University of
HUDDERSFIELD

AN INVESTIGATION INTO THE ANTI-WEAR CAPABILITES OF LOFRIX

Research conducted at the University of Huddersfield by The Centre for Efficiency and Performance Engineering

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Authorised for release on behalf of the University of Huddersfield

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1. INTRODUCTION

This research project was undertaken to determine under controlled conditions the efficacy of the Lofrix friction and wear reduction oil additive. A number of industry based case studies have been written about the beneficial effects of Lofrix. The claimed benefits (http://www.lofrix.com/case-studies/index.html) include, extended oil life, reductions in oil temperature, reductions in noise levels and reductions in energy consumption. When a product such as Lofrix is evaluated in operational conditions there are many influencing variables at work, which can result in overstating or understating any benefits. Therefore, in order to obtain a controlled empirical view of the impact of Lofrix it was decided to conduct a formal test in a laboratory where fewer influencing variables exist. The methodology to be applied would include the use of a Friction Brake Test Machine.

2. METHODOLOGY

The methodology was to repeat three times a 60 minute test with different amounts of Lofrix mixed with a base oil. Then to compare the data generated against the base oil without any Lofrix added.

The data recorded would consist of noting the amount of time in seconds that it took for fluid film lubrication at a surface contact area to break down; identified by an increase in noise and a momentary increase in the amperage being consumed by the Friction Brake Test Machine. Although somewhat subjective the increase in noise associated with lubrication failure is very noticeable. Time was recorded using a stopwatch and the amperage was measured using a 'Plug-In Power and Energy Monitor'. The momentary increase in amperage occurred in a matter of seconds and the focus was to measure time and then amperage, so caution must be exercised in respect of the amperage data.

A measured amount (60 millilitres) of oil was placed in the oil bath (Figure 1) in which the bottom of the spindle was immersed, this oil then lubricates the contact surface between the spindle and the roller.

Figure 1 Oil Bath

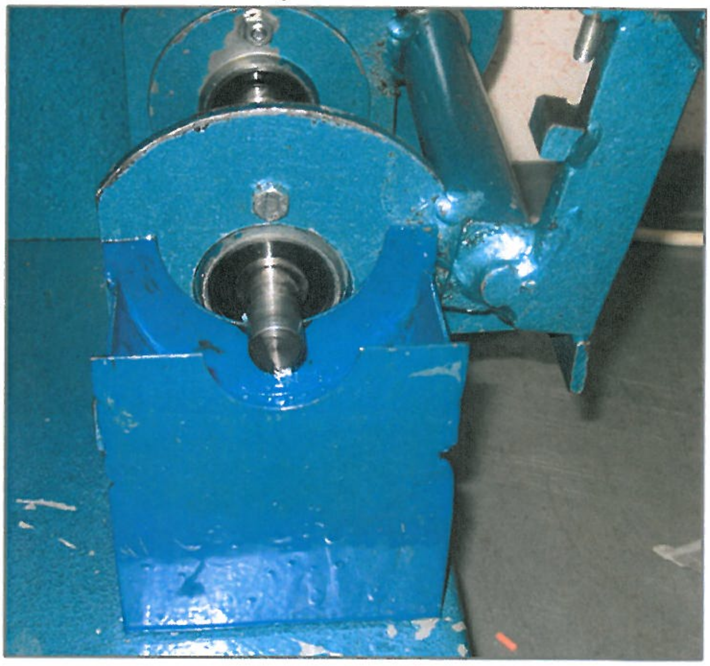
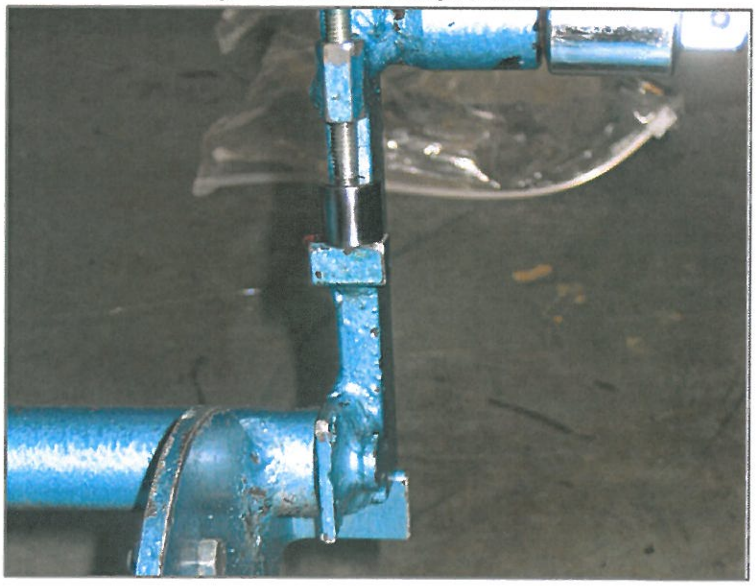


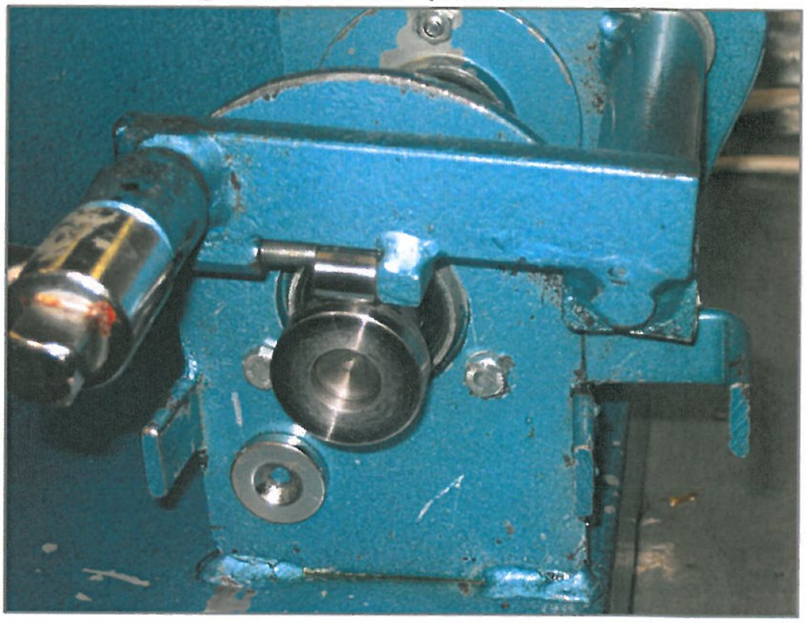
Figure 1 shows the oil bath and drive shaft, prior to the spindle being fitted to the drive shaft and the oil being placed into the oil bath. In the top right hand corner of Figure 1 the clamp that holds the roller in position can be seen. In Figure 2 a roller can be seen clamped into position.

Figure 2 Roller clamped to arm



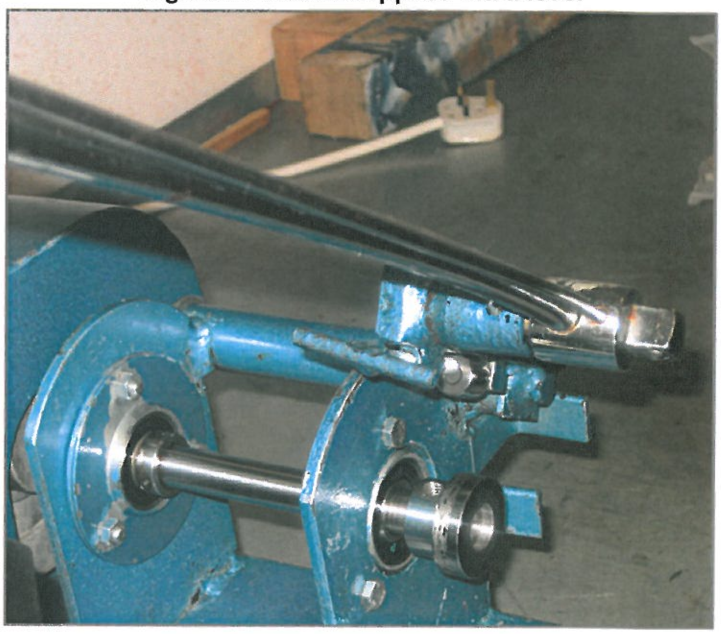
This roller is placed under a constant load, which exerts pressure at the contact surface. Initially the contact surface is very small, but as wear takes place the contact surface area increases as the surface of the roller becomes worn. The average pressure of 6.0958 Tonnes/cm^2 exerted by the constant load has been calculated using a Hertzian Stress Calculation, which is shown in Appendix 1 and which clearly represents very harsh wear conditions. Figure 3 shows the spindle and roller coming into contact. The oil bath has been removed for picture clarity.

Figure 3 Roller and spindle in contact



Pressure is applied to the roller via a lever mechanism with a weight attached to the end. In this case the weight was six kilograms. The lever was a torque wrench with a square socket drive attaching it to the section of metal that clamped the roller in position, which is shown in Figures 3 and 4.

Figure 4 Pressure applied via a lever



The tests were performed using Millers Millmax 68 Hydraulic Oil as the base oil and Lofrix was added to fresh base oil in solutions made up in weight by weight concentrations of 0%, 1%, 2%, and 3%. The oil bath was cleaned after each test run.

The spindles were made from EN8 steel and the spindles were rotated at 720 revolutions per minute (RPM). The rollers were made from EN8 and were case hardened. The shaft was driven by an electric motor connected via a tensioned 'V' belt.

The tests were each run for 60 minute periods and then stopped.

3. TEST DATA

The data from the tests is shown in Table 1 under two headings. The first is the amperage that was recorded on or around the time that the oil broke down. The second is the time in seconds from the start of a test to the time that the oil broke down.

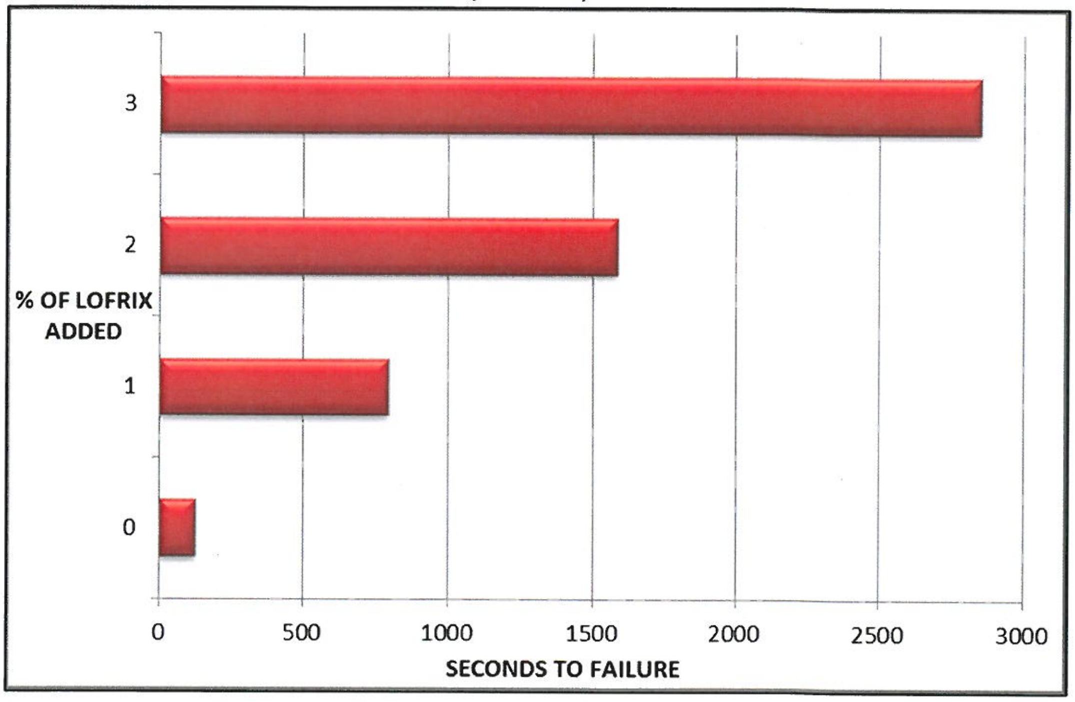
Table 1 Test data

0% Lofrix	Test 1	Test 2	Test 3	Average
Amps	3.67	3.65	3.55	3.62
Time to noise (s)	134	115	130	126
1% Lofrix	Test 1	Test 2	Test 3	
Amps	3.01	2.99	3.05	3.02
Time to noise (s)	690	813	885	796
2% Lofrix	Test 1	Test 2	Test 3	
Amps	2.91	2.96	2.93	2.93
Time to noise (s)	1,470	1,603	1,690	1,588
3% Lofrix	Test 1	Test 2	Test 3	
Amps	2.86	2.91	2.90	2.89
Time to noise (s)	2,880	2,760	2,910	2,850

Analysis of the data in Table 1 indicates a reduction in amperage and a significant increase in the time to failure (i.e. when the oil breaks down). The increase in time to the oil breaking down is quite clear and significant. Adding 1% Lofrix to the base oil increased the time to oil breakdown by a factor of 6.3 that is the oil lasted 6.3 times longer before failing. Adding 2% Lofrix to the base oil further increased the time to oil breakdown to 1,588 seconds, thereby increasing the life of the oil by a factor of 12.6. Finally, when the Lofrix content is increased to 3% the time to oil breakdown increases to 2,850 seconds; an increase to a factor of 22.6.

Clearly these are significant increases in the ability of the oil to handle the pressure exerted by the lever effect. It can be seen in Chart 1 that after the initial large saving achieved by the addition of 1% of Lofrix thereafter every additional 1% has approximately doubled the time to failure.

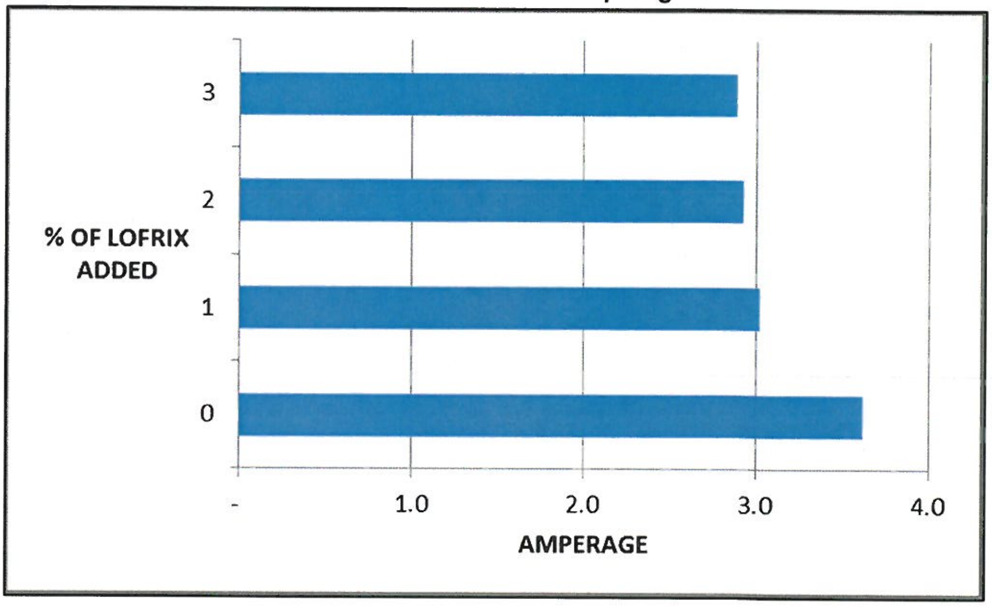
Chart 1 Time (seconds) to failure of oil



The reductions in amperage are indicative of energy savings with the caveat that because the increase was only momentary – when the oil breakdown actually happened – the accuracy of the amperage data has to be treated with caution. However, taking this caveat into account there is clearly an emerging trend of a reduction in amperage. This is an area that needs to be considered for further exhaustive research. Similarly as shown in Chart 1 it can be seen in Chart 2 that after the initial large saving when 1% Lofrix is added the saving reduces. The initial reduction in amperage is of the order of 16.5% and then the rate of change (improvement) reduces by 2.5% at 2% and then to 1.1% at 3% of Lofrix.

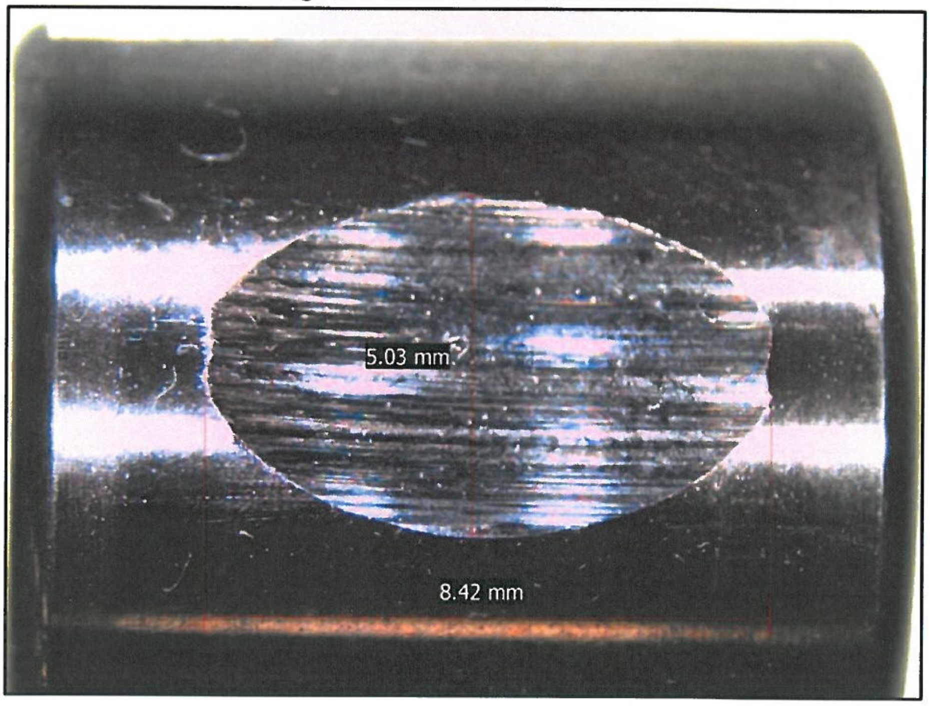
The amperage and time to failure as indicated by the 'time to noise' data recorded in the twelve tests and shown in Table 1 was found to have a fairly strong correlation of - 0.781. Even stronger correlation was found between the amount of Lofrix mixed with the base oil and amperage at -0.856 and the time to noise at 0.986.

Chart 2 Reduction in amperage



A clear visual indication of the impact of Lofrix on the base oil can be seen by examining the roller bearings. Figure 5 shows the length and width of the wear scar when the oil had no Lofrix added.

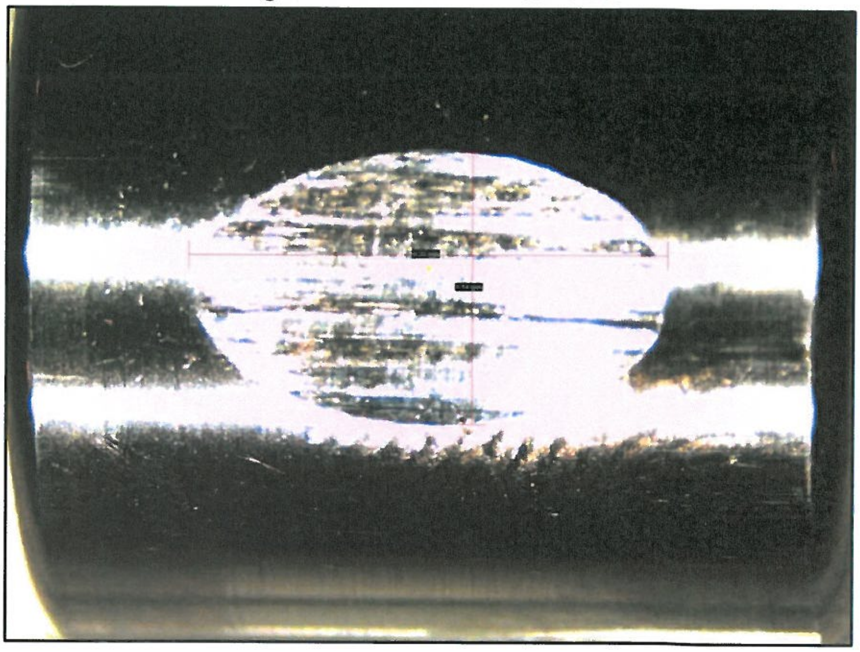
Figure 5 Wear scar at 0% Lofrix



The scar is 8.42 millimetres in length and 5.03 millimetres in width.

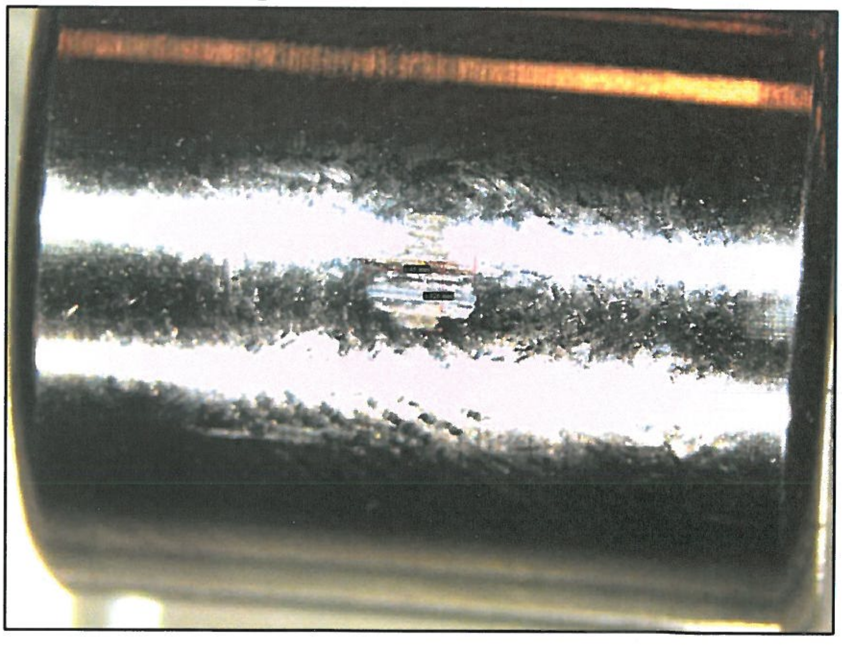
The sooner the oil breaks down and loses its ability to effectively lubricate the two surfaces in contact, the greater the wear that takes place, because the test is running to a fixed time interval. This can be seen quite clearly when comparing the sizes of the wear scars on the rollers. Figure 6 below shows a marked reduction in the size of the scar.

Figure 6 Wear scar at 1% Lofrix



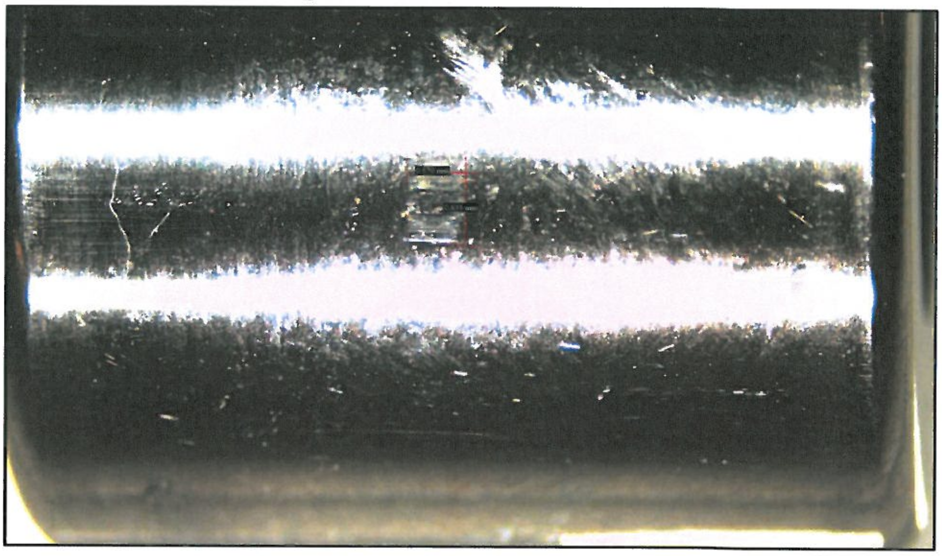
The scar is 6.20 millimetres in length and 3.54 millimetres in width. As the amount of Lofrix added to the base oil increases further reductions in the size of the wear scar can be seen (Figures 7 & 8).

Figure 7 Wear scar at 2% Lofrix



With 2% Lofrix in the mix the wear scar has reduced to 1.45 millimetres in length and less than one millimetre in width (0.926 of a millimetre). This improvement continues as the amount of Lofrix is increased by another percentage point to 3% as shown in Figure 8.

Figure 8 Wear scar at 3% Lofrix



Both the length and width of the wear scar are now less than one millimetre, which is a substantial difference when compared to having no Lofrix added to the base oil. These measurements are compiled in Table 2.

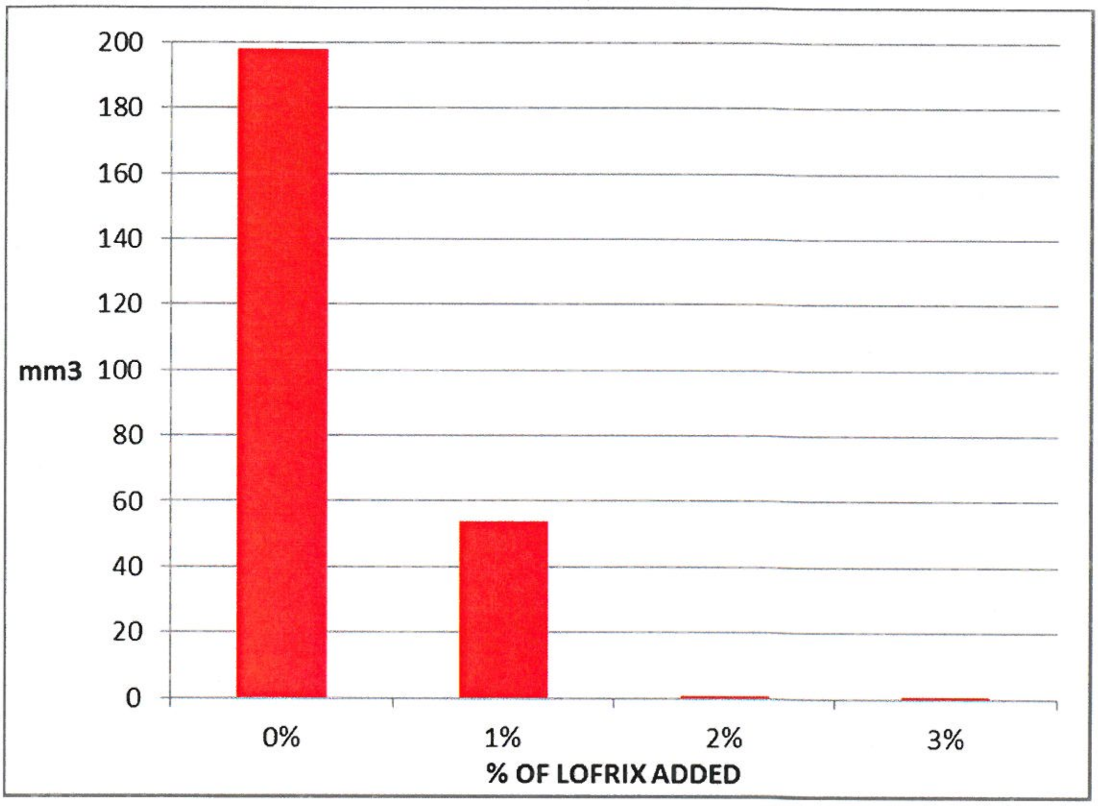
Table 2 Compilation of wear scar data

CONDITION	LENGTH	WIDTH
0% Lofrix	8.42	5.03
1% Lofrix	6.20	3.54
2% Lofrix	1.45	0.926
3% Lofrix	0.670	0.835

There are certainly clear reductions in the size of the wear scar, which of course is influenced by the time it took for the oil to breakdown. What this clearly indicates in this investigation is that the addition of Lofrix to the base oil has had a positive impact. The damage caused following the breakdown of the lubricating oil has been significantly reduced as shown by the reduction in the length and width of the scar. With three percent of Lofrix added to the base oil the length of the scar has been reduced to 0.67 of a millimetre – 92% reduction in scar length. At the same strength the width of the scar has been reduced to 0.835 of a millimetre, which equates to an 83.4% reduction in scar width.

Whilst the associated data and pictures of the wear scars provide some indication of the impact of adding Lofrix the final analysis compares the volume of material removed from the roller. The calculated figures represented in Chart 3, are very impressive and the calculations are shown in Appendix B.

Chart 3 Calculated wear volume mm^3



There has obviously been a major reduction in the amount of material removed.

4. CONCLUSION

This evaluation has found that under very harsh wear conditions adding Lofrix to a base oil has delayed the oil from breaking down and therefore causing wear on the contact surfaces. The gradual increase in the amount of Lofrix added to the base oil has seen the delay in the oil breaking down increase. This has had a major impact on the size of the scar on the roller after a fixed period of testing and reinforces the hypothesis that adding Lofrix to the base oil increases its ability to substantially reduce wear.

Furthermore, whilst caution must be exercised in respect of the amperage data there appears to be an emerging trend of reduction in amperage. This

could be indicative of a reduction in machine energy consumption with associated reductions in energy costs and emissions.
The results from this initial research project do support the findings reported in the industrial case studies and strengthen the case for further long term research.

5. RECOMMENDATIONS AND AREAS FOR FURTHER RESEARCH

Key areas for further research that have been identified include accurately and continually recording amperage during repeat tests. This will help to determine any improvements in energy consumption. Additional validation with other engineering materials will reinforce the results found in this evaluation and could identify any sensitivities involved. For example, would harder material benefit more? Similarly, testing with different oils will provide a more in-depth understanding of the benefits of Lofrix, such as being able to use lighter and less expensive oils.

Co-efficient of friction tests should also be incorporated into further research to establish the exact impact of Lofrix upon the co-efficient of friction. This will provide quantitative evidence of reductions in friction. A reduction in friction should also create less heat and monitoring component and oil temperature will provide additional data to inform on the impact of Lofrix upon any base oil.

A further variable to explore is the applied loads. Is Lofrix more or less effective under low, medium and harsh wear conditions? Finally, damage to a person's hearing is an industrial injury and noise reduction strategies are a cost to industry. Reductions in noise levels have been reported by users and future tests should also consider incorporating the use of noise or acoustic sensors.

6. APPENDIX A: HERTZIAN STRESS CALCULATION

L1=545	lever length or distance between load and pin (mm)
L2=60	distance between bearing load and pin (mm)
W=6*9.81	loading (N)
F=(W*L1)/L2	force at roller(N)
R1=7	radius of roller (mm)
R2=17.5	radius of rotor (mm)
Re=(R1*R2)/(R1+R2)	reduced radius of curvature (mm)
E1=2.1*10^5	MPa (N/mm2)
E2=2*10^5	MPa (N/mm2)
v1=0.3	poisson's ratio 1
v2=0.3	poisson's ratio 2
Ee=2*E1*E2/(E2+(1-v1^2)+E1*(1-v2^2))	Modulus elasticity reduced(MPa orN/mm2)
t=90	contact angle

ko=((1/R1)^2+(1/R2)^2+2*(1/R1)*(1/R2)*cos(2*t))^0.5/(1/R1+1/R2);

k1=1.7	from chart
k2=0.7	from chart
a=k1*(3*F*Re/Ee)^(1/3);	
b=k2*(3*F*Re/Ee)^(1/3);	
pmax=3*F/(2*pi*a*b)	N/mm^2
pav=F/(pi*a*b)	N/mm^2

pmax = 897.0006 N/mm^2

pav = 598.0004 N/mm^2

pmax=3*F*100/(9.81*1000*(2*pi*a*b));%Ton/cm^2

pav=F*100/(9.81*1000*(pi*a*b));%Ton/cm^2

pmax =9.1437 Ton/cm^2

pav = 6.0958 Ton/cm^2

7. APPENDIX B: CALCULATION OF WEAR VOLUME

Since only one cylinder suffers wear, we need to calculate only half the ellipsoid volume.

$$V_{wear} = 1/2\left(\frac{4}{3}\pi abc\right)$$

Where *a* and *b* can be measured from wear scar using a calibrated microscope, and

$$c = R - \cos(\theta)\,R$$

Where $\theta = \mathrm{Sin}^{-1}(a/R)$

Table 3 Volume removed calculation

% LoFrix	0	0.01	0.02	0.03	
a	8.42	6.2	1.45	0.67	mm
b	5.03	3.54	0.926	0.835	mm
R	17	17	17	17	mm
theta	0.518	0.373	0.085	0.039	deg
c	2.232	1.171	0.062	0.013	mm
Wear vol (mm^3)	197.956	53.824	0.174	0.015	mm^3