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Effects of Bearing Cleaning and Lube Environment on Bearing Performance

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Abstract

Running torque data of SR6 ball bearings are presented for different temperatures and speeds. The data are discussed in contrast to generally used torque prediction models and points out the need to obtain empirical data in critical applications.

Also, the effects of changing bearing washing techniques from old, universally used CFC-based systems to CFC-free aqueous/alkaline solutions are discussed. Data on wettability, torque and lubricant life using SR3 ball bearings are presented. In general, performance is improved using the new aqueous washing techniques.

Introduction

Torque prediction has been accomplished over the years using various models, followed by actual data at as close to application environments as possible. In space applications, this prediction becomes very critical.

Torque models don't vary that much. They include various combinations of three terms. First, a term for the mechanical drag torque, second, a term for retainer drag, and finally and most important, a term for viscous lube effects [1, 2, 3]. A typical model used in the bearing industry is equation 1.

$$T_t = \frac{\mu\, T^{4/3}\, d^{1/3}}{n^{1/3}\, \sin^{1/3} B_t} \frac{\left(1 - \partial^2\right)^{1/3}}{E^{1/3}} \left(K_t + \frac{P \bullet K_r}{d}\right) + S + L \qquad (1)$$

where μ = friction coefficient (0.1 typical), T = thrust load, d = ball diameter, n = number of balls, B_t = contact angle, E = modulus of elasticity, ∂ = Poisson's ratio, K_t and K_r are contact ellipse functions ($K_t = 1.3$, $K_r = 0.11$ for 52% curvatures, P = pitch diameter,
$S = 15000\, d^{2/3}/p^2$, and

$$L = 6800\, P^{5/3}\, d^{4/3}\, n^{2/3} \left(1 - \frac{d \bullet \cos B_t}{P}\right)^{2/3} \bullet (\text{inner ring speed})^{2/3} \bullet Q \bullet \left(\frac{\text{viscosity} \bullet \text{specific grav}}{100}\right)^{2/3}$$

The whole first term, the mechanical drag, is friction coefficient times thrust load times area. The second term, "S", is the retainer term, a function of ball size and pitch diameter. Finally, the term, "L", the viscous drag term is made up of bearing speed to the $^2/3$ power, a lube quantity factor (Q) that equals one except when lube is starved (at which time it equals 0.4), and a lubricant property term that multiplies viscosity times specific gravity. In general, this model is dominated by the viscous drag term,

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especially at higher speeds and higher viscosities caused by lower operating temperatures.

In Figure 1, the torque curves versus speed for the test bearing, SR6RHH7, are plotted for some of the lubricants tested in this work. It can be seen that increased torque follows increased viscosity and specific gravity, the dynamic viscosity, and then gets multiplied by speed. The nature of this investigation was to see how good these models are in light of the critical nature of space mechanisms.

In the second part of this paper, data on ball bearing performance such as low speed torque, steel wettability, and lubricant life in actual ball bearings are presented where the test variable is final assembly wash before applying lubricant. MPB's old Freon / aqueous / Freon dry system was compared to our new aqueous ultrasonic, rinse, air dry system.

Test Procedures and Results

Grease Testing

Actual torque results of the SR6RHH7 ball bearings were obtained using a variable-speed torque tester based on the same principles as the MPB RT2C, MIL-STD-206 running torque tester. The mechanical elements of the tester were put in an environmental chamber to obtain the temperature results. All bearings were run in for a pre-determined time and tests were repeated to ensure and demonstrate repeatability.

Bearings were lubed with the test greases to $^1/_3$ full. Speed was varied from 1000 to 5000 rpm and a constant 35.5 N (8 lb) axial load was used. Bearings were tested at 24°C (75°F), 1.7°C (35°F), -17.8°C (0°F), and -29°C (-20°F). The greases reported on here are:

Halocarbon 25-10M ----	polychlorotrifluoroethylene
Mobil 28 --------------------	clay thickened synthetic hydrocarbon base
Rheolube 2000 -----------	organic gel thickened synthetic hydrocarbon
Aeroshell 5 -----------------	clay thickened petroleum base
Mobilith SHC220 ---------	lithium soap thickened synthetic base
Braycote 803 RP ----------	perfluoroether

The results of the room temperature tests versus speed are shown in Figure 2. The torque curves are all under the model prediction and are quite independent of speed in this range. Only the heavy, low-viscosity Halocarbon reverses the trend.

The Mobil 28 and Rheolube 2000 appear close to predicted but are flat with respect to speed. It is possible the shearing of the lubricant is not so prevalent as predicted. It is also possible that another effect, not modeled, is lowering the torque with increasing speed, counteracting the lube shearing effect.

Figure 3 is a collection of traces from Mobil 28 greased bearings to illustrate bearing to bearing consistency. Also the independence of torque versus speed is quite graphic.

The bearings were run in for a minimum of 30 minutes eliminating any beginning grease distribution differences.

In Figure 4, the Mobil 28 viscosity change with temperature is put into the model, resulting in dramatic increases in torque predicted with lower temperature and higher speed. In actual tests, the second graph in Figure 4, the data do show an increase in torque at 1000 rpm due to the test temperature induced viscosity change. As speed is increased, however, torque is not affected and went down at 1.7°C (35°F) as speed increased. This poses a question, "Is the extra shearing force being offset by other mechanical effects, such as lube availability in the contact zones going to marginal, i.e., less lube sheared?" The torque traces in Figure 5 show this trend.

Also in Figure 4, the Rheolube 2000 data show similar results. It follows the model, increasing at 1000 rpm as the test temperature is decreased. Also, speed effects are negated or masked by unmodeled phenomena. Another interesting observation, shown in Figure 6, is the occasional chatter or hash, which is not predictable in a model.

The torque of Mobilith SHC220 grease is also shown in Figure 4. In this case, almost no sensitivity to speed and test temperature was found. It seems the dynamic character of a grease is not all in its viscosity and density. The nature of additives, blending and other factors are not modeled. Figure 7 contains the actual torque traces. The test was also run at -45°C (-50°F) where only torque hash was encountered. This grease was the best tested under the conditions imposed.

Finally, the base oil of Mobil 28 was tested under the same conditions as the greases to observe the effect of the thickener and additives. The amount of oil was obtained using a standard practice of flooding, then a centrifuge at 800 G's for 3 minutes. Figure 4 illustrates the torque is very similar to the grease. Other experience has shown that smaller instrument ball bearings or thin section bearings with small balls do show an increase in torque hash with grease versus oil.

<u>Washing</u>

In the washing experiments, a smaller, SR3R ball bearing was used. All bearing components were washed during manufacturing using distilled mineral spirits. These components were assembled into bearings, split into groups and assembly washed appropriately. The old system was Freon ultrasonic, aqueous ultrasonic, and Freon-based drying. The new system is aqueous ultrasonic, rinses, and air drying.

In the goniometer disc wettability tests, 440C stainless steel discs were washed in hexane after surface preparation. Then the discs were processed through the standard assembly wash that utilizes freon products or through the new assembly system that is aqueous-cleaning based. The standard oil drop angle was measured after fifteen minutes and after 24 hours. The old freon process gave oil drop angle of 8 to 9° while the CFC-free washed discs were at 4 to 6°; the new wash procedure is more wettable. These results did not change after 24 hours.

The ball bearing life test uses the following parameters and log normally distributed statistical analysis was applied to 37 freon-washed failures and 10 aqueous-washed failures.

440C rings and balls
stainless ribbon retainer
lubrication: 1000/1 hexane and Mil-L-6085 oil, starved or parched lube running conditions
test speed: 12,000 rpm
test load: 13 N (3 lb) axial
Failure mode -- lubricant breakdown quantified by bearing cartridge accelerometer limit

Figure 8 represents the weibull data of the ball bearing life test. The circle data points represent the CFC-washed bearings and the triangles are the CFC-free washed bearings.

This test is an accelerated lubricant life tester for instrument bearings. Instrument bearings are usually lightly loaded such that lubricant breakdown or polymerization is the failure mode. The resulting associated torque increase causes bearing performance degradation. This tester is designed to accelerate that condition by limiting the available lubricant to just a thin, non-replenishable, partial EHD film.

The data show a L50 life of 4.74 hours for the new CFC-free washing system versus 2.25 hours for the old Freon-based system. Statistics say the data supports a conclusion of difference, that is, the new wash is better than the old wash at the 99 % confidence level.

The running torque tests were performed to MIL-STD-206 requirements on SR3R stainless bearings washed both ways. The tester used is manufactured by MPB and is an industry standard that measures torque at 2 rpm under a standard light load, using a transducer. The specimens were run lubed with a standard oil and also dry to pick up subtle effects. Results were statistically compared.

In the torque tests, four groups of ten bearings were torque tested. They were:

CFC washed, oil lubed
CFC washed, dry
CFC free (aqueous) washed, oil lubed
CFC free (aqueous) washed, dry

The first three groups showed no significant difference in the running torque levels and all averaged about 5000 µg•m (Figure 9). The fourth group, the aqueous washed, dry, was significantly higher with 99% confidence. In fact, the repeatability was also bad in this group as evidenced by a sample trace at the bottom of Figure 9. Here it is obvious the bearing is hanging up. Normally, one would not test bearings dry and it is not per the MIL STD to do so, but our experienced bearing people have historically always used the "dry" mode for analytical reasons. Now, in this case, it seems the extra clean nature of the metal parts does not allow this.

Silicon nitride balls are quite popular in instrument bearings for their lubricant extending ability. It is reported that the dissimilarity of the ball and race materials inhibits interactions that cause lubricant breakdown [4, 5]. Some customers have asked if running silicon nitride ball hybrids without lubricant is possible and most suppliers have been reluctant to approve this.

This thought about the dry nature of CFC-free (aqueous) washed parts caused the following observation. Figure 10 shows the condition of ceramic balls in an SR3 full complement (no retainer) bearing after a running torque test at 2 rpm in the dry condition. The balls have scratched themselves while rubbing against each other. Further tests have proved that the washing in the old freon-based wash system eliminated the scratches, leading to speculation that there is a slight oil residue after freon washing. Introduction of a retainer, separating the balls, stops the scratching.

Conclusions

Grease Testing

1. In torque models, the lube term is dominant. For example, Mobil 28 at 1000 rpm at room temperature is 3.7 g•cm due to mechanical drag, 0.17 g•cm due to retainer drag, and 6.25 g•cm due to the lube term. And as speed increases or temperature decreases, it gets even more dominant. In the moderate speed and temperature range investigated here, this dominance is not evident.

2. Some greases are inherently more independent of speed and temperature changes than others of comparable viscosities and other modelable properties. In this testing, Mobilith SC 220 was the most independent.

3. This work will lead to an investigation of retainer type and grease amount in an effort to understand if either of these variables are affecting or negating the viscosity term.

Bearing washing performance

1. Based on the consistent results of goniometer angle, life tests, and torque tests, it is felt that the new aqueous-based assembly washing systems do not cause any degradation in bearing performance and may lead to enhanced lubricant life in some situations.

2. Torque perturbations and scratching of silicon nitride balls in the dry, full complement bearing condition suggests that aqueous washing may be eliminating a minor or thin film of some nature left behind by the old Freon-based cleaning system. The performance testing still indicates that this film was not beneficial to normally lubricated bearing performance.

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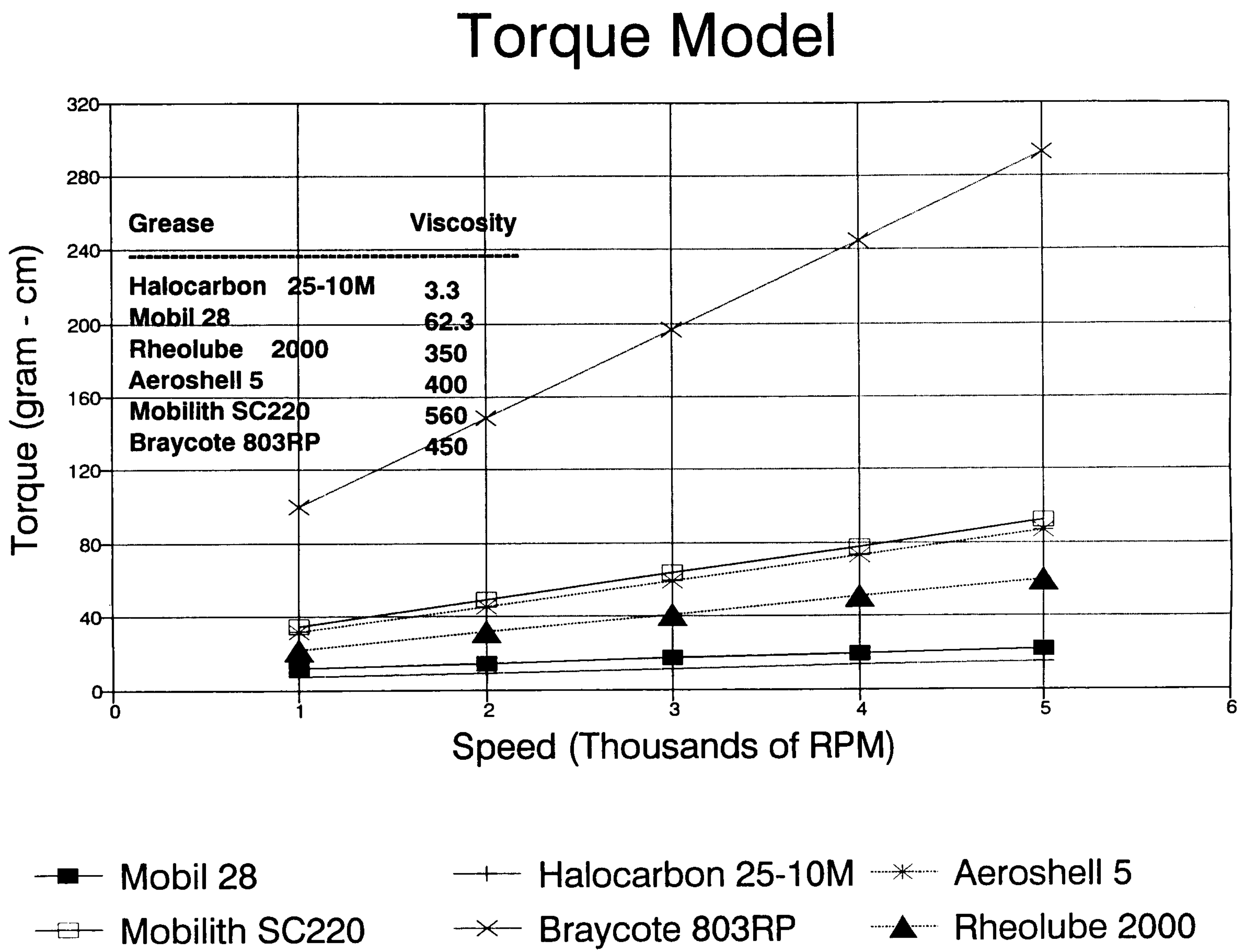
Torque Model
Torque (gram - cm)
Speed (Thousands of RPM)
320
280
240
200
160
120
80
40
0
1
2
3
4
5
6
Grease
Viscosity
Halocarbon 25-10M 3.3
Mobil 28 62.3
Rheolube 2000 350
Aeroshell 5 400
Mobilith SC220 560
Braycote 803RP 450
Mobil 28
Halocarbon 25-10M
Aeroshell 5
Mobilith SC220
Braycote 803RP
Rheolube 2000

Figure 1

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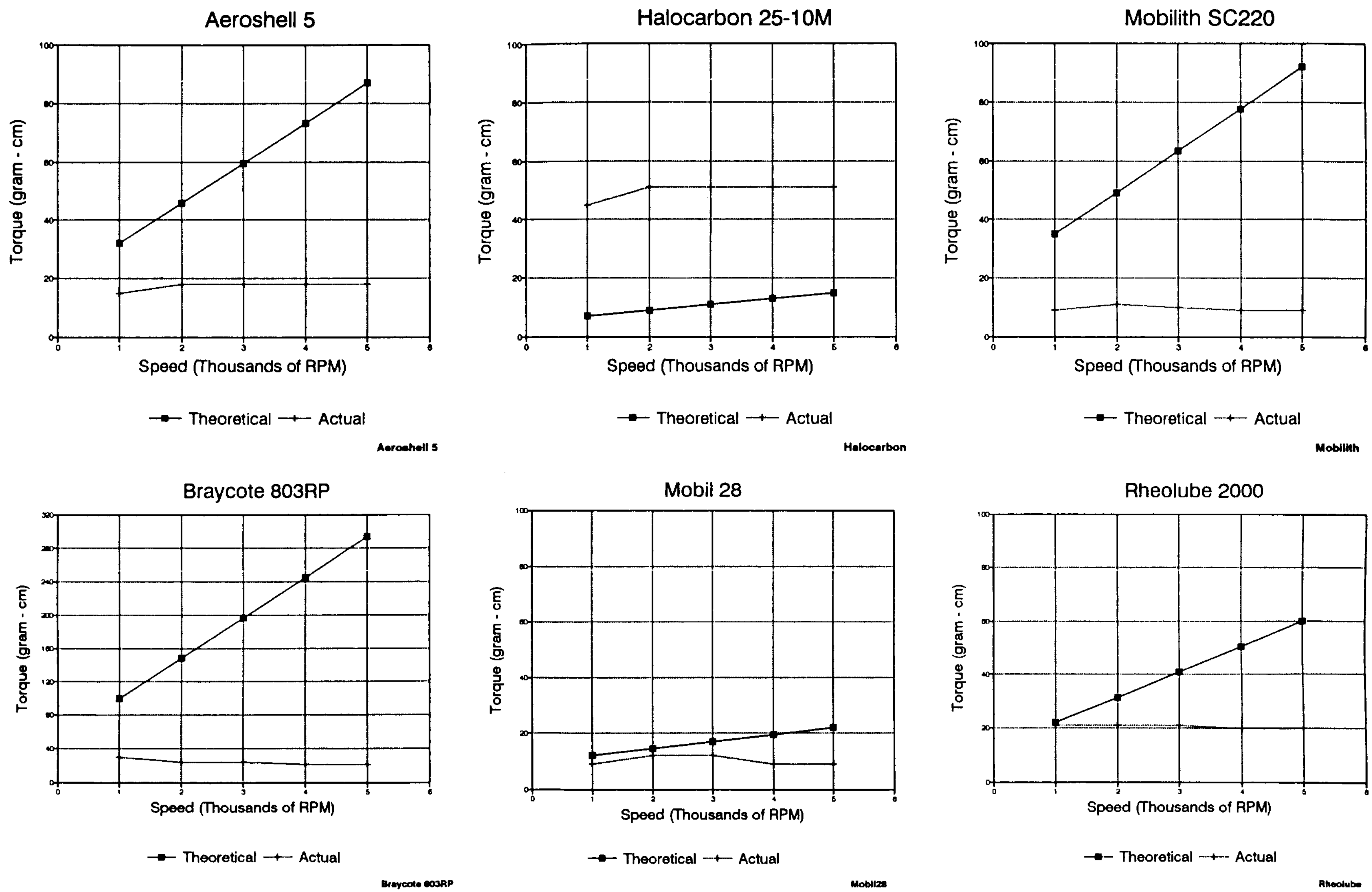

Torque vs. Speed at Room Temperature

Figure 2

MOBIL 28 GREASE CONSISTANCY

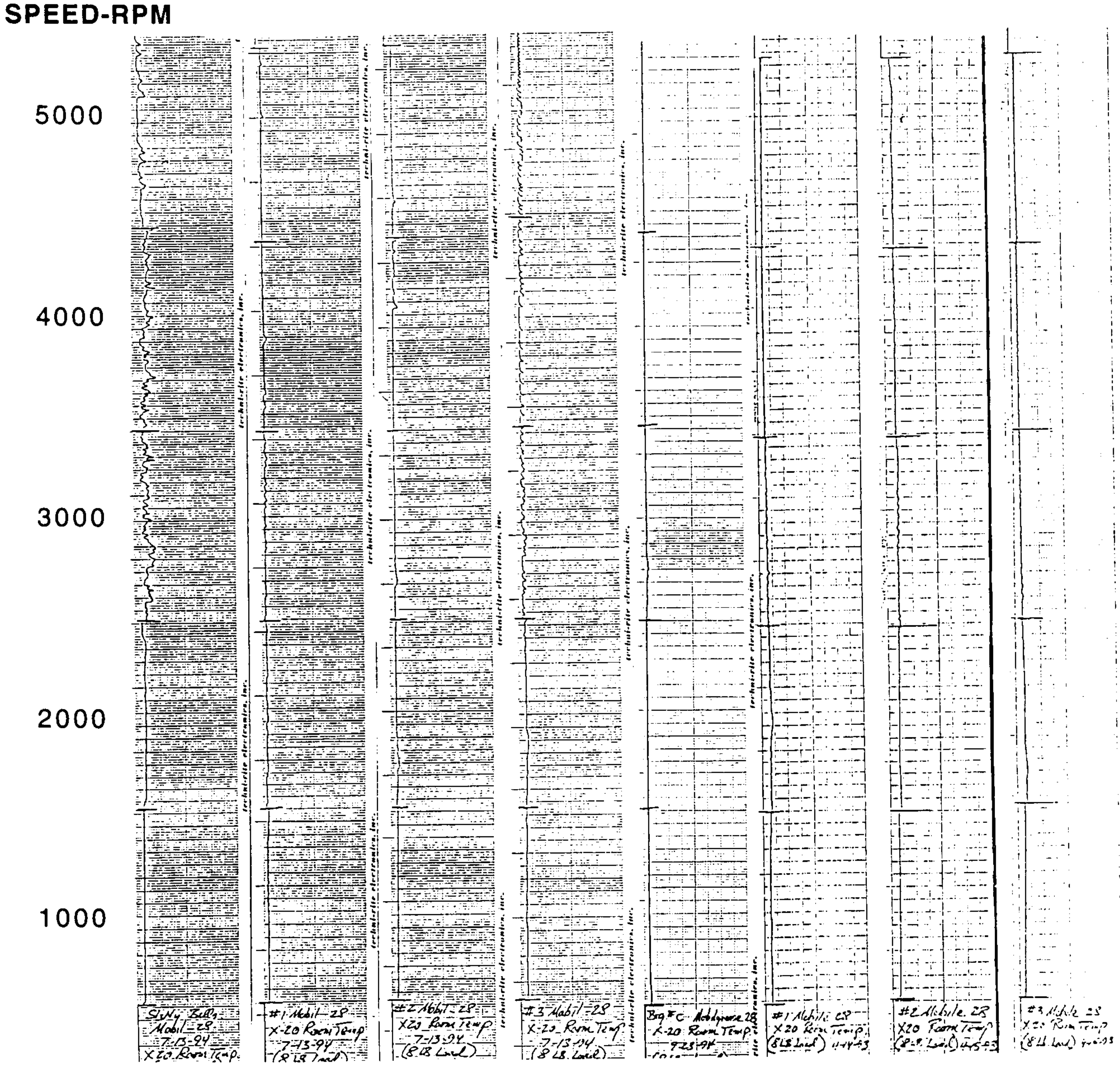

FIGURE 3

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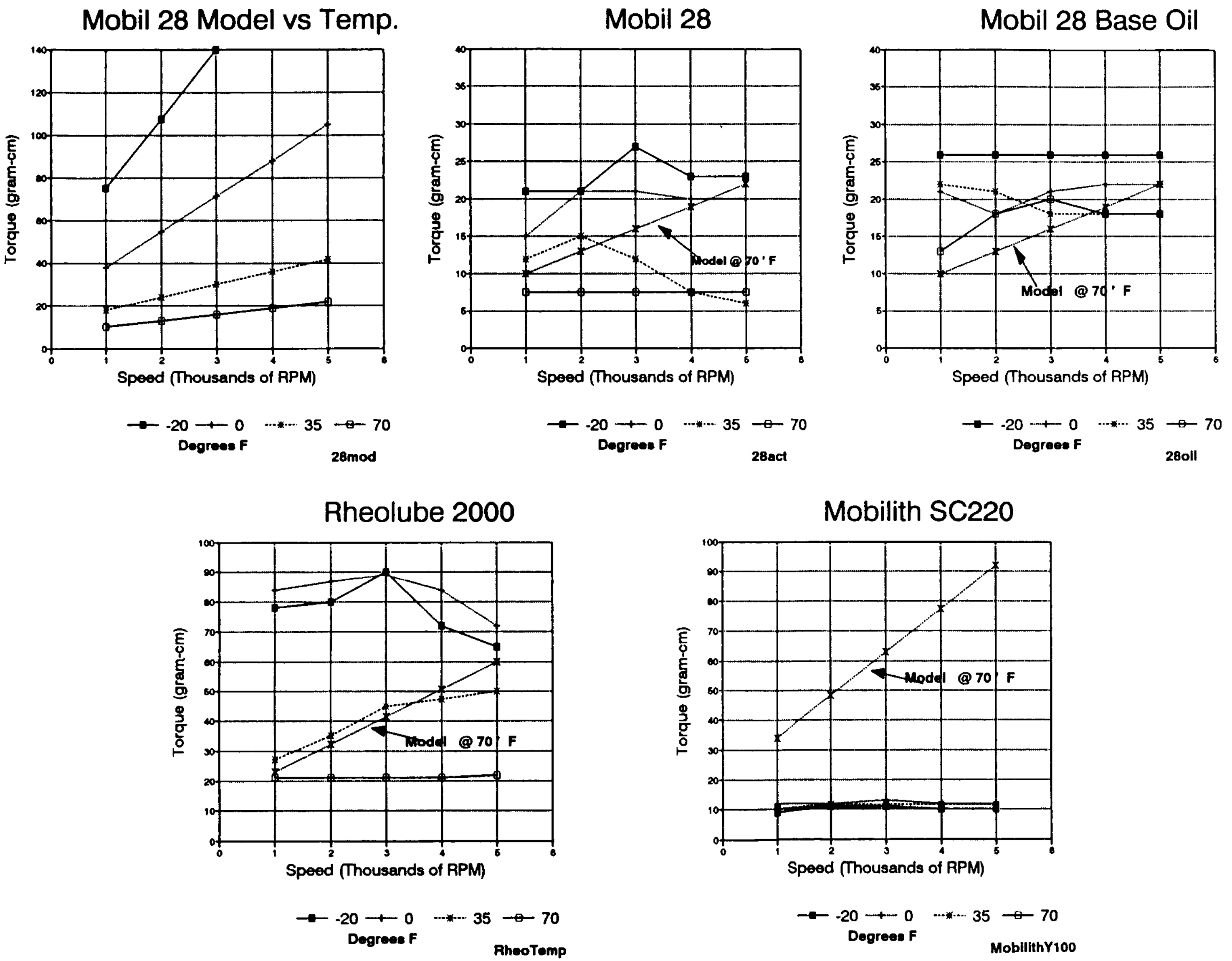


Torque vs. Temperature

Figure 4

MOBIL 28 TORQUE VS TEMPERATURE

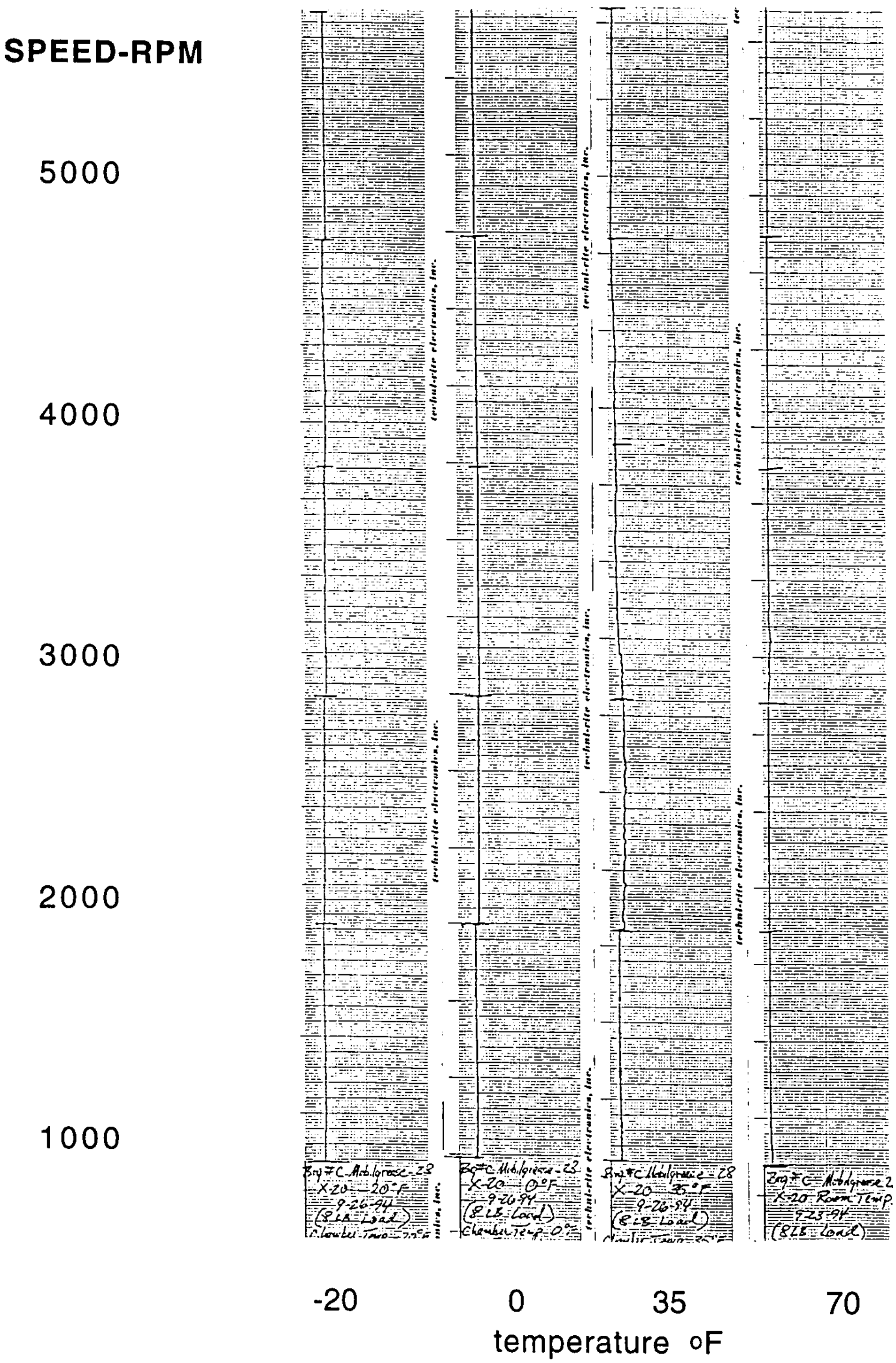


FIGURE 5

RHEOLUBE 2000 TORQUE VS TEMPERATURE

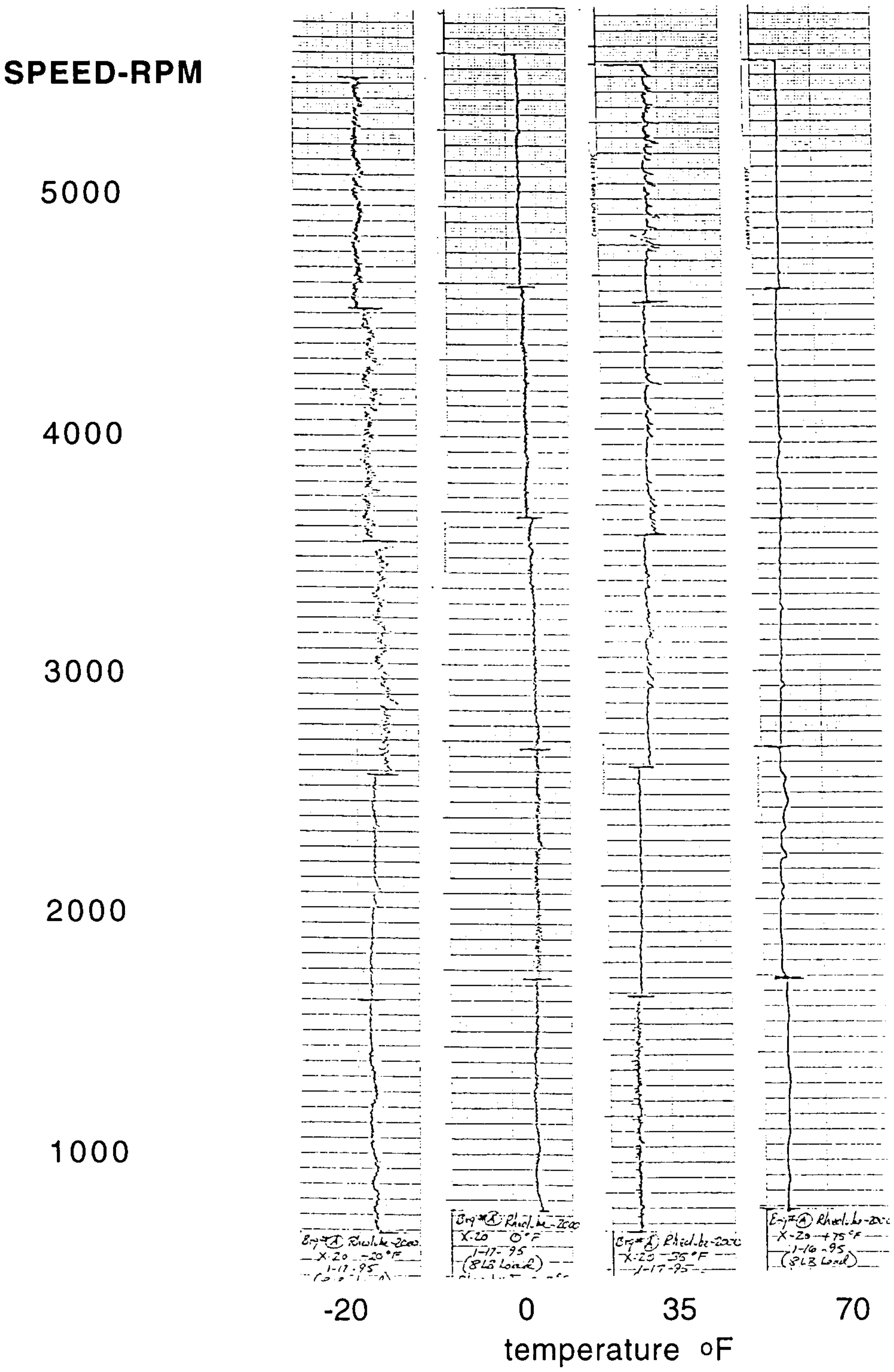


FIGURE 6

MOBILITH SC 220 TORQUE VS TEMPERATURE

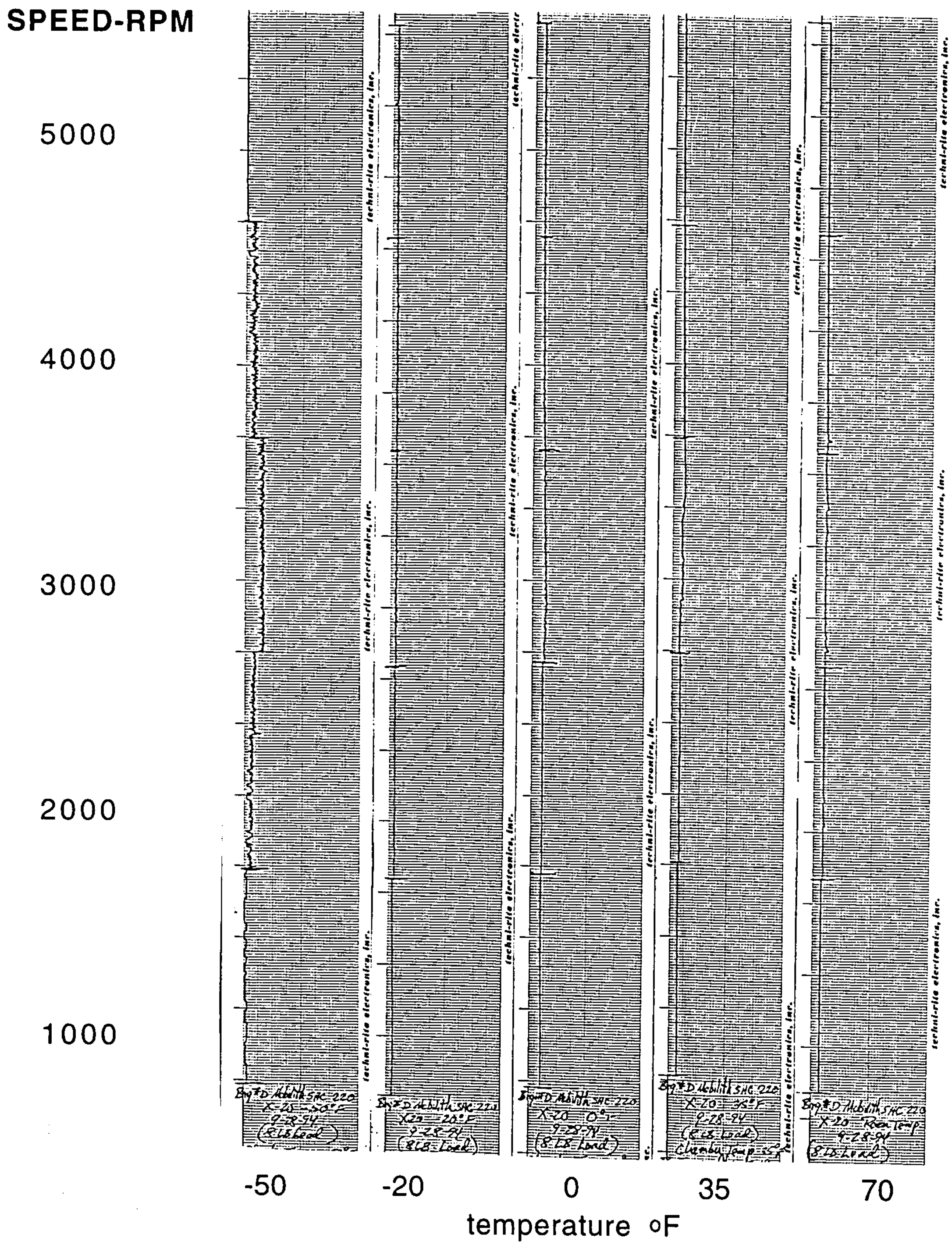


FIGURE 7

BEARING LUBRICANT LIFE TEST

CFC CLEANED BEARINGS (0) VERSUS CFC FREE CLEANED BEARINGS (Δ)

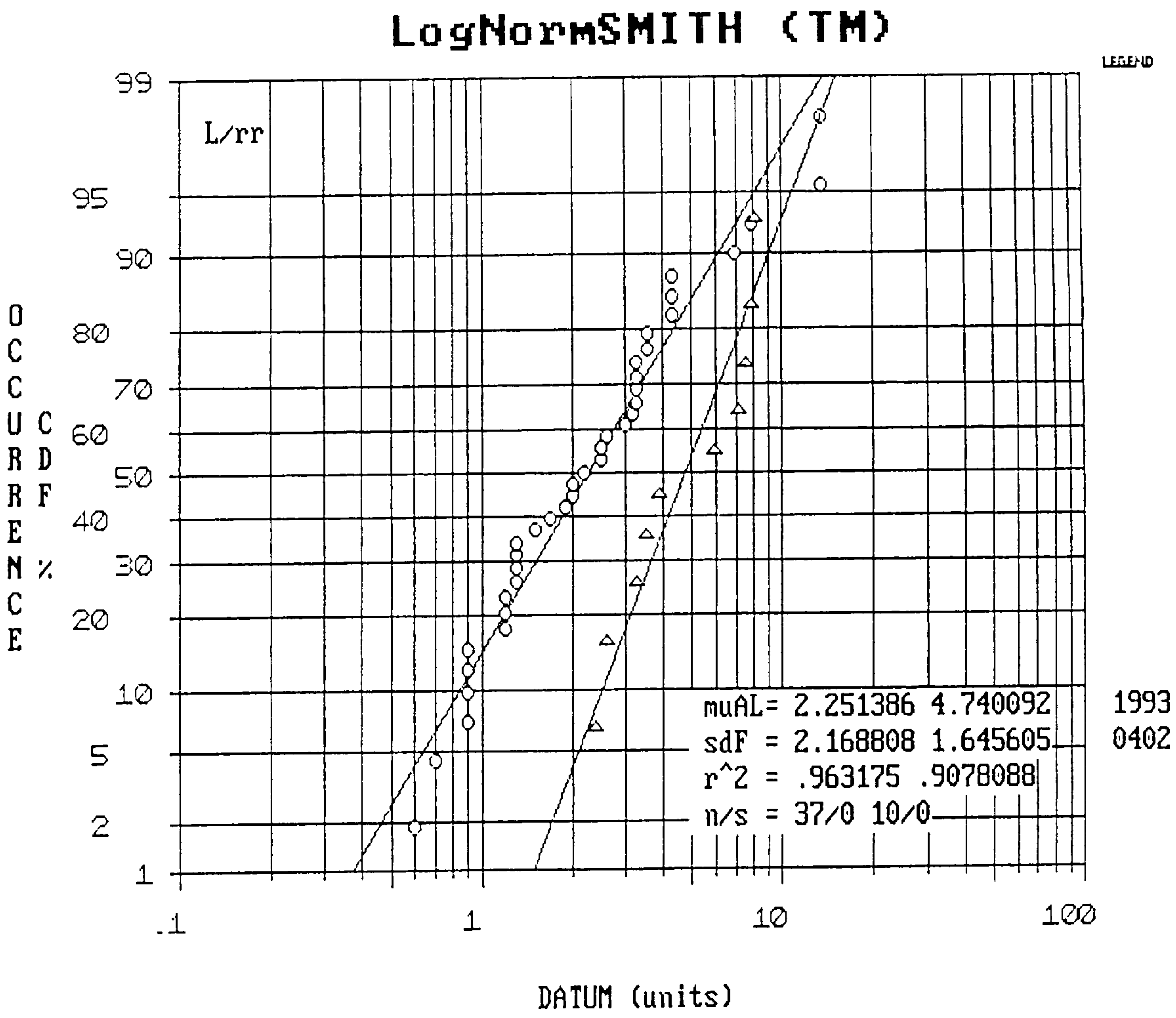
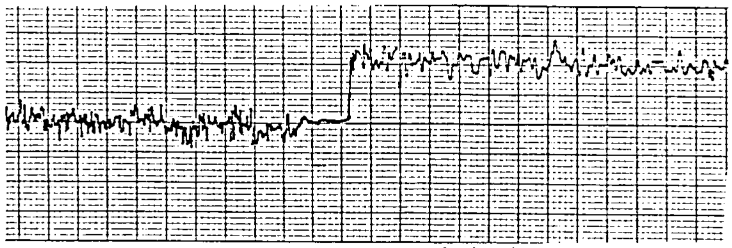
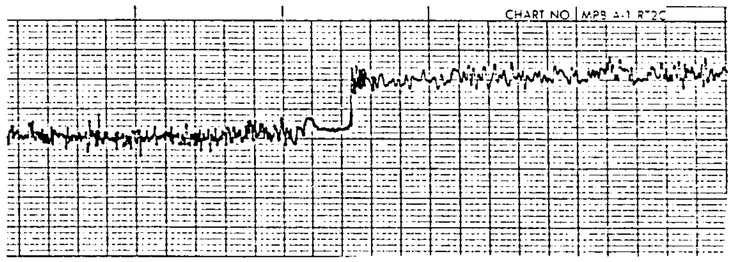


FIGURE 8

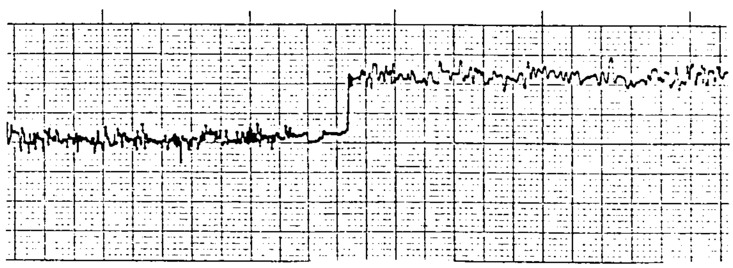
CFC CLEANED BEARINGS VERSUS CFC FREE CLEANED BEARINGS



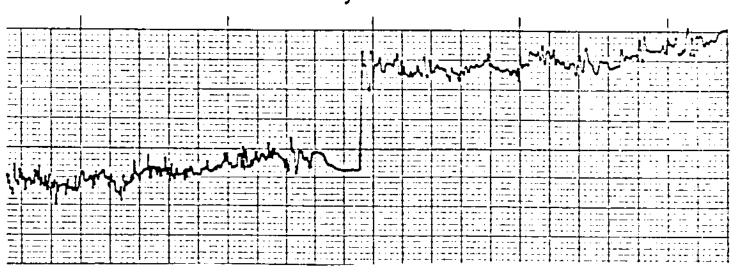
Old wash system -- Lubed



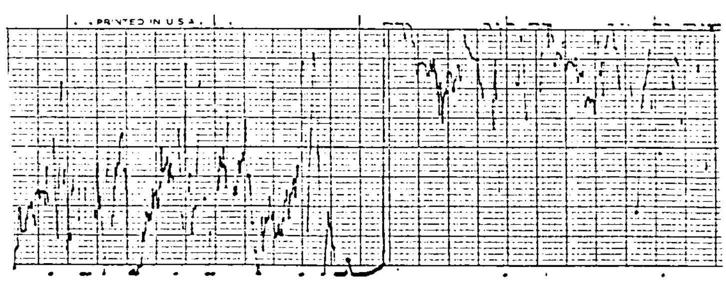
Old wash system -- Dry



New wash system -- Lubed



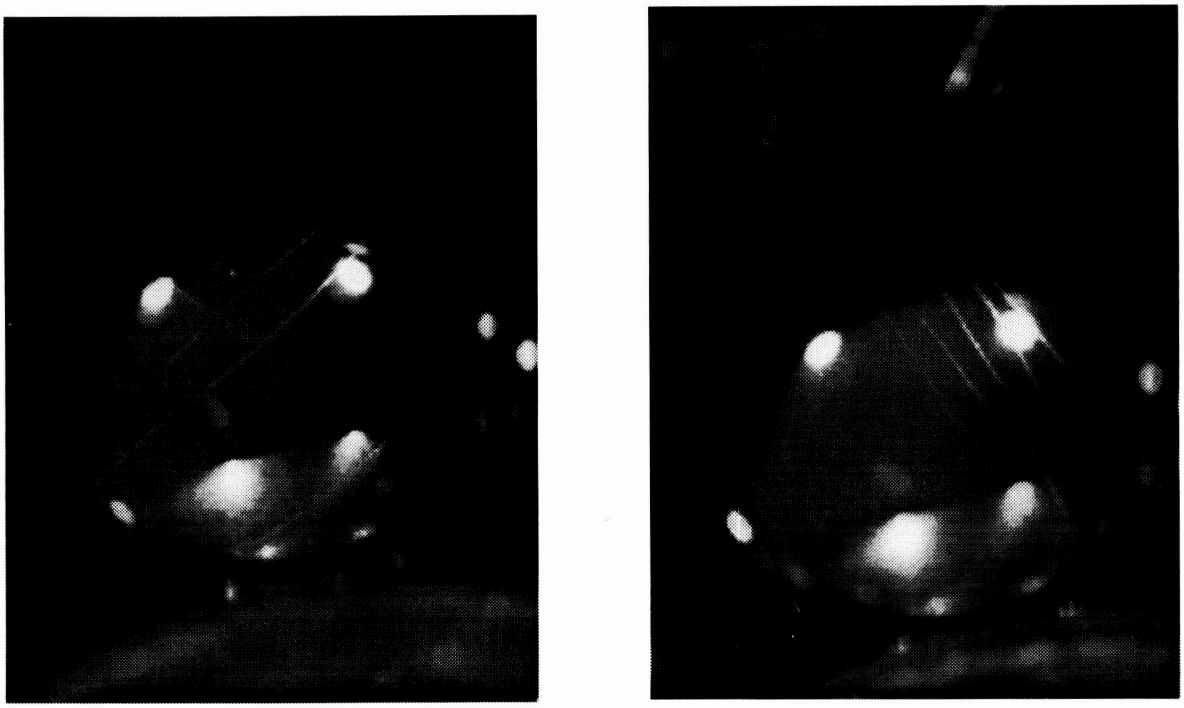
New wash system -- Dry



New wash system -- Dry
Sample of erratic behavior

FIGURE 9

CERAMIC HYBRID SR3 BEARING



NEW WASH SYSTEM
CERAMIC BALLS SHOWING BALL TO BALL SCUFFING DAMAGE

FIGURE 10