THE TRIBOLOGY OF GEARS FOR SATELLITE APPLICATIONS

K T Stevens

ESTL, Risley, UK

ABSTRACT

The paper describes an investigation into the tribological performance of gears for use in space satellite mechanisms. The work has been aimed at producing basic friction and wear data and also at proving particular material/lubricant combinations for specific mechanism applications.

Plasma nitrided gears gave low wear, even when running dry against themselves and are recommended in preference to through-hardened or precipitation hardened steel gears. Grease, oil and dry lubrication (MoS_2 or lead) all reduce wear to varying degrees.

MoS_2-filled polyimide gears have the lowest wear rate of the three plastic materials examined and can be used in light duty, low precision applications. Oil-filled porous polyimide performed poorly.

Tests on light anodised aluminium alloy gears running against themselves showed that they failed by cracking of the hard, thin coating but the performance was improved by running with MoS_2 or operating against plastic gears.

For gears in general, the tooth wear rate was found to be proportional to $(\text{load})^{1/2}$ and to be independent of speed.

Key Words: Gear, Vacuum, Plasma Nitriding, Plastic, Anodized Aluminium, Lead, MoS_2.

1. INTRODUCTION

Gears are now in common use in space satellite mechanisms. They are needed in solar array drives, pointing mechanisms, deployment systems and in robotic arms. As torque demands increase, the use of heavy, powerful motors to achieve high reliability via direct drive becomes unacceptable and the mechanism designer must consider employing a gear box.

The gears can be in many forms. For instance, a single, perhaps large reduction, pair; a purpose-built train of gears; or a commercially available instrument-type gear box. They all bring the same problems; the questions of lubrication and wear in a thermal vacuum environment and the problems of manufacture and precision.

Before choosing a gear system, it is important to consider the duty. This can be, for example, high load, low speed continuous operation (eg a solar array drive); a continuous oscillating motion, perhaps involving only one or two teeth (such as the trimming of a pointing mechanism); an infrequent, perhaps high speed, continuous operation (such as a deployment mechanism); or complex combinations of continuous or reversing motions (such as robotic arms).

This paper sets out to describe the materials and lubricants which are available for gears in space and discusses the tribological behaviour of a number of systems. References are made to specific case histories involving tests designed to simulate the operation of actual satellite mechanisms and some advice is given on optimising the heat treatment and hardening of gears.

2. GEAR MATERIALS

Depending on the duty required, a wide range of materials can be considered. Some are listed in Table 1.

Proceedings of the First European Symposium on 'Space Mechanisms & Tribology', Neuchâtel, Switzerland, 12-14 Oct. 1983. (ESA SP-196, publ. December 1983).

TABLE 1 GEAR MATERIALS

Material	Alternatives	Comments
Steel (For high loads)	i. Unhardened	Tribological problems, particularly with stainless steel
	ii. Hardened: Through-hardened	Distortion problems during manufacture
	Case-hardened	Eg, nitrided. Requires a nitriding steel, eg Nitralloy. Less distortion
	Precipitation-hardened	Hardening at lower temperatures with low distortion. Preciptation hardening steels or maraging steels.
Platings (usually on steel gears) (For medium loads)	i. Chrome plate ii. Electroless nickel (hardened) iii. Electroless nickel + PTFE	Advantage of a low application temperature. A medium carbon steel, tough substrate is advantageous.
	iv. Ion plated layers	Eg. TiN
Aluminium Alloys (Low loads, need for	i. Untreated	Severe tribological problems
	ii. Anodized iii. Zinal treatment	Surface hardening to improve wear resistance
Titanium Alloys (higher loads, light weight)	i. Untreated ii. Oxidised iii. Nitrided	Severe tribological problems Eg H.E.F. Tifran process Produces a surface layer of Titanium nitride.
Bronze (medium loads, usually run against steel)	i. Lead Bronze ii. Phosphor Bronze iii. Aluminium Bronze	Bronze gears can be run without lubrication
Plastics (light loads, low precision)	i. Polyacetal. ii. Polyimide iii. Cotton-phenolic laminate iv. Oil filled porous plastic	Plastic gears can be run without lubrication, usually against steel or aluminium.

2.1 Steal Gears

To optimise tribological performance, whether dry or lubricated, there is a need to produce a hard surface on the gear teeth. This can be achieved by through-hardening (heating to over 800°C, quenching and tempering) either a medium carbon alloy steel or, perhaps, a martensitic stainless steel (such as AISI 440C). Hardness values of 700VPN are practicable. Since there is considerable distortion, the final gear profile must be created after hardening and, when using the fine pitch teeth typical of space gears, this requires a difficult 'crush grinding' process.

An alternative method is to case-harden the steel and one of the most attractive ways is to 'plasma nitride' (also called 'ion nitriding'). This is a nitriding process carried out in a partial vacuum and it produces a clean, hard surface at processing temperatures as low as 470°C. Distortion is very much reduced and a very wear resistant surface (with a hardness up to 1200VPN) can be obtained (Fig 1). It is necessary to use a high alloy steel (such as a Cr/Mo/Al 'Nitralloy' steel) and the best response to nitriding occurs if the material is in a hardened and tempered condition. This also provides a tough substrate to support the hardened layer (typically 0.2mm deep) on the surface. Distortion can be further minimised by including a stress-relieving stage (at a temperature higher than the intended nitriding temperature) after any initial machining and before the final tooth profile is created.

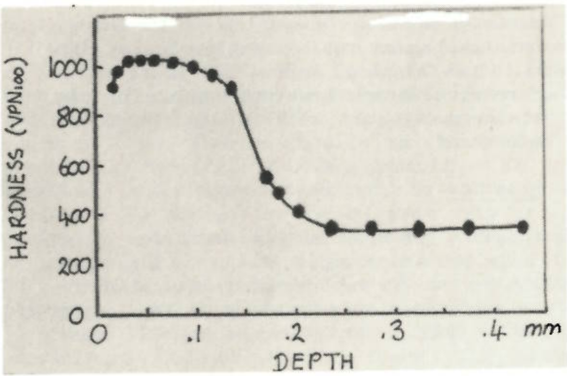


Figure 1 Hardness Profile Obtained with Plasma Nitrided Nitralloy Steel

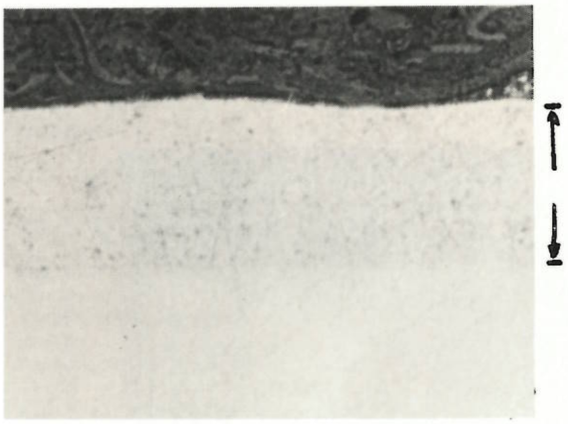
Hardening at even lower temperatures (with a consequent lower distortion) can be achieved using precipatation hardening steels (typically 14% Cr, 5% Ni, 2% Mo, 0.06% C) or maraging steels (typically 18% Ni, 9% Co, 5% Mo, 0.01% C). Treatments typically involve the soaking of the finished gear in an inert atmosphere for 4 hrs at 400°C. Hardness values upto 800VPN can be achieved.

2.2 Platings

The electrolytic processes are not ideally suited for gears because the tooth tips are preferentially coated and the geometry is modified. Hence, any coating (for instance, chrome plate) should be kept to a small thickness; usually less than 0.01mm. To support the layer, the preferred substrate is a medium carbon steel.

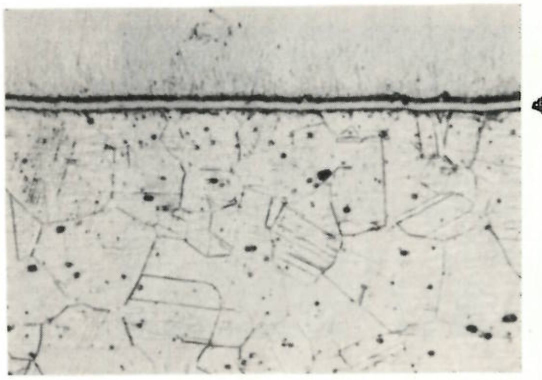
An electroless process has the advantage of depositing a completely even coating. A thickness of 0.05mm is suitable but the original gear must be made undersize or the gear centres adjusted to allow for the increased dimensions. Electroless nickel (93% Ni, 7% P) can be post heat-treated for 1hr at 400°C to increase the hardness from 550VPN (as deposited) to 1000VPN. There are also newer coatings incorporating PTFE into electroless nickel which offer the possibility of self-lubrication (Fig 2).

The modern techniques of ion plating and sputtering offer the possibility of applying ceramic layers (eg TiN, TiC, ZrN) to the gear teeth. However, they are very thin (0.5 to 3 microns, Fig 3) and it is essential to support them with a tough substrate. Plasma nitriding followed by ion-plating is a possible two-stage process to increase the wear resistance of a gear.



x320

Figure 2 Electroless Nickel/PTFE Coating on Stainless Steel



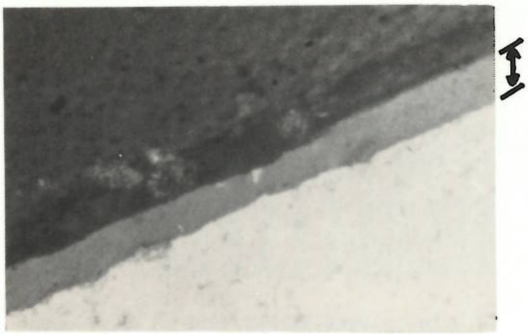
x1000

Figure 3 A Thin Coating of TiN On Hardened Steel

2.3 Aluminium and Titanium Alloys

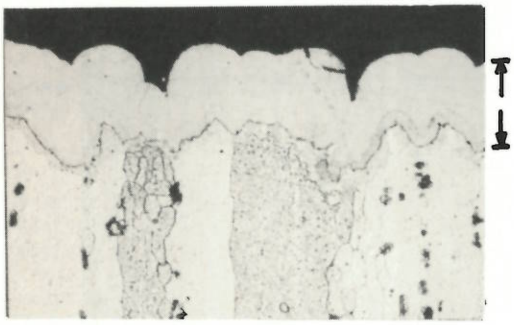
The need to save weight may force consideration of aluminium or titanium alloy gears. Neither material is acceptable tribologically in an untreated condition and it is necessary to harden the surface. For aluminium, it is possible to hard anodise, producing a layer typically 10 microns thick (Fig 4) with a hardness near 500VPN. another possibility is the H.E.F. (France) Zinal treatment which involves a two stage process; electroplating with coating comprising Indium and other elements followed by a heat treatment to create diffusion and to form hard, intemetallic phases at the surface (Fig 5).

It has recently been demonstrated (for instance, at Birmingham University) that titanium alloys can be plasma nitrided, producing a surface layer of TiN (Fig 6). Wear tests on a Falex machine have proved very successful and the technique is promising. The treatment temperature is high (over 900°C) but there are no phase changes to produce large distortions. Following such a treatment there will be a loss of core hardness and it is necessary to post heat-treat by quenching from 850°C and ageing to give a tough, hard substrate, (for example, near to 400VPN hardness with an IMI 318 alloy).



x420

Figure 4 Ematal Hard Anodised Layer of Aluminium Alloy



x420

Figure 5 Zinal Treated Aluminium Alloy

x420

Figure 6 Plasma Nitrided Titanium Alloy

2.4 Bronze

It is usual to employ bronze gears in conjunction with steel gears to give a smooth-running couple. Whilst lead bronze is a praticularly good self lubricating material it is usually considered to be too soft for gears (unless the duty is very light). The normal material is an ordinary tin-bronze but, for harder and more wear resistant gears, phosphor bronze or aluminium bronze can be used. For low precision applications, the harder grades of aluminium bronze can be cast. It is also common to use sintered bronze for gears which either retain lubricant on the surface or can be vacuum-impregnated with a space-approved oil beforehand.

2.5 Plastics

Plastic gears can only be considered for light load, low precision applications. They will normally be run dry against stainless steel or against anodised aluminium alloy. The latter combination will, or course, be all light weight.

There is a wide variety of plastics available with good tribological properties and a lot of data has been produced for their friction and wear in air. For gears, there is a need for stiffness and hardness as well as for low friction and wear and this limits the choice to some degree. For instance, PTFE, unless it is heavily reinforced, is not suitable. Other materials, for example nylon, absorb large amounts of moisture and are not attractive for use in vacuum. Materials well suited for gears are polyacetal (perhaps with carbon fibre or glass fibre reinforcement for extra stiffness) and polyimide (with a solid lubricant additive such as MoS_2). High pressure laminates (for example, cotton fabric phenolic similar to those used for bearing cages in space) are particularly tough and resiliant and are well suited for gears. They must be machined from sheet and the surface quality of the gears will not be as good as that obtained with polyimide or polyacetal. The latter is particularly easy to machine and is therefore, suitable for fine pitch gears. In general, materials containing graphite cannot be expected to perform well in vacuum, even if wear data suggests a good performance in air.

Finally, it is also possible to consider the use of a porous plastic impregnated with a liquid lubricant. An oil used in this way for space satellite applications should have a low vapour pressure and it is unlikely that any products available commercially (such as oil-filled polyacetal) will be suitable. It would be necessary to machine gears from unfilled porous plastic and then to vacuum impregnate with the required oil.

3. GEAR LUBRICANTS

Since a conventional approach of running the gears with an oil dip or splash is usually undesirable in space the simplest approach is either to operate dry or to lubricate with grease or solids.

Any grease used in a satellite mechanism must have acceptable outgassing properties to avoid any unwanted contamination on things like optical surfaces or other critical areas. The number of products available is comparatively few and two excellent greases, BP2110 (mineral oil based) and BP 8135 (tre-ester based) are no longer manufactured. There are greases based on KG80 oil (Kendal, a highly refined mineral oil with a vapour pressure better than 10^{-8} Torr at 20°C) and on Braycote 815Z oil (Bray Oil, a fluorinated oil with an exceptionally low vapour pressure of 10^{-13} torr at 20°C. An equivalent oil is Fomblin Z25, manufactured by Montecatini -Edison in Italy). There are also greases based on other fluorinated oils such as those produced by Montecatini Edison (Fomblin Y VAC greases) and those produced by Kluber (Barrieta ES grades) or by Du Pont (Krytox greases). Whatever grease is used, it must be packed around all the teeth and isolated by a creep barrier painted in a circle around the gear. After a long period of operation in space it can be anticipated that, if precautions are not taken, there will be sufficient oil loss by creep to completely ry out the grease.

An alternative to fluid lubrication is dry lubrication and thin films (1 or 2 microns) of lead or MoS_2 will give protections against wear. The perfomance of these materials has been well proven in ball-bearings at the European Space Tribology Laboratory (ESTL), (Ref 1) and both coatings can be applied to gears; lead by evaporative ion plating and MoS_2 by sputtering. MoS_2 can also be applied in a bonded film; for instance Molykote 321R has exhibited good friction and wear properties in sliding tests at ESTL (Ref 2).

Instead of applying the lubricant to the gears before operation it is feasible to use an idler gear (perhaps PTFE) to transfer a film to the teeth during operation. However, it is difficult to control the amount of transfer and there is a danger that tooth clearances will be reduced and that the running torque will increase.

4. TRIBOLOGY OF GEARS IN VACUUM

A work programme is on-going at the European Space Tribology Laboratory to investigate the friction and wear of a range of gear materials and lubricants in vacuum. The tests are aimed at two specific objectives.

i. To obtain general tribological data on a range of gear materials under dry and lubricated conditions. The information will enable ESTL to advise ESA and its contractors on all aspects of gear problems.

ii. To prove the satisfactory performance of gear materials and lubricants in particular applications which arise within the ESA programme.

The results reported here originate from a mixture of work packages performed against both these objectives so, in some cases, the information is widely applicable whilst, in others, it is very specific to certain operating conditions.

To date, the tribological investigations have been performed with steel, aluminium alloy and plastic gears. No work has yet been conducted with bronze or titanium gears. All tests have been performed at laboratory ambient temperature (21°C).

4.1 ESTL Test Gears and Equipment

4.1.1 Equipment

The ESTL test equipment consists of a 'four-square' arrangement of gears inside a vacuum chamber with the pinion shaft being driven by a variable speed motor via a ferrofluidic feedthrough (Fig 7). This last is a proprietary device using a seal of magnetic particles suspended in a lubricant which is then maintained in the gap by a magnetic field. The system is evacuated by a sorption pump and ion pumps with pressures less than 10^{-7} torr being attained at 20°C.

The gear loading is applied by 'winding-up' the lower wheel gear (mounted on bearings) with the spring wire before clamping all the gears on their respective shafts. A known wind-up torque is achieved by using either a spring balance or a pulley and weights to twist the wire. The frictional torque reaction from the gear assembly when in motion causes the mounting block and wheel gears to rotate backwards around the drive shaft. This motion is restrained by a force transducer and the output from this is used to monitor the torque. This torque is made up of the frictional reaction from the gears and from the friction in the support ball bearings. These support bearings are dry-lubricated (PTFE/MoS_2)/Glass Fibre scrificial cages) and their torque has been measured in a separate experiment using a gear set with a zero-wind-up torque.

The gear assembly is surrounded by a thermal shroud which can be heated or cooled by fluids fed from thermostatically controlled baths. Then, by a process of radiation heat transfer, the test temperature can be set an any value between -50°C and +80°C.

4.1.2 Gears

The details of the ESTL gears are as follows:

Pinions 41 teeth, PCD = 30mm, 10mm wide
Wheels 120 teeth, PCD = 90mm, 3mm wide

Both have a module 0.75 tooth size. They are manufactured accroding to British Standard 978 Part 1 (1968) - Class C.

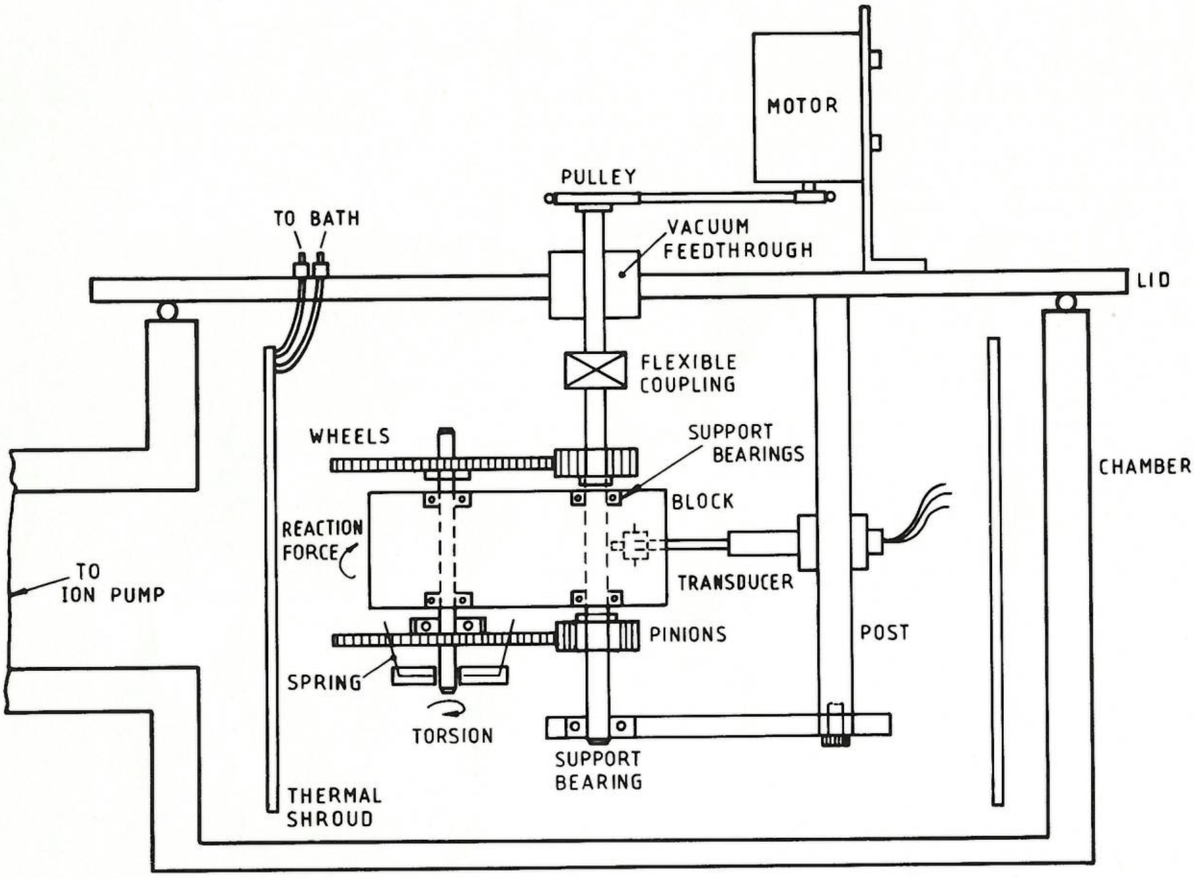


FIGURE 7 GEAR TEST RIG

4.2 Test Results

4.2.1 Steel Gears

A summary of the tribological data obtained with steel gears is given in table 2.

The results shown originate from a mixture of tests, some intended to simulate specific satellite applications and others intended to investigate basic tribological behaviour.

Tests 1 and 2 simulate the operation of a solar array drive (1rev per day for 10 years) and demonstrate the benefit of lead lubrication in reducing both friction and wear. The material combination was the initial choice of the contractor (prompted by good test results reported by Courtney, Ref 3) and it was clear that, if run dry, considerable damage could be expected on the 440C pinions. Fig 8a shows the extent of this wear and illustrates the improvement when ion-plated lead was used as a lubricant. In fact, a combination of nitrided steel against itself proved to be more satisfactory with lead lubrication again being beneficial (Test 7 and 8, table 2). Tests 3-6 were performed in an oscillating mode and simulated the trimming of an antenna pointing mechanism. The intention was to grease-lubricate (Braycote 3L-38RP) and the accelarated test showed no detectable wear (test 4). In practice, however, during prolonged operation in space, oil creep would occur and the grease would dry out. Even so, tests 5-6 demonstrated that the grease filler alone (PTFE) or very small traces of the base oil ($2\mu l$) were able to significantly reduce the wear compared to that obtained running dry (test 3). Again, if nitrided pinious were used instead of 440C (test 9), less wear resulted.

Since nitrided steel against itself appeared to be a satisfactory combination, tests 10-15 were performed to investigate the tribological performance more fully.

TABLE 2 TRIBOLOGICAL PERFORMANCE OF STEEL GEARS

Test No	Materials		Tooth load (N/mm)	Pinion Speed (rpm)	lubricant	Running torque ($NM \times 10^{-4}$)	Total* Pinion Revs	Tooth flank wear rate (mm depth per encounter)	
	Pinions	Wheels						Pinions	Wheels
1	Through Hardened 440C	Plasma Nitrided EN40B	17 (320)¥	10	Dry	90 220	10^5	2×10^{-7}	No measureable wear (NM)
2	"	"	17	10	Lead	60	10^5	5×10^{-8}	NM
3	"	"	17	10	Dry	300	3×10^5	1×10^{-7}	NM
4	"	"	17	10	Grease	63	3×10^5	NM	NM
5	"	"	17	10	PTFE	240	3×10^5	3×10^{-8}	NM
6	"	"	17	10	Oil(trace)	165	3×10^5	1×10^{-8}	NM
7	Plasma Nitrided EN40B	Plasma Nitrided EN40B	17	10	Dry	100 240	10^5	8×10^{-8}	5×10^{-8}
8	"	"	17	10	Lead	70	10^5	2×10^{-8}	NM
9	"	"	17	10	Dry	165	3×10^5	2×10^{-8}	2×10^{-8}
10	"	"	7(200)	50	Dry	80	3×10^4	1.3×10^{-7}	NM
11	"	"	7	50	Dry	80	2×10^6	1.0×10^{-9}	1×10^{-9}
12	"	"	7	300	Dry	NA#	7×10^6	1.1×10^{-9}	1×10^{-9}
13	"	"	7	1000	Dry	NA	2×10^7	1.1×10^{-9}	1×10^{-9}
14	"	"	3.5(150)	1000	Dry	NA	3×10^7	7×10^{-10}	NM
15	"	"	21(350)	1000	Dry	NA	4×10^7	1.7×10^{-9}	2×10^{-9}
16	"	"	7	50	MoS_2	30	3×10^4	1×10^{-8}	NM
17	"	"	7	300	MoS_2	NA	2×10^5	1×10^{-8}	NM
18	"	"	7	300	MoS_2	NA	2×10^6	1.6×10^{-9}	1×10^{-9}
19	FV520B	"	7	300	Dry	NA	10^6	1×10^{-7}	NM
20	G125	"	7	300	Dry	NA	10^6	2×10^{-8}	NM
21	Elec Ni /PTFE	Elec Ni /PTFE	7	300	Dry	NA	2×10^6	2×10^{-7}	2×10^{-7}

Key: *Wheel revolutions will be ⅓rd of this value #Torque readings not taken at higher speeds
¥Tooth contact stress in N/mm^2 is given in the brackets.

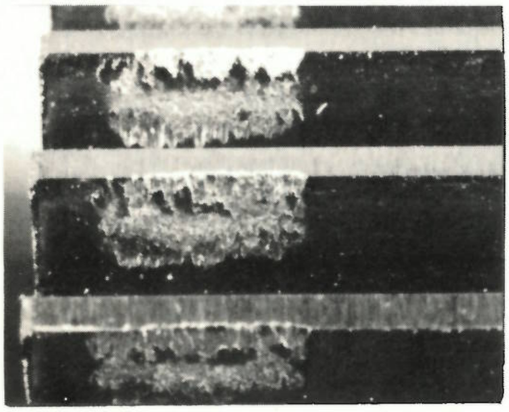
Three points emerged:

i. Initially, there is brief period of high wear rate (test 10) when the gears run-in. The wear depth on the teeth reaches 0.001mm.

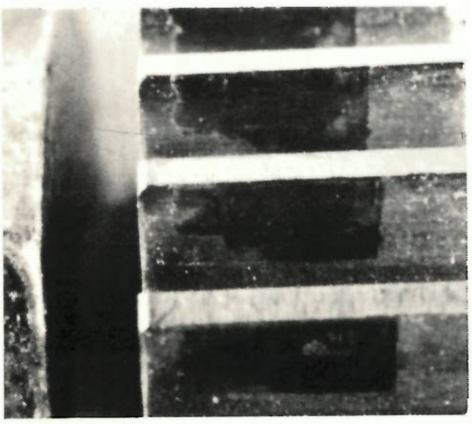
ii. Thereafter, the wear rate falls (test 11) and is unaffected by the speed (tests 12 and 13).

iii. The wear rate is roughly proportional to $(\text{load})^{1/2}$ (tests 13, 14 and 15).

A feature of this sequence of tests was the smooth running (a steady torque) that was maintained and the smooth, polished wear on the teeth. This is illustrated in Fig 9 which shows the appearance at the end of the test when wear had progressed to a depth of 0.05mm, about ⅓rd of the way through the hard, nitrided layer.



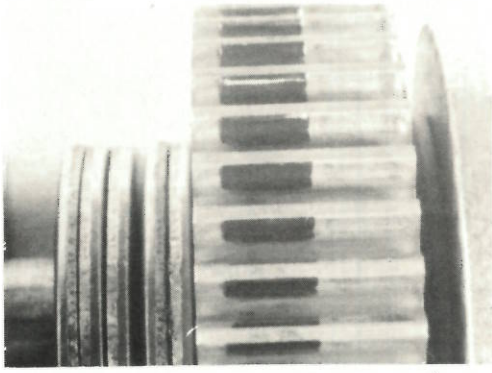
A) Dry x8



B) Lead Lubricated x7

Figure 8 Wear of Hardened AISI 440C Gears

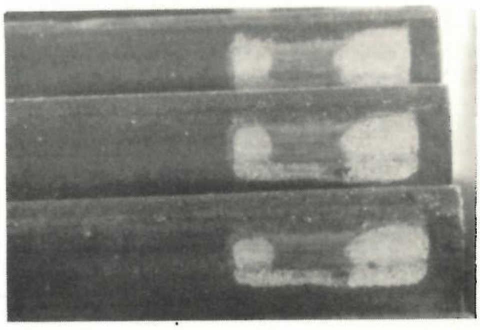
During the operation of gears, the tooth contact is a combination of rolling and sliding, pure rolling occurring at the tooth pitch point. The actual sliding velocity is low and, even at 1000rpm, averages only 0.5m/sec. In pure sliding it would be expected that wear rate would proportional to load but, in the roll/slide situation in gears, it appears to be proportional to the Hertzian stress (ie, proportional to $(\text{load})^{1/2}$). With the ESTL gears, a tooth load of 7N/mm is equivalent to a Hertzian contact stress of 200N/mm^2.



x4

Figure 9 Smooth, Polished Wear on Plasma Nitrided Gear

In tests 16-18, a 2 micron thick layer of sputtered MoS_2 was used to lubricate the gears. It provided protection for about 2×10^5 tooth encounters after which most of the lubricant had been lost from the contact area (Fig 10). Further running gave a wear rate similar to that found in the dry test.



x6

Figure 10 MoS_2 - Lubricated Nitrided Gear

FV420B is a precipitation hardening steel (14% Cr, 5%Ni, 3% Mo + W, 0.05% C) and pinions were aged at 400°C to give a hardness of 550VPN. The wear rate (test 19) was high (Fig 11) and the material is not recommended for space gears. A better performance was achieved with the G125 maraging steel (test 20, Fig 12). This is again a precipitation hardening steel (18% Ni, 9% Co, 5% Mo, 0.01% C) and was age hardened to 750VPN at 400°C to give a wear performance similar to that of through-hardened 440C but without problems of distortion.

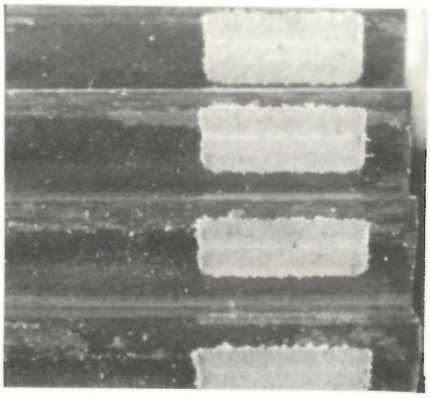


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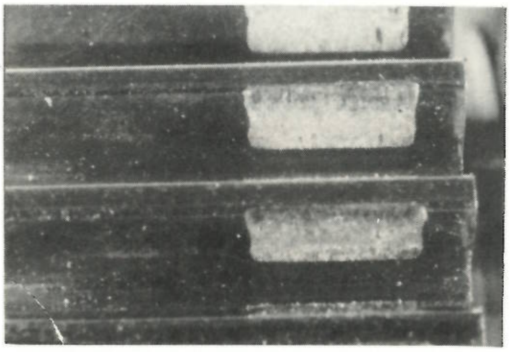
Figure 11 Wear of FV5208B Gear



x6

Figure 12 Wear of G125 Maraging Steel Gear

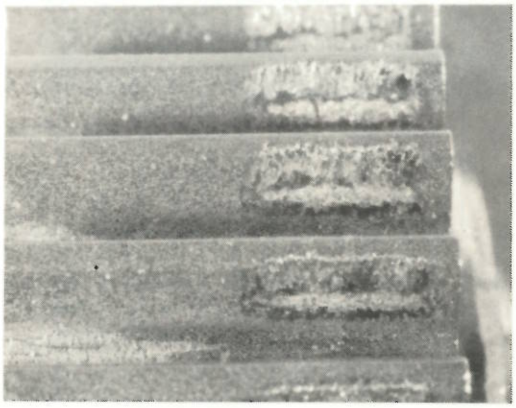
Finally, the electroless nickel/PTFE coating, when applied to stainless steel gears, gave a poor performance (test 21). The wear rate was a factor of 200 greater than that found with nitrided steel and the tooth surface was roughened and torn (Fig 13).

4.2.2 Aluminium Alloy Gears

Gears were manufactured in Aluminium alloy type ISO 6082 and treated by an anodising technique (Ematal, Contravees, Switzerland) and by the Induim-based electroplating/diffusion process (Zinal, H.E.F. France). The base material was used in two conditions; annealed (hardness 65VPN) and age-hardened (hardness 105VPN).

A summary of the test results is contained in Table 3.

The first of the tests was designed to simulate the operation of a deployment mechanism. The required duty was short (10^3 revs) and the Ematal-treated gears survived without measureable wear. Fig 14 shows that the tooth surfaces were marked but that no cracking or removal of material could be detected.



x6

Figure 13 Worn Electroless Nickel/PTFE Gear

x40

Figure 14 Slight Wear on Ematal Treated Aluminium Alloy Gear

TABLE 3 TRIBOLOGICAL BEHAVIOUR OF ALUMINIUM ALLOY GEARS

Test No	Materials		Tooth load (N/mm)	Pinion Speed (rpm)	lubricant	Running torque ($NMx10^{-4}$)	Total pinion Revs	Tooth flank wear rate (mm depth per encounter)	
	Pinion	Wheel						Pinion	Wheels
1	Ematal Annealed Al Alloy	Ematal Annealed Al Alloy	10 (150)#	0.2	Dry	290	10^3	NM	NM
2	"	"	7(125)	5	Dry	NA	4×10^3	Coating cracked	Coating cracked
3	"	"	1.5(60)	5	Dry	NA	$2.3x10^4$	"	"
4	"	"	0.3(25)	50	Dry	NA	$2.3x10^5$	"	"
5	Ematal Hardened Al Alloy	Ematal Hardened Al Alloy	7	50	Dry	210	$1.8x10^5$	"	"
6	"	"	7	50	MoS_2	100-200	$2.4x10^6$	"	"
7	MoS_2 Filled polyimide	Ematal Annealed Al Alloy	7	1000	Dry	NA	10^7	Zero, plastic transfer	3×10^{-9}
8	Zinal Annealed Al Alloy	Zinal Annealed Al Alloy	7	50	Dry	NA	$7.5x10^4$	Coating cracked	Coating cracked

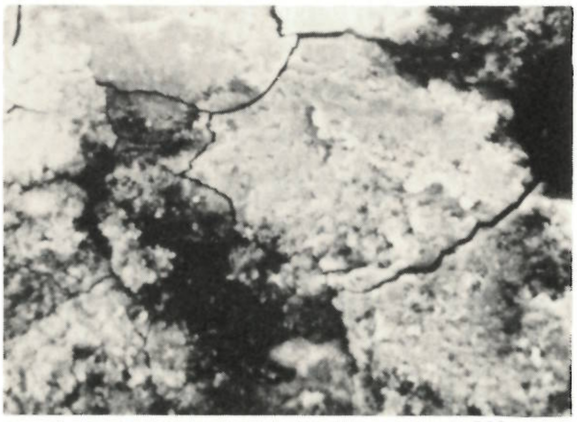
Key: #Tooth contact stress in N/mm^2

Extending the tests further (test 2 to 4) revealed that, when run dry, failure of the coating occurred quite quickly; not by wear but by cracking and flaking. The onset of this failure is illustrated in Fig 15 and continuation of the test beyond that point quickly results in exposure of the soft substrate and very rapid wear. Within a few thousand revs the teeth are worn completely and the wheel gears are no longer driven. The time to failure increases as the load is decreased and there was considerable improvement when a hardened substrate was used (test 5), the life being increased from 4×10^3 revs to 1.8×10^5 revs. However, even this life is relatively short and it is obviously unwise to run anodised aluminium gears without lubrication for long periods.

It is clear that one reason for failure is collapse of the hard, brittle anodised layer into the soft substrate under the compressive stress of the gear contact. The improvement with a hardened substrate would, therefore, be expected. However, the high shear forces experienced during dry rubbing are equally, if not more, important as illustrated when Ematal-treated wheels were run against plastic (MoS_2 filled polyimide) pinions. In that test (test 7) there was no failure, the Ematal picking up a transfer film (Fig 16) and the plastic gear wearing steadily but smoothly (Fig 17). The wear rate of the plastic was 3 times that observed when running dry against plasma-nitrided steel (Section 4.2.3).

x50

A) Tooth Wear Scar



x800

B) In Upper Region

Figure 15 Failure of Anodised Coating

x3

Figure 16 Polymer Transfer to Ematal Treated Aluminium Alloy Wheel Gear

Another way to reduce the shear forces is to use a lubricant. With a sputtered MoS_2 film (test 6) on Ematal-treated hardened aluminium alloy the life was increased from 1.8×10^5 revs to 2.4×10^6 revs, a factor of 15

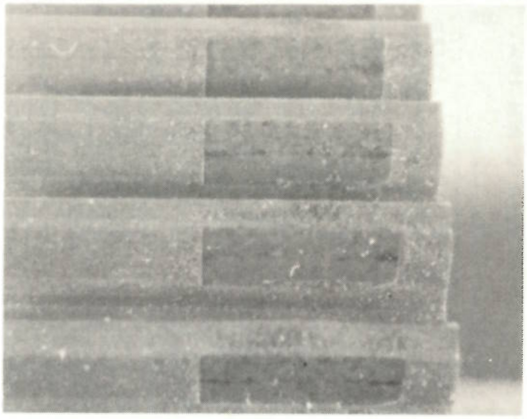


Figure 17 Wear of Polyimide/MoS_2 Pinion, Run Against Ematal

improvement. At the end of the test, the MoS_2 had been worn off the gear teeth, the torque had increased and the coating showed the first signs of cracking.

Finally, the one test conducted so far with the Zinal treated gears (test 8) exhibited the same type of failure but the life (7.5×10^4 revs) was much greater than that found with Ematal (4×10^3 revs). More tests are planned using a hardened substrate and also lubrication.

4.2.3 Plastic Gears

Three types of plastic have been tested:

i. Carbon fibre (20%) filled polyacetal manufactured by the United Kingdom Atomic Energy Authority, Harwell (UK).

ii. Vespel SP31, an MoS_2 (15%) filled polyimide manufactured by D U Pont (USA).

iii. Vespel SP8, a porous (25%) polyimide, again by D U Pont, and impregnated with a low vapour pressure mineral oil, BP110.

The test results are summarised in table 4.

Tests 1 to 7 demonstrate the wear performance of the carbon-fibre reinforced polyacetal pinions at various speeds. Tests 11 to 17 are the equivalent experiments with Vespel SP31 gears. Both were, in fact, continuous experiments where the speed was increased in steps and the progress of wear with running distance is shown in Fig 18. The following conclusions emerge:

i. There is no evidence of any change in wear rate with speed for either material. At 1000rpm the sliding speed averages only 0.5m/sec and J K Lancaster (Ref 4) has published work showing that changes in wear rate are not observed with polyacetal until the speed reaches more than 1m/sec (when flash temperatures reach the melting point of the plastic)

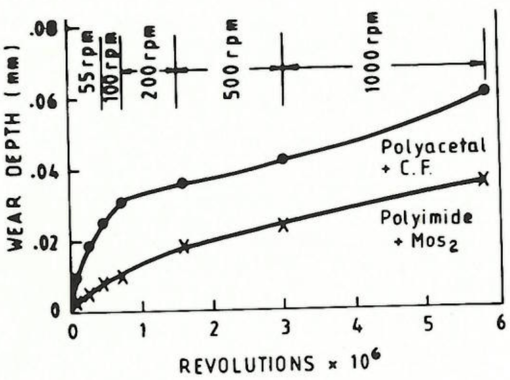


Figure 18 Effect of Speed on the Wear of Polyacetal and Polyimide Gears

ii. The running-in wear is greater for the polyacetal than for the polyimide.

iii. The steady state wear rate (21 N/mm load) is about 5×10^{-9} mm depth/ tooth encounter forboth materials.

Fig 19 shows the appearance of the polyacetal gear after completion of 6×10^6 revs (after test 7), the scar being divided into three areas; the pitch point (where the gears roll) and the sliding areas above and below. Fig 19b shows there is no reduction in wear at the actual pitch point. This may be because the wear of plastics tends to be related to PV (pressure x velocity) and, even though the sliding speed (V) is low at the pitch point, the actual contact pressure (P) is a maximum. Either side of the pitch point, P decreases but V increases.

A(x15)

Figure 19
Tooth Wear Scar
On Polyacetal Gear

B)x50

The effect of load is demonstrated by tests 7 to 10 (polyacetal) and 17 to 20 (Vespel SP31). Like the results with steel gears, the wear rate tends to be proportional to about $(\text{load})^{1/2}$. However, the overall wear rate is higher; for instance, at a load of 7N/mm, it is 4 or 5 x 10^{-9}mm/encounter compared to only 1 x 10^{-9} for nitrided steel gears. At the same load, the tooth contact stress using a plastic pinion is much less than when using a steel pinion; 35N/mm^2 compared to 200N/mm^2 with a load of 7N/mm.

Tests 21 to 24 show, perhaps unexpectedly, that oil-impregnated porous polyimide (Vespel SP8) actually has an inferior performance to the dry, unimpregnated plastic. The difference occurred during the running-in period (First 10^5 revs) when the wear rate of the oil-filled gear was high (Fig 20). Most of the wear debris, in the form of a plastic/oil mixture, remained in the region of contact and compacted in the roots of the teeth (Fig 21). This produced a period of rough running and accelerated wear. In the longer term (4 x 10^6 revs) the wear rates in the dry and lubricated cases were equal (4x10^{-9} mm/ encounter), a value similar to that found with the MoS_2-filled polyimide. However, unfilled polyimide has a rather high coefficient of friction (0.6) and the MoS_2 acts to reduce this rather than to reduce the wear.

4.3 Discussion and Ideas for Further Work

The tribological study has yielded useful data on friction and wear with different gear materials. Tooth wear rates, in terms of depth/encounter, can be used to estimate the increase in gear backlash under given conditions. Since the effect of load and speed is known, then some degree of extrapolation is possible but the performance under higher loads cannot be taken for granted. For instance, plastic gears are limited by the degree of tooth bending and, for maximum stiffness it is necessary to consider materials like the carbon-fibre reinforced polyacetal and the cotton-phenolics. To estimate likely wear rates it will, of course, be necessary to calculate the Hertzian contract stress between the gear teeth in question and to compare it with the data given for the ESTL test gears.

For nitrided gears, there will also be an upper load limit when there is a danger of the hard layer cracking as the softer substrate deforms. The present ESTL test rig is rather limited in load capacity and will be uprated to allow tests at higher loads. In this respect, the maraging steel offers a possible solution to very high load situations (eg, robotic arms). This steel responds well to plasma nitriding and can, therefore, give the desired combination of a hard, tough substrate with an even harder, wear resistant surface and the performance of gears made in this way will be studied at ESTL in the future. The low distortion is also an attractive feature.

In the four-square test arrangement there is no actual torque transmitted through the chain, the load being created by wending up the gears against themselves. The torque required to rotate the four gears is equal to the frictional losses at the two gear contacts plus the frictional loss in the support bearings. The torque figures given in tables 2 and 3 have been corrected for the bearing losses and can be used to calculate the power loss in the gears. Since the

TABLE 4 TRIBOLOGICAL BEHAVIOUR OF PLASTIC GEARS

Test No	Materials		Tooth load (N/mm)	Pinion Speed (rpm)	lubricant	Total Pinion Revs	Tooth flank wear rate (mm depth per encounter)	
	Pinion	Wheels					Pinion	Wheels
1	Carbon fibre inpolya-cetal	stainless steel	21# (60)	50	Dry	5×10^{4}	1×10^{-7}	Zero
2	"	"	21	50	Dry	3×10^{5}	7×10^{-8}	Zero
3	"	"	21	50	Dry	5×10^{5}	7×10^{-8}	Zero
4	"	"	21	100	Dry	7×10^{5}	2×10^{-8}	Zero
5	"	"	21	200	Dry	16×10^{5}	5×10^{-9}	Zero
6	"	"	21	500	Dry	30×10^{5}	6×10^{-9}	Zero
7	"	"	21	1000	Dry	60×10^{5}	5×10^{-9}	Zero
8	"	"	7(35)	1000	Dry	30×10^{5}	3.5×10^{-9}*	Zero
9	"	"	3(23)	1000	Dry	30×10^{5}	2×10^{-9}*	Zero
10	"	"	1(13)	1000	Dry	30×10^{5}	1×10^{-9}*	Zero
11	Vespel SP31	Stainless Steel	21 (50)	50	Dry	5×10^{4}	1×10^{-8}	Zero
12	"	"	21	50	Dry	3×10^{5}	7×10^{-9}	Zero
13	"	"	21	50	Dry	5×10^{5}	7×10^{-9}	Zero
14	"	"	21	100	Dry	7×10^{5}	9×10^{-9}	Zero
15	"	"	21	200	Dry	16×10^{5}	6×10^{-9}	Zero
16	"	"	21	500	Dry	30×10^{5}	5×10^{-9}	Zero
17	"	"	21	1000	Dry	60×10^{5}	4×10^{-9}	Zero
18	"	"	7(30)	1000	Dry	30×10^{5}	3×10^{-9}*	Zero
19	"	"	3(20)	1000	Dry	30×10^{5}	2×10^{-9}*	Zero
20	"	"	1(10)	1000	Dry	30×10^{5}	1×10^{-9}*	Zero
21	Vespel SP8	Stainless Steel	7	200	Dry	1×10^{5}	4×10^{-8}	Zero
22	"	"	7	200	Dry	40×10^{5}	4×10^{-9}	Zero
23	"	"	7	200	BP110	1×10^{5}	8×10^{-8}	Zero
24	"	"	7	200	BP110	40×10^{5}	4×10^{-9}	Zero

Key: *Steady state wear rate after running-in wear completed.
#Tooth contact stress in N/mm^2

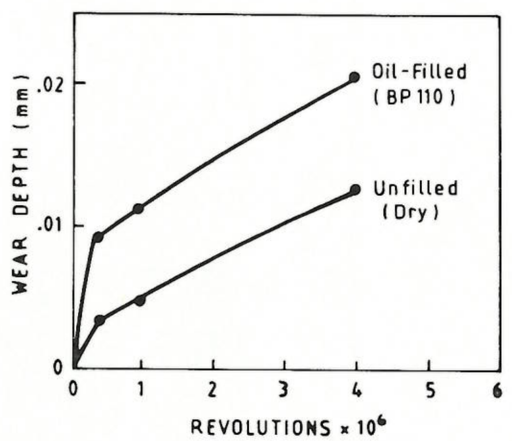
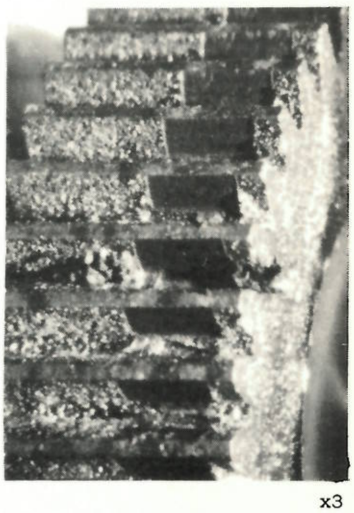


Figure 20 Wear of Oil-Filled Polyimide Gear



x3

Figure 21 Tooth Wear and Debris With Oil-Filled Polyimide Gear pinion radius is 15mm, the frictional loss in % is:

$$\frac{\frac{1}{2}\text{ (running torque in Nm)}}{15 \times 10^{-3}\text{ mm}} \times \frac{100\%}{\text{Tooth loaded in N}}$$

so that, for instance, the power loss with nitrided steel against AISI 440C is 2%, whereas, for lead lubricated nitrided steel against itself, it is only 0.5%.

In many respects, the work by Courtney (Ref 3) in the USA has useful parallels with the work at ESTL. He found that nitrided steel vs 440C gave low wear but he did not look in detail at a nitrided vs nitrided combination (on the basis that like - against - like is not usually a good tribological situation). The ESTL work has shown, in fact, that nitrided pairs can be run with low wear and that the fabrication process for the gears is easier than that required when working with through-hardened materials. In this respect, some useful experience has been gained; for instance, it has been found that a stress-relieving stage (at a temperature higher than the intended nitriding temperature) should be performed after the rough machining has been completed. This very much reduces the risk of distortion at the nitriding stage. It has been found that nitriding improves the corrosion resistance of the steel, an important fact if the gears are stored in air without lubrication prior to launch.

Courtney also found that light-anodised aluminium alloy gears (case depth 10 microns) failed by cracking. However, he also showed a much better performance with thick anodised coating (80 microns) and this could be a useful result to check in further ESTL tests. Equally, hardened titanium alloy (by plasma nitriding) is a possible solution for light weight system and this too will be evaluated.

With copper alloy gears, Courtney obtained a poor performance with phosphur bronze and Beryllium copper but, as yet, the ESTL testing has not covered these materials. It may be that aluminium bronze will give more wear resistance and it is hoped to evaluate this material in the future.

Finally, with gears operating in space, there is a problem of lubrication. Grease is a possible solution and the ESTL tests have shown that it can be effective. However, the long term problems of oil loss (creep, evaporation) remain and it is difficult to simulate this by experiment. Ultimately, the gears may run dry and the wear rate and torque under such conditions must be considered by the mechanism manufacturer as the 'worst case' and the design modified accordingly. Dry lubricatiuon is also possible and the ESTL tests have demonstrated the benefit of the in- house processes of ion-plated lead and sputter-coated MoS_2. However, the test periods have been relatively short and there remains a need to optimise the process and to assess their ultimate durability on gears. In addition, it is important to remember that neither lead nor MoS_2 perform well in air.

5. CONCLUSIONS

The main conclusions are:

1. For medium or heavy load applications plasma-nitrided steel gears are recommended. Without lubrication these gears gave a wear rate which was a factor of 10 lower than that obtained with through-hardened or precipitation hardened steel gears.

ii. For light duty, plastic gears can be used in combination with stainless steel gears. The lowest wear was obtained with MoS_2-filled polyimide but even this material gave a wear rate five times higher than that obtained with nitrided steel. Oil-filled porous polyimide gave a poor performance, the oily wear debris remaining trapped in the tooth contact area and increasing the friction.

iii. Aluminium alloy gears, either treated by the Ematal anodising process or by the H.E.F. zinal process, showed severe limits in their performance when operating against themselves. Failure was by cracking of the hard, thin surface coating. A greatly improved performance resulted when the frictional shear forces were reduced, either by using MoS_2 as a lubricant or by operating the aluminium gears against plastic gears.

iv. With both steel and plastic gears the wear rate was found to be proportional to $(\text{load})^{1/2}$ and to be imdependent of speed.

v. The wear rate of plasma nitrided gears was reduced by either lead or MoS_2 lubrication. The benefit was observed to last over at least 10^5 revs in the particular conditions of the ESTL tests; but some improvements from changes of in coating parameters could well be possible. Lubrication with grease reduced the wear to near zero but the long-term effectiveness of this method is difficult to assess because of lubricant loss by creep and evaporation.

vi. An electroless nickel/PTFE gave a very poor wear performance and is not recommended for space gears.

6. REFERENCES

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2. Dr A Thomas, 1982. The Friction and Wear Properties of some MoS_2 Spray Lubricants. ESA(ESTL) Report No 50.

3. Courtney W J, 1970. Evaluation of Wear Resistance of Fourteen Gear Materials, Final Report, NASA Contract No NASA-9590.

4. Lancaster J K. Estimation of the Limiting PV Relationship for Thermoplastic Bearing Materials.

DISCUSSIONS

E. Condé (CNES):

Have you made the same studies with two normal ball-bearings, same dimension but with angular contact?

R.A. Rowntree (ESTL):

If you are referring to thin-section bearings, no! However at ESTL we have extensive experience of stiffness, torque and conductance of normal section bearings.

R. Birner (MBB):

Could you comment on pre-rolling phase torque characteristics (i.e. characteristic angle and equivalent stiffness) of these bearings as compared to standard bearings?

R.A. Rowntree:

*I have not measured the pre-rolling characteristics of these thin-section bearings. I would expect however that as for conventional ball-bearings, equilibrium rolling would commence at an angle equivalent to a distance of 1.5*a*, where 'a' is the semi-minor axis of the Hertz ellipse of contact.*

M. Maillat (LSRH):

Could you give us an idea about machining and mounting of the 4-point contact bearings? Are they 2-hole bearings?

R.A. Rowntree:

The bearings were supplied by the Kaydon Corporation, Michigan. Each race is a single component, they are not split. The bearing housings and seatings of the experimental rings were machined to ABEC-9 precision.

P.J. Whiteman (MSDS):

Is there an optimum cross section for the 'L' section clamping ring, and optimum number of clamping bolts for any size of 4-point contact bearing?

R.A. Rowntree:

For the ESTL design of clamp ring (made before the effect of axial clamping was known) reducing the section in contact with the bearing will lower bearing sensitivity to axial clamping. Within reasonable limits the number of clamp ring bolts should not matter, it's the axial clamping force on the bearing race which is more important.

H. Ludwig (Hughes):

You stated that all ball spin is on one race in your analysis. Later in your presentation you compared your analysis to the 4-point contact data. If you used all or most balls in contact, the data did not compare. One of the reasons may be that in the bearing all four contacts are spinning.

In order to make a fit, 25% of the balls were assumed to be in contact. These grade and bearings would have high-grade balls (∿5 or 10). With this high grade of balls it is doubtful that only 25% of the balls would be loaded. It is suggested that another explanation is in order.

M.J. Todd (ESTL):

Because of space allocation, the analysis in the paper was confined to ordinary angular contact bearings but of course where there are 4-contact points per ball then spin/roll occurs in general at all contacts. Our computer software includes appropiate changes to handle 4-point cases.

Even with small very high-precision bearings, it is our experience (and that of the ball bearing manufacturers) that, under light or moderate loading, you cannot assume that the full complement is load-carrying at any time. Certainly in large diameter, thin-ring bearings with large ball complements and under light/moderate loads it is easily demonstrated that a very significant number of balls is playing no part in load support. I would cite here the Kaydon 4-point bearing of 7 in ID and having 87 balls (which we used for conductance/torque measurements). Though this bearing was of the highest precision class made by Kaydon and was carefully mounted on seatings of well within ABEC-9 standard, it was found that, even under the highest axial clamping force, only an estimated 20-30 of the balls were in contact with both raceways.

A more detailed dry torque theory would require the use of statistical data on ball diameter variance, raceway form errors and surface finish in order to provide a contact load/area distribution for the ball set from which torque could then be computed. However, the effort required to do this seems not worthwhile since one is usually interested in only rough estimation. For more critical cases or when unconventional bearings like the 4-point were being considered then we would advise torque measurement under known load/thermal conditions.

By normalising one measured value to theory one could, as I have shown in the paper, have good confidence in the prediction of torque under other thermal states.

G. Wallis (Sodern):

What is the global accuracy of the torque prediction computation? How is it sensitive to parameters not well known (for instance friction coefficient or BB conformity)?

M.J. Todd:

Because of the natural variation and mean sliding friction coefficient for any dry or boundary lubricant, because of the uncertainty in how many balls are load-supporting and, of course, because the theory assumes catalogue bearing conditions then the computation of torque must be regarded as a guide only.

In certain favourable cases involving high-precision bearings with small ball complements and thin film lubricants of known (measured) friction coefficients we have found that our theory agrees to within 10-20% of the measurements.

For large, flexible ring bearings like the four-point contact type which contain many balls, the majority of which do not carry load, then the theory is less accurate primarily for that reason.

However, by making use of the 'rule of thumb' I mention in the paper then a modest accuracy (of the order of 30-40%) can be retained.

R.H. Bentall (ESA):

Do you think that the primary cause of the degradation observed with the Fomblin oil is due to the loss of oil from the bearing and the chemical degradation consequently a symptom of that loss - rather than the reverse?

If this is so is it reasonable to search for a solution by maintaining the oil in the bearing, by geometry or anticreep barriers and repeating the tests?

K. Stevens (ESTL):

I think the phenomenon of the drop in DC torque (loss of oil from the ball/race contacts) and that of the degradation are separate. The drop in torque occurs only at high speed and is, therefore, probably related to some odd rheological property of the oil. At low speeds, the DC torque remains relatively steady but, because the lubrication regime is boundary, the ball/race contacts give rise to active surfaces which then polymerise the oil.

If, at high speed, some way could be found to keep the oil in the ball/race contacts (thereby maintaining EHD) the source of degradation would be removed. It is difficult to suggest a way of doing this without knowing the rheological properties of the oil.

H. Ludwig:

I found this paper very interesting. I would like to make a few comments. First, most gear systems are stepper motor driven. It would be better to use a stepper drive in your test. Secondly, you stated that liquid lubricated gearboxes are not practical for space use. The TDRS and Space Telescope spacecraft have many liquid-lubricated gearboxes using harmonic drives and Bray 3L38RP. Finally, it would be of great interest to compare the results of your best candidates to the currently used gear materials and lubricants.

K. Stevens:

If the stepper motor imposes extra, impulsive loads on the teeth, I agree that it might be important. My feeling is that such loadings would be low and have minimal effect on wear. A stepping drive might well add sufficient rapid cycling of the load to produce fatigue-like failure - particularly with thin anodised AI Alloy.

The Bray 3L38RP is a grease - I suggested that oil lubrication would appear to be impractical.

H. Boving (LSRH):

We at LSRH strongly believe, as the author, that thin ceramic type coatings like TiC a TiN on the bearing races and/or balls, to separate the metallic components from coming in real contact, increase the oil perofrmance and lifetime. In fact, tests run together by LSRH and RMB, with bearings using CVD-obtained TiC coatings on the races or on the balls (24000 rpm, air environment, hydrocarbon-type oil) the oil lifetime increase was tremendous, compared to standard uncoated bearings. The interested reader will find more about these tests under the following refs: *Lub. Eng.*, Vol. 37, 9, 1981, 534-537 and Vol. 39, 4, 1983, 209-215.

The bearing tests you described were performed under a preload of 40 N, which is rather low. What will be, according to you, the oil behaviour when the preload is 300 or 400 N?

K.T. Stevens:

Since the EHD film thickness is insensitive to load I would not expect the level of loading to be a critical test parameter. However, under boundary lubrication conditions (say 100 rpm), there is no doubt that a higher ball/race load will produce more steel wear (micro-pitting, in my experience). This would produce more oil degradation.

J.J. Capart (ESA):

With the very conclusive evidence provided by Mr. Baxter and yourself, is it not better to invest in a search for a suitable alternative, possibly with a slightly less outstanding vapour pressure?

K.T. Stevens:

The Fomblin Y Series Oil, 'YVAC 40/11' would be my next choice. The viscosity at 20°C is 400CS and the vapour pressure is 10^{-11} Torr (not as good as Z25). However, the viscosity index of the Y40 is low, so it does not have the advantages of Z25.

In similar ball bearing tests the Y40 oil performed well with no sign of degradation or wear and it would be my next choice for a space lubricant now that the highly refined mineral oil, BP110, is no longer produced.