grow into a murmur of dissent not so much against the concepts of classical hydrodynamic theory in its rightful domain, since this had already been proven beyond reasonable doubt, but in that the newly crowned 'prince' governed exclusively in <u>all</u> tribological realms, specifically in certain conformal contact situations but much more particularly in the realm of the counter-conformal contact, as exemplified by lubricated gear teeth. Here, if anywhere, it seemed, the apparent failure of Martin's analysis to predict realistic oil film thickness by classical hydromechanics was a domain where a new pretender to that particular throne might reasonably raise his standard in revolt. In reality, nearly thirty years were to elapse before, to coin a metaphor, oil, and not the fatty acid molecule, was to be poured on the troubled waters of the counter-conformal conflict in the shape of an analytical approach worthy of the pragmatism of Reynolds. Proposed by Grubin and Vinagrodov, Moscow 1949, it combined classical hydrodynamic theory with the elasticity of the contacting materials and the effect of pressure on the lubricant's viscosity and predicted, at last, finite fluid film thicknesses albeit of some one or two orders less than those that their compatriot, Petrov, had first brought to the notice of the world some eighty-three years previously; eighty-three years, in truth, from a 'Russian thick film' to a 'Russian thin film'[46].

1.18 The 'Great Schism'

Such a point in the early 1920's therefore, might justifiably be referred to by the tribological historian, as the point of the 'Great Schism', where an arbitrary division of the lubrication spectrum arose. The first camp, the area of full fluid lubrication, frequented by the industrial bearing user and designer in a search for greater efficiency and reliability in the face of ever mounting bearing duty, progeny of the classical hydrodynamic tradition, secure in the knowledge of their analytical lineage and the genealogy of their illustrious antecedents the experimentalists; the second, the new embryonic school of the surface contact investigators whose banner, born initially by the boundary lubrication workers, would eventually rally together a new generation of studies in the physics and chemistry of thin films and dry surfaces, contact mechanics (including friction and later wear processes), lubricant and material testing procedures, metallurgy and surface topography, together with a rapid development in surface techniques and lubricant and additive formulations.

1.19 The Concept of 'Oiliness'

By 1925, the schism might be said to be fully developed, though certain experimentalists had a 'foot in both camps'. Kingsbury, himself, as far back as 1903, had concluded that there was a friction reducing property in a lubricant under boundary friction which was separate and distinct from viscosity; he called this 'oiliness' and his was among the first attempts to describe how certain lubricants appeared to develop resistance to film rupture under these extreme conditions. The definition of 'oiliness' adopted by the Society of Automotive Engineers (S.A.E.) is indicative of the uncertainty of the phenomenon; for them oiliness was a term signifying, to quote: "differences in friction greater than can be accounted for on the basis of viscosity when comparing different lubricants under identical test conditions" i.e., that lubricants can possess boundary friction capability entirely independent of their viscosity. Water, for example, had appreciable viscosity, but practically no 'oiliness' in the concept of those times.

1.20 Hydrodynamic Limits - The Work of Stanton

Other workers, however, had mixed feelings about precisely when and where boundary lubrication conditions could be said to apply. A Lubricants and Lubrication Inquiry Committee had been set up in England by the D.I.S.R., in 1920; the general concensus was that, in gear teeth at least, the high localized stresses precluded hydrodynamic action and that boundary lubrication would pertain. One member of the Committee, Dr. T. E. Stanton was prepared to swim against the tide of enthusiasm of the 'molecular chain school' and realizing the impracticability of measuring pressures between rotating gear teeth simulated such conditions by employing a conformal bearing with large radial clearance[47]. Using a pressure gauge, (like his famous predecessor, Beauchamp Tower), he first measured pressures around a small load-carrying arc, recording peak pressures of some 3½ ton/in^2 and obtained the theoretical pressure profile for the contact configuration from hydrodynamic theory. The degree of correlation was outstanding and did much to reinforce confidence that hydrodynamic action could still be manifest under such severe conditions. This early experiment undoubtedly provided inspiration for the subsequent work on elastohydrodynamic action in gear teeth that was to commence immediately after the Second World War. Ironically, Stanton himself believed that boundary lubrication applied to piston rings after carrying out experiments in 1925; his sliding velocities were, however, much slower than later workers' who suggested that hydrodynamic films operated for most of the stroke, except at the extremities of travel, where boundary films were manifest[48].

1.21 Outcome

It is not proposed within the confines of this discussion, to continue looking at the chronology of tribological progression further, since it has been illustrated how early liaison between the disciplines developed towards the end of the nineteenth century and how the Great Schism evolved towards the first quarter of the twentieth, which tended to dissolve it. Clearly, factors such as the rise to preeminence of the internal combustion engine brought a certain further separatism in compartmented lubricant speicifications and testing procedures, together with a continuance of the two separate viscosity concepts, the efflux time specification, principally employed by the oil industry, and the absolute or dynamic viscosity concepts used exclusively by hydrodynamicists and bearing designers.

1.22 Factors Assisting Early Liaison

What is important, however, is to distill, on the basis of the chronology that has been discussed, the factors which up to 1925, either singly or in combination, resulted in the close liaison between workers in the field, since some of these early formative movements towards a multidisciplinary approach might gainfully be considered in the context of the present day problems of compartmented specialization. These factors may be roughly generalized as follows:

(1) The realization by experimentalists, theoreticians, industry, learned societies and government laboratories that common lubrication problems existed with the development of steam power and the

willingness of each section to cooperate with the other in attempts at common solutions. This might be termed the 'Problem Recognition and Cooperative Factors'.

(2) The formation by the learned societies of special committees and groups, both to foster cooperation between a cross-section of workers in the relevant areas by serving jointly on various panels and to disseminate information by meetings and the publication of papers. In certain cases, as already seen, such groups were prepared to support relevant research by direct funding. This might be termed the 'Support and Dissemination Factor'.

(3) The emphasis on the interrelationship between theory and practice as exemplified by the dual roles undertaken by academician/engineers such as Hirn, Petrov, Thurston, Reynolds, Stribeck, Goodman and Michell, to name a few. This flexibility of the individual, both to be prepared to assimilate practical experience from personal contact with experimental apparatus and, at the time, to apply high-level theoretical knowledge to the solutions of practical engineering problems, would appear to be a hallmark of this period when technical innovation and analysis flowered in the tribological field as never before. This might be termed then, the 'Individual Duality Factor".

(4) The establishment of centers for the training and education of specialists in disciplines specifically relevant to the problems arising with the coming of the new technology and the orientation of such studies to the practicalities of solutions in the real world of engineering and industry. Such a step represented a partial break with tradition and, for the time being at least, the destruction of the academic 'White Tower' syndrome, which had bedevilled certain educational institutions. This may be called "The Orientated Training Factor".

2. THE INTERLINKING OF THE DISCIPLINES

Following the end of the Second World War, a rapid interlinking between the disciplines began to take place in the field of Tribology. This is attributable to five primary causes, three technical, one economic, and one conservational. Chronologically, these may be categorized as follows:

(1) The rapid post-war development of chemical additives for lubricants and the increased application of synthetic fluids, solid lubricants, and plastic materials.

(2) The successful development of the elastohydrodynamic theory for counter-conformal contacts and its experimental vindication.

(3) The increasing sophistication of techniques for surface examination and analysis, parametric measurement and data recording and processing.

(4) The economic implications of an overall multi-disciplinary approach to tribology following the findings of the Jost Report in 1966.

(5) The current emphasis on material and energy resource conservation arising as an aftermath of the world oil crisis.

These factors will be reviewed in order to see their influence on the interaction of the disciplines.

2.1 Additives, Synthetic and Solid Lubricants and Plastics

As previously noted, boundary lubrication studies began in earnest in the early 1920's and by the mid-thirties large quantities of vegetable oil and animal fats were being used by the petroleum industry. Such compounds are rich in fatty acids with their active polar group, a prime prerequisite in forming the metallic soap films on sliding surfaces with dramatically reduced friction coefficients. Since, for the most part, the constituent compounds of mineral oil are nonpolar, they make poor boundary lubricants. Thus, the inclusion of fatty acids in lubricating oil and greases, gave them an added lubricating dimension.

Heavy sliding conditions induce high surface temperatures and when these rise above about 150°C metallic soap films decompose. This problem arose with the advent of the so-called 'Extreme Pressure' (E.P.) lubricant, basically compounds of sulphur, phosphorous, and chlorine. Strictly, these should be termed 'Extreme Temperature' (E.T.) lubricants, since they react chemically with the metal surfaces only at higher temperatures to produce low coefficient layers by a process of controlled corrosion. A wide range of other ingredients was also in use by 1937 in this type of application, among these being lead soaps.

Prior to the war then, attention had been focused on reducing friction and wear by additives. After the war this was turned to improvements in the performance of the lubricant itself.

The rapid change of the viscosity of mineral oils with temperature was occupying the motor industry at this time. 5W, 10W and 20W oils were initiated around 1950 to assist starting in sub-zero temperatures, but this led to high consumption in normal running. By 1952, the first multigrade oils were being introduced; high molecular weight polymetric materials such as polymethacrylate and polyisobutylene being added to the lubricant base stock to inhibit the natural drop in viscosity with increasing temperature. In addition to these V.I. (Viscosity Index) improvers, came a wide range of other additives such as antioxidants, corrosion inhibitors, detergent/dispersant additives, pour point depressants and anti-foam additives, these being developed and extensively employed for use in both oils and greases. A wide range of literature is available on the subject of additives, excellent reviews being presented in References 49 and 50.

The invention of the gas turbine towards the end of the war, resulted in an intense period of activity in the field of synthetic lubricants in its aftermath. Zorn, in 1939, had experimented with castor oil, but storage deterioration presented problems [51].

The first successful materials were the organic diesters, reaction products of organic acids and alcohols, a promising compound being di (2 ethyl-hexyl) sebacate, a derivative of sebacic acid and originally used as a rubber plasticizer. With the rise of engine power, bearing temperatures rose causing some problems in gear lubrication in the turbines. This was overcome by further research, complex esters (derivatives of polyethylene glycol, sebacic acid and alcohols) being used to thicken the diesters. Inevitably, however, with bulk lubricant temperatures in excess of 150°C, thermal breakdown at a position known as the 'β' hydrogen, (Figure 3), in this material took place and an entirely new material, trimethylopropane-tripelargonate (a derivative of pelargonic acid) was developed, known, because of the absence of the 'β' hydrogen in its structure, as a 'hindered ester', (Figure 3). The bulk capability of this material was in excess of 200°C. Ester type lubricants currently serve as the 'bread and butter' lubricant for the aviation industry.

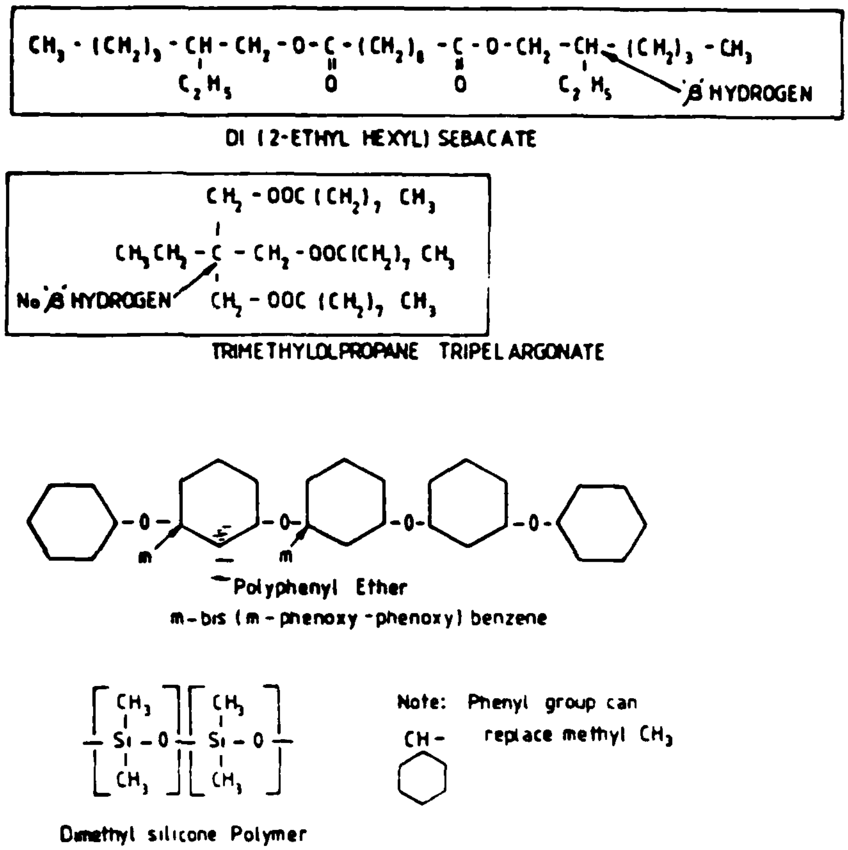


FIGURE 3 LUBRICANT CHEMISTRY EVOLUTION

A third generation arose with the coming of the supersonic 'Concorde' with bearings operating in the region of 260°C, together with gear lubrication problems at high temperatures. The economics of developing entirely new lubricants proved daunting and the pelargonates were 'stretched' by employing new additives to inhibit thermal degradation of the base fluid and the older additives themselves. A new concept in mechanical testing of the lubricants and additives was employed at this time in which test rigs simulated actual working conditions in the turbine or complete turbine components were, themselves, used as mechanical test rigs. This direct mechanical testing enabled a wide range of lubricants and additives to be examined directly on a 'trial and error' basis which represented a breakaway from the traditional 'benchtop' measurements of lubricant properties for specification purposes and enhanced the importance of mechanical testing in overall lubricant performance evaluation.

Military requirements, principally for supersonic fighter aircraft, pushed thermal demands on fluids to the extremes of the temperature spectrum in the late 1960's and new materials were produced, principally polymers of aromatic phenols such as polyphenyl ether, (Figure 3), or aliphatic polymers containing inorganic silica in place of carbon, (Figure 3). The latter had some of the best V.I.s available but were poor boundary lubricants especially on ferrous surfaces and did not readily accept additives. The former, while not having this drawback, were exceedingly viscous in cold conditions and had to be 'diluted' with ester type oils. Both, however, had very good high temperature capability; in the region of 300 - 370°C.

Other synthetic fluids were also developed for applications in specific lubrication situations, such as silicate esters for low temperature lubrication and corrosion prevention (compounded with other synthetics), or as base stock for high temperature hydraulic fluids. Phosphate esters were used as fire resistant hydraulic fluids in mining, aircraft, and ships. Polyglycol ethers were employed in cutting and machining operations and in certain boundary lubrication conditions or, in its water soluble form, for brake fluid and specialized hydraulic oils. Halogenated compounds (for use in high flammability process situations) and Silanes (for high temperature hydraulic oil and grease base stock) began to find use. Thus the work of the organic and inorganic chemist, combined with the study of surface contact mechanics and mechanical testing, followed from the evolution of additives and synthetic fluids.

Rheology was added to the spectrum in the study of the behavior of grease, particularly in the combination of synthetic oils with clay and other bases to produce high temperature greases of the 'Bentonite' type.

Solid lubricants, in the form of graphite and molybdenum disulphide, were employed in tribological applications during the nineteenth century, but research on these and other materials escalated in the period 1950-1965, particularly in the U.S.A., with the challenge of the space age (environments of extreme temperatures in rocket engines and hard vacuum in space.) A wide range of loose powders, metals, oxides, molybdates, and tungstates were investigated in addition to layer lattice salts. Mixtures of graphite with soft oxides and salts in a variety of environments were researched, as were coatings of oxides of lead and silica in 'duplex' structure ceramics and ceramic-bonded calcium fluoride.

Thus, disciplines of powder metallurgy, inorganic chemistry, surface failure mechanics, friction materials science, ceramic composite material science, and surface physics were combining with surface examination and analysis techniques to spread the multidisciplinary umbrella. A comprehensive treatment of non-conventional lubricants is given in reference 52.

Nonmetallic bearing materials were already being discussed in 1937 with regard to rubber and graphite, but the largest application by far, has been in the use of polymetric materials[53]. These divide into two classes of plastic; thermoplastic and thermosetting. In the former category, nylon found initial use but in terms of minimum friction no commercial plastic surpasses polytetrafluoroethylene (P.T.F.E.), formulated early in the 1950's, followed by polyamides (1959) and polyacetal, polypropylene and polyethylene in the later years.

Thermosetting materials such as phenolic resins were introduced in combination with fabrics for so-called synthetic resin bearings and were used in German rolling mills as early as 1928, water lubricated bearings of this type being also used in similar applications in the U.S.A. in the 1930's[54].

The importance of plastic bearings is their ability to operate without a lubricant and in the food, textile, and processing industries they are extensively used for this reason. In the domestic market, they find application in lightly loaded mechanisms and their general inertness makes them suitable in environments where metallic corrosion is a problem and in medical applications. Additionally, solid or synthetic lubricants may be incorporated in the plastic to enhance its tribological performance. P.T.F.E. is also finding increasing usage, itself, as a 'filler' in powdered metal bearings, in addition to the normal fluid and solid lubricants used in these units. Thus plastics and powder metallurgy must rightly be included in our multidisciplinary spectrum.

2.2 Elasto-Hydrodynamics

It might be justifiably said that no single area of twentieth-century tribological study has yielded such profound success, both in the theoretical attack on the problem and in the vindication of the theory by experimental results, than in the area of elasto-hydrodynamic lubrication (E.H.L.).

In certain respects, it recalls the heyday of Petrov, Beauchamp Tower, Reynolds, and Sommerfeld in welding together theory and practice into a unified whole. It differs in one important respect, however, in that in the last quarter of the twentieth century, experimentalism had begotten theory, while some fifty years after the converse was to hold; elasto-hydrodynamic theory spawning the experiments. Two things, however, were certain; first, that through all the vicissitudes of the 'counter-conformal conflict' classical hydromechanics had emerged triumphant, and second, as a corollary to this, the 'Great Schism' of twenty-five years earlier had, at last, been healed. No doubt the chairman of the 'Committee on Tribology', Peter Jost, had such thoughts in mind when in a jocular misquote of Karl Marx, he addressed a Tribology Conference with the appeal: "Tribologists of the world unite! You have nothing to lose but your molecular chains![55]"

How then had such a transition from the days of Martin's theory come about? In 1936 and 1938, Peppler had considered elastic deformation effects between gear teeth, but concluded, erroneously, that fluid pressure could never exceed the required Hertzian contact pressure[56,57]. Gatcombe in 1945, analysed pressure viscosity effects in the oil film but although this increased the thickness predicted by Martin, it did not satisfy the criterion of a fluid film above the height of asperity roughness[58]. Hersey and Lowdenslager in 1950 produced similar results to this[59]. Blok in 1952 indicated that, for an exponential pressure viscosity relationship, a limiting film thickness giving infinite fluid pressure arose if the surfaces were considered iso-elastic; for this condition the predicted load was some two to three times Martin's load; while earlier, in 1950, Poritsky had suggested a Hertzian type pressure distribution with a parallel film, modified by deformations at inlet and exit to permit finite gradients of fluid pressure, (Figure 1)[60,61].

Prior to most of this work, however, almost at the end of the war, a Russian, Ertel, produced a rationale of the problem in two intuitive steps of masterly insight. First, that with an exponential law for pressure/viscosity effects in the two dimensional Reynold's equation, a practically parallel film results, i.e., the pressure in the Hertzian zone is the same whether a lubricant is there or not and second, outside this zone, the fluid pressure in not high enough to change the shape of the conjunction from that of the dry contact. Using these concepts together with the Hertzian equations for the contact geometry, Grubin in 1949, produced an analysis, primarily of the inlet conditions, and by fitting an algebraic expression to a family of iterated integral curves, obtained a single formula for the mean dimensionless film thickness in the contact zone;[46]

$$H_o = \frac{1.95\ (GU)^{8/11}}{W^{1/11}}$$

It is seen that essentially the film thickness is hardly affected by load, the fluid behaving almost like a solid under the high pressure. This equation at once predicted film thicknesses which were orders of magnitude greater than Martin's theory and established the probability of full fluid films in the contact area.

In a qualitative discussion of the pressure distribution, Grubin postulated the existence of a 'pressure spike' arising from a sudden convergence of the surfaces in the exit region, (Figure 1).

Characteristics of the elasto-hydrodynamic contact predicted by Grubin were confirmed by Petruesevich in 1951 for a limited range of numerical solutions, but, specifically, his solutions produced both the contraction in film thickness at exit and the pressure spike, sometimes referred to as the 'Grubin/Petruesevich' spike [62].

An extended numerical solution of the problem for a wide range of operating conditions was obtained by Dowson and Higginson in 1959, using iteration[63]. A detailed account of this work, together with a comprehensive review of the chronology of elasto-hydrodynamic lubrication may be found in the book by these authors, Reference 64.

For minimim film thickness calculation i.e., at the exit contraction, the Dowson-Higginson formula is given by:

$$H_m = \frac{h_m}{R} = \frac{2.65\ U^{0.7} G^{0.54}}{W^{0.13}}$$

For frictional and other calculations involving the major area of contact, the Grubin formula is applicable. A comprehensive study of isothermal solutions of the problem was given by Dowson and Whitaker in 1964[65].

Thermal effects were considered by Sternlicht, Lewis, and Flyn (1961) and Cheng and Sternlicht (1964)[66,67]. Rheological effects were studied by Bell in 1961, Crook (1963), and Dyson (1970). Christensen had considered surface roughness effects (1962) in normal approach contacts while Dawson (1961) studied disc/pitting[68-72].

A series of elegant papers had been published by Crook (1958-1963) in which he measured film thicknesses in the conjunction by electrical capacitance, a similar technique being used by Dyson (1966) with excellent correlation being obtained[73, 74]. Similar work was done by Sibley (1960) using X-ray collimation techniques[75]. Optical methods were used for this purpose by Kirk (1962) and Cameron who, using interferometry techniques, studied a large range of contact situations[76,77]. This method is given in some detail by Ford et. al. in Reference 78.

One of the most dramatic measurements was the pressure profile in the conjunction by Kannel (1964), using Maganin' (a material whose resistance varies with pressure) being deposited as a thin strip on one contact surface[79]. Cheng and Orcut (1966) and Hamilton and Moore (1971) extended this work with certain pressure spikes being obtained[80, 81]. Temperature measurements have been made by many of the above workers using trailing or buried thermocouples and strip transducers. Infrared techniques have been used by Hamilton and Moore and Prof. Winer and his colleagues in recent years[82].

Thus, a wide range of disciplines: Hydrodynamics, Elasticity, Fatigue, Thermodynamics and Heat Transfer, Fluid Physics, Rheology, Surface Topography, Electronics, Transducers and Instrumentation, Heat Radiation, Optics and Surface Contact Mechanics had been collectively interwoven through the study of elasto-hydrodynamics in a concerted multidisciplinary approach.

2.3 Surface Studies, Measurement, Monitoring and Processing

The sheer scale and scope of the explosion of methods and devices for surface examination and analysis, parametric measurements, data recording and processing, coupled with the advent of digital and analogue computers in the fields of theoretical studies, data storage and process control, places any extensive treatment in depth beyond the bounds of the present discourse. A broad outline only, of this realm of multi-disciplinary interaction, within our Tribological spectrum, must therfore, necessarily suffice.

In essence, this section may, quite arbitrarily, be broken into the following groups, bearing in mind their tribological relevance:

(1) Friction and Wear, Surface Examination, and Measurement
(2) Mechanics and Properties of Materials and Lubricants
(3) Measuring Techniques and Parametric Measuring Devices
(4) Mechanical Testing, Bench Testing, and Field Evaluation
(5) Condition Monitoring Applications.

It is emphasized that this is entirely subjective, since there is obviously no lack of overlap between the areas chosen.

2.3.1 Friction, Wear and Surface Examination Analysis

From a cursory examination of the subjects of friction and wear, it might appear paradoxical that while these topics are clearly associated, studies on the former began as far back as 1699, with the historical work of Guillaume Amontons, while studies on the latter topic have only really accelerated in the period following the Second World War[83]. Two factors which may help to explain this dichotomy are as follows. First, where power loss in tribological situations could be mitigated, this would, in general, be opted for. Mechanical integrity (or the lack of it) manifests itself much more dramatically than wear integrity and since frictional power loss in machines was of prime importance (especially when input power was limited) studies would be necessarily directed to overcoming the all too obvious outcomes of these problems. Second, a greater appreciation has been manifest, both during and after the war, of the importance of long-term reliability and life of machine components and therefore attention has been directed to the importance of surface studies.

To further as well as implement these studies, a range of new techniques and equipment has evolved which broadens, even further, the scope of multidisciplinary tribology.

During the thirties, the stylus profilometry methods were invented which, after the war, became linked to the digital computer enabling three dimensional topographical models of surfaces to be produced; an example being the work of Edmonds described in Reference 84.

Optical methods of surface examination had, of course, been used by metallurgists for many years, but, with the marketing of the 'Quantimet' image analyzer in 1963, surface contours could be examined and analyzed in still greater detail. Optical interferometry itself had also been used to aid optical surface examination but by far the greatest advance in surface examination and particle morpholgy came with the invention of the electron microscope (Scott and Scott 1957) and the scanning electron microscope (Salt 1970). A further breakthrough came with the electron probe analyzer (Cay and Quinn 1972) in which a collimated electron beam causes the excited surface atoms to produce characteristic X-rays, which may be detected by a spectrometer giving, essentially, the chemical constituents of the material. For surface examination, low energy electron diffraction (L.E.E.D.) and Auger analysis (Cheng 1971 and Buckley and Pepper 1972) have been used. Wear studies by Jones (1976) illustrate the utility of the method[85].

The generation of hydrogen peroxide (H_2O_2) at newly machined surfaces was explained by Kramer (1950) on the basis of the so-called exo-electron emission from the oxide films forming on the surfaces and this phenomenon has been used by March and Rabinowicz (1976) for incipient fatigue investigations using a rolling four-ball machine[86, 87].

Electrical conductance and contact resistivity techniques have been used by numerous workers for the investigation of surface asperity contact mechanics and the influence of surface oxide and other gaseous and solid contaminants on surface interaction[88, 89]. Such work has, also, much importance in the field of electrical engineering, such as switch and brush contact problems, in addition to mechanical engineering applications. Since friction and wear are always manifest when surfaces physically interact in rubbing contact (welding and tearing being considered to be incurred at the asperity peaks) a great deal of attention has been directed, both experimentally and analytically, to the mechanism of such interactions. These subjects have included surface metallurgy, surface physics, surface topography, surface chemistry and chemical kinetics, surface and asperity heat transfer, contact mechanics, material elasticity and plasticity, gas chromotography and chemical spectroscopy, in addition to a wide variety of surface treatments, metallurgical, mechanical and chemical. Clearly, in this work, the emergence of new techniques and equipment for surface examination will, as already indicated by precedent, have a vital role to play in future tribological development, among these being the likely evolution of laser and holographic techniques both in static and dynamic situations.

2.3.2 Mechanics and Properties of Materials and Lubricants

With the inclusion of stress and elasticity into tribological analysis, the importance of techniques to measure the behavior of materials subject to plastic and elastic deformation in a range of relatively severe environments exemplified by temperature, pressure, high overload, transient shock, vibration, and general excess duty have become increasingly important.

In the field of direct and indirect strain measurement alone, standard techniques such as strain gauges, mechanical and optical methods, photoelasticity, photostress, and brittle lacquers have been reinforced with the developement of solid-state equipment. Metrology, itself has advanced with increasing use of electronic comparators, air gauges and a variety of displacement sensors. Lasers are among the new tools finding increasing use in this area, while X-ray/crystallography techniques have been employed in the study of crystal lattice movement, interfacial slip and residual stress measurements[90].

Fluid physics have been used in the study of lubricant behavior under high pressure, particularly from the viscometric standpoint, while analytical work has been carried out on the rheology of liquid and semifluid lubricants with reference to their visco-plastic behavior and relaxation time[91]. Studies of lubrication in semi-turbulent and full turbulence regimes have been extensive while the effect of extremely high shear rates (of the order of 10^6 sec^{-1} and above) has received attention as the bearing peripheral surface speeds have increased[92, 93].

With the extended use of hydraulic equipment, the behavior of fluids under cavitation conditions is a subject of considerable study together with the propensity of lubricants to entrain and release gases[94]. Lubricant and additive behavior at very fine levels of filtration (<0.5 microns) has received attention, as have situations in face where ultra-flat surface (<0.1 micron) waviness has been achieved[95].

2.3.3 Measuring Techniques and Parametric Measuring Devices

Probably no other field of technology has had such an impact on tribological activity as the revolution that has taken place in the field of electronics during the post-war years. For purposes of exposition, the subject may be divided into the area of data measurement and the area of data recording and processing. In the former, the development and increasing sophistication of instrumentation transducers for fluid pressure and film thickness measurement, micro thermocouples and thermistors, load cells, linear and angular displacement transducers, accelerometers, flow measurement devices and particle counters has enabled the experimental tribologist to quantify, with a hitherto unknown accuracy and diversification, the manifest interactions of tribological phenomena over a multifarious spectrum of activity.

In certain areas, micro-miniaturism had added a new dimension to what was previously considered possible, a particular example being the foil strain gauge, enabling these devices to be incorporated into 'custom built' transducer units for particular applications. The author and his colleagues have been active in this field for several years, a particular outcome of this work being the discovery of the phenomenon of fluid 'tensile stress' in journal bearings[96]. Thus, invention and the development of new manufacturing techniques have ever increasing ramifications in the progression of tribological discovery.

Likewise, in the field of data recording and processing, the invention of the transistor and a variety of solid state devices opened up an entirely new domain of data recording and processing almost overnight, particularly in the field of signal amplification. Operational amplifiers, initially used in analogue computers, have been employed in conjunction with tribological transducers for signal conditioning, giving compact and highly stable systems[97, 98]. Transient signal recording was enhanced with storage oscilloscopes and other devices such as ultraviolet recorders and digital storage recorders, high speed chart and tape recorders and with the coming of the microprocessor, automated testing data storage and processing, coupled with robotic control, opened up an entirely new field of tribological possibilities.

In the field of theoretical tribology the coming of the digital computer has enabled problems, otherwise intractable to classical analysis, to be circumvented or refined and finite difference techniques, which lend themselves admirably to computer processes, have been extensively exploited[99]. The so-called 'mini' and micro computer adds further possibilities of diversification. The author has found computer graphics to be an exceedingly useful tool in examining pressure envelopes in hydrodynamic journal bearings, where groove configuration effects may be directly portrayed, Figure 4.

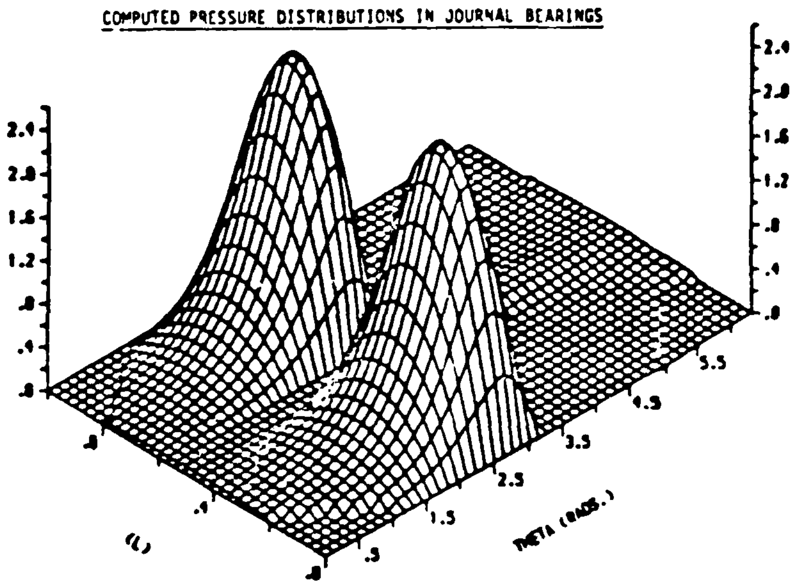
2.3.4 Mechanical Testing, Bench Testing and Field Evaluation

This area is so vast that nothing but a cursory glance can be made within the confines of the present exposition. Bench testing, and to a lesser extent, mechanical and other rig testing, grew from the need to standardize lubricants and materials between organizations at both national and international levels. The oil companies and bearing manufacturers themselves, from the basic necessity of product uniformity, initiated standardized procedures; the former with, primarily bench tests, the latter (particularly the manufacturers of rolling contact bearings) with product batch-testing in mechanical test rigs. A great deal of tribological information of a multidisciplinary nature arose from this source, while the formation of test bodies such as the American Society of Automotive Manufacturers (A.S.A.M.) and the Institute of Petroleum (I.P.) helped to disseminate this information and standardize test procedures. Government institutions and the armed forces in various countries, also laid down close specifications and quality control for bearings, gears, materials, and lubricants.

Notwithstanding the value of laboratory testing, it was quickly realized that simulation is no substitute for actuality, and a growing trend became manifest, particularly after the Second World War, for either testing products in a true working environment (as has been outlined in the mechanical rig-testing of the 'Concorde' lubricants) or through cooperation between a product manufacturer and a product user in extensive 'field' trials. This had both the advantage of operations in a real environment under a wide spectrum of working conditions and of providing information, on a statistical basis, of product performance. Through such organized data acquisition, information is compiled for both diagnostic and prognostic purposes, enabling, in many cases, early remedial action to be initiated.

2.3.5 Condition Monitoring Applications

In situations where high mechanical and wear integrity are demanded, or plant economics mitigate against complete shutdown or service withdrawal, condition monitoring may be



(a) Circumferential Oil Groove. $L/_D = 1$, $\epsilon = 0.7$.

(b) Twin Axial Oil Grooves. $L/_D = 1$, $\epsilon = 0.7$

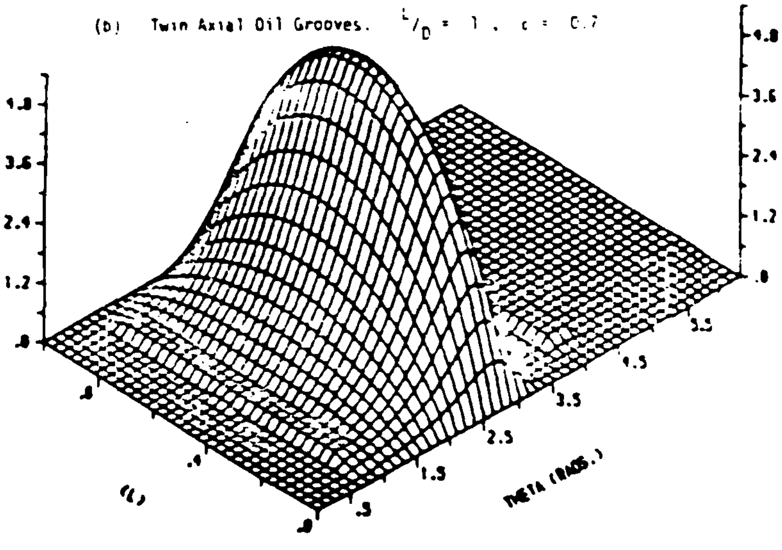


FIGURE 4 COMPUTED PRESSURE DISTRIBUTIONS IN JOURNAL BEARINGS

employed in which 'in-situ' information is extracted from the machine, or its components, on a temporal 'in-service' basis. In the tribological realm, this procedure can be of extreme importance, particularly in assessing wear integrity situations. A variety of techniques exist to establish if wear is taking place; in what specific areas (if this is established) and if operating conditions fluctuate, in what situation is the condition manifest or accentuated. Apart from techniques of direct or semi-direct observation such as optical, chemical or physical methods, performance changes within the tribological system can sometimes be assessed, either quantitatively or qualitatively, through increased power losses, changing noise levels or thermal effects. This alone involves a wide cross-section of inter-disciplinary techniques. Wear debris and particle contamination is receiving particular attention in this sphere and it would appear that automated equipment and micro-processor control are areas where future development may well be extended as the overall economics of machine reliability and efficiency are more fully appreciated.

This third section on the interlinking of the disciplines therefore, has indicated the vast extension of tribological technology and activity under the multidisciplinary umbrella particularly in the post-war years, and the impetus that advances in technology and invention have initiated.

The exponential growth curve mentioned in the opening section of the discussion is thus seen to be no misnomer in describing the rate of change which is taking place. Clearly, the implications are that some methodology with regard to categorization must be applied, however loosely, if the modern day tribologist is not to end up, as has already been intimated, in tending his own particular 'tree' to the exclusion of the rest of the 'tribological wood'. This characterization and methodology will be examined later in the discussion.

2.4 The Economic Implications of Multidisciplinary Tribology

In the chronolgy of tribological coteries, certain meetings or formations of special groups are remembered for their significance. For overall epic proportions the great 1937 'Discussion on Lubrication and Lubricants' organized in London by the Institution of Mechanical Engineers must rank as an historical landmark. Some 600 delegates registered, 136 papers were given, and an exhibition of lubricants, bearings, materials, test machines, and research equipment, held at the Science Museum, was visited by some 18,300 persons in two weeks.

In post-war years, this impetus was continued by the 1957 'Conference on Lubrication and Wear' organized again by the Institution with over 1,000 delegates from 21 different countries participating; 104 papers being given. In a prolific decade of meetings from 1963 to 1972, the Institution promoted a vast spread of tribological papers, engendering a growing tide of multidisciplinary participation in research and development work.

The economic aspect, particularly in maintenance and replacement cost reduction and savings commensurate with increased life of machinery, was beginning to make itself apparent generally and studies in this area in particular were already underway in certain of the Eastern-Bloc countries, notably East Germany.

In 1964, the British Minister of Education and Science, Lord Bowden of Chesterfield, requested H. Peter Jost to '... consult with persons and bodies to establish the present position of lubrication, education and research in this country, and to give an opinion on industry's needs thereof', terms of reference whose parlance echo, with a certain faint nostalgia, the palmy days of 'The Mechanicals' and Beauchamp Tower!

On this basis a Lubrication Engineering Working Group of fourteen members under the chairmanship of Peter Jost was inaugurated, holding ten meetings and eight investigatory sessions to hear expert evidence.

Views were sought from wide and diverse sources in the United Kingdom and abroad, some 400 colleges and universities, industrial companies, corporated, and government research establishments and laboratories, government organizations and private individuals being consulted. From this mass of information, the Group formed the opinion that, at a conservative estimate, a potential saving to British Industry of some £515 million pounds per annum could be achieved arising from improvements in education and research at all levels, from the industrial shop floor to the university post-graduate researcher.

Early assessment of the entire gamut of the investigation led to the irrevocable conclusion that the word 'lubrication' held such subjective connotations that an entirely new generic term was required. This was not merely an exercise in semantics, but a genuine effort to produce a single work encompassing the entire range of the interlinking multidisciplinary fields of activity. With commendable scholastic zeal the English Dictionary Department of the

Oxford University Press was approached and, with their help, a single word 'Tribology' was chosen. The word, derived from the Greek 'tribos' (rubbing), was considered by some, at the time, to be the converse of what good lubrication practice was trying to acheive. Nevertheless, it was after due discussion, adopted - not only by the Group, but, with a dignified and wholly appropriate passage of time, by the concensus of industrial opinion. Its definition was as follows:

> 'Tribology is the Science and Technology of interacting surfaces in relative motion and the practices related thereto.'

The Report of the Working Group entitled 'Lubrication (Tribology) Education and Research - A Report on the Present Position and Industries' Needs (H.M.S.O. 1966), provided within its innocuous blue covers, the seeds of an economic bombshell.

The Jost Report, as it subsequently came to be called, made certain recommendations. In essence, these were as follows:

Educational

General and specialized courses at all levels from the shop floor upwards should be initiated, including the enlargment of undergraduate and postgraduate training; this would include specialized M.Sc. courses.

Research

Institutes specializing in Tribology should be established for basic and applied research, postgraduate instruction and industrial liaison. A two-way bridge with industry should be formed; for example through contract research and industrial research training.

General

An information center in Tribology should be established and a design and practice handbook published. The Institution of Mechanical Engineers was invited to extend the membership and activities of its 'Lubrication and Wear Group'.

In the light of the economic implications of the Jost Report, it is not surprising that, unlike so many good working committee reports before, it did not simply acquire dust on some ministerial shelf. Governments may be accused of lethargy in matters concerning the overall public 'good' but when pecuniary considerations of significant magnitude arise their 'response time' is little short of miraculous. So it proved to be, but, leaving aside the overtones of a £515 million carrot, the Working Group were to be congratulated on framing their report in so cogent a manner that any 'stick-waving' amounted to little more than a gesture to ritual. By 1972, most of the recommendations of the 1966 Report had been completely implemented.

The rest of Europe was not slow in following this initial impetus. In 1973, an International Tribology Council was formed with Peter Jost as its first President and by 1974 some twenty-one countries were represented on the Council. Thus Tribology flourished throughout Europe, its multidisciplinary nature pervading all quarters and at all levels in a spreading and interliking network. The long-term effects of this cannot, even at this time of writing, be fully assessed but the economic implications are manifestly obvious. Temporarily this genesis could not have been more opportune in the light of the world energy crisis that was, so swiftly, to follow.

2.5 The World Oil Crisis and the Conservation of Energy

Although to the casual observer the face of the petroleum oil industry may appear outwardly sanguine, the passage of time has seen violent changes reflected in its complexion, Simpson, Reference 100.

Perhaps the very nature of its parturition and dynamic evolution has given it a natural predilection for drama on the grand scale, but no single event since the war years has so thoroughly shattered the economic quiescence of the West than the onset of the world oil crisis. Almost overnight, it seemed the attitude of nations to the energy hoards stored in their economic coffers changed from that of inebriate spendthrifts to that of Draconian misers.

In the halcyon days before the 'OPEC' spectre sat at the feast, concepts such as 'Energy Conservation', 'Total Energy', and 'Total Technology' bore little or no credence. Suddenly, however, such strange terms became vibrant with meaning as this newly acquired truth penetrated into the entrenched bastions of governmental and industrial hierarchy, those august bodies establishing, with a hitherto uncharacteristic swiftness, that, at least as far as their fossil energy reserves were concerned, they were living on 'borrowed time'.

Yet this, in fact, was no newly created credo, no nascent economic theory of energy relativity. The semantic trappings might be different, the body of the argument carried, for certain tribologists at least, a decidedly familiar ring. Petrov, in the long paper of 1883 on 'Friction in Machines and the Effect of the Lubricant' produced formulae for estimating power loss and showed how a proper selection of lubricants dramatically reduced wastage of energy. He was awarded the Lomonosoff Prize of the Imperial Russian Academy of Sciences for his pains, and there the matter rested. Robert H. Thurston, impressed by the need for greater economy in the use of power, published in 1885 an extended edition of an earlier book entitled 'Friction and Lost Work in Machinery and Millwork', which went largely unnoticed by his fellow engineers in America. His, if anyone's, was a lone voice crying in the wilderness of these exuberant years of early oil exploitation.

Mayo Hershey, the doyen of American lubrication during the years of its meteoric expansion, had written in papers of 1933 and 1949 on the importance of the study of power loss in any engineering education[101, 102]. Through his now classical paper of 1936 'The Oil-Shed Fallacy, Attacking the Problems of Lubrication by Rational Methods', his inner feelings on the matter became abundantly clear[103]. Though the presentation is in a jocular vein and makes both humorous and delightful reading, the underlying current of concern and sincerity cannot pass unnoticed.

If any sort of conviction was lacking, however, a paper given by a European authority, Dr. Georg Vogelpohl, (1951) to the Third World Petroluem Congress at The Hague, must surely have shattered all illusions. In a study of world energy production and dissipation, Vogelpohl estimated that one third to one half of the world's energy production is consumed in friction. Even to the converted, this makes startling reading. The opening pages of Prof. Dudley Fuller's book: 'Theory and Practice of Lubrication for Engineers' contains further material on the energy question and is worthy of scrutiny[104].

In conclusion, it is evident that any future prognosis must wait for the outcome of current diagnosis. What is abundantly clear, however, is the role that multidisciplinary tribology must play if any sort of satisfactory outcome is to be acheived. To quote Mayo Hersey on the necessity for a broad base of scientific and technical education in the field of lubrication: "... it is a kind of pyramid, the attainable height depending on the extent of the base." It is the individual blocks that make the multidisciplinary base of our pyramid, however, and we must now turn to these in order to see how they interact and how then might be concisely assembled.

3. THE INTERACTION OF THE DISCIPLINES

3.1 The Surface Concept

We have, through our study of some of the chronology of Tribology and the several factors that, either acting singly or in combination, have influenced the interlinking of the disciplines, now reached a foundation point where it is possible to describe the interaction of the disciplines within specific boundaries. With Tribology reduced to an overall concept of proximate surface interaction, we may conveniently use this orthodoxy to illustrate the interaction of the disciplines themselves, at least as an initial starting point. Such illustration has already been used by Barwell and may, with certain modifications, be extended here to encompass a wider spectrum[105].

3.2 Function of a Lubricant

The author, in lectures to his students, has referred to the concept of 'Function of a Lubricant' which may, broadly, be defined by four categories.

(1) To eliminate or reduce wear between surfaces in relative motion.
(2) To reduce friction between such surfaces.
(3) To afford protection of the surfaces in hostile environments.
(4) To effect transfer of heat from the surfaces.

With regard to (1), we have already seen that we may eliminate wear entirely if surface asperity contact or particle contamination are precluded. Alas, with (2), however, we can never entirely eliminate friction, since every fluid film in viscous shear will dissipate energy.

Hostile environments may follow broad classes, examples being excessive load and speed, temperature extremes, chemical and abrasive environments, general contamination, radiation and space environments.

The importance of heat transfer is clerly self-evident; the lubricant and surfaces playing, in actuality, dual roles both being, in themselves, sources of generating heat (by viscous friction and/or asperity contact), while simultaneously acting as agents for its dissipation. If the former outweighs the latter, then a 'thermal spiral' is initiated and the surfaces seize.

3.3 The 'Surface Entity'

As a preliminary step let us postulate a hypothetical pair of surfaces and direct our thinking to the phenomenon of their interaction. Our hypothetical working surfaces may be characterized by the sequential order of their production and utilization e.g.,

(1) Production of Bulk Material
(2) Surface Heat Treatment (if any)
(3) Final Finishing Operations
(4) Application of Lubricant to Surface
(5) Bearing Working Environment and Duty.

Considering these (in the above) we will examine each as follows:

(1) The bulk material properties (mechanical strength, ductility, etc) will depend on the mechanical forming and heat treatment. Bulk intercrystalline stress will be manifest.
(2) Surface heat treatment may produce different metallurgical structures with, possibly, complex residual stresses in the surface giving inter-crystalline-changes in the component's outer skin.
(3) Final finishing operations dramatically affect tribological characteristics. Essentially, both the overall surface profile and surface topography will be determined by the methods employed at this stage. Additionally, a distorted skin microstructure will result with further lattice crystalline stress and, possibly, work hardening. Chemical amalgams also form, particularly with additive-loaded cooling fluids, in the elevated temperature environment between the metal removal agent and the component. Oxide films will quickly arise on the freshly cut area, permeating into the microstructure, together with contaminants such as water vapor, gases, and other chemicals all adding to the original amalgam. This type of surface can, in fact, react electronically itself, giving the possibility of inciting the lubricant's oxidation at the common interface by catalytic action. Thus, the component's surface skin will have entirely differnt frictional, hardness and strength properties from that of the original 'parent' metal before a lubricant is even used.
(4) Application of the lubricant to the surfaces immediately initiates further chemical reactions. Surface adsorbed layers form from reactions with

additives, such as metal passifiers and corrosion inhibitors, while fatty acids produce long-chain polar compounded layers over the whole surface, giving further chemical complexity.

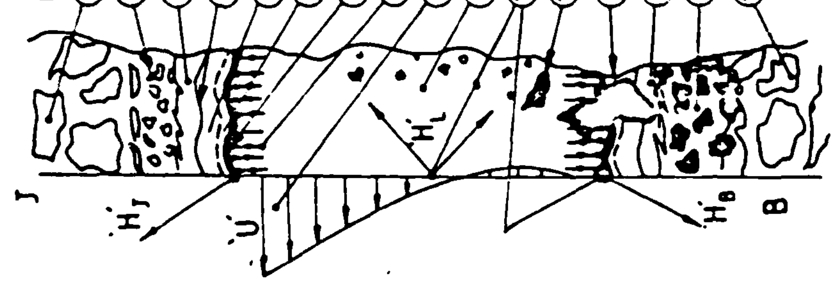
(5) Bearing working environment clearly modifies further our tribological picture. At higher temperature the initial surface chemistry is modified. E.P. additives, if present, will generate a controlled corrosion; asperity contacts modify overall surface profile and topography and, in their act of contact, generate heat locally with concomitant mechanical and chemical changes to the working surfaces. High loads produce elastic distortion and internal stresses which may, as in the case of pitting, adversely affect the whole mechanical integrity of the surface itself. High speeds increase viscous shear stress, reduce additive stability and produce further heat which both lowers the lubricant viscosity and causes thermal distortions. Debris, particles, chemical and gas entrainment mechanically and chemically affect the working surfaces and inhibit the action of the lubricants and additives. Embedded particles produce friction and wear of bearing surfaces, while cavitation may initiate surface damage in babbit or other soft materials.

Through this labyrinth of acting and reacting phenomena, therefore, we may now begin directly to establish the interaction of the disciplines themselves in relation to their applicability to the various phenomena. Figure 5 illustrates a schematic of our hypothetical surfaces for a hydrodynamic or elasto-hydrodynamic condition and gives immediately the required correlation of the disciplines in one entity. This we will designate, for further reference, 'The Surface Entity'.

4. MULTIDISCIPLINARY THINKING

4.1 The Importance of a 'Rational'

In the light of the phenomenist approach we have employed in the schematic of Figure 5 to illustrate the interaction of the disciplines through the interaction of the phenomena, it would clearly be of value to the tribologist anxious to have an eye on the 'tribological wood', if an overall rational, or 'tribological philosophy' could be established, not only in uniting the disciplines within a single entity but to establish a basic conceptualization of that entity in both its generalized form and function. Further, this philosophy could



No	DESCRIPTION	DISCIPLINE
1	Bearing Material.	Metallurgy, Mechanics of Materials.
2	Heat Treatment Structure.	Metallurgy, Production Heat Treatment.
3	Distorted Microstructure from Working.	Crystalogaphy, Electron Physics.
4	General Envelope of Surface.	Metrology, Production Engineering.
5	Surface Asperity Demarcation Zone.	Surface Topography, Metrology.
6	Oxides, Adsorbed Gas and Contaminants.	Inorganic Chemistry, Surface Physics.
7	E P and other Reactive Additive Layer.	Organic and Surface Chemistry.
8	Fatty Acid Polar Chain Layer.	Boundary Lubrication, Organic Chemistry.
9	Fluid Viscous Shear Generating Heat.	Hydrodynamics, Viscometry.
10	Generated Fluid Pressure.	Hydrodynamics, Instrumentation.
11	Heat Transfer to Lubricant and Surfaces.	Heat Transfer, Temperature Measurement.
12	Particles and Gas in Entrained Fluid.	Microscopy, Monitoring, Fluid Physics.
13	Embedded Particle Contamination.	Contact Mechanics, Friction and Wear.
14	Mechanical and Thermal Distortion.	Elasticity, Plasticity, Thermal Physics.
15	Soft Babbit (Duplex Structure).	Metallurgy, Materials Science, Microscopy.
16	Rigid Backing Material.	Mechanics of Materials, Materials Testing.

FIG. 5

FIGURE 5 INTERACTION OF DISCIPLINES IN THE ELASTO/ HYDRODYNAMIC REGIME of the Hersey number, until we reach the point of minimum friction coefficient sometimes referred to as 'Leloup's Point' or the 'unlatching point' by workers such as Gümbel[106, 28]. To the left of this point, asperity contact (i.e., Regime B) will be initiated, a progressively larger portion of the load being carried by this form of contact as the value of the Hersey number decreases, see Figure 6.

This situation is modelled by Figure 7 where we have a condition of mixed friction, part hydrodynamic, arising from the shearing of the pressurized lubricant trapped between the asperities, and what might loosely be described as 'boundary lubrication' at the interfaces of the sliding asperities. This situation is exceedingly complex since, insofar as the asperities are concerned, their deformation may be either elastic or plastic and their lubrication effected either by metallic soap films, E.P. additive action (if the temperature is high enough) or both, combined with 'chemical amalgam lubrication' from the initial surface films.

What is known, a priori, is that at these moving asperity conjunctions intense heat flashes are manifest, surfficiently hot to locally weld asperities in a transitory manner before they are torn apart, thus creating further crystal lattice deformation in the surface skin of the material.

4.3 The Interaction of Friction, Wear and Temperature

In experimental work on the above phenomenon, the author employed a unique design of a Four-Ball machine capable of continuously monitoring wear, friction, and contact temperature as the test proceeded[107]. Figure 8 (a trace of results using an 'E.P.' loaded oil) shows an initially rapid rate of wear and temperature rise with time, the former quickly becoming essentially constant while the latter reduces to a uniform level as the formation of the wear scar decreases the contact pressure. Heat soak into the bulk material follows, until the bulk temperature reaches a level for the E.P. additive to react chemically with the surface; the formation of the E.P. film causing an immediate drop in friction and therefore contact temperature. As the latter is lowered, the E.P. reaction ceases, the protective film is worn away and the cycle is repeated in a series of 'saw tooth' profiles, Figure 3. Thermocouples placed in the 'skin' of the materials (approx. 5 x 10^{-3} ins. below the contact) showed, additionally, violent temperature oscillations while bulk temperature response, itself, was relatively gradual, Figure 9. This tends to support the Flash Temperature concept proposed by Blok (1937), in that there are two components of temperature, a 'skin' and a 'bulk', in this type of

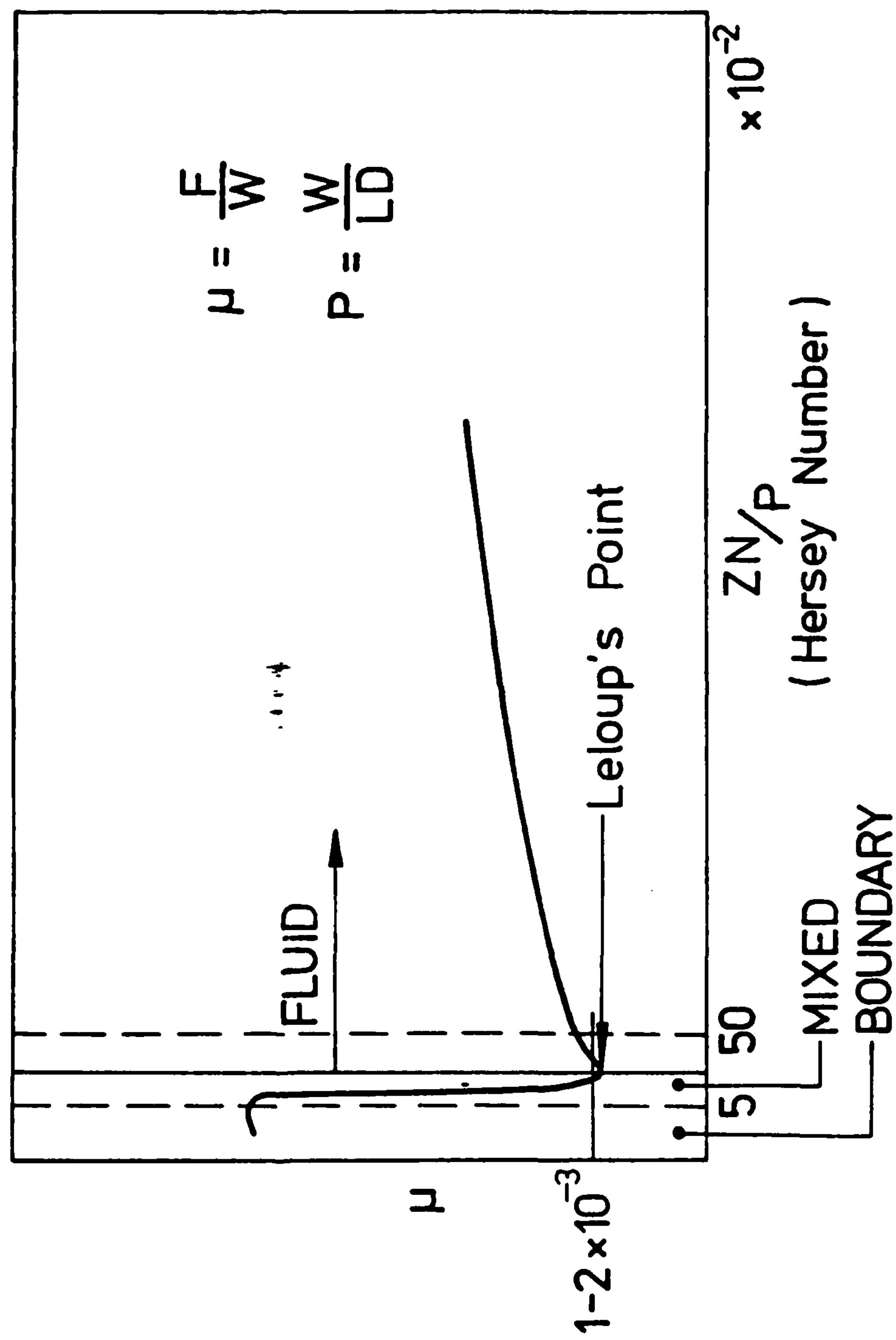


FIGURE 6 STRIBECK CURVE

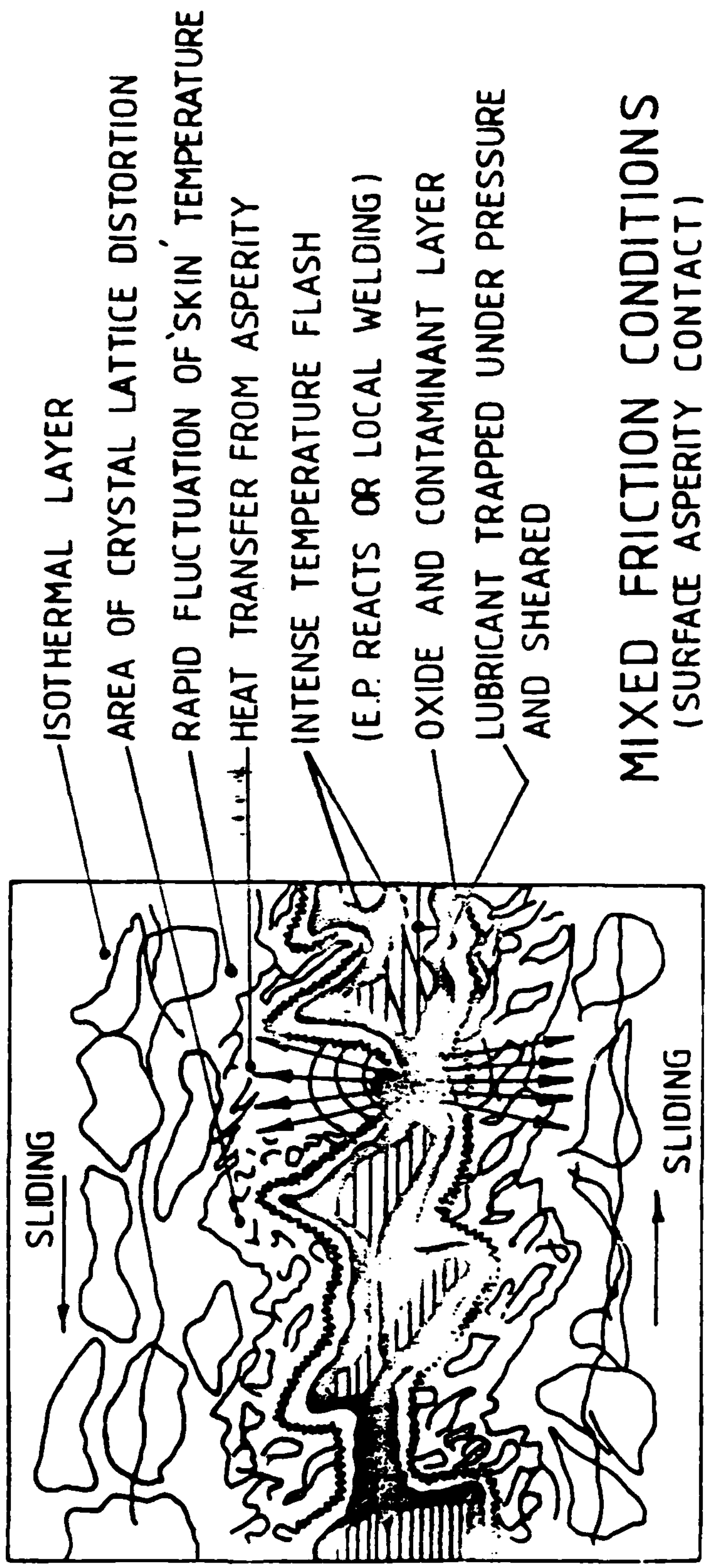


FIGURE 7 MIXED FRICTION CONDITIONS (SURFACE ASPERITY CONTACT)

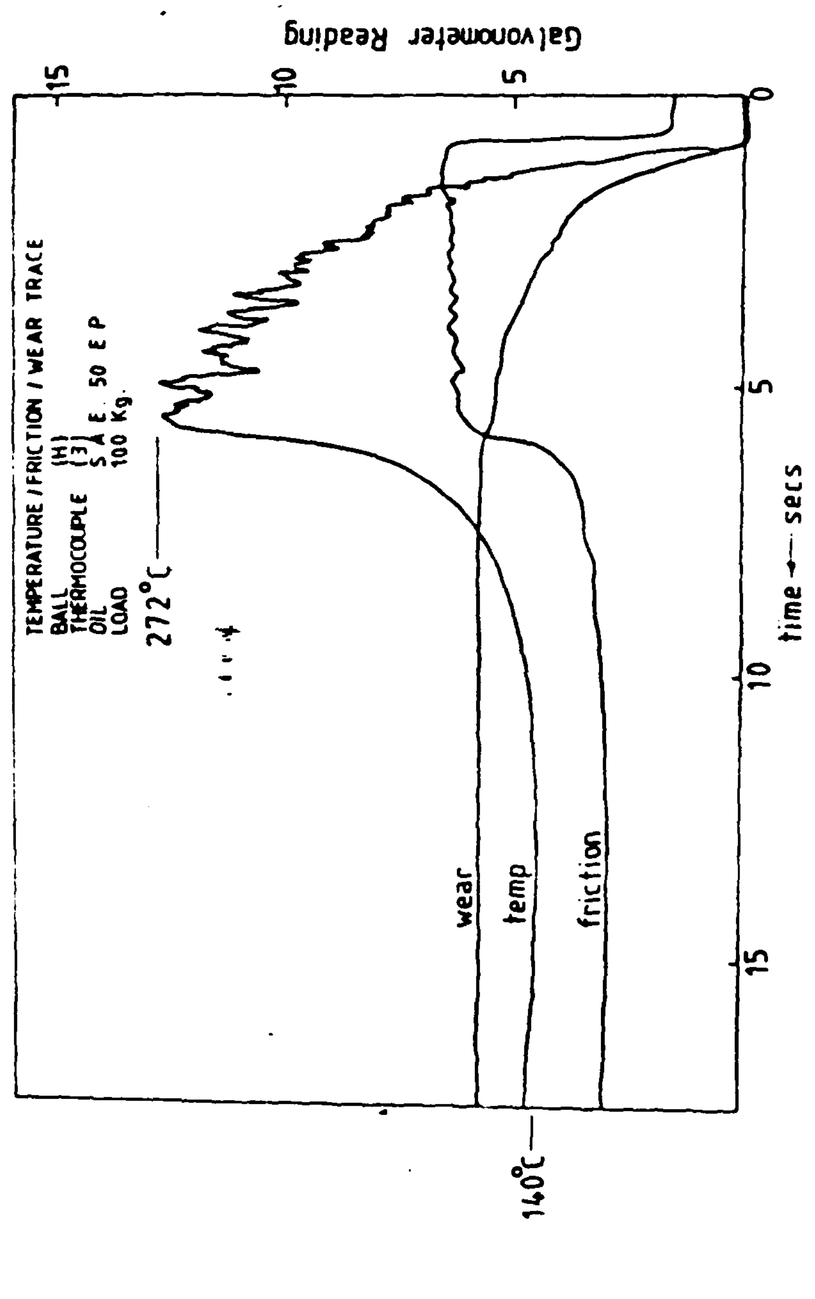


FIGURE 8 TYPICAL TEMPERATURE/FRICTION/WEAR TRACE FOR EXTENDED TEST RUN WITH AN E.P. LOADED OIL

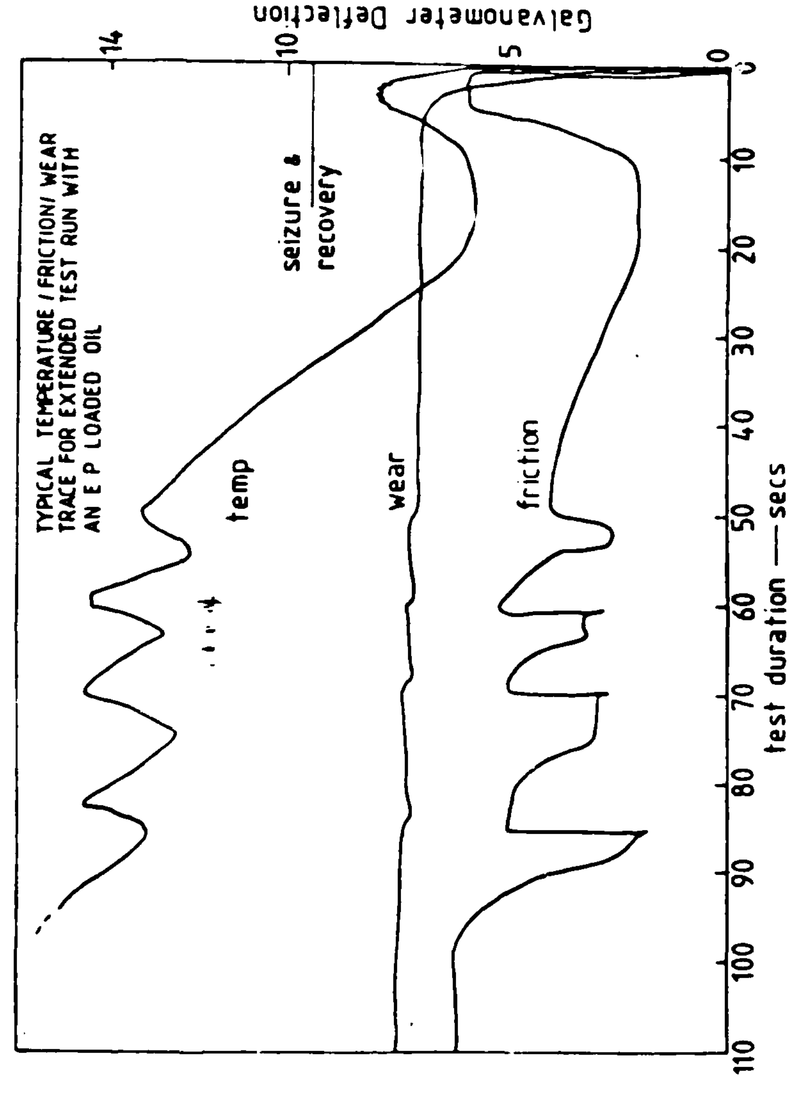


FIGURE 9 TEMPERATURE/FRICTION/WEAR TRACE

then be extended to embrace the question of the creation of that entity, and its subsequent development and utilization in an overall holistic methodology (including the maintenance of its efficient operation).

To do this, however, we must proceed in a series of logistic steps producing, from an initial fundamentalism, a progressional scheme of conceptual development.

4.2 The 'Micro' and the 'Macro' Models

As intimated above we must begin any logistical process from some generalized approach. Let us start, then, by dividing our surface contact into two geometrical categories:

(1) Conformal Surfaces
(2) Counterconformal Surfaces

which will be considered to be capable of operating in two distinct regimes:

(A) Hydrodynamic or Elasto-hydrodynamic
(B) Surface Asperity Contact or Mixed Friction Regime

Regime A - Using Figure 1 we may, therefore, initially define the confines of our exoteric boundaries (those which lend themselves to normal processes of synthesis) for either conformal or counterconformal conditions which we will term the 'Macro Model'. From the relationship between the configurations and idealized form of its components, we may establish, through a knowledge of geometry, trigonometry, kinematics, mechanics and dynamics, the outer function or 'External Manifestations' (E.M.) of the system.

Returning to Figure 5, our esoteric multidisciplinary 'Surface Entity' which we may term the 'Micro Model' of the system [containing its own inner, interlinked functions or 'Internal Manifestations (I.M.)] we proceed to place this, conceptually, at any relevant point within the conjoining surfaces of our 'Macro Model'. Thus, we immediately establish a unified whole, representing an extension of our concepts into one entity, possessing within itself both 'Internal and External Manifestations'.

Regime B - The next step on the logistical path is to return to our 'Surface Entity' or 'Micro Model' of Figure 5 and imagine a gradual, but progressive, dimunition of the fluid film between the surfaces. A corollary to this is to consider a translation of the working condition along the abscissa of the 'Stribeck' curve leading to a reduction in the magnitude contact[108]. Thus, the parametric (and therefore the multidisciplinary) nature of these interactions may be inferred from a study of such results, at least on a qualitative basis.

Returning to our 'Micro Model' we can, therefore, for a general liquid-lubricated contact, examine its 'Internal Manifestation' in the various regimes of lubrication (specified by translation through Hersey numbers) while simultaneously inducting it into our 'Macro Model'.

For visco-elastic lubricants the 'hydrodynamics' of the 'Surface Entity' model (as conceived) will not be altered, since heat will be generated whatever the lubricant. In such cases, as with solid lubricated or dry contact conditions, the friction coefficient itself may be used, instead of the Hersey number, to conceptualize parametric changes.

4.4 The Tribological Entity (T.E.)

The entity we have thus created represents an apperceptive paradigm or conceptual tribological model, by means of which we may interpret, through its 'External Manifestations' (its E.M.'s), the interaction with its outer environment over a multiplicity of situations and, as a result of this prescribed role, its 'Internal Manifestations' (its I.M.'s) arising from the reactive and interlinking phenomena between the proximate surfaces, characterized and assessed through the relevant disciplines and their own multiple interaction.

This created entity the author will term the 'Tribological Entity', later to be expanded into a larger overall complex embracing, as has already been intimated, not only the entity, per se, but its production development and utilization. It would be convenient at this stage to first categorize the disciplines specifically, on the basis both of the discussion so far and in the light of the ultimate object being considered.

4.5 The Categorization of the Disciplines

It is to be emphasized at the outset that any categorization of disciplines assembled with the object of utilizing such an aggregate in the process of tribological thought (in the multi-disciplinary sense) must be, by its very nature, entirely subjective. The author, while being aware of this, makes no apology for the schematic to be presented but makes this point initially, not to avoid objections but at least to mitigage extraneous digression. Methodology, in any field, is subjective in nature, any proponent being influenced both by his individual psyche and the conditioning process of his personal milieu; the author being, certainly, no exception to this universal verity. With this small caveat in mind, we will categorize the disciplines into eight primary 'cells' or 'blocks' with one 'Service Discipline'. These will be termed 'Central Disciplines of Tribology' and are defined as follows:

1. Discipline of Mechanical Engineering
2. Discipline of Materials Science
3. Discipline of Production Engineering
4. Discipline of Tribo-Engineering
5. Discipline of Chemistry
6. Discipline of Chemical Engineering
7. Discipline of Physics
8. Discipline of Electrical and Electronic Engineering together with:
9. Discipline of Mathematics (Service Discipline)

Each discipline may be delineated in terms of fields or subjects relevant to tribologial applications. In the context of the ground that has already been covered in the exposition so far, and in the light of the material to be presented, we may conveniently place this list in the form of an appendix. This also has the advantage of compactness in the interest of a clearer cognitive presentation, see Appendix 1.

A second subclass of disciplines is also to be recognized in which peripheral tribological activity has some relevance and which, in turn, motivates special tribological thought. These may be termed 'Peripheral Disciplines of Tribology' as follows;

1. Applied Medicine and Bio-Engineering
2. Management and Product Entity Planning
3. Energy Conservation
4. Ecology
5. Geo-Mechanics

4.6 The 'Hersey Pyramid'

Having now erected a model of the initial concept of a 'Tribological Entity' and defined the interdisciplinary cells or blocks in schematic form, our next task is to begin to integrate them into the 'Hersey Pyramid' referred to earlier in the text. At the risk of laboring the metaphor, we may describe the 'blueprint' of the structure as the <u>concept</u> of the 'Tribological Entity' and the disciplines as the building blocks.

Our next task is to produce a method jointly utilizing these to produce a working actuality, with all its manifest ramifications. This will be treated as an 'Application Methodology' requiring a basic elucidation.

5. APPLICATION METHODOLOGY

5.1 Concept of 'Extended Systems'

Before describing the details of the methodological approach, it is necessary to extend our concept of the 'Tribological Entity' itself.

We must recognize that, in applying an overall multidisciplinary approach to the problem of 'interacting surfaces in relative motion' and to take the 'Jostian' definition of Tribology to its definitve conclusion, 'the practices related thereto', it is necessary to extend our concept of an appropriate system beyond that of the narrow constraint of 'bearing' or 'gear' which our initial conformal and counterconformal illustration may have, however inadvertently, implied.

A workpiece, turning against a cutting tool and surrounded by an environment of cutting fluid, is clearly not a bearing, although a pair of proximate surfaces covered with articular cartilage and operating within a closed environment of synovial fluid might be considered as such; the likely generic description adopted however, certainly by the medical profession, would be 'animal joint'! The importance of the coefficient of friction between the surfaces of the objects in which we encase our feet and the contacting surfaces which support our weight is not readily obvious to the majority of our fellow men unless environmental conditions, such as a polished floor or a children's winter sled, bring these matters to their abrupt attention.

From the foregoing, it is seen that the number of various systems falling under the aegis of our definition of 'Tribological Entity' is likely to be exceedingly large.

5.2 'T.E.' 'E.M.' and 'I.M.'

In an initial appraisal of an approach to a methodology, we will, for brevity, use 'T.E.' for 'Tribological Entity' and 'E.M' and 'I.M.' for its 'External and Internal Manifestations'. We see that the 'T.E.' exists to perform its 'E.M.', but in order to do so, it must be capable of maintaining the integrity of its 'I.M.' within the confines envisaged for it by its creators. This gives us, then, an extended concept of the 'T.E.' as a self-maintaining entity within certain constraints, among which we may place both temporal constraint and service or maintenance constraints in addition to the normal design and production ones.

In the case of the cutting tool, for example, the 'T.E.'s 'E.M.' is to produce workpieces to whatever criteria have been specified; maximum throughput, minimum cost or predetermined tolerance, relative to its own specified integrity. To accomplish this, the 'I.M.' may involve maximum life of the 'T.E.' through, say, minimum friction and wear, together with minimum service and replacement costs for both tool and lubricant. All these considerations are inextricably bound with design parameters (such as tool geometry, material and lubricant choice and machine speed and feed) and must, in our present paradigm, be characterized by our 'Surface Entity' concept and background procedural planning (service and replacement costs, design adaptability, procurement feasibility) together with background factors such as product and market research and process optimization.

From this simple example we begin to see the relevance of certain subsections of the disciplines we have listed in our schematic of Appendix 1.

5.3 The Overall Tribological Entity ('O.T.E.')

In characterizing the 'T.E.' we have initially assumed a singularly specific system. There is no reason, however, why this concept should not be extended to embrace a series of tribological systems in terms of an 'Overall Tribological Entity' (O.T.E.) characterized by an 'Overall External Manifestation' (O.E.M.)

The 'O.E.M.' of a gearbox for example is, basically, to produce speed changes between its input and output power transmitters and/or to change their direction of operation. This system may contain complementary 'T.E.'s such as bearings, gears, fluid seals and slideways, together with external peripheral equipment serving the main unit, with their own subsets of 'T.E.'s e.g., pumps, valves, centrifuges and pressure actuators. The tribologist must therefore consider such a system as an integrated whole, conceptually linked with actual tribological problems arising as a result of the 'O.E.M.'.

Nor should the concept be restricted to a single machine complex. In optimizing the efficient lubrication of a steel mill, for example, or in setting up a tribological research laboratory, the modern tribologist must examine a further gamut of factors, not all technical; examples being overall economics, health and safety constraints, ecological factors and homo-ramifications.

5.4 The Function Factor

An application methodology which has seen extensive use at the Cranfield Institute of Technology over a wide spectrum ranging from specific industrial consultancy to the planning and commissioning of a complete tribological research laboratory, integrates the concept of the 'O.T.E.' to a number of interrogative factors termed 'Function Factors', which the author has found, may be gainfully employed where some form of 'O.T.E.' has, in the broadest sense, to be 'created and sustained'. The factors, which may be used either singly or in combination, are categorized as follows:

1. The Design Function Factor (D.F.F.)
2. The Production Function Factor (P.F.F.)
3. The Service Function Factor (S.F.F.)
4. The Market Function Factor (M.F.F.)
5. The Homo Function Factor (H.F.F.)

Each factor may be delineated in a series of subclasses. For brevity, as with the subclasses of the disciplines, we will place these for reference in appendix form, Appendix 2. Their elucidation and utilization will be illustrated in the following sections.

6. A MULTIDISCIPLINARY APPROACH IN DESIGN, DEVELOPMENT, AND MANUFACTURING OF AN O.T.E. AND ITS SUSTAINMENT

6.1 Methodological Background

The methodology to be enumerated in attempting to fulfil the assignment inferred in the title to this section, is not hypothetical. Its basis is one of simple pragmatism which, through dint of application and a process of evolution over a number of years, has assumed, rightly or wrongly, the trappings of a methodology. Its foundation rests on three 'Definitions' and three 'Enumerations' i.e.,

<u>Definition</u>

Step 1. The Definition of the 'Requirement'
Step 2. The Definition of the 'Problem'
Step 3. The Definition of the 'Constraints'

Enumeration

Step 1. The Function Factors
Step 2. The Disciplines Involved
Step 3. The Overall Analysis, Decisions and Results

6.2 The Use of Case Studies

The initial two specifications, although appearing hardly worthy of scrutiny, are considered by the writer to be of cardinal importance. To illustrate this point two case studies will be given, covering each.

Case 1. A requirement arose, during the manufacture of micro-transducers for a tribological research rig, for lengths of micro-bore (5×10^{-3} ins. dia) tubing to be cut. The author and his colleagues went to considerable pains to overcome the problem but without success, the severed end always becoming deformed in spite of a variety of different approaches. In examining a sample of the tube, forwarded by the manufacturers, the author suddenly realized that both ends were cleanly cut. A telephone call elicited the technique and the problem was immediately solved, although the manufacturers agreed they had, themselves, spent time and effort on the solution. This illustrates an erroneous definition of the requirement: 'We must solve this problem' instead of 'We must get this problem solved'. Time and effort may often be better conserved by lateral thinking, rather than longitudinal.

Case 2. The second example had much more serious ramifications involving the definition of a problem. A manufacturer of industrial machines placed a problem involving gas sealing between moving surfaces with a tribological organization. An analysis was made involving computer modelling of the hydrodynamic contact between the seal and the sliding surface. Recommendations were proposed and implemented, but the problem remained. At a later date the author happened to be involved, in a consultative capacity, on the problem and requested actual components from the machine instead of the drawings that had previously been furnished. Examination of the surfaces revealed extensive wear, the problem being, in actuality, one of surface contact mechanics and not hydrodynamics.

We see, from these two simple case studies, the importance of establishing, as quickly as possible, both the initial requirements and the nature of the problem. Clearly the whole concatination of procedural effort is totally negated if either of the initial definitions proves to be erroneous.

Proceeding then on the assumption that proper definitions have been forthcoming, we apply Step 3. to our problem, the 'Definition of Constraints'.

6.3 Constraints

Constraints may be both complex and multifarious, but it has been found useful to attempt to establish a distinction between what may be termed 'Primary' and 'Secondary' constraints. The classification of these will in general depend on what is being attempted and will differ from one project to another.

To avoid obfuscation, we will list the constraints in twelve broadly definable areas and leave the reader to extend or particularize them to his own subjective satisfaction.

6.3.1 Constraint Classifications

(1) Feasibility, (2) Environment, (3) Performance and/or Development, (4) Complexity, (5) Economic, (6) Spacial, (7) Weight, (8) Number, (9) Manufacturing, (10) Maintenance/Service, (11) Market, (12) Homo (e.g., Skill, Knowledge, Effect).

6.4 The Methodology Process

Having listed our constraints we must introduce them into our methodology, together with the 'Function Factors'. In general, we consider <u>all</u> constraints but in particular, we reduce them, in most applications, to a specific few; in fact, the fewer the better since, as we shall gather from the case studies to follow, initial constraints often engender the formation of new ones.

The methodology is essentially an iterative process in that, having stated the terms of reference of the problem and specified 'Primary' and 'Secondary' constraints, we form an 'initial decision'. This being done we scan the 'Function Factors' interrogatively and note down their relevance to our requirements. It is sometimes convenient at this stage to include the Step 2. Enumeration; 'The Discipline Involved' as we proceed or, alternatively, leave this until later. We may conveniently list the steps as follows:

1. State Terms of References through Definitions
2. Specify Primary and Secondary Constraints
3. Make Initial Decisions
4. Initiate 1st Function Factor Scan (Appendix 2)
5. Produce Intermediate Decisions
6. Specify New Constraints (if any)
7. Initiate 2nd Function Factor Scan
8. Repeat to Final Decisions
9. Intiate Actions as Necessary
10. Examine Results and Produce Conclusions.

The process is conveniently illustrated by two typical case studies, one for an 'O.T.E.' and one for a 'T.E.'. (Note: Conditions considered especially important carry underlining; thus A10).

7. CASE STUDIES

7.1 Case Study No.1, Type 'O.T.E.'

Subject: 'To Plan and Commission an Advanced-Level Tribological Research Laboratory'

Constraints

Primary - (6)* Spacial (Given Floor Area)
(8) Number (Maximum Rig Variety)
(3) Performance (High Quality Research)

Secondary - (5) Economic (Budget Limit)
(3) Development (Research Variation)
(10) Service and Maintenance (Minimum Cost)

Intial Decision

1. Small Size of O.T.E.'s (For Maximum Number)
2. High Integrity (Reliability, Low Maintenance Cost)
3. High Precision (Reliability and Quality of Results)

1st Function Factor Scan

D.F.F. (A1)** Mechanical Integrity
(A4) Rig Adaptability
(A7) Quick Dismantling
(A8) Extended Life
(A10) Good Development Potential
(A12) Minimum Costs

P.F.F. (B6) Internal/External Choice
(B8) Good Quality Control
(B10) External Limitations (Small Rig Availability, Small Transducers, High Costs).

* See Constraint Classification PSI.
** See Appendix 2

S.S.F. (C1) Signal Conditioning, Recording Equipment
(C3) 'In Service' Rig Development
(C7) No Field Data
(C9) Special Monitoring Instruments
(C11) Mainly Student Skill
(C12) Minimum Service and Maintenance Costs

M.F.F. (D1) Micro-Miniature Transducers Not Available
(D4) Manufactured Transducer and Signal Conditioning Equipment Expensive.

<u>Intermediate Decisions</u>

1. (A1 - A12) Design Own Test Rigs
2. (B6 - B11) Build Own Test Rigs. Manufacture Rig Components Internally. Buy Ancillaries Externally.
3. (B8, <u>B10</u>, C9, C12, <u>D1</u>, <u>D4</u>) Design, Build and Develop own Micro-Miniature Transducers.

<u>Decision Constraints</u>

Primary 1. <u>Feasibility</u> (Transducer Design, New Concepts)
2. <u>Manufacture</u> (Physical Size of Transducers, High Accuracy of Rig Components)

Secondary 1. Available Student/Technician Skill
2. Complexity ('Simple' Transducer Designs Required)
3. Economic (Signal Conditioning Equipment Costs)

<u>2nd Function Factor Scan</u>

Subject: 'Micro Transducers and Instrumentation'

(A5) Assess Design of Micro Transducers and Signal Conditioning Systems
(B1) (B9) Assess Overall Manufacturing Costs of Transducer Designs
(B1) (B10) Assess Total Cost of Manufactured Signal Conditioning Systems
(C11) (D1) (D9) Initiate Small Student Projects on Transducers

(C1) (C9) Assess Internal Instrumentation Potential (Manufacture of Equipment)
(B6) (B10) (C6) Research 'Market Capability'/Cost Situations
(B6) Good Probability of Student Interest and Enthusiasm
→ Motivation. Possibility of Students Training other Students
→ All Round Homo/Economic Benefit.

Final Decision Proceed as Intermediate Decisions, But Build Signal Conditioning Equipment Internally.

Results Test Rigs Built. Transducers Designed and Built. Students Acquire High Degree of Instrumentation Knowledge and Training. Good Student Research Facilities at Relatively Low Costs. Rapid Expansion of Laboratory. Diversification of Research. Economical Contract Research Capability. Unification of Multidisciplinary Methodological Approach.

Disciplines Utilized or Involved

(I) 1, 4, 6 (II) 1 (III) 1, 3, 4 (IV) 5, 6, 8, 9, 10, 11 (VII) 3, 12 (VIII) 1, 2, 3, 5.

7.2 Case Study No. 2. Type 'T.E.'

Subject: 'To Investigate the Performance of Porous Bush Journal Bearings in Domestic Fan Heater Units in the Light of Premature Bearing Failures'

(Work undertaken for U.K. Government - 'British Gas')

Constraints

Primary (3) Performance (Wear Integrity)
(5) Economic (Speicifed Contract Budget)
(3) Development (Units Already in Mass Production - i.e. Minimum Modification)

Secondary (9) Manufacturing (Fixed Procedure)
(2) Environment (High Temperature: 100°C plus)

Initial Decision

(1) Investigate Bearing Operating Performance
(2) Look at Possibility of Environment Simulation
(3) Research Manufacturer's - Fan Heater Units

1st Function Factor Scan

D.F.F.	(A1)	Unknown Mechanical Integrity
	(A2)	Unknown Excess Duty
	(A4)	Limited Design/Manufacture Flexibility
	(A5)	Basic Design Simple
	(A7)	Limited Dismantling Efficacy
	(A8)	Seeking Approx. 15,000 Hrs. Service Life
	(A9)	Likely Long-Term Employment
	(A11)	Geometry and Material Fixed
	(A12)	Essentially Minimum Cost
P.F.F.	(B1)	Mass Produced
	(B2)	Little Flexibility
	(B5)	Lubricant Options, Lubricant Impregnation Flexibility
	(B7)	None
	(B8)	Little Information - Need to Investigate
	(B9)	Low Costs Imperative
	(B10)	Units and Bearings Freely Available
S.F.F.	(C1)	Felt Lubricant Reservoirs
	(C2)	None
	(C3)	None
	(C4)	Fan Units Interchangeable in Service. (Difficulty in Changing Bearings)
	(C5)	None (Complete Fan Unit Replaced)
	(C6)	Unsatisfactory - Requires Definite Improvement
	(C7)	Minimal
	(C9)	Parameters to be Decided
M.F.F.	(D1)	Improved Reliability
	(D2)	Large Fixed Demand
	(D3)	Possible Development in Other Areas
	(D4)	Limited - Since Units Cheap
	(D5)	Relatively Low Unit/Volume Cost
	(D6)	Negligible
H.F.F.	(F1)	High Utilitarian Appeal
	(E6)	User Complaints (Service Life, Noise During Failure) Customer Satisfaction Important

Intermediate Decisions

1. Build Test Rig for Mechanical Evaluation
2. Batch Test Statistically (12 Units/Test)
3. Accelerate Test Time by Overload
4. Test Complete Unit 'In-Situ'
5. Use Extended Tests (Full 24 Hours if possible)
6. Monitor Bearing Temperature and Friction Continuously

7. Attempt to Simulate Environment
8. Restrict Transmitted Vibrations from Test Rig to Test Units.

2nd Function Factor Scan - Subject: 'Test Rig'

D.F.F.
- (A1) Long Term Wear/Mechanical Integrity
- (A2) Testing Overload Capability
- (A4) Bearing Interchangeability in Fan Unit
- (A5) Basically Simple Design - Continuous Running Capability
- (A6) Ease of Fan Unit Changes
- (A7) Minimum Dismantling Time
- (A10) Acquired Results Considered for Other Applications
- (A11) No Extraneous Forces of Vibrations to be Transmitted; Accurate Monitoring on a Continuous Basis
- (A12) Overall Costs within Budget

P.F.F.
- (B2) Minimum Internal Commitment
- (B6) Manufacture Components Externally
- (B8) High Quality Control on Manufacturing (Unit Interchangeability)
- (B9) Rig Costs to be an Absolute Minimum (Component Quotations Required)
- (B10) None - Manufacturer Willing to Supply Bearings and Fan Units, Gratis.

S.F.F.
- (C1) Data Logger and Tape Punch for Monitoring
- (C3) Rig Adaptability in Service
- (C5) Bearing Interchangeability
- (C9) Continuous Monitoring of Bearing Temperature and Torque - Fixed Load and Speed
- (C10) Engage a Full-Time Research Officer.

H.F.F.
- (E2) Minimum Manual Attention
- (E6) Specialist Research Officer

Final Decisions on Rig

1. Multi-Unit Test Rig - Extended Tests under Fixed Overload
2. Operate Loaded Units on Air Bearings (No Vibration or Extraneous Friction)
3. Proceed with Mechanical Testing Only, Initially. (On Basis of Cost and Complexity)
4. Fully Automate Parametric Monitoring. Computer Plot and Process Data.

Result from Rig Tests

Rig showed wear integrity to vary between batches of bearings. Not necessarily load dependent. Higher friction and temperature noted in certain cases with lighter loads, but correlated with load when lubricant impregnation carried out in laboratory.

Final Decision

Quality control of lubricant impregnation during manufacture suspect. Likely to be prime cause of the reported premature failures.

Results

New research initiated on impregnation techniques and lubricant and additive optimization. New numerical solution of porous bush bearing lubrication produced and published[109].

Further research investigations in this field currently proceeding, including Mechanical Modelling. Extensive multi-disciplinary interaction. Methodology qualified.

Disciplines Utilized or Involved, (See Appendix 1);

(I)	1, 2, 3, 5, 6, 7, 8
(II)	1, 2, 4, 5, 7, 9
(III)	1, 3, 4
(IV)	1, 2, 3, 4, 5, 7, 9, 10, 11
(V)	1, 2, 4, 5, 6
(VII)	2, 3, 8, 12
(VIII)	1, 2, 3, 4
(IX)	1, 2, 3, 5, 6

8. CONCLUSION

On the simple premise that human nature remains relatively immutable throughout the processes of time, although allowing that expressionistic or behavioristic patterns are undoubtedly conditioned by events in any particular milieu, we can by an investigatory examination of historical epochs in which progress, innovation, and a general flowering of human cooperation and endeavor arose, elicit an overall appreciation of the various motivating forces contributing to such a renaissance. Conversely, by a like process we can observe, in general terms, factors contributary to the antithesis of such a desirable homogeneity. These are the facts that history, itself, makes plain.

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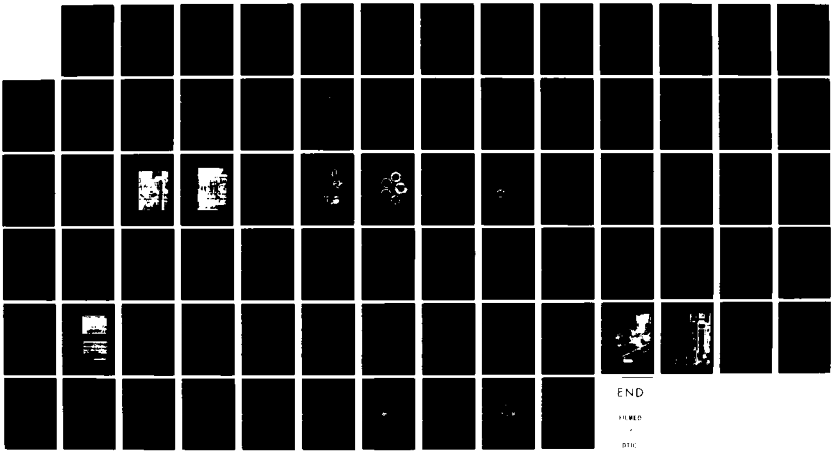

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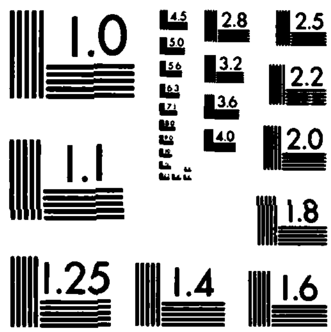

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

Technology is considered by the author to be no exception to this rule and he has therefore, throughout this expose, stressed the importance of obtaining insight into the recognition of our current situation from a study of the history of tribological development. Segregative attitudes chiefly arise when the overall commonality of problems becomes lost or obscured.

We are all, consciously or unconsciously engaged, insofar as the tribological field is concerned, in multidisciplinary action of some sort and it is, perhaps, through the creation of some basic paradigm that our appreciation of this fact is heightened. This the author has attempted to do and, using the methodology and case studies outlined, has indicated the possibility of generating an overall holistic approach to tribology. It is hoped that this may, with due allowance, assist tribologists, (through multidisciplinary insight rather than direct revolution or schism) towards that most desirable process, to recapitulate the 'Jostian' metaphor, of finally casting off their 'molecular chains'.

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10. NOTATION

C	= bearing radial clearance
D	= diameter of shaft/bearing
e	= bearing eccentricity
F	= frictional force
G	= materials parameter
h	= local fluid film thickness
h_o	= film thickness at maximum pressure
h_m	= minimum film thickness
H	= $hm/_R$, dimensionless film thickness
L	= bearing length
N	= rotational speed
p	= local fluid pressure
P	= W/LD, specific pressure
R_1, R_2	= local radii of curvature, $\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$
U, U_o, U_1	= surface velocities
V	= surface velocity
W	= load on bearing

x	= coordinate in direction of U
y	= coordinate through fluid film
z	= coordinate in direction of V
ε	= e/c, bearing eccentricity ratio
θ	= angular coordinate
η	= fluid viscosity
μ	= $\frac{F}{W}$, coefficient of friction

11. APPENDICES

11.1 Principal Interdisciplinary Fields of Tribology

11.2 Schematic for Methodology in the Design, Production, and Marketing of an Overall Entity

Appendix 1

PRINCIPAL INTERDISCIPLINARY FIELDS OF TRIBOLOGY

I. DISCIPLINE OF MECHANICAL ENGINEERING

SUB-SECTIONS OF DISCIPLINE

1. Machine and Element Design
2. Hydrodynamics and Fluid Mechanics
3. Heat Transfer and Thermodynamics
4. Mechanics of Materials
5. Metrology and Surface Topography
6. Mechanical Testing and Condition Evaluation
7. Environmental and Field Evaluation Methodology
8. Mechanical Modelling Techniques
9. Vibration Technology

II. DISCIPLINE OF MATERIALS SCIENCE

SUB-SECTIONS OF DISCIPLINE

1. Material Integrity Science
2. Metallurgical Engineering
3. Polymer, Ceramic, Mineral and Composite Materials
4. Friction Material Science
5. Surface Failure Mechanics
6. Contamination and Corrosion Science
7. Hard and Soft Material Science
8. X-Ray Crystallography and Electron Microscopy
9. Analysis Techniques: 'Ferrography', Traces, Surface Topology, 'Quantimet' etc.

III. DISCIPLINE OF PRODUCTION ENGINEERING

SUB-SECTIONS OF DISCIPLINE

1. Material Removal Technology: Cutting, Grinding, Lapping, Stamping etc.
2. Material Displacement Technology: Forming, Forging, Drawing, Rolling, Extruding, Spinning, Rivetting.
3. Material Shaping Technology: Casting, Moulding, Fabrication, Powder Metallurgy.
4. Special Process Technology: Electro-Chemical, Ultra-Sonic and Spark Machining Technology, Surface Plating, Friction Welding, Surface Treatment and Coating Technology, Heat Treatment, Chemical Treatment.
5. Tribological Aspects of Production Methodology.

IV. DISCIPLINE OF TRIBO-ENGINEERING

SUB-SECTIONS OF DISCIPLINE

1. Lubrication Function and Regime Specification
2. Lubricant and Material Specification Methodology
3. Lubricant and Cutting Fluid Applications Technology
4. Tribological Technology in Hostile Environments
5. Tribological Systems Technology
6. Tribological Field Appraisal Methodology
7. Tribological Malfunction and Diagnosis Assessment
8. Special Applications Tribology
9. Application Constraint Specification
10. Ancillary Element/Systems Technology
11. Categorised Field Applications

Appendix 1
(Continued)

PRINCIPAL INTERDISCIPLINARY FIELDS OF TRIBOLOGY

V. DISCIPLINE OF CHEMISTRY

SUB-SECTIONS OF DISCIPLINE

1. Liquid Lubricant (Petroleum & Synthetic) Technology
2. Additive Formulation and Development Technology
3. Solid and Semi-Solid Lubricant Technology
4. Lubricant Analysis, Evaluation and Testing Methodology.
5. Surface Contact Chemistry (including Surface Kinetics).
6. Lubricant and Surface Contamination Chemistry
7. Polymer and Special Tribological Materials Chemistry.
8. Surface Process Development Chemistry
9. Toxicity, Health and Ecological Monitoring Methodology.
10. Chemical Analysis Techniques: Spectography, Surface Chemical Analysis, Gas Chromotography, Specific Tests etc

VI. DISCIPLINE OF CHEMICAL ENGINEERING

SUB-SECTIONS OF DISCIPLINE

1. Petroleum Lubricant Production Technology
2. Synthetic Lubricant and Additive Production Technology
3. Solid and Semi-Solid Lubricant Production Technology
4. Polymetric and Special Tribological Material Production Technology
5. Process Fluid Production Technology
6. Lubricant Handling, Dispersing and Storage Methodology
7. Lubricant and Materials Marketing Management

VII. DISCIPLINE OF PHYSICS

SUB-SECTIONS OF DISCIPLINE

1. Rheology
2. Surface Contact Physics
3. Optics and Electro-Optics
4. X-Ray Defraction and Crystallography
5. Irradiation Technology
6. Acoustic Emission and Condition Monitoring
7. Laser and Holographic Technology
8. Contamination, Filtration and Particle Monitoring
9. Vacuum Physics
10. Extreme Temperature Physics
11. Radiation and Atomic Physics
12. Transducer and Instrumentation Physics.

VIII. DISCIPLINE OF ELECTRICAL AND ELECTRONIC ENGINEERING

SUB-SECTIONS OF DISCIPLINE

1. Electrical Instrumentation and Transducer Technology
2. Signal Transmission and Data Acquisition Technology
3. Signal Conditioning and Display Methodology
4. Electrical Contact Science
5. Electronic and Solid State Engineering
6. Magnetic and M.H.D. Bearing Technology
7. Electrical Process Technology
8. Electrical Discharge Science
9. Electrical Analogue Simulation

Appendix 1
(Continued)

PRINCIPAL INTERDISCIPLINARY FIELDS OF TRIBOLOGY

IX. DISCIPLINE OF MATHEMATICS (SERVICE DISCIPLINE)
1. Classical Analytical Methods
2. Finite Difference and Finite Element Methods
3. Analogue and Digital Computer Methods
4. Computer Aided Design Techniques
5. Statistical and Probability Methods
6. Mathematical Modelling Techniques
7. Mathematical Comparology

I. DISCIPLINE OF MECHANICAL ENGINEERING
II. DISCIPLINE OF MATERIALS SCIENCE
III. DISCIPLINE OF PRODUCTION ENGINEERING
IV. DISCIPLINE OF TRIBO-ENGINEERING
V. DISCIPLINE OF CHEMISTRY
VI. DISCIPLINE OF CHEMICAL ENGINEERING
VII. DISCIPLINE OF PHYSICS
VIII. DISCIPLINE OF ELECTRICAL AND ELECTRONIC ENGINEERING

Appendix 2

SCHEMATIC FOR METHODOLOGY IN THE DESIGN, PRODUCTION AND MARKETING OF AN OVERALL ENTITY

A. DESIGN FUNCTION FACTOR (D.F.F.)

1. Mechanical/Wear Integrity
2. Excess Duty Potential
3. Energy Conservation Efficacy
4. Design/Manufacture Flexibility
5. Design Complexity Assessment
6. Specific Component Accessibility
7. Component Dismantling Efficacy
8. Component/Entity Life Requirement
9. Component/Entity Obsolescence Norm
10. Entity Development Latency
11. Specific Design Constraints
12. Influencing Economic Factors

B. PRODUCTION FUNCTION FACTOR (P.F.F.)

1. Production Potential
2. Manufacturing Commitment
3. Material Commitment/Availability
4. Machine Availability/Adaptability
5. Process Availability/Adaptability
6. Manufacturing Options
7. Material Options
8. Quality Control Capability
9. Production Economic Constraints
10. External Component Availability/Cost
11. Entity/Component Production Time
12. External Processes/Delivery Constraints

C. SERVICE FUNCTION FACTOR (S.F.F.)

1. Peripheral Service Requirement
2. Component Maintenance Requirement
3. In-Service Adaptability
4. Service Interchangeability
5. Unit Replacement Capability
6. Statistical Field Integrity Norm
7. Field Service Data Availability
8. Field Induction Methodology
9. Entity/Component Monitoring Requirement
10. Service Environmental Constraints
11. Service Skill Constraints
12. Service Economic Status

D. MARKET FUNCTION FACTOR (M.F.F.)

1. Market Research Specification
2. Overall Market Requirements
3. Latent Market Potential
4. Market Option Status
5. Entity Marketing Cost (Unit and Volume)
6. Entity Operating Cost Evaluation

E. HOMO-FUNCTION FACTOR (H.F.F.)

1. Utilitarian and Psychological Appeal
2. Ergonomic Efficacy
3. Health and Safety Constraints
4. Ecological Acceptability
5. Dispensing and Handling Constraints
6. Homo-Liability Ramifications

APPENDICES

Appendix A

PRESSURIZED BEARINGS

D. Koshal
Liverpool Polytechnic

W. B. Rowe
Brighton Polytechnic

1. INTRODUCTION

Introducing the subject of pressurized bearings it is important to distinguish between hydrostatic bearings more generally known as externally-pressurized, and pressure-fed hydrodynamic bearings which are alternatively termed self-acting bearings. The two types of journal bearings are illustrated in Figure 1. The term hydrostatic usually refers to externally pressurized liquid bearings rather than gas bearings. Hybrid bearings combine hydrostatic and hydrodynamic features. The following discussion is concerned with both liquid and gas bearings.

1.1. Hydrodynamic Bearings - Advantages and Disadvantages

The hydrodynamic bearing is supplied with liquid under pressure at a point near the maximum film thickness. Rotation of the journal draws the lubricant into the heavily loaded region and separates the two surfaces. Although such bearings may be fed under pressure, the point of entry is normally opposite the applied load or near the point of maximum film thickness. Hence the pressure does not act to support the load, in fact, quite the opposite occurs. The reason for pressure-feeding is to ensure that a sufficient volume of oil or other fluid enters the bearing clearance. This is necessary to ensure sufficient oil to maintain a full film throughout the loaded portion of the bearing and also to prevent overheating. Since the load support mechanism depends

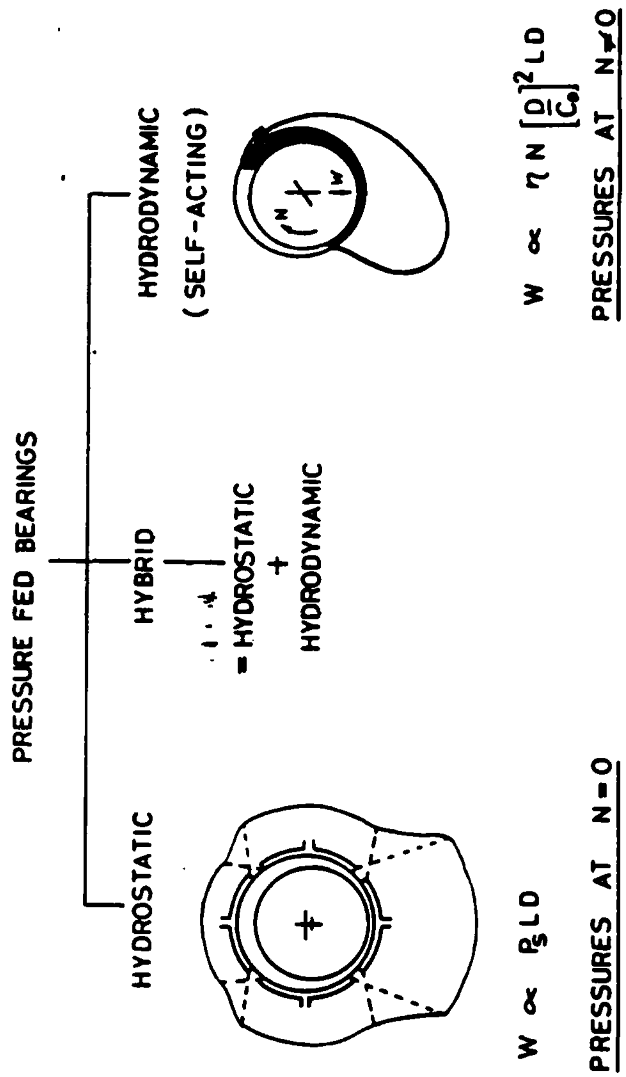


FIGURE 1 TWO TYPES OF JOURNAL BEARINGS

entirely on movement of the bearing surfaces towards the thin film region, it follows that there is no fluid-film load support at zero speed. Starting and stopping hydrodynamic bearings under load will inevitably lead to wear. The bearings illustrated in Figure 2 are all examples of self-acting or hydrodynamic bearings which may or may not be pressure-fed. The attraction of most hydrodynamic bearings of which there are a wide diversity of types, is the compactness and simplicity of the system for a high load-carrying capacity at appropriate speeds. Possible problems at high speeds which may be overcome with hydrostatic bearings, are whirl instability and high running temperatures. Gas bearings have been reported at extremely high speeds although the operating film pressures are much lower.

1.2 Hydrostatic Bearing - Applications

The hydrostatic bearing provides fluid-film load support by a completely different mechanism. As shown in Figure 1, the fluid is supplied to a number of entry ports around the bearing from a source at a supply pressure P_s, which must be high enough to support the load. The pressures in the bearing film are controlled so that the highest pressures act to oppose the applied loads. As a result of the pressurized supply and the associated control devices, the fluid-film support does not depend on speed and such bearings may operate at any speed down to zero with full fluid-film separation of the surfaces. The ability to operate at zero speed and high speeds with any load capacity determined by the supply pressure is one of the principal advantages of hydrostatic bearings. In addition, high stiffness may be achieved and in the case of liquid lubricated bearings, the stiffness can be designed independently of the load. This allows the designer to determine the bearing performance to suit the requirements of the machine. Other features of hydrostatic bearings are low starting torque, zero friction at zero speed, absence of start-up wear, high accuracy of location and smoothness of movement, good dynamic stability and cool operation with suitable design. The disadvantages are the requirements for more expensive and bulky hydraulic equipment, control restrictors, and effective filtration to prevent blockages in the supply. Some examples of liquid hydrostatic bearings are shown in Figure 3 and will be discussed in more detail in connection with the design of bearing systems.

Hydrostatic bearings have been employed very successfully for many years in a number of low-speed machines which require high load support and low friction in order to achieve high precision in positioning. In the USA several radio telescopes have been supported on hydrostatic bearings. A notable

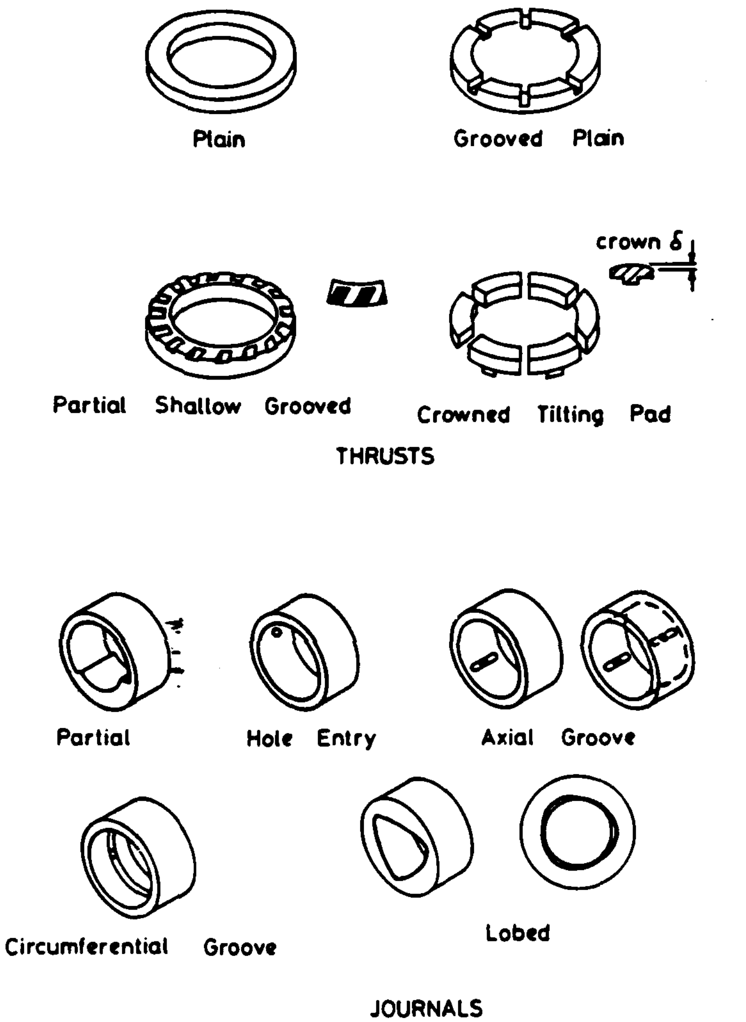


FIGURE 2 SOME EXAMPLES OF HYDRODYNAMIC BEARINGS

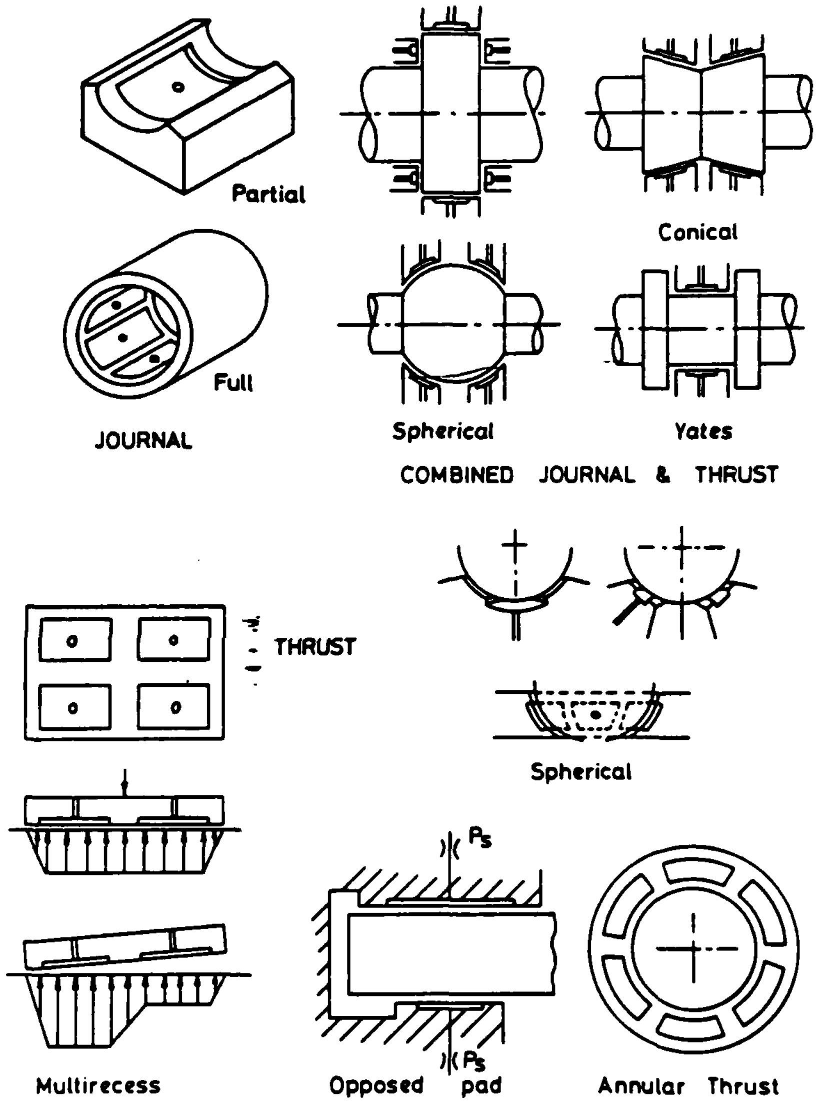


FIGURE 3 EXAMPLES OF HYDROSTATIC BEARING CONFIGURATIONS

example includes the 63 m (210 ft.) diameter, Goldstone radio antenna. Hydrostatic bearings have also been widely applied in machine tool slideways and spindles for low wear, high positioning accuracy, and high bearing stiffness. It is important that the bearings of machine tools are not subject to wear which reduces the resistance to chatter in metal-cutting operations and makes it difficult to maintain machining tolerances and production rate. Such features are particularly important for automatically controlled machine tools. Hydrostatic spindle bearings have been employed in a range of metal-cutting, grinding, and measuring machines at low and high speeds.

Other applications include experimental apparatus such as support bearings for bearing test-rig journals and dynamometers, as well as bearings and seals in hydraulic motors where a ready source of pressurized oil is available. A feature of hydrostatic bearings which could be important in some applications is a strong vibration damping action which has been applied for noise damping.

1.3 Externally Pressurized Gas Bearing Applications

Externally pressurized bearings may also be supplied with air or other convenient gas. A similar range of thrust and journal bearings may be designed, although gas bearings are usually designed as plain bearings for stability. Two typical gas bearing configurations are shown in Figure 4. Gas bearings offer very low friction and are often operated from the workshop compressed air supply. Careful filtration is essential.

Externally pressurized gas bearings have many of the advantages of liquid lubricated hydrostatic bearings including the possibility of constructing extremely high precision movements. Exceptionally low friction even at high speeds is a distinctive feature of E.P. gas bearings which tend to be completely cool running. A feature which may be of importance is the freedom from contamination when employing a clean inert gas. Slot-entry bearings have the additional advantage that they are commercially available from the Horstmann Gauge Company.

E.P. gas bearings are more often operated with lower pressures than liquid hydrostatic bearings for convenience and safety. This means that lower loads can be supported for the same size of bearing. E.P. gas bearing applications include high-speed dentists drills, turbine flow-meters, gyroscope bearings, grinding machine spindles, and roundness measuring machines. In each case, the virtual elimination of friction and wear is of overriding importance for the achievement of accuracy of positioning, reliability, or very high speeds.

1.4 Hybrid Bearing Applications

When a hydrostatic bearing operates at speed there is extra load support from the fluid-film due to hydrodynamic effects. The bearing is then said to operate in a hybrid manner.

A similar effect can be achieved with externally pressurized gas bearings, although the application of hybrid gas bearings has not received the same degree of consideration as hybrid liquid bearings.

Hybrid bearings may be designed to optimize both the hydrostatic and the hydrodynamic performance to achieve high load-carrying ability economically. Hybrid bearings, when designed appropriately, perform as superior hydrodynamic bearings at speed with the attractive features of hydrostatic bearings at low and high speeds. These features include good load capacity and stiffness independent of speed, low start-up torque and absence of wear, high accuracy of location and smoothness of motion, and good dynamic stability and cool operation when appropriately designed. The additional features are high overload capacity at high eccentricity, the ability to employ higher viscosity oils at high speeds, and the tolerance of wider variations in manufacturing clearance. The hybrid bearing is also superior to both axial groove and circumferential groove hydrodynamic bearings where heavy dynamic loading is applied in widely varying directions. The disadvantage of hybrid bearings is the same as for hydrostatic bearings; it is necessary to provide auxiliary hydraulic equipment. However, it is possible that a lower pressure system will suffice in view of the high overload capability. Hybrid bearings avoid recesses for maximum hydrodynamic effect and hence, in appearance and construction have more in common with externally pressurized gas bearings as shown in Figure 4.

There are obvious applications where plain hybrid bearings would have performance advantages. These applications are in high-speed machines where hydrodynamic bearings tend to suffer from whirl instability as in generator sets and turbines. Machine tools for intermittent cutting operations, or where shock loads and heavy overloads may occur on occasions, are also suitable applications.

A special class of hybrid bearing is the jacking bearing, in which the bearing is jacked hydrostatically under pressure to separate the bearing surfaces and hence avoid wear and high

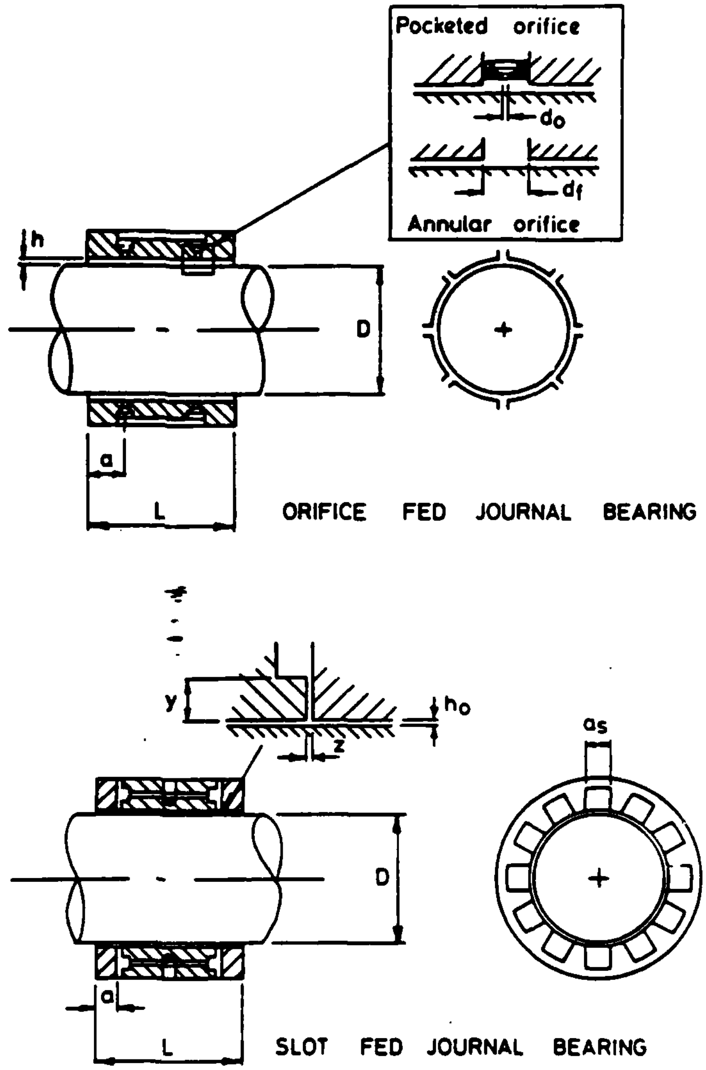


FIGURE 4 TYPICAL GAS BEARING AND PLAIN HYBRID BEARING CONFIGURATIONS

friction under starting and stopping conditions. At speed the supply pressure may be reduced or even switched off to allow the bearing to operate purely hydrodynamically.

2. EXTERNALLY PRESSURIZED BEARING SYSTEMS

A hydrostatic bearing essentially involves two resistances in series. The fluid flows through an orifice restrictor or other control resistance and then through the resistance formed by the bearing clearance.

The basic principle of a hydrostatic bearing is illustrated in Figure 5. A fixed displacement pump supplies oil to a capillary restrictor at a constant supply pressure, P_s, which is controlled by a relief valve. The pressure is reduced through the capillary and arrives at the entry to the bearing at recess pressure, P_r. The pressure further reduces as the oil flows through the restriction caused by the bearing gap, h. If the bearing gap decreases by a distance, δ, the restriction to flow through the bearing increases and the recess pressure increases. The relationship between bearing clearance and recess pressure will depend on such features as the nature of the control device, in this case a capillary, and parameters such as the supply pressure and the oil viscosity. An important aspect of hydrostatic bearing design is careful attention to filtration. The main filter is often placed in the return line from the relief valve to the tank. This filter operates to extract particles whenever the pump is switched on. A blocked orifice or capillary restrictor can easily occur and bearing failure may result. A line filter of the type used in carburetor fuel lines is a simple and useful additional protection for the restrictor.

Figure 6 indicates the principal types of externally pressurized bearings. In each case, the bearing may be designed for liquid or gas operation although the detail design will require to be quite different.

Liquid and gas hydrostatic bearing configurations mainly fall into three categories; journal bearings which allow shaft rotation, thrust bearings which allow sliding on a flat plane, and combined journal and thrust bearings which allow shaft rotation with axial constraint. Some examples of these three groups of liquid lubricated bearings are shown in Figure 3.

A common oil-lubricated journal bearing configuration is the cylindrical bearing containing 4, 5, or 6 recesses. Each recess must be controlled by its own restrictor. Recesses are not an essential feature of hydrostatic journal bearings and

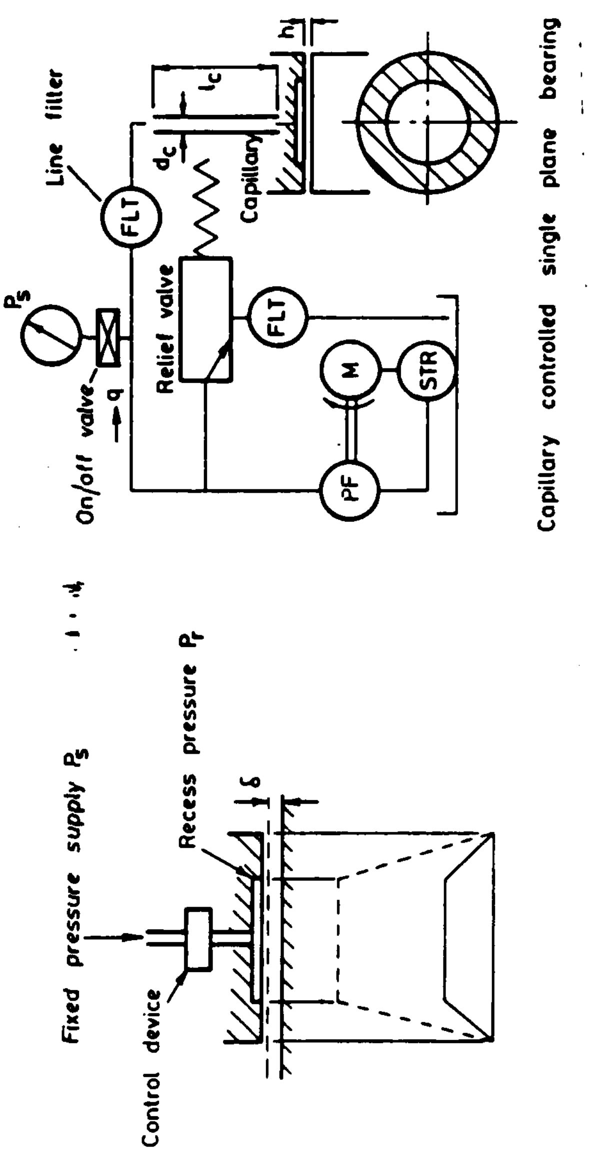


FIGURE 5 BASIC HYDROSTATIC BEARING SYSTEM AND TYPICAL PRESSURE DISTRIBUTION

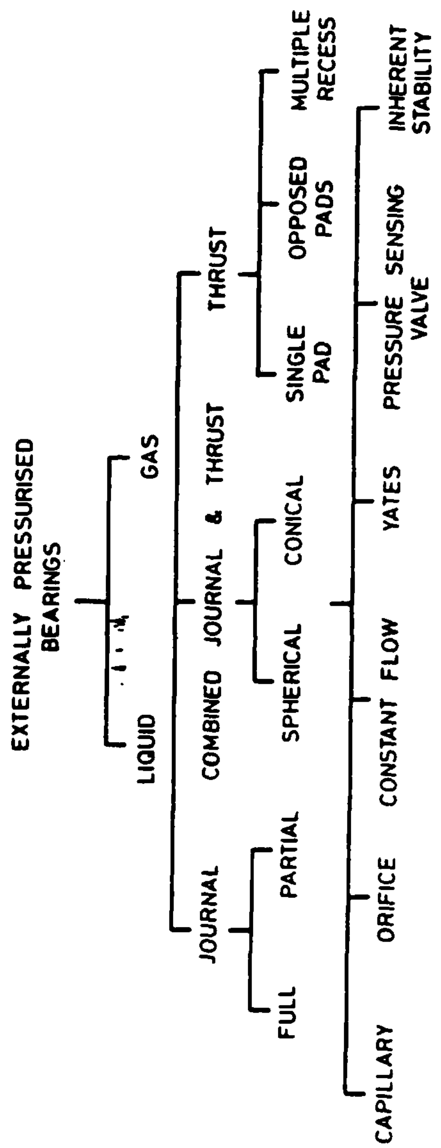


FIGURE 6 EXTERNALLY PRESSURIZED BEARINGS

the alternative plain bearing configurations may offer advantages for manufacturing simplicity and reliability.

For very large oil-lubricated journal bearings and high speed journals, the designer may well experience problems in specifying reasonable tolerances for manufacture. There may also be problems of high power consumption and flow-rate. Both problems may be reduced by employing partial journal bearings. Usually a pedestal will contain two partial bearings angularly disposed to cradle the journal.

Most journal bearings require axial constraint and a common method is to employ a thrust flange supported between two annular recessed pads. Thrust bearings for journals tend to require relatively high flow-rates due to the problems of achieving sufficiently wide bearing lands in the space available. This is a problem which may be partially overcome by designing a caliper arrangement instead of the usual 360° thrust pads. An economical alternative is the conical configuration which can withstand radial and axial loads. The spherical arrangement allows for misalignment in addition to combined radial and axial loads. In this case, it is necessary to decide whether this advantage is worth the extra manufacturing complexity. The Yates bearing achieves radial and axial loads in a simple and economical way. The oil which leaks from a conventional journal bearing has to escape through the thrust bearings. It has been found that such a system allows quite substantial axial loads to be supported in addition to normal radial loading.

The simplest example of a flat sliding bearing is the circular or rectangular single-recess pad. However, a single plane thrust pad must always be held down by a positive force. Where the load reverses in direction, as in some machine tool slideway systems, opposed pads would be employed. The thrust pad with a single recess has virtually no resistance to tilt. This may be an advantage in a spherical bearing which is designed to allow free rotation in any direction, but for most machines pads must be arranged in a suitable pattern to ensure alignment of the bearing surfaces. The rectangular multi-recess bearing pad for a linear slide and the multi-recess annular thrust pad for rotary movement, may both be employed where tilt resistance is required. Figures 7 and 8 show an example of a recent application of an annular thrust pad to support the analyzing magnet of the Nuclear Structure Facility at the SRC Daresbury Laboratory. This bearing was designed in collaboration between Liverpool Polytechnic and Daresbury.

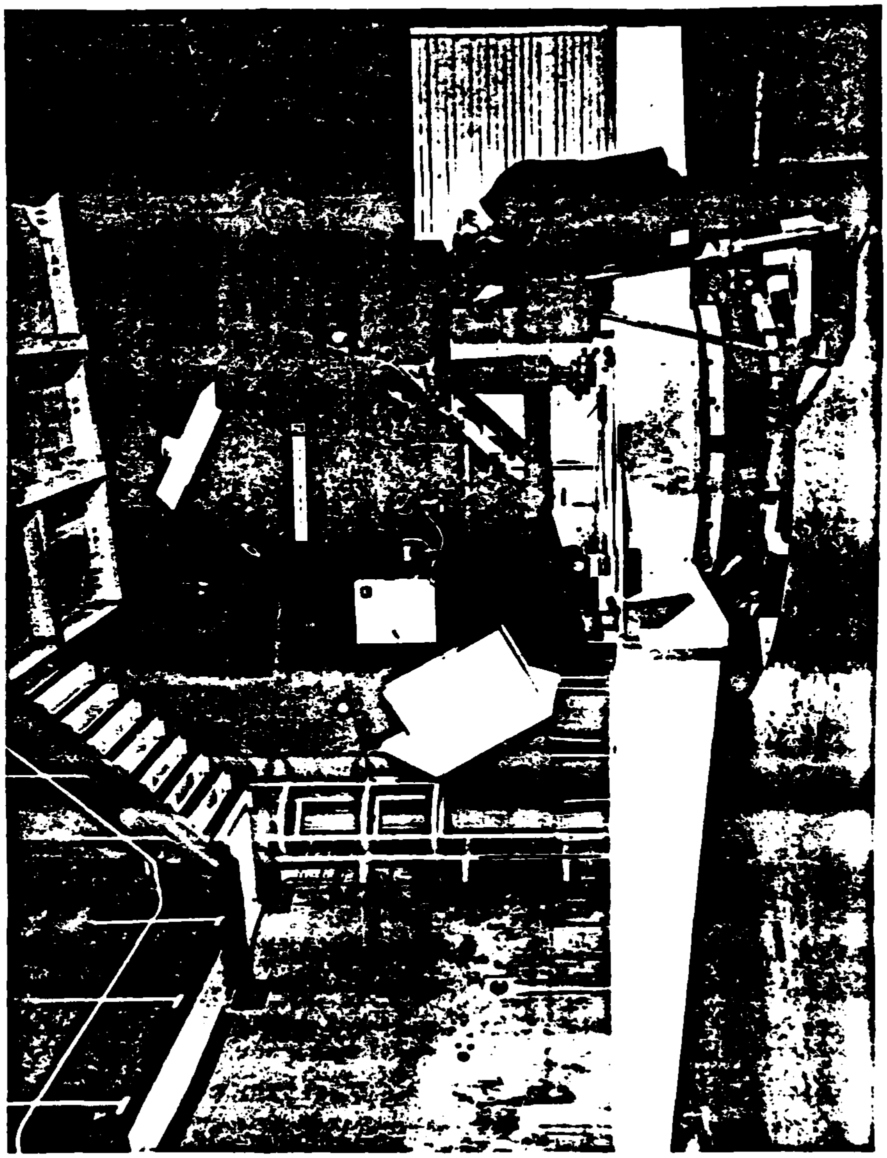


FIGURE 7 THE ANALYZING MAGNET AT THE BASE OF THE NUCLEAR STRUCTURE FACILITY AT THE SRC DARESBURY LABORATORY SUPPORTED ON A HYDROSTATIC BEARINGS

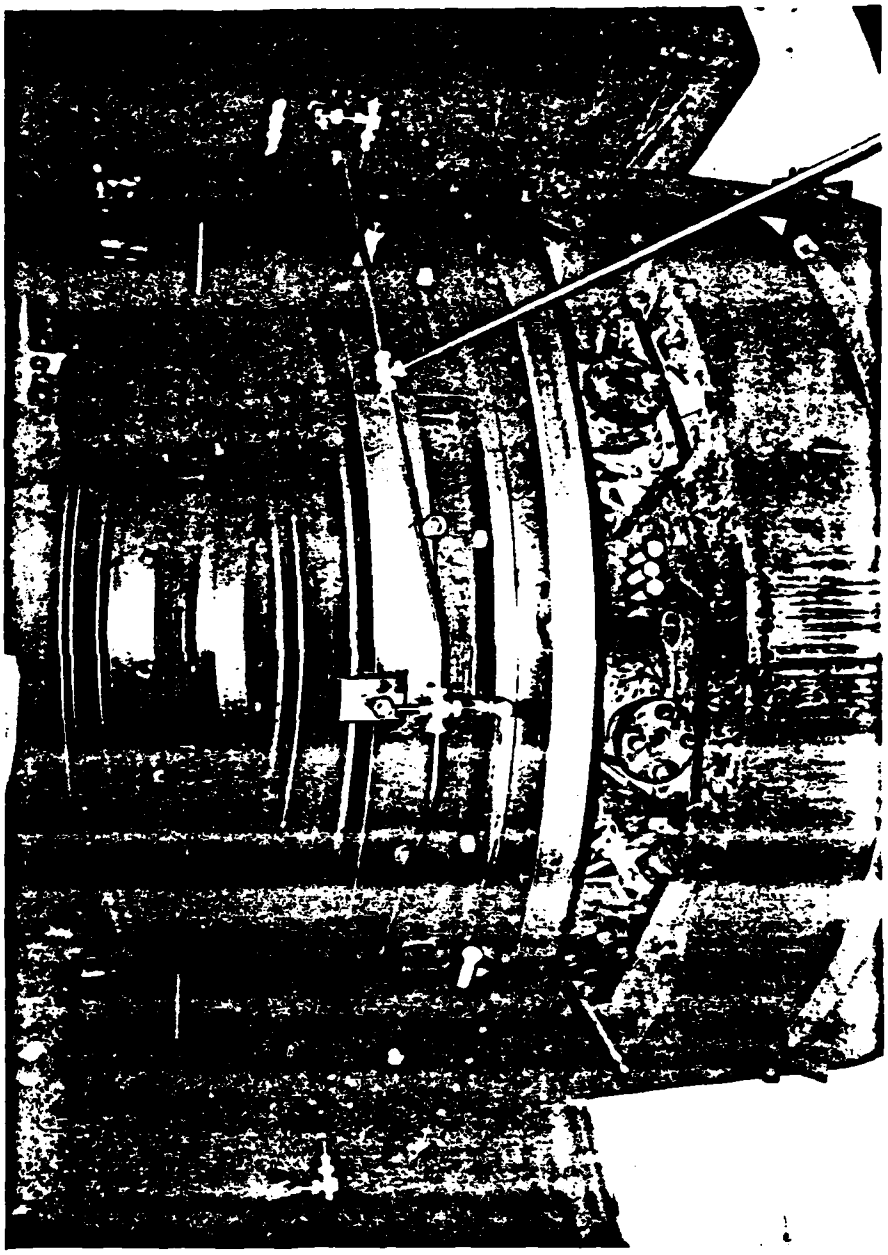


FIGURE 8 ANNULAR THRUST PAD BEARING FOR DARESBURY LABORATORY

The magnet bends the beam of ions through 90° towards the experimental equipment. The diameter of the bearing is 1.8 m (72 in.); it carries a load of 53 tonnes (53 tons) and its surface is flat to an accuracy of 5 microns (0.0002 in.). The operating requirements involved maintaining the vertical center-line of the apparatus within 0.1 mm (0.004 in.) radius at a height of 6.3 m (21 ft.) above the bearing face, while the magnet may be rotated to direct the beam into any one of three experimental areas. This was successfully achieved with a supply pressure P_s, of 1.1 MN/m^2 (162 lbf/in^2) and a pressure ratio $\beta = 0.5$. Removable capillary tubes were used for bearing control for reasons of simplicity and ease of cleaning.

The machining and construction of thrust pads is usually a simple matter. For journal bearings, several methods have been employed for producing recesses. These include milling or grinding, which is difficult for small and medium size engineering bearings of less than 250 mm (10 in.) diameter. Other methods include electrical discharge machining and fabrication. The latter method is the simplest in the absence of EDM facilities. Examples of the fabrication of journal bearings is illustrated by Figures 9 and 10. Figure 9 shows the parts of a recessed journal bearing and Figure 10 the parts of a slot-fed journal bearing suitable for hydrostatic or hybrid operation with either liquid or gas lubrication.

Figure 6 also indicates a range of control principles which may be employed in hydrostatic bearing design. Some method of control is essential, as already explained, and a separate control device must normally be employed for each bearing recess. External devices may include capillary or slot restrictors, orifices, constant flow valves or pumps, and pressure-sensing valves. Some bearing configurations have inherent control due to a shallow recess or the particular combination of journal and thrust bearings as in the Yates bearing. The stability of the bearing film clearance and the stiffness of the bearing film are both dependent on the pressure/flow-rate characteristics of the control device. Some pressure/flow-rate characteristics are illustrated in Figure 11. It will be seen that flow-rate through a capillary is directly proportional to the pressure difference along the capillary and hence the flow-rate is inversely proportional to recess pressure. This gives the lowest oil film stiffness as indicated by line 1 on the film thickness versus load ratio diagram. The orifice, line 2, and the constant flow supply, line 3, both yield stiffer bearings and a pressure sensing valve, line 4 may be tuned to achieve a region of infinite or even negative static oil film stiffness. In practice, negative stiffness is not recommended as it can lead to limit-cycle

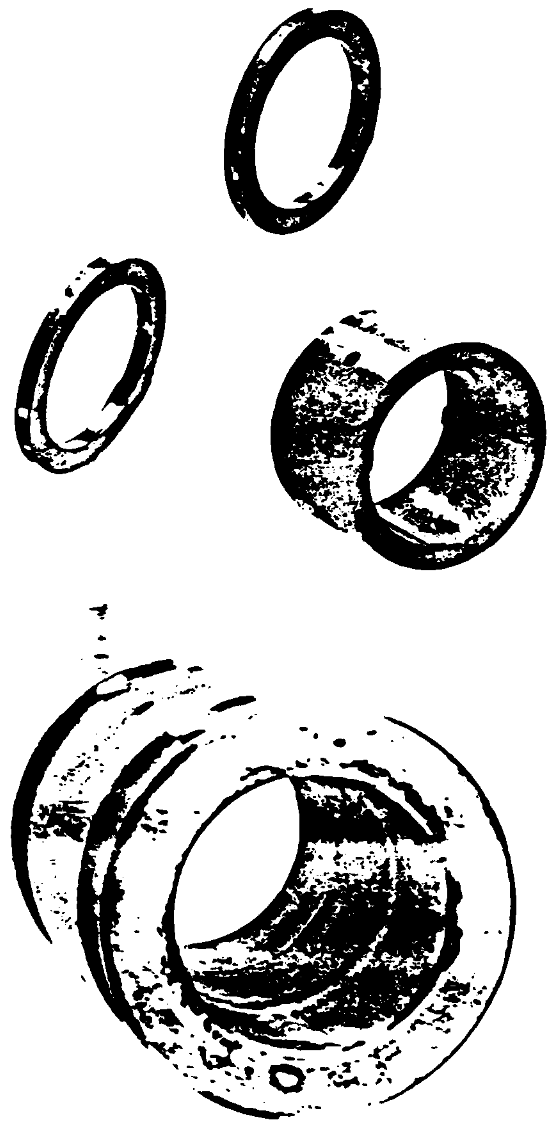


FIGURE 9 A FOUR-RECESS HYDROSTATIC JOURNAL BEARING

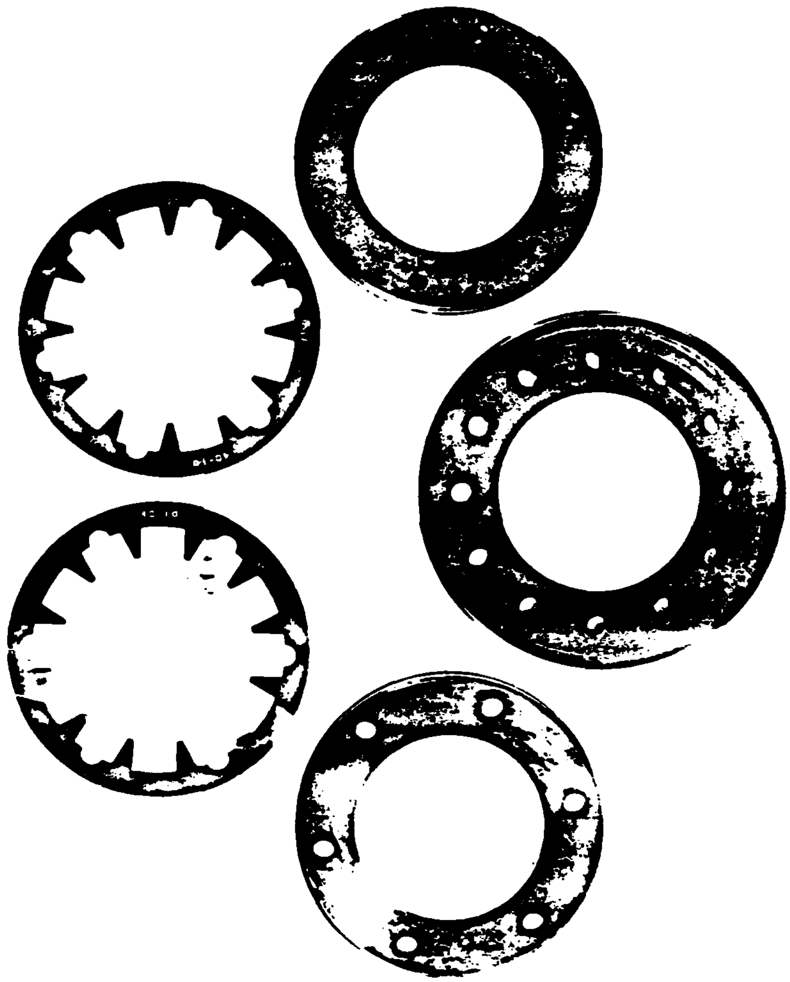


FIGURE 10 A SLOT-ENTRY JOURNAL BEARING FOR GAS OR LIQUID OPERATION

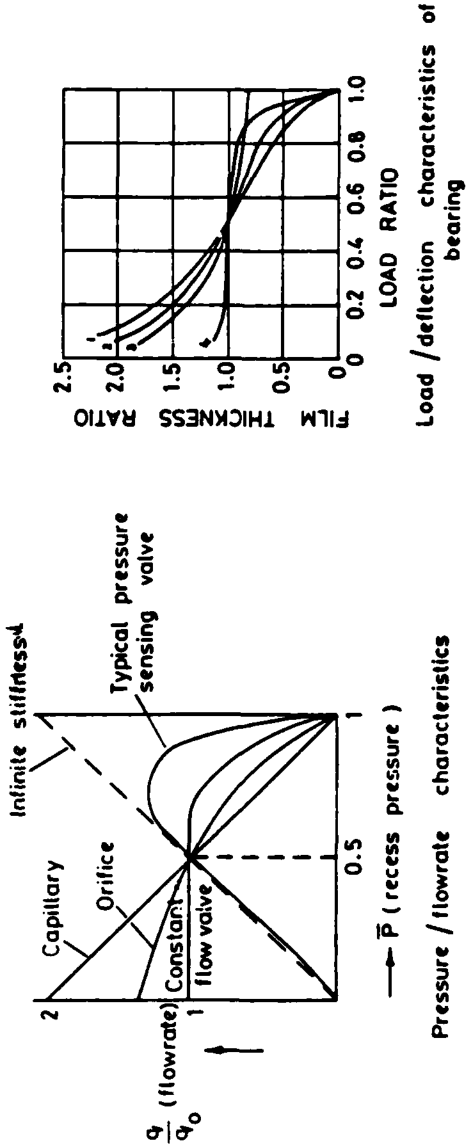


FIGURE 11 EFFECT OF TYPE OF CONTROL DEVICE

oscillation of the bearing system. It is also found that with bearings which are more highly tuned for high stiffness, it is necessary to maintain a more strict control on other parameters such as oil temperature and bearing clearance. For this reason, capillary restrictors are recommended for most purposes. A simple form of pressure sensing valve, applied to a journal bearing, is illustrated in Figure 12. In this example the control device is a double diaphragm valve. When a recess pressure increases in reaction to an applied load on the bearing, it causes the diaphragm to deflect. This reduces the restriction in the supply line and increases the flow-rate. The flexibility of this device may also be an advantage for preventing the build up of silt in the restrictors. Figure 12 shows this principle applied to a 6-recess bearing employed in a high removal - rate centerless grinding machine. Three such valves are necessary to control the six recesses. The novelty of the system illustrated is the use of the bearings for grinding force measurement and also for dynamic wheel balancing. Force measurements could also have been achieved with capillary control although the extremely high bearing stiffness was considered an advantage for achieving accuracy.

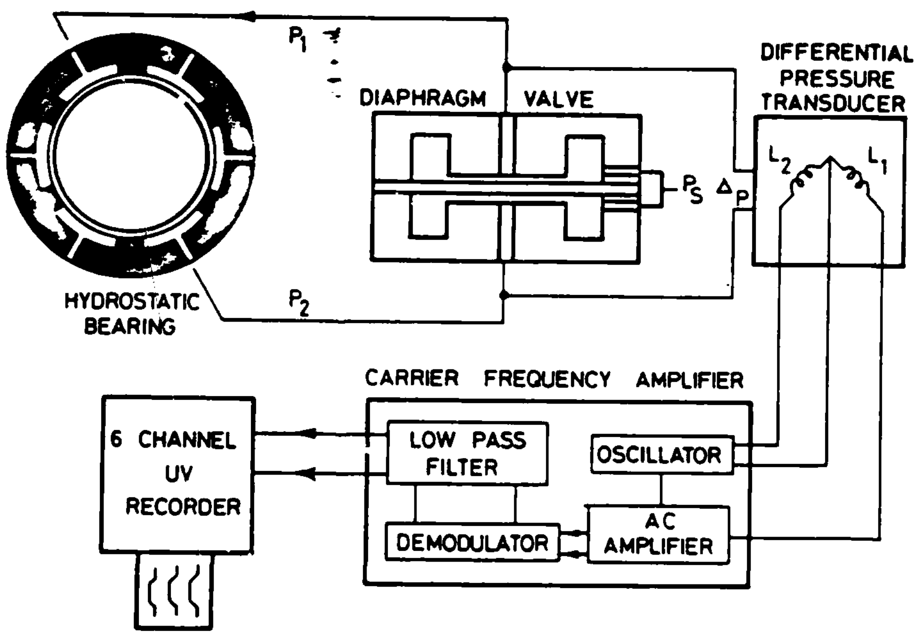


FIGURE 12 FORCE MEASUREMENT AND BALANCING WITH HYDROSTATIC BEARINGS

As previously mentioned, the detail design and construction of externally pressurized gas bearings tends to be quite different from the design of liquid lubricated hydrostatic bearings. Whereas many liquid-fed bearings involve substantial recesses as shown in the previous diagram, such recesses are likely to prove unacceptable for a gas bearing. Gas bearings tend to be prone to a vibrational instability known as pneumatic hammer if large gas volumes are stored between the restrictor and entry to the bearing clearance. For this reason orifice and lost restrictors are more commonly employed than capillary restrictors or other valves which may be less stable. Considerable research has been carried out on porous pad restrictor inserts which potentially yield high bearing stiffness and good stability if the problems of manufacture and controlling the porosity are overcome.

Externally pressurized gas bearings also tend to differ from liquid hydrostatic bearings in the materials employed. It is essential that all materials should be corrosion resistant and hence stainless steel and brass are used extensively. Greater precautions are also necessary to allow for a possible gas or power supply failure since the bearings operate dry. In the case of gas-supply failure, drive motors must be isolated and stopped quickly. A pressurized gas reservoir offers some protection by allowing time for braking and run-down of the motor.

3. DESIGN OF EXTERNALLY PRESSURIZED BEARINGS

The bibliography in Section 5 of this discussion will direct the reader to sources of design data. The following notes are intended to give an indication of load-carrying capacity of plane pads and cylindrical journal bearings. For brevity, the following guides are necessarily over-simplified. However, greater accuracy and sophistication can be achieved by reference to the published data.

3.1 Design Procedures for Hydrostatic Journal Bearings

The geometry of a typical bearing is shown in Figure 13. The land width, 'a', and the recess pressure, P_r, are the main parameters which govern the flow-rate from a recess. The total flow-rate from the bearing is given by:

$$q = \frac{P_r \pi D h_o^3}{6 a \eta}$$

At higher speeds more flow will be required to maintain a cool bearing. As a guide, it is usually recommended to design the bearing for minimum total power dissipation where the total power is equal to pumping power plus friction power.

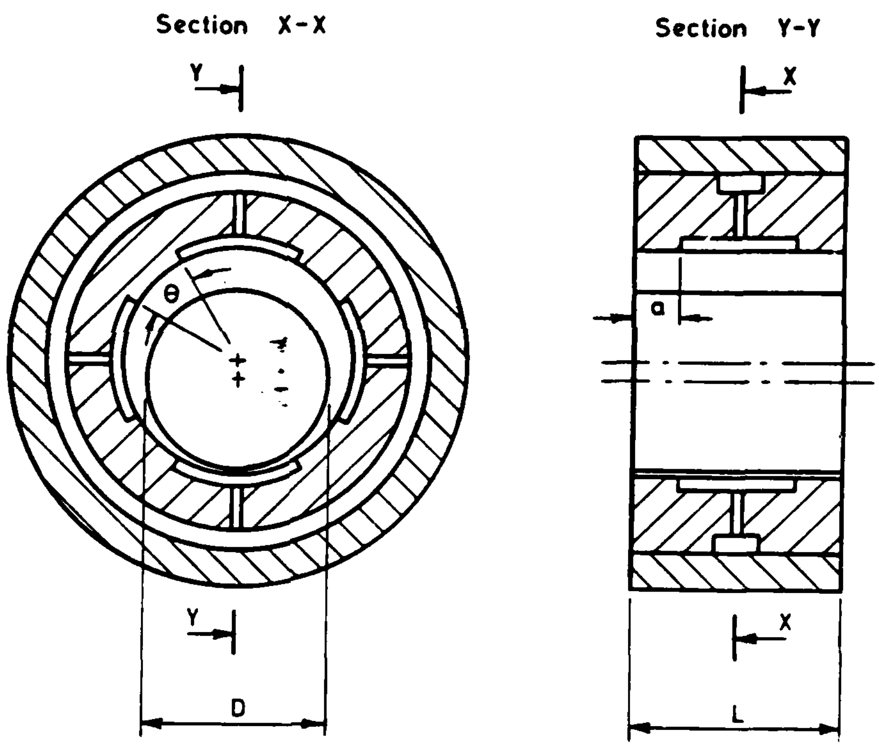


FIGURE 13 THE GEOMETRY OF A CAPILLARY - CONTROLLED FOUR-RECESS HYDROSTATIC JOURNAL BEARING

Design for minimum power is assumed in the flow-chart, Figure 14, and forms a basis for the selection of bearing parameters For minimum power, the ratio of friction power to pumping power should lie in the range

$$1 \leq K \leq 3$$

An approximate load capacity is given by

$W = \frac{1}{4} P_s LD$ where $P_r = \frac{1}{2} P_s$

Stiffness

$$\lambda = \frac{\frac{1}{2} \times P_s LD}{h_o}$$

Temperature rise (^oF) = $0.015 \times P_s$ (lbf/in^2)

Temperature rise (^oC) = $4 \times 10^{-6} \times P_s$ (N/m^2)

For the widest tolerances on bearing clearance, the minimum clearance condition should not be less than 2/3rds the maximum clearance condition. Figures 15 and 16 are design charts which give some indication of the selection of clearance and other design parameters. The recess pressure at the maximum clearance (at K = 1), is adjusted to make $P_r = 0.4\ P_s$. This gives $P_r = 0.7\ P_s$ and K = 3 at the minimum clearance. At the minimum clearance condition the temperature rise may be doubled.

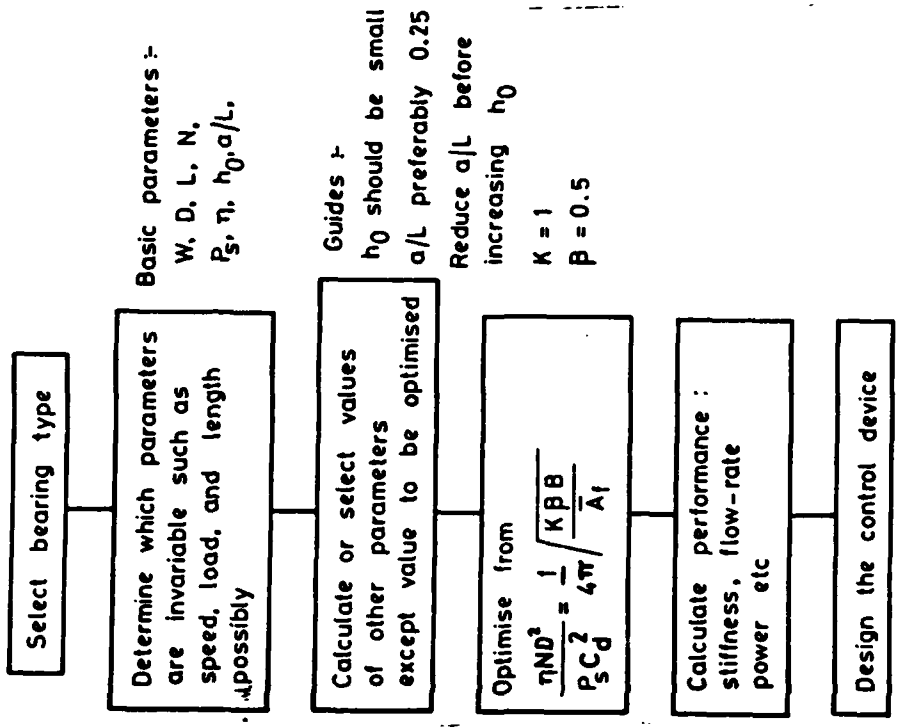


FIGURE 14 FLOW CHART ILLUSTRATING THE BASIS OF DESIGN PROCEDURES FOR JOURNAL BEARINGS

3.2 Design of Hybrid Journal Bearings (Non-recessed)

Figures 17, 18, and 19 indicate the type of bearing loads achievable with hybrid bearings. To obtain greater load capacity from hybrid bearings, the speed may be increased or the equivalent effect may be obtained by setting K = 12 in the flow chart for hydrostatic bearings. Figure 17 compares a plain hybrid bearing, at a relatively low-speed (K = 1), with a capillary controlled recessed journal bearing operating at the same speed corresponding to the minimum power condition. The hybrid bearing is also compared with an axial groove hydrodynamic bearing. The hybrid bearing supports the highest loads up to an eccentricity ratio of 0.8. Figures 18 and 19 give an indication of load capacity at higher speeds corresponding to higher values of K.

3.3 Hydrostatic Plane Pads (Recessed)

Design procedures follow similar considerations to those for journal bearings, so that the following approximate guides apply:

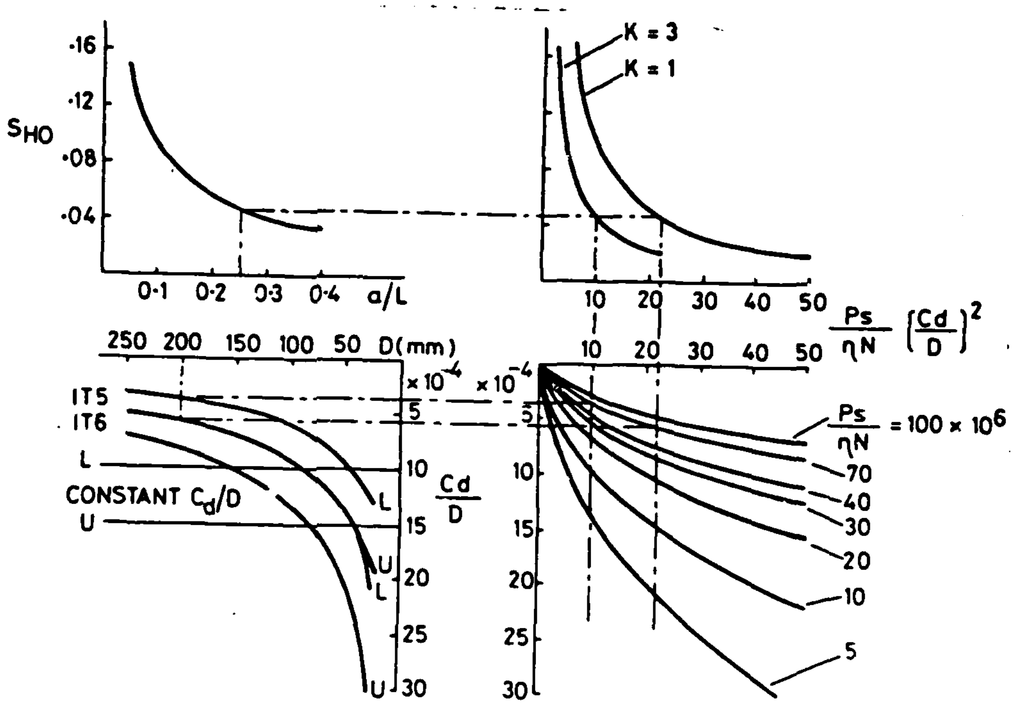


FIGURE 15 RECESSED JOURNAL BEARINGS - TOLERANCES

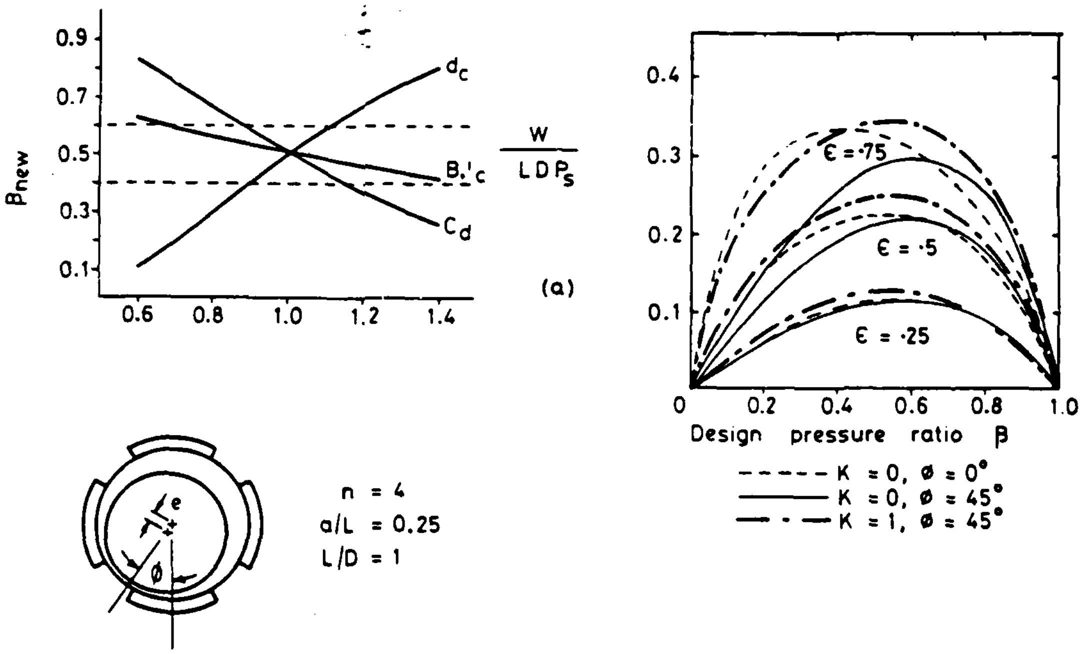


FIGURE 16 PERFORMANCE OF FOUR RECESS HYDROSTATIC JOURNAL BEARINGS

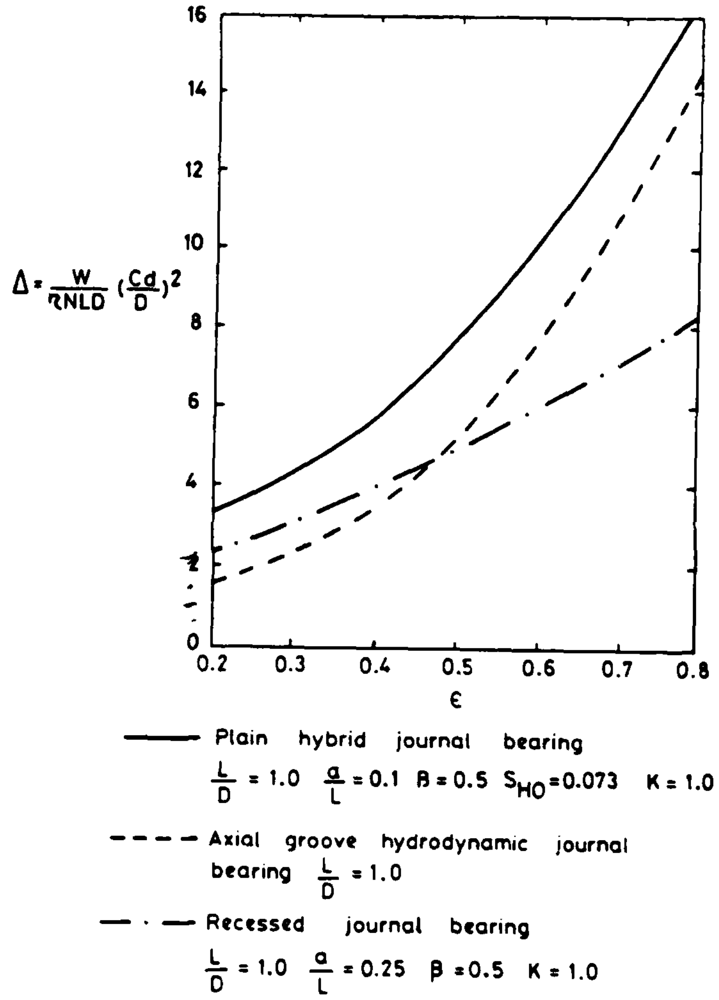


FIGURE 17 COMPARISON OF SLOT HYBRID JOURNAL BEARING WITH AXIAL GROOVE HYDRODYNAMIC JOURNAL BEARING

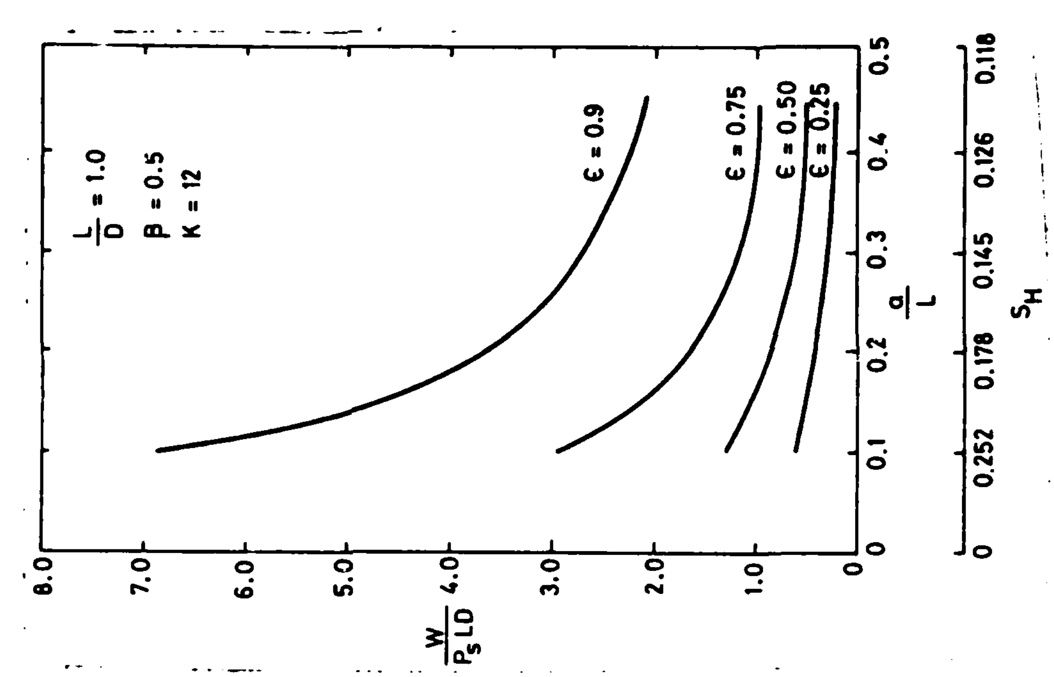


FIGURE 18 SLOT HYBRID BEARING

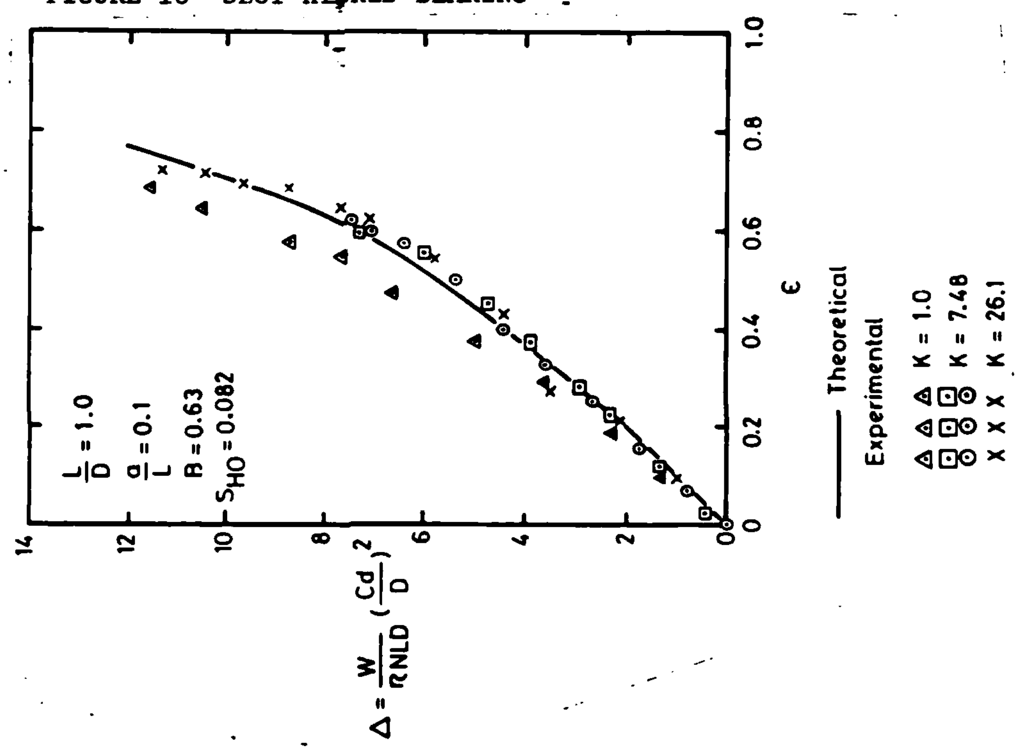


FIGURE 19 VARIATION OF INVERSE SOMMERFELD NUMBER WITH ECCENTRICITY RATIO

Recess pressure

$$P_r = \tfrac{1}{2} P_s$$

Load

$$W = P_r \times \text{Effective area}$$

Effective area = ½ x (total area + recess area)

Stiffness

$$\lambda = \frac{1\tfrac{1}{2} \times W}{h_o} \qquad \text{..... (Capillary)}$$

$$\lambda = \frac{2 \times W}{h_o} \qquad \text{..... (Orifice)}$$

Flow rate

$$q = \frac{P_r h_o^3}{12\, a\eta} \times \text{(mid-land perimeter of bearing)}$$

3.4 Design of Externally Pressurized Gas Bearings - Plane Pads

The load capacity of plane pad gas bearings is subject to greater errors in calculation than liquid bearings. The load capacity depends on a number of factors including the pad shapes, the inlet configuration, and the type of restriction. Experimental results have been given by Wunsch of the National Engineering Laboratory for square pads and these results may also be applied as an approximate guide for circular pads. The design employed by Wunsch involved an orifice with a small inlet pocket as in the journal bearing, Figure 4. The size of the pocket was found to be important. Increasing the pocket size increases the static stiffness and the load carried by a particular size of pad. However, the dynamic stiffness deteriorates and the likelihood of pneumatic hammer is increased. For this reason, it is important to avoid deep pockets. Orifices are typically inserted in plugs of 3.175 mm (⅛ in.) diameter at a depth of 0.075 mm (0.003 in.) to 0.225

mm (0.009 in.) to form a small pocket. A small pocket is a considerable advantage for load and stiffness. The following results were obtained for square pads

$$W_{max} = 0.2\ P_g^{1.1} a^{0.8} \qquad \text{..... (Pocket orifice)}$$

$$W_{max} = 0.13\ P_g^{1.1} a^{0.8} \qquad \text{..... (Inherent orifice)}$$

where P_g = gauge pressure

a = pad area

A stable air-bearing at the natural frequency tends to act like a simple mass-spring system. It was found by Scholes and Wunsch that the natural frequency could be calculated approximately from the static stiffness and the mass.

$$\omega_n = \frac{\lambda_{st}}{m}$$

On thrust bearings, the natural frequency can be increased by a double-sided system of opposed pads.

Air flow depends strongly on clearance and hence involves the question of the minimum clearance which can be maintained taking account of the necessity for tolerances and structural deformation under load.

An example of typical flow requirements for square pads, as described above, varying in size from 1600 mm^2 (1 $in.^2$) to 6400 mm^2 (4 $in.^2$) supplied at pressures up to 0.54 MN/m^2 (80 lbf/in^2) guage may be found in the NEL Report by Nimmo. It was found that these bearings consumed up to 0.28 l/s (0.6 $ft.^3$/min) of free air. In each case, the orifice diameter was 0.5 mm (0.020 in.).

3.5 Design of Externally Pressurized Gas Bearings - Journals

The basic configuration is illustrated in Figure 4, for a double-entry bearing i.e. 2 rows of restrictors at $\frac{a}{L} = 0.25$. An alternative for small $\frac{L}{D}$ ratio bearings is the single-entry configuration i.e. 1 row or restrictors at $\frac{a}{L} = 0.5$. The following is an approximate quide.

n = 8 Number of pocketed orifices per row (orifice entry)

n = 12 Number of inlet slots (slot-entry)

K_{go} = 0.4 Gauge pressure ratio

$\frac{L}{D}$ = 1.0 Length/diameter ratio of bearing

$W = \frac{1}{3} \times (P_o - P_a)\ D^2$ ..... Single entry

$W = 0.4 \times (P_o - P_a)\ D^2$ ..... Double entry

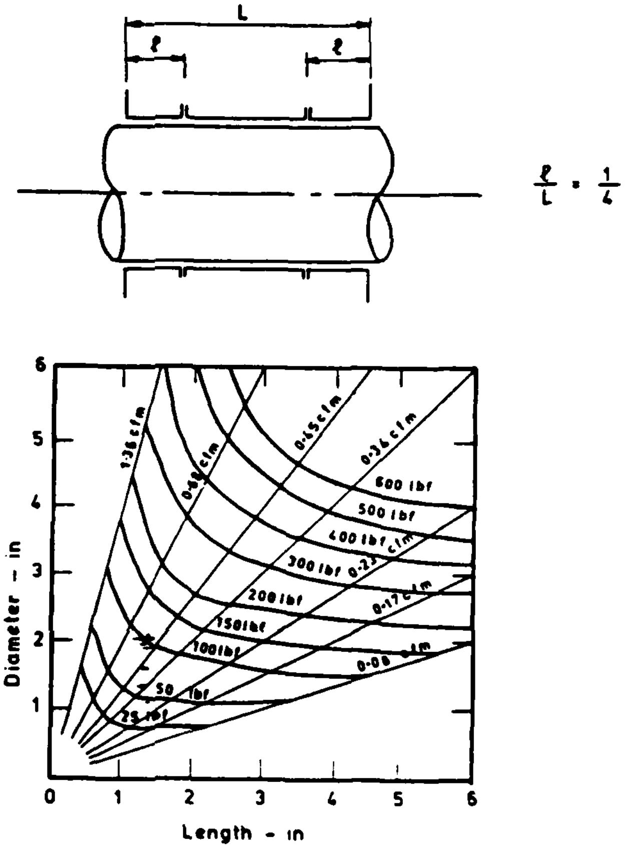
Typical flow-rate =

0.12 l/s (0.25 $ft.^3$/min) free air with C_d = 0.025 mm (0.001 in.) at 0.54 MN/m^2 (80 $lbf/in.^2$) gauge supply pressure.

Further information for double-entry bearings is given in Figure 20.

4. NOTATION

a, 1	Bearing land width
a_s	Width of slot restriction
C_d	Diametrical clearance
d_c	Capillary diameter
e	Eccentricity
h	Film thickness
h_o	Design value of film thickness
l_c	Capillary length
n	Number of recesses, orifices, slots etc.
P_r	Recess pressure
q	Bearing flow-rate
y	Length of flow-path through slot restriction
z	Film thickness in slot restriction
D	Bearing diameter
K	Design parameter = Ratio of Friction Power to Pumping Power



Load capacity of Air Journal Bearings.
Two rows of 8 pocketed orifices at 1/4 stations. E = 0·5

Air flow (free air) given for C_d = 0·001 in

For other clearances flow-rate proportional to C_d^3

Load capacity for $P_o - P_a$ = 100 lbf/in^2

For other supply pressures W proportional to $P_o - P_a$ Stiffness = $\frac{4W}{C_d}$

FIG. 20

FIGURE 20 DOUBLE ENTRY BEARINGS

L	Bearing length
N	Rotational speed in rev/s
P_g	Gauge supply pressure
P_s, P_o	Supply pressure
S_h	Design parameter $= \frac{\eta N}{P_s} (\frac{D}{c_d})^2$
S_{ho}	Value of S_h when $K = 1$
W	Bearing load supported by fluid film
β	Pressure ratio P_r/P_s
Δ	Inverse Sommerfeld Number $= \frac{W}{\eta NLD} (\frac{c_d}{D})^2$
ε	Eccentricity ratio e/h_o
η	Dynamic viscosity
λ	Bearing film stiffness

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Appendix B

TRIBOLOGICAL INVESTIGATIONS OF THE CONTACT MECHANICS IN A ROTARY POSITIVE DISPLACEMENT MACHINE

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Cranfield Institute of Technology

1. INTRODUCTION

This is an initial report of a tribological study undertaken on the sliding contact mechanics of a rotary positive displacement machine. The overall objective of this study was to enhance the operating life efficiency of the machine.

The innovative mechanical design relies on the sliding contact between a tribological pair of elements, namely, a polymeric and a cast iron element.

The overall operating efficiency of the machine is susceptible to any wear taking place in the sliding contact zone, which alters the running clearances in the machine. Any tribological solution to minimize this aspect would, therefore, enhance the overall performance of the machine.

2. CONTACT CONFIGURATION

The physical configuration of the sliding contact between the two rotating elements may be imagined as a cylinder (cast iron) meshing orthogonally with two planer discs (polymer). Specifically, the assembly represents a cast iron worm gear meshing with polymeric pinions.

The cast iron rotor, driven at synchronous speed, drives the pinions at a fixed speed ratio. However, the linear velocity at any point on the meshing surface varies throughout the contact path.

Essentially, the two rotating elements form a noncontacting seal system to contain the working fluid, 'FREON', for the short period that the two are in contact. Thus, the efficiency of the seal is governed by the wear between the elements.

3. OUTLINE OF THE PROBLEM

The sealing action was designed to operate hydrodynamically with a finite fluid film of the transported fluid. The fluid film was not single phase, gas or liquid, but was a two-phase mixture thus negating a 'true' hydrodynamic action.

In practice, however, asperity contact between the two rotating surfaces is sufficient to cause the initial condition of a nominal straight line contact of the polymer pinion to change to a finite area contact. The initial wear is very evident at the early stages of the machine operation. The problem can thus be stated as follows:

(1) To establish the actual operating conditions in the contact zone.

(2) On the outcome of (1) to modify the material selection or specification to maximize operating life and efficiency.

(3) To provide, if possible, some form of boundary lubrication.

4. DESIGN APPROACH

To understand the complex contact condition, it was necessary to obtain 'in situ' data from the machine operating at its actual running condition. Past experience of the wear in complex sliding contact indicated the limitation of an analytical approach[1]. Any analysis (be it a theoretical study or via a separate model simulation) was unlikely to produce meaningful results or the required tribological optimization.

At its most basic level, a simple planer tribological pair of surfaces of cast iron and polymer operating in conditions of mixed friction and using 'FREON' as a lubricant is not readily amenable to analysis. In the machine itself, both the geometry and the kinematics of the contact zone were complex. Influencing factors include the variation of the surface sliding speed and the phase change of the working fluid. Another factor was the behavior of such a tribological pair at the relatively low temperature ($-10^{o}C$) since a literature search had produced limited information on such conditions.

Clearly, since a multiplicity of interacting variables existed in the contact zone, with varying degrees of tribological relevance, it was necessary to decide on the most significant. Another primary consideration was whether any signal output could be measured and transmitted from the transitory surface contact during the actual operation of the machine.

The difficulty in physically achieving this, lies in four distinct areas:

(1) The general inaccessibility of the contact within the machine.

(2) The importance of keeping any machine modification to an absolute minimum both from the cost and time aspects.

(3) The necessity of transmitting the signals from a rotating component to the monitoring devices.

(4) The spacial restrictions within the contact area proper, necessitating the use of miniature transducers.

5. INSTRUMENTATION

5.1 Choice of Instrumentation

Although several specific variables were available for measurement in the sliding contact, temperature and fluid pressure measurement were considered to be of prime importance. The temperature would indicate the severity of the surface friction; the pressure would indicate the variation of the pressure within the transported fluid. The pressure would also indicate the efficiency of the contact geometry in producing the quasi-hydrodynamic film.

Spacially, the installation of the micro-thermocouples presented less of a design problem than that of placing micro-pressure transducers in the contact zone. Temperature, as a variable, was therefore selected for a preliminary measurement. The choice of one initial parameter also reduced the overall cost. The problem resolved itself, therefore, into the development of a system for monitoring the transducer signal from the rotating pinion, together with the sealing arrangement and the machine modifications.

5.2 Instrumentation System

The complete instrumentation circuit is shown in Figure 1, and illustrates the signal transmission, display, and storage. Basically, the signal from the temperature sensors was passed from the rotating component via high quality silver/silver graphite slip rings which were air cooled to minimize thermal drift. Maximum peak-to-peak noise in these units is 50 μV/mA at 10,000 r.p.m. Any 'Seebeck' effects were minimized by mounting rotating junctions on a junction disc integral with the pinion shaft and rotating in ambient air.

Signals from the stationary side of the slip rings were fed, via a junction box, to a cathode ray oscilloscope for display and photography. The signal was also fed to a 'Fluke 2200 B' sixty channel data logger for signal conditioning and data storage. The makers specify an accuracy of $\pm 0.1^{\circ}C$. An optical pick-up was used, in conjunction with a disc, to provide a time datum for the oscilloscope traces.

5.3 Temperature Sensors

Thermocouples were manufactured in the laboratory from copper/constantan wire 0.127 mm (0.005 ins.) diameter. The basic design requirements for the temperature sensors were as follows:

(1) Small tip mass to delineate rapid temperature transients.

(2) High mechanical integrity and fatigue strength at the sliding contact.

(3) Good output characteristics to minimize signal/noise ratio.

5.4 Method of Installation of the Sensors

The contact edge sensors were installed by milling five grooves (four in the leading edge and one in the trailing edge) in one of the teeth of the polymer pinions, the grooves being 1.0 mm wide x 8 mm long x 0.5 mm deep. Each sensor was cemented in place using an epoxy resin adhesive and the excess smoothed to the profile of the pinion, upon hardening. Tapping holes, drilled in the pinion, led the sensor wires out through the middle of the shaft to the slip rings.

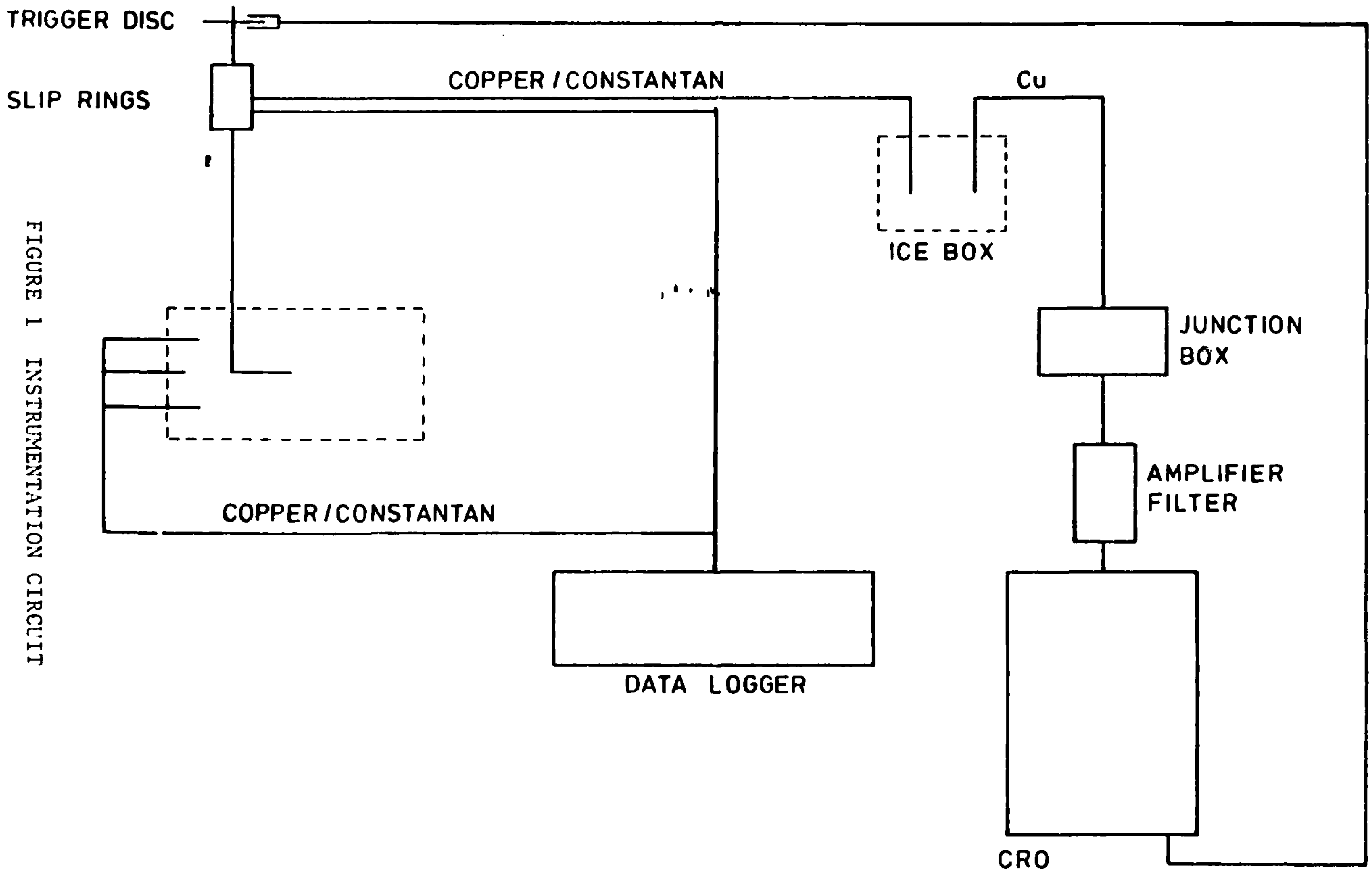

FIGURE 1 INSTRUMENTATION CIRCUIT

751

Additionally, a reference sensor was placed in the pinion to datum the mean temperature of the 'FREON'.

5.5 Testing Stages

From Reference 1 it was already known that, for ferrous surfaces in sliding contact, a two-zone temperature distribution was measured. This consists of a surface skin effect and a bulk material effect. The former gives temperature transients within a skin some 0.5 mm (0.020 ins.) thick while the latter shows a gradual change of temperature with time. There was no evidence to suggest that this effect would be manifest in polymeric material.

Testing was, therefore, initiated in two stages. Initially, the edge sensors were located 1.5 mm (0.060 ins.) below the edge for the first tests. A second pinion had its sensors mounted flush with the contact surface; the thermocouple beads being honed back to the contact edge. This arrangement would, in effect, sense the temperature level of the contact surface proper.

6. RESULTS

Tests were run for over 200 hours during which time the output signals from the sensors were monitored and photographed. Plate 1 shows a typical surface thermocouple trace at the initial and final periods of the tests. The trigger, in each case, is the lower trace. From these, the temperature transients during each cycle are seen to occur during the 10 ms of engagement.

The test run with the sensors placed below the contact line showed no transients at all, a mean constant temperature of -15°C being recorded throughout the test.

It must again be emphasized that this work reports merely the preliminary findings. Further refinements have been achieved and will be presented at a later date. However, the essential nature of the contact mechanics, as far as the temperature effect is concerned, is considered to have been established.

7. DISCUSSION

The most noticeable facts emerging from a study of the photographic traces (Plate 1) may be stated as follows:

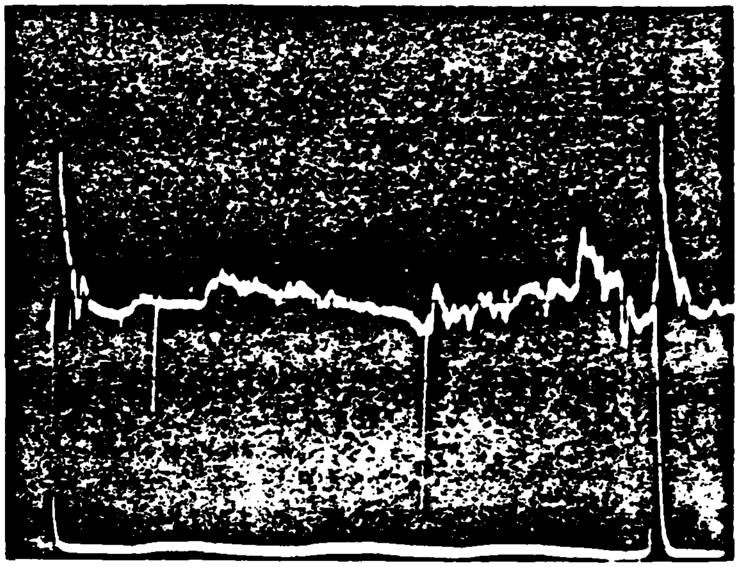

PLATE 1A. PROFILE AFTER 1 HOUR.

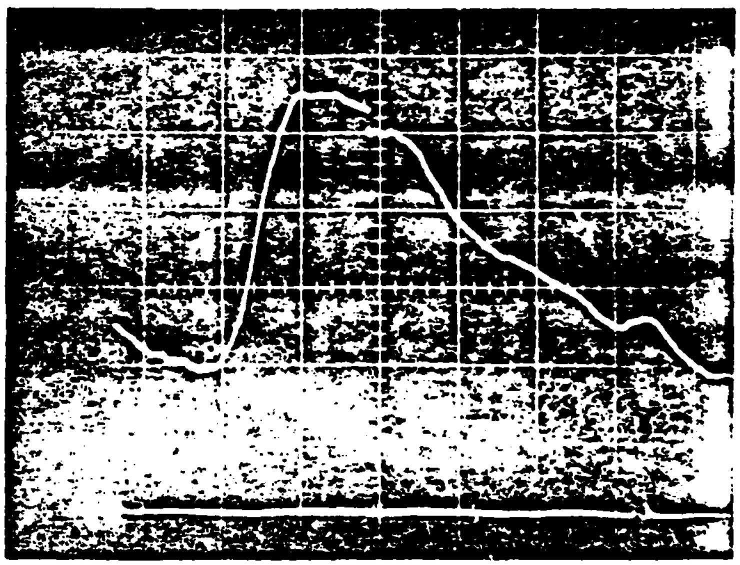

PLATE 1B. PROFILE AFTER 200 HOURS.

PLATE 1 TEMPERATURE PROFILES

(1) As with ferrous surfaces, there is a 'skin' effect with the polymer material, showing rapid temperature fluctuations during the initial running period, Plate 1. A.

(2) As in Reference 1, there is a distinct change in both the magnitude of the temperature and the general form of the temperature profile with time. Plate 1. B. shows that after 200 hours running the surface temperature fluctuations in the skin have disappeared, the temperature profile merely reflecting the changes of temperature of the transported fluid.

(3) The 'bulk temperature', as with the ferrous surfaces, does not reflect the rapid skin transients experienced during the initial running-in, and remains largely constant throughout the tests. This reinforces the concept that surface transient effects convey heat to the bulk material in a gradual manner.

Figure 2, taken from Reference 1, shows curves of temperature plotted against distance below the contact surface for a variety of test duration times. From this, two basic points emerge:

(1) The subsurface temperature gradient is markedly affected by the duration of the running time; the longer the time, the less the gradient.

(2) In the surface skin proper and in the bulk material some distance below the skin, the temperature gradient is linear, while in the transition zone between the two, the effect in nonlinear.

Essentially, the 'heat sink' mechanism mentioned earlier may be seen to apply, heat constantly being generated within the surface skin transferring itself to the thermal reservoir of the bulk material, the temperature gradient between the two being reduced (and thus the rate of heat transfer) as the mean temperature of the thermal reservoir is raised. This results in a smaller concomitant increase in heat transfer if the friction (and therefore the heat generated) remains constant.

This mechanism appears to apply for ferrous materials but the same could be true for a polymeric material with a lower heat dissipation property.

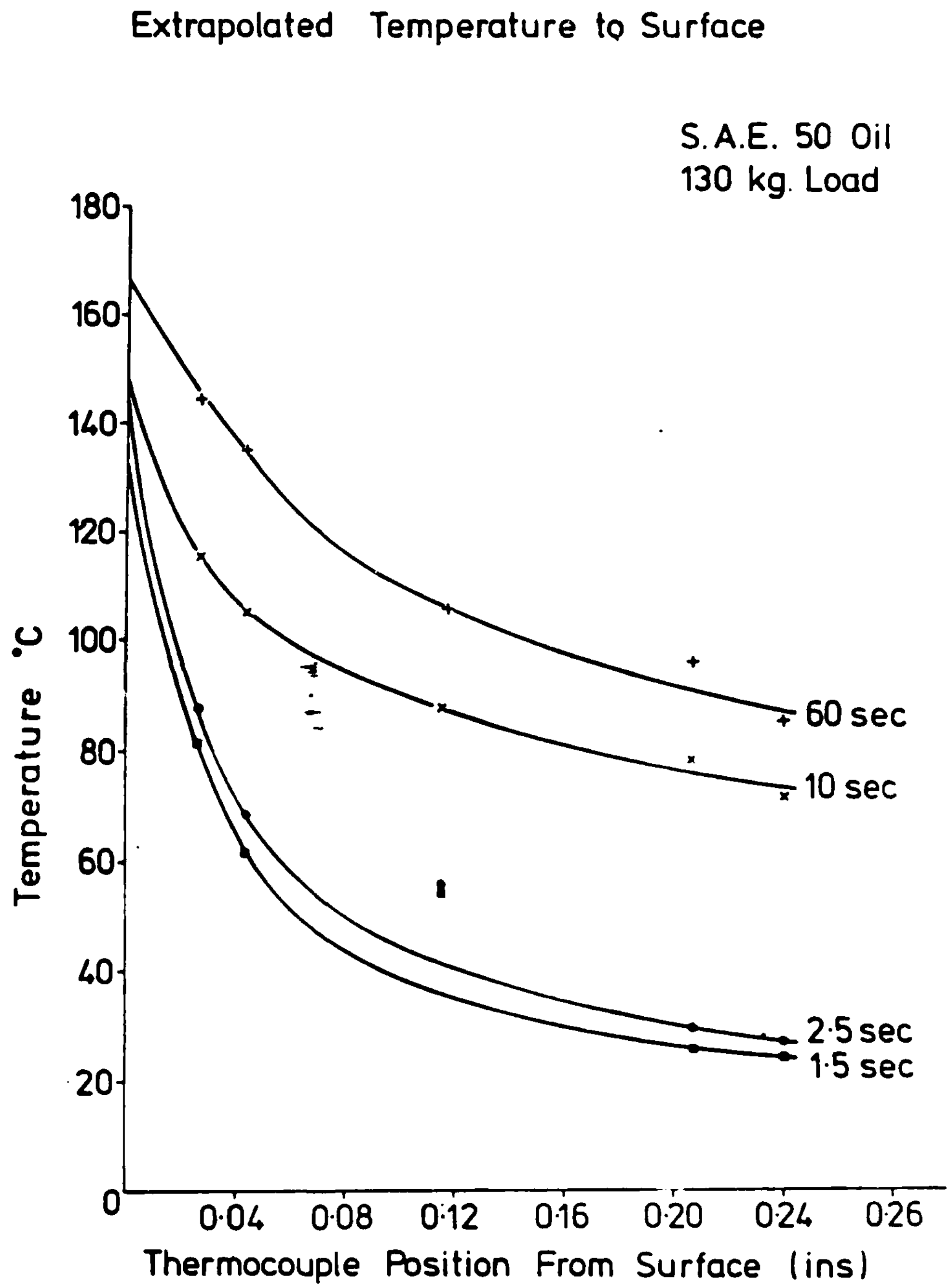

FIGURE 2 EXTRAPOLATED TEMPERATURE TO SURFACE

Considering Plate 1 from this perspective, we may postulate a similar mechanism for the polymeric material contact. Initially, the pinion makes a nominal line contact with the ferrous surface of the rotor. Specific pressures are high and rapid temperature fluctuations are manifest in the contact region, frictional heat being concentrated on the asperities as they interact with one another in the regions of high local pressure. This produces localized temperature spikes. Clearly, the frictional heat intensity is a maximum as the surface asperity area is small, giving low heat transfer.

This heat flux is rapidly dissipated, not internally to the bulk material of the polymeric pinion, while it remains at a constant temperature, but rather to the two phase mixture in the contact zone. Some heat transfer to the rotor surface, (because of its better thermal conductivity) is also probable. However, as the asperities are rapidly worn away, the effective area of the tooth flank that 'sees' the rotor groove increases, producing better heat transfer to the polymer pinion. Concomitant with this, two effects are manifest; first, specific pressures decrease with increasing area, thus decreasing local contact frictional energy. Second, a higher proportion of the load is carried by the quasi-hydrodynamic fluid film in the contact zone, thus decreasing frictional energy generation yet further.

Plate 1. B. shows the run-in surface absorbing (and therefore retaining for a longer period) a higher proportion of the total heat now generated in the contact by the transportation of the 'FREON'. This gives an exponential type decay, the heat transfer being largely governed by the heat transfer coefficient of the polymeric material.

This effect is modelled in Figure 3 which indicates the direct effect of the changed contact area on the proportion of heat transferred to the pinion material in terms of the change in the temperature profile seen in Plate 1, A and B. Initial measurement of the amount of wear at the contact edge of the pinion showed an average wear scar of some 0.5 mm (0.020 ins.) width (after 200 hours of running) from the nominal line contact at the start of the tests.

8. CONCLUSIONS

It has become clear from this preliminary study, of the striking overall similarity in the surface mechanics of the polymeric material and cast iron to the situation of two hardened ferrous surfaces[1]. The similarity of the two zone temperature distribution in these two tribological systems is

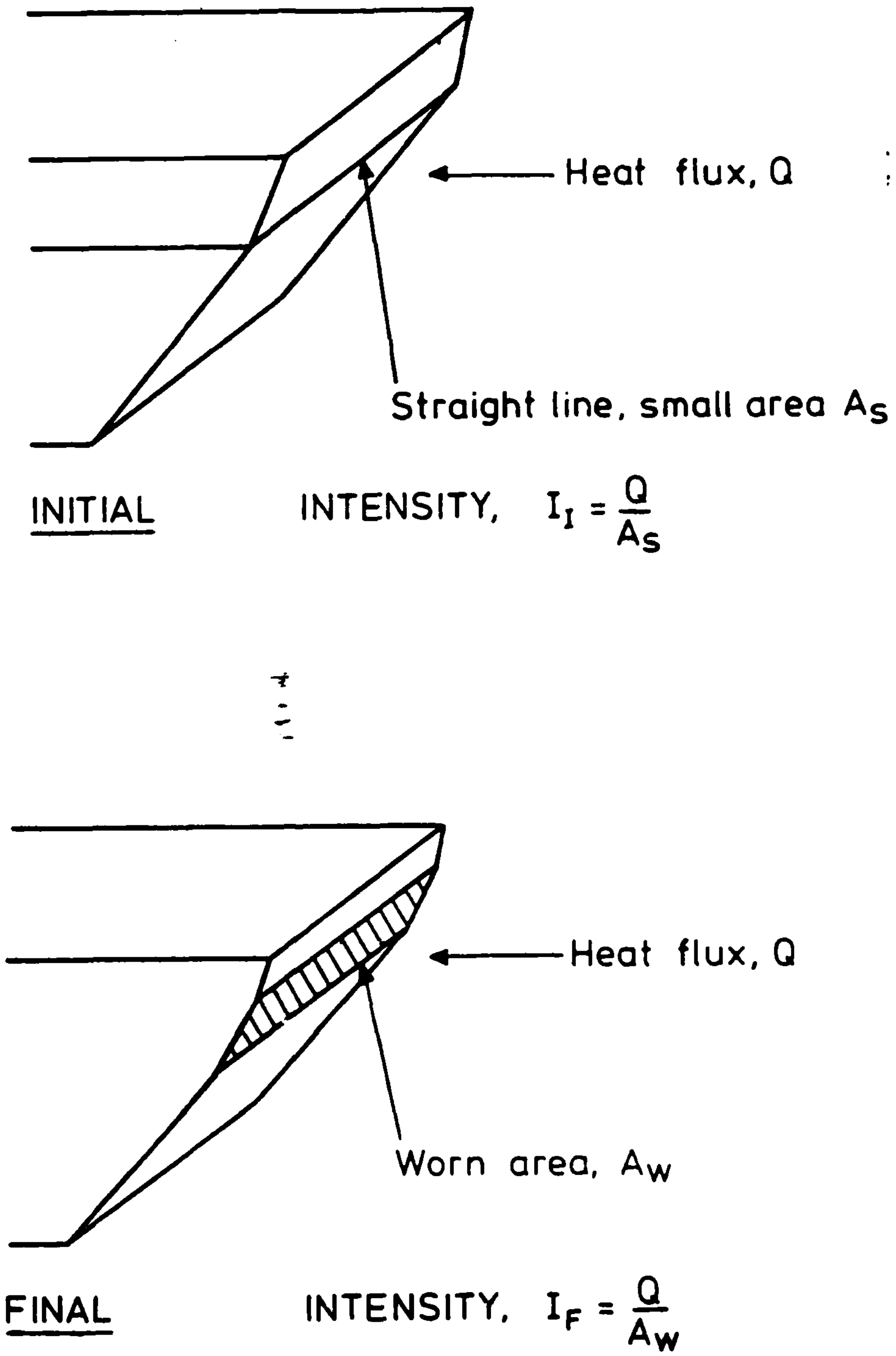

FIGURE 3 IDEALIZED CONTACT CONFIGURATIONS

the most surprising aspect of this study. This suggests that the surface asperity wear, and its related 'skin heat transfer' effect, may be manifest in a wide variety of contact conditions.

The major experimental challenge of monitoring, displaying and processing the transducer signal from the contact zone with the machine operating under actual operation has been achieved. Further experimental results are encouraging and it is hoped, with enhancement of the data acquisition system, a better understanding of the contact mechanics will be obtained.

The project illustrates the importance of a multi-disciplinary tribological approach towards improving the operating life and efficiency of an industrial machine. These improvements will enhance cost effectiveness, for this type of rotary positive displacement machine.

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Appendix C

THREE DIMENSIONAL TOPOGRAPHICAL DESCRIPTIONS OF SOLID SURFACES

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1. INTRODUCTION

The discussion describes a surface measuring system developed for accurate three-dimensional surface characterization and the quantification of surface degradation due to wear or any other deformation process. The system is based on a stylus profile-tracing instrument and incorporates an automatically controlled parallel-profile tracing technique, with micro-computer data handling and processing. The advantages of such a technique for three-dimensional assessment of surfaces, as against the more usual two-dimensional assessments, are highlighted in the typical quantitative and qualitative surface representations obtained from the system.

2. NOMENCLATURE

m Mean absolute profile slope over sampling length, radians

R_q Root-mean-square roughness over sampling length, μm

R_k Kurtosis of the ordinate distribution density

R_s Skewness of the ordinate distribution density

R_z Ten-point height, μm

R_{max} Maximum peak-to-valley height within a sampling length, μm

t_p Bearing ratio at a depth p (expressed as a percentage of R_{max}) below the highest peak, %

λ_a Average wavelength, μm

3. SURFACE ASSESSMENT

The need for accurate surface descriptions is well appreciated in the field of tribological research. Consequently, during the last two decades the expenditure of considerable effort has resulted in important progress with respect to (i) the choice of parameters which truly and uniquely characterize a surface, as well as (ii) the invention, development, and commercial production of reliable, highly sensitive equipment for measuring these parameters. Problems concerning element (i) occur because of the large number of parameters available, some of which are interrelated and may not uniquely specify a surface. Also in making the choice, careful consideration must be given to the particular application[1]. Most previous and present surface analysis techniques are based on the digitized interpretation of recorded profiles, obtained from stylus instruments performing single profile traces. The value of the technique has been greatly enhanced by the application of parallel-profile tracing[2 - 5]. More truly representative surface descriptors, emerge from this technique as a result of multiple parallel tracings, and averaging the recorded data. Added to this is the ability to compute topographic rather than only single profile data. Surface topographical descriptions in the form of isometric and contour maps, may also be produced.

This discussion describes a three-dimensional surface measuring system employing parallel-profile tracing and presents examples of its output, which are relevant to the examination of surfaces before and after they have been subjected to friction and wear.

4. MEASURING SYSTEM

The system developed at Cranfield Institute of Technology has been based on a Talysurf 4 stylus profile-tracing instrument. It incorporates an automatically controlled, three-dimensional relocation stage, and a micro-computer data handling and processing facility.

The relocation stage, Figure 1, consists of two moving tables: one which traverses parallel to the locked stylus arm and the other indexing perpendicular to the arm thereby enabling parallel tracing to be performed. The former is driven through a pulley arrangement and reduction gearbox by means of a D.C. motor. Slow speeds of traverse (i.e. 1mm/sec or 0.25mm/sec) can be chosen according to the degree of surface roughness likely to be encountered and fast return speed is used in the reverse direction, i.e. when the stylus has been raised off the surface. The straightness accuracy of the slide-way is to within 1µm over the complete 150mm range of traverse. An optical encoder, coupled into the reduction gearbox, enables accurate positioning of the traversing table. Calibration results using an interferometer yield a basic encoder step length of 1.532µm ± 0.004µm over the full range of traverse. The sampling interval has a minimal value of 3.06µm (corresponding to every second encoder pulse) or can be any multiple of this up to 61.2µm. The indexing table, mounted on the lower traversing table, is driven by a stepper motor. This gives a minimal step-interval between the parallel traces of 2.501µm ± 0.034µm over a maximum 25mm traverse.

Lifting of the stylus arm, in order to raise the stylus off the surface to permit the fast return, is achieved by a motor-driven lifting arm connected to a light stirrup placed around the stylus carrying arm. No restrictions of the normal operation of the stylus instrument, i.e. single traces and meter cut-off options, have been incorporated into the design. Disconnection of the lifting stirrup and unlocking of the stylus arm enables the instrument to be used in its normal mode. A kinematic relocation facility, using the three-ball technique, is incorporated on top of the traversing table. Relocation of this table and the indexing table to within ±1µm is possible by means of the positional display units on the associated electronic control console. This console consists of several modules, each interlinked to provide a fully automatic operation cycle. Pre-settings of the required sampling/parallel step intervals and number of traces, together with the traverse position at which sampling will commence and end, are made on the console. A general view of the system is shown in Figure 2.

Ouput from the stylus head is fed directly to a North Star Horizon micro-computer through an 8-bit ADC input channel and recorded onto floppy discs. Data handling is software-controlled; the unit possessing a capacity of 16 x 10^3

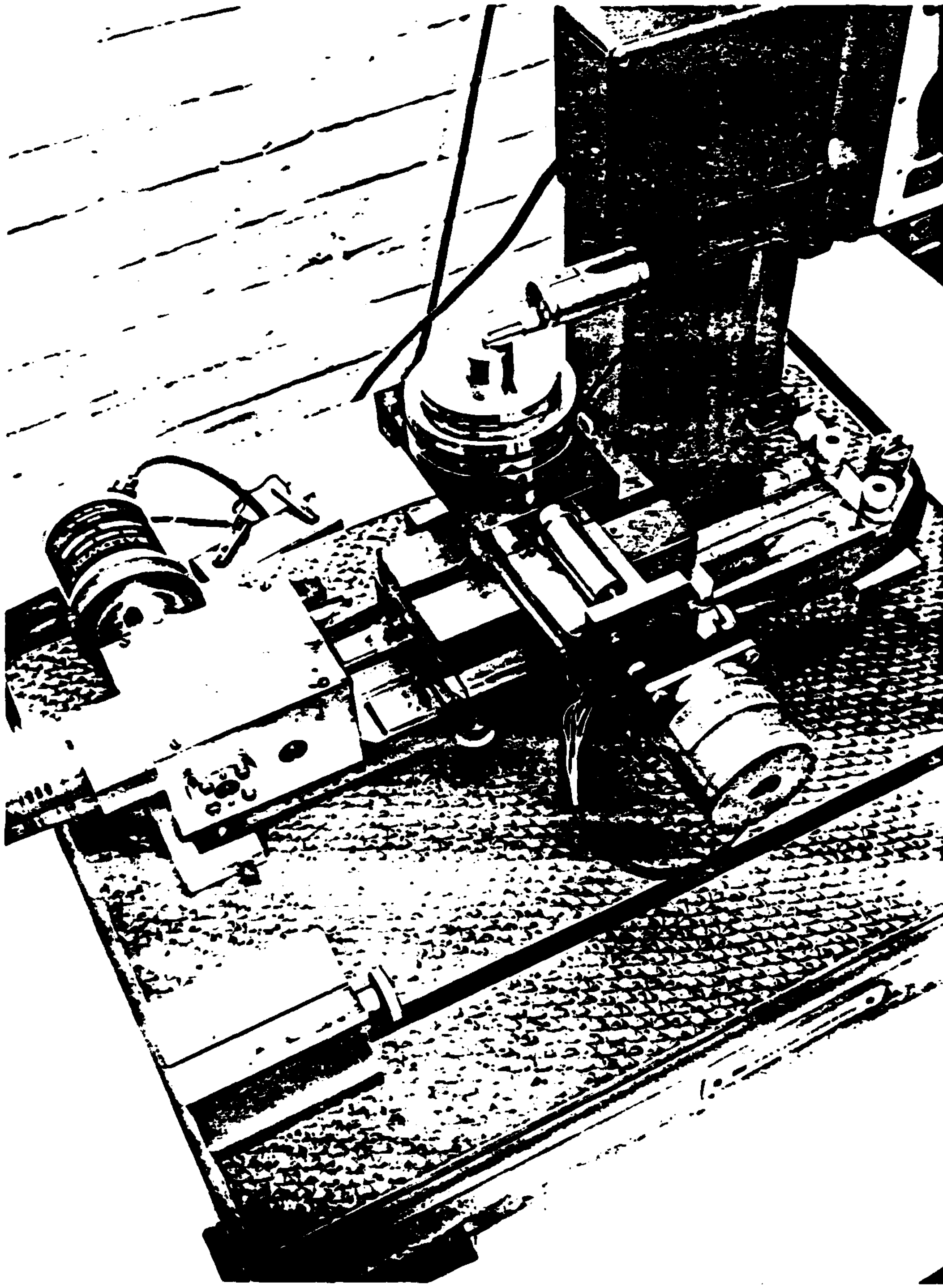

FIGURE 1 THREE-DIMENSIONAL RELOCATION STAGE

FIGURE 2 VIEW OF THE MEASURING SYSTEM

readings per stylus trace and up to 250 ensemble averages. At this stage some standard surface parameters may be evaluated on the micro-computer which provides an output to a line printer. For detailed statistical analyses and qualitative representation of the surface in the form of contour maps, data are transferred via a direct link to a GEC 4070 interactive computer, Figure 3.

5. RESULTS AND DISCUSSION

5.1 Surface Parameters

The effects of the measuring procedure on certain defined parameters for a surface are shown in Table 1. In particular, a quantitative illustration of the effect of performing parallel tracing and averaging the profile data may be observed. For turned and ground surfaces, some of the standard surface parameters computed on the micro-computer at various ensemble averages are given. R.M.S. roughness, ten-point height and mean surface slope for both surfaces show most distinct changes on increasing from single profile analyses to averaging from two parallel traces. Similarly the average wavelength parameter, which is particularly sensitive to the wear of the surface, significantly changes according to how many traces are analyzed. The skewness and kurtosis parameters, which have been used to assess the running-in of contacting surfaces show different effects as the number of profiles increases[6]. For both surfaces the skewness values show only minor changes, the ground surface exhibits a slightly negative skew and the turned surface a small degree of positive skew, see Figure 4. This does not occur for the recorded kurtosis parameter in that, for the ground surface at one profile trace, the implied distribution is platykurtic changing to strongly leptokurtic on increasing the number of traces taken. Consideration must be given (when assessing such parameters) to the problems of large scatter due to random sampling[7]. The power spectrum, Figure 5, becomes more evenly distributed as the number of profiles taken increases. It has been suggested that the power spectral density function derived by Fourier analysis, as presented here, is an insensitive indicator of wear and that the Walsh power spectrum may be more appropriate when attempting to quantitatively describe wear[8].

The commonly-used bearing area curves for both specimens, Figure 6, provide an interesting comparison between topographic and profile measurements. A total of 100 parallel profile traces, each trace containing 128 samples, makes up the array used for the computation of topographical curves. As would be expected, the single profile assessment taken along

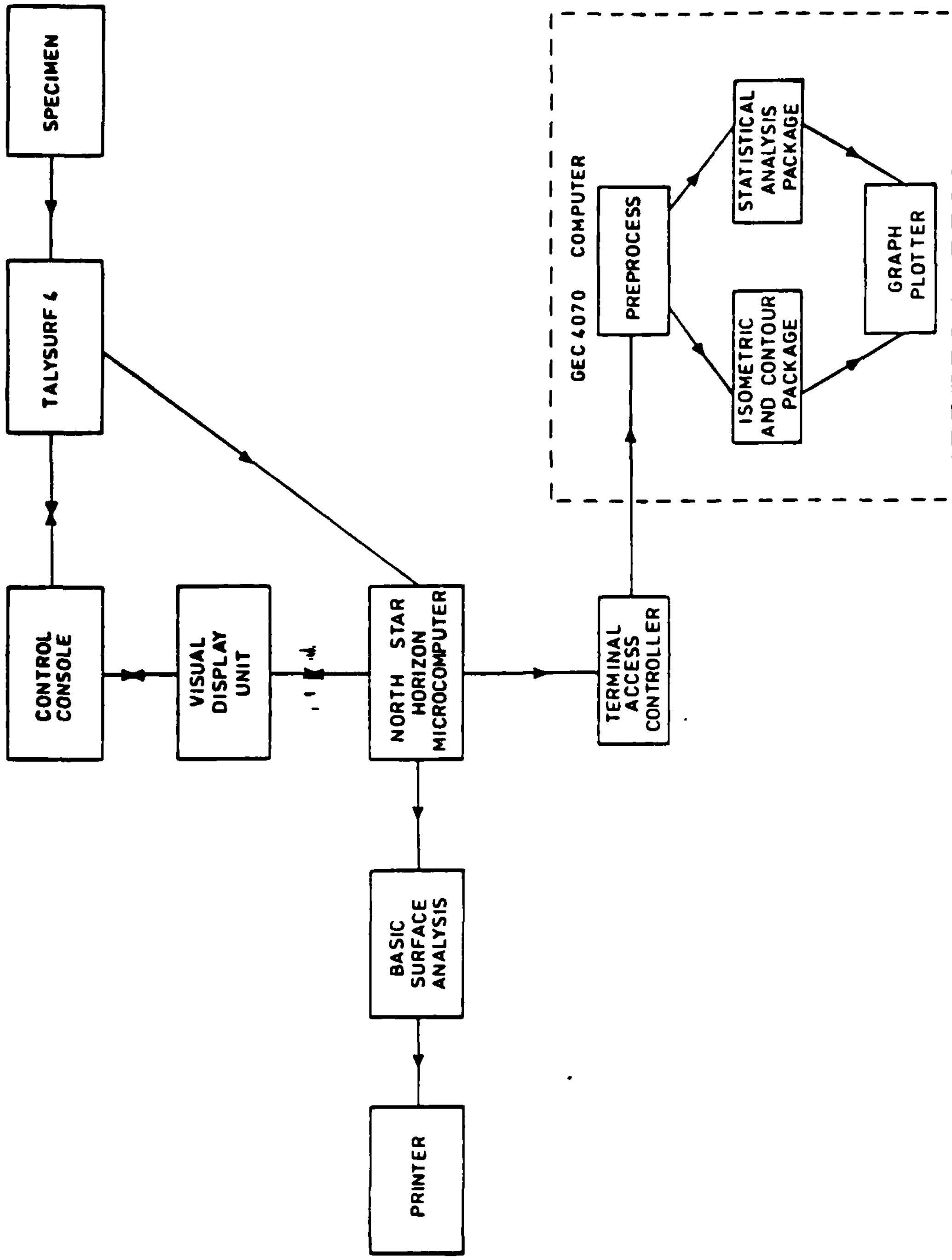

FIGURE 3 DATA HANDLING AND PROCESSING SEQUENCE OF OPERATIONS

Surface Examined	Parameters	R.M.S. Roughness R_q	Ten Point Height R_z	Mean Surface Slope m	Average Wavelength λ_q	Skewness R_s	Kurtosis R_k
Ground-Surface (along lay) parallel step interval of 12.5 μm	1 profile trace	0.286	1.68	0.042	42.78	-0.37	-5.30
	2 profile traces	0.848	4.58	0.181	29.44	-0.39	6.50
	5 " "	0.937	4.13	0.190	30.99	-0.45	7.72
	10 " "	0.829	4.62	0.144	36.17	-0.53	19.4
Turned-Surface (along lay) parallel step interval of 25 μm	1 profile trace	16.05	62.52	0.335	301.0	0.19	-7.06
	2 profile traces	17.54	85.01	0.990	111.3	0.33	-3.90
	5 " "	17.44	71.76	0.950	115.4	0.31	-4.72
	10 " "	17.89	79.79	0.995	112.9	0.32	-0.18

ALL ANALYSES: 2048 SAMPLES AT A 3.06 μm SAMPLING INTERVAL

TABLE 1 SURFACE TEXTURE PARAMETERS AS EVALUATED AT VARIOUS ENSEMBLE AVERAGES

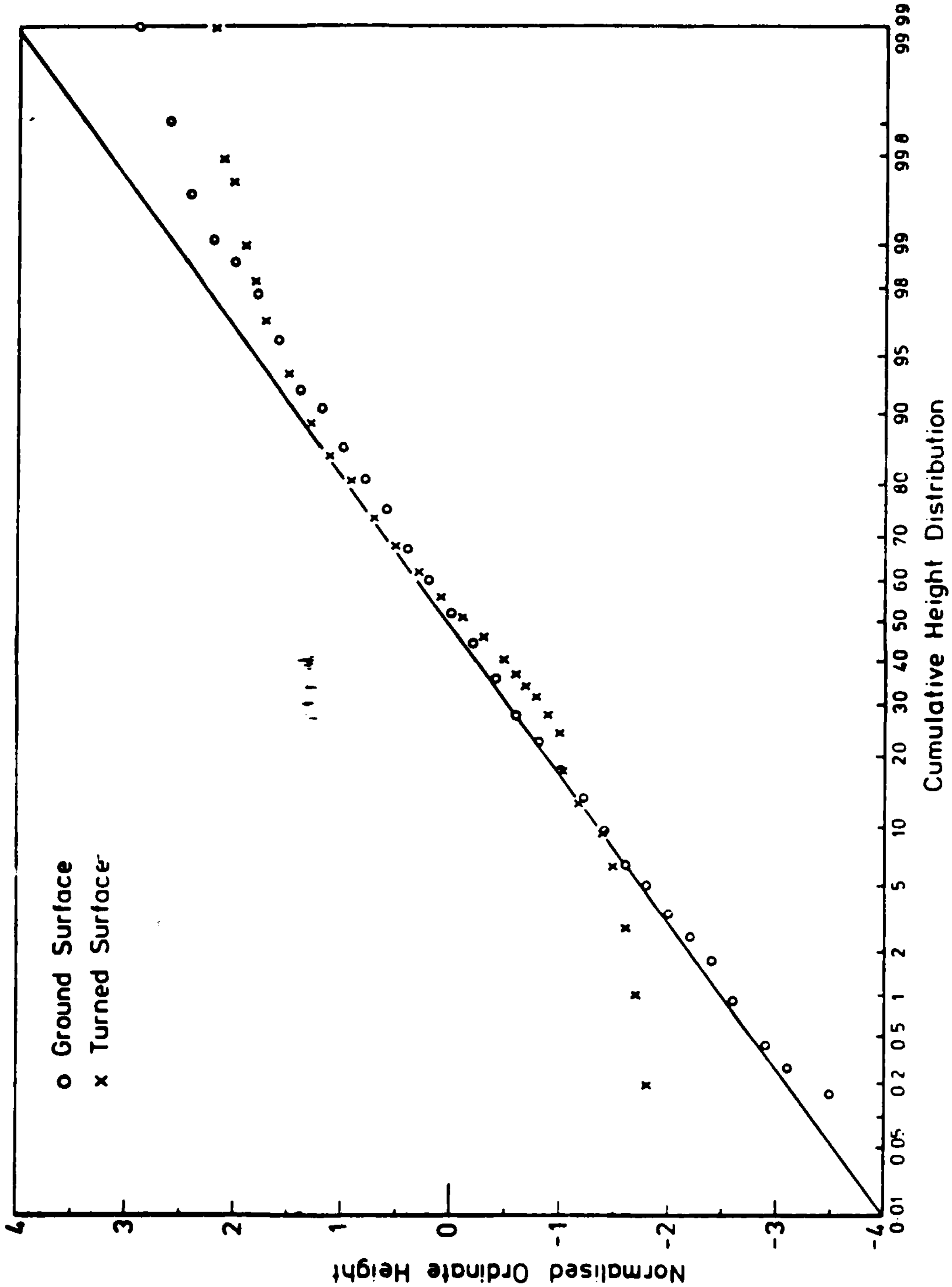

FIGURE 4 CUMULATIVE ORDINATE HEIGHT DISTRIBUTIONS

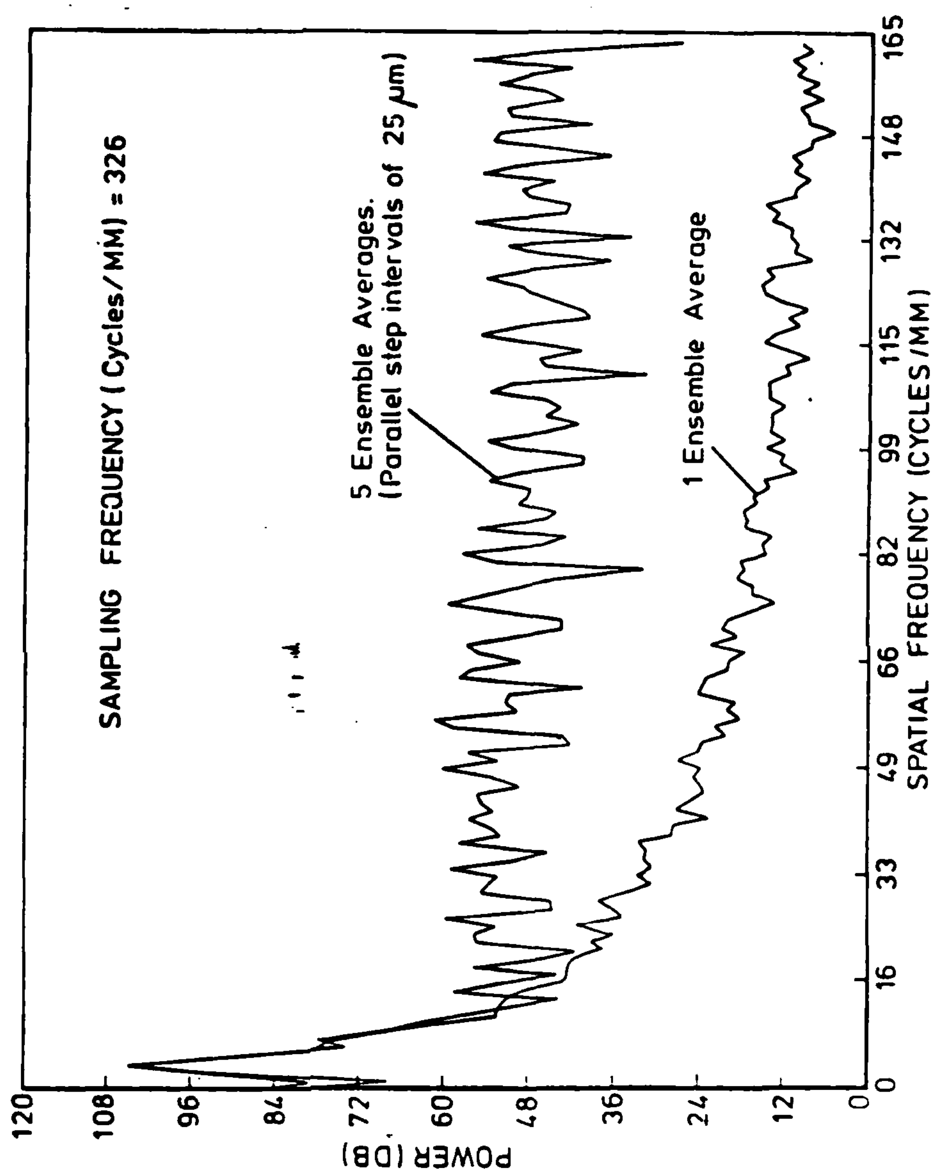

FIGURE 5 POWER SPECTRUM FOR A TURNED-SURFACE FOR TWO DIFFERENT ENSEMBLE AVERAGES

the lay of the periodic turned surface gives a far better description of bearing area than the assessment from a profile, taken at random, across the surface lay, Figure 6a. For the more random ground specimen, topographic, single profile and ten profile assessments are collated and compared in Figure 6b.

5.2 Surface Maps

An isometric view of a turned surface is depicted in Figure 7. This covers 1.57 x 1.25mm^2 of nominal area, made up from 100 parallel traces at a step interval of 12.5 μm, each trace containing 128 samples taken at 12.24μm intervals. The associated surface height distribution map, Figure 8, contains ten contour levels each at an interval of 8.6μm. Smoother surfaces may be described using the experimental measuring system described; Figure 9 illustrating a lapped surface of 0.021μm r.m.s. roughness. The assessment time required for surface tracing, data transfer, and plotting over such an area is typically about 90 minutes.

Such plots can assist in quantifying surface degradation due to wear or any other deformation process. The relocation facility enables the accurate assessment of the same area of surface before and after damage has been inflicted. Inverted maps (i.e. with the valleys represented as peaks) may be produced to assist in the examination of local deformations within valleys[9].

6. CONCLUSION

The experimental facility described provides accurate measurements and pictorial representations of surfaces in three-dimensions. Comparison of single profile assessments and averaged multiple profiles clearly highlights the advantage of this three-dimensional profilometric technique for the characterization of surfaces. Such systems enable useful quantitative assessments of surface wear to be obtained as well as to examine the relationships between profile and surface statistics.

To reduce computational time, the future development of the micro-computer software is now envisaged. Also, a completely software-controlled operation by the micro-computer is planned.

7. ACKNOWLEDGEMENT

The authors are indebted to the Science Research Council for support of this project.

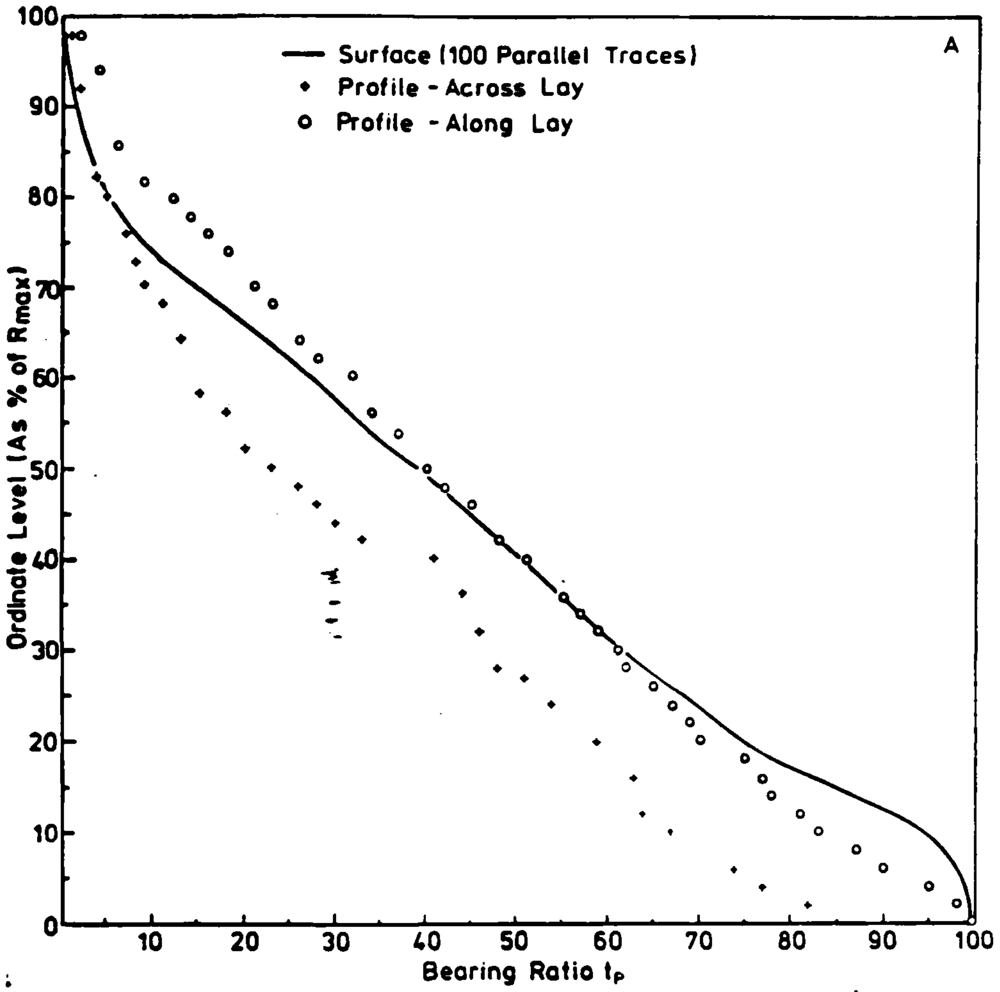

FIGURE 6(A) TOPOGRAPHICAL AND PROFILE BEARING AREA CURVES - TURNED-SURFACE BEARING CURVES

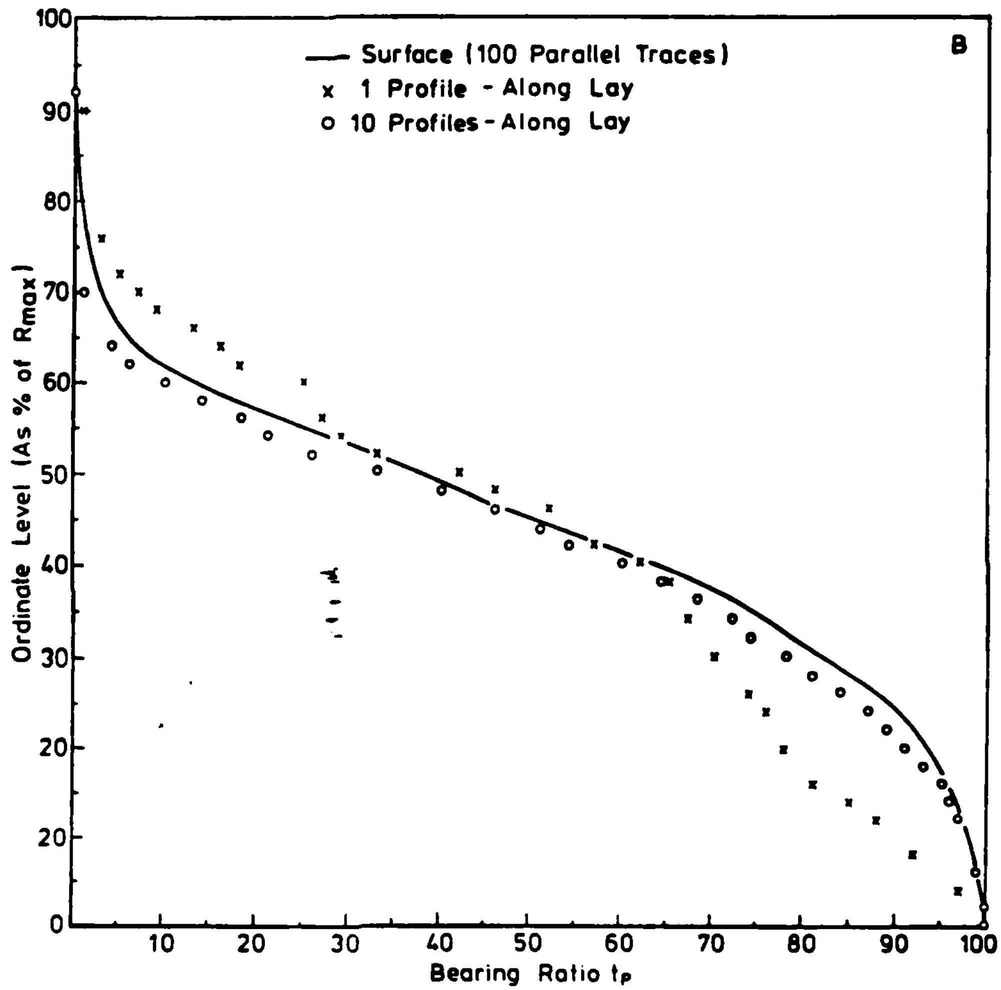

FIGURE 6(B) TOPOGRAPHICAL AND PROFILE BEARING AREA CURVES - GROUND-SURFACE BEARING CURVES

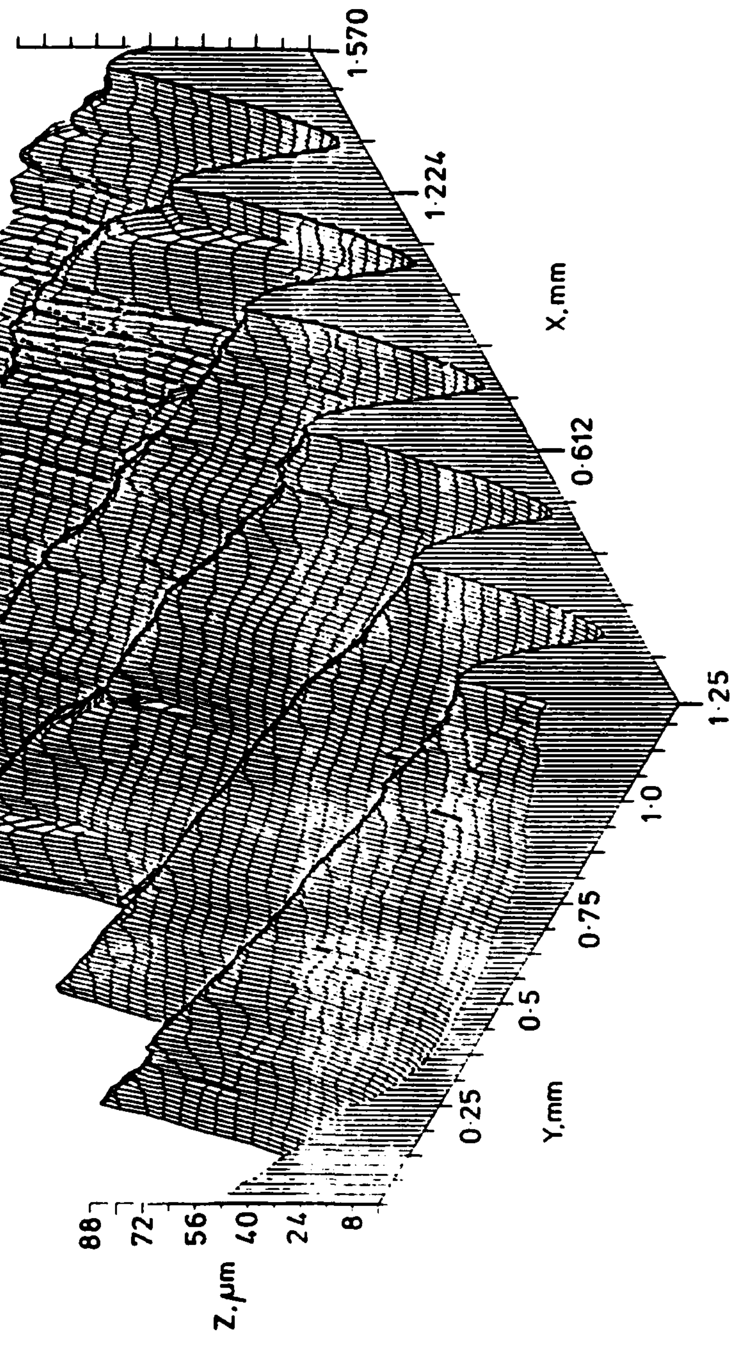

FIGURE 7 ISOMETRIC VIEW OF A TURNED-SURFACE

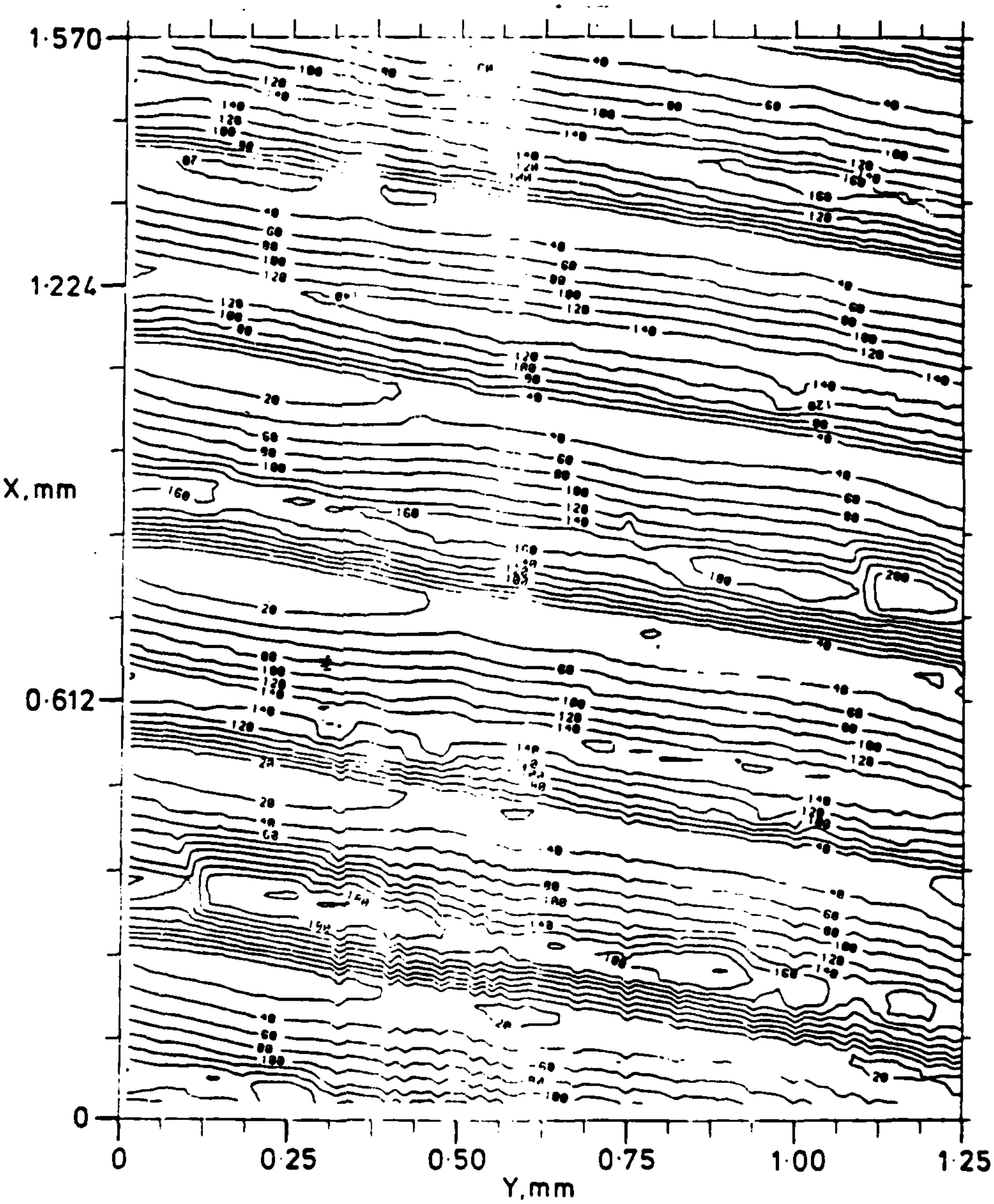

FIGURE 8 CONTOUR PLOT OF THE TURNED-SURFACE DEPICTED IN FIGURE 7

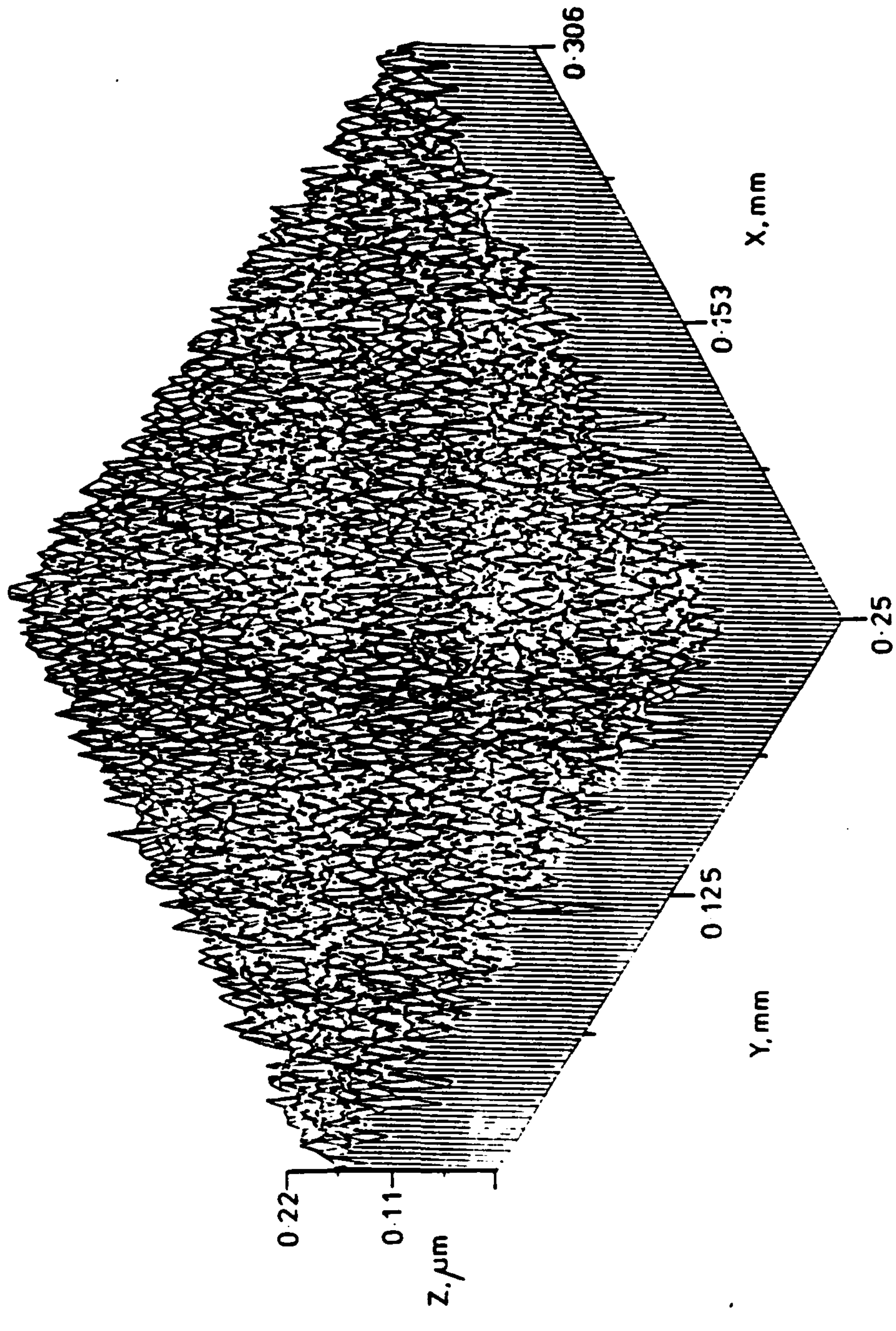

FIGURE 9 ISOMETRIC VIEW OF A LAPPED SURFACE

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