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Assessment of Frictional Characteristics of Gear Oils in Rolling element Bearings

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Abstract:
Industrial Gear Oils are also used to lubricate the support rolling element bearings at the supports in the gear boxes. Besides the large quantum energy consumption in the gear contacts, a significant amount of energy gets expended at the bearings. These bearings work under a variety of conditions depending upon the application and the industry in which these are used. An attempt is made to assess the energy efficiency of these gear oils using the FAG FE8 Bearing test rig. This paper describes the simulated studies carried out using this rig on various gear oils to assess the factors that influence the energy consumption in the rolling element bearings.

1.0 Introduction:
In order to understand the friction in rolling element bearings, it is necessary to understand the mechanism of lubricant film formation in these. Rolling element bearings operate in the classic EHL regime of high pressures and relatively high rolling speeds. Film thickness generation is governed by the contact geometry, kinematics, and the bulk lubricant properties. In general, the elastic deformations of the contacting surfaces are much greater than the lubricant film thickness.

Friction in Rolling Element Bearings working under a rolling contact is comparatively small in magnitude compared to other machine components, but still contributes to the energy losses encountered in service in Rotating equipments. These frictional losses are significantly dependent upon the lubricant used in the bearings. Although this quantum can be assessed under simulated contacts in similar geometrical configurations, a controlled bearing test gives a proper insight and correlation with the Energy efficiency of the lubricant in the rolling element bearings.

The study of frictional losses in oil lubricated bearings is an area wherein lubricant manufacturers have shown increased interest. Although the rolling element bearings exhibit low frictional characteristics (hence the popular name “antifriction bearings”), the oil characteristics influences the energy consumption pattern of any equipment using rolling element bearings to a great extent. Standard test methods do not facilitate the evaluation of this property of the lubricating gear oil under a simulated bearing contact and will only offer an insight into the contact friction under simulated sliding point contact or a simulated rolling contact. The rolling element bearings operate predominantly under rolling contact and the coefficient of friction is of a small magnitude dependent upon contact temperatures and the film thicknesses existing at the contact points. Although improvement of the frictional characteristics of the oils for reduction of energy consumption at the gear contacts also works towards the improvement of the EE characteristics at the bearing contacts, there is a need to assess the same to assess the factors responsible for the energy consumption at the bearing contacts. Rolling element bearings work under a mixed to EHD regime of lubrication depending upon the speeds-loads and temperatures. There is a need to do an assessment under a variety of conditions simulating those under which the actual gear boxes operate.

For this purpose, a FE8 test equipment manufactured by FAG Germany was used to assess the frictional characteristics of the oils and greases. It evaluates the rolling element bearing as a tribological system simulating a variety of operating conditions as existing in field. This equipment can study the mechanical properties of lubricants under different bearing geometries under varying conditions of loads, speeds, temperatures and environments.(1,2). DIN Standards for this purpose assesses the wear characteristics of lubricating oils and greases(3). An earlier publication (6) by the same group described the study on this equipment to measure the friction moment under a tapered roller bearing configuration for different greases. In that paper, the effect of factors such as the grease chemistry and its base oil type and viscosity on the bearing friction was studied. These results were able to study the factors responsible for the development energy efficient greases for use in rolling element bearings.

FAG FE8 test rig as per DIN 51819 test standards various parts (**3-5**) is used for the studies on the wear performance of greases and oils under a simulated bearing contact under several conditions. It forms a part of the specification requirements for several lubricating oil specifications such as paper machine oil, wind turbine lubricants and also different grades of gear oils. For Oils, there are two types of tests specified (1) FE8 wear test D 7.5/80-80 run for 500 hours at 7.5 rpm, 80KN Load (2) FE8 endurance test D 75/80-80 run for 1500 Hours at 7.5 rpm, and 80 KN.

In this paper, an attempt has been made to use the parameters like Friction moment exhibited by the oil with the oil lubrication-recirculation system connected to the test rig, and the temperature rise to quantify the energy efficiency of the oil under several sets of conditions. Some of the typical test conditions for these are under

1) FE8- 7.5 rpm, 80 Hours, DIN 51 819-3 for the wear of rolling elements
2) FE8 - 750 rpm, 500 hrs, 120 deg C, 0.1 liter per minute per bearing of oil (with and without ingress of water)

Short duration tests of 80-500 hrs with oil or grease lubrication have been generally run to evaluate the anti-wear characteristics of the greases as per the DIN standard. It is also possible to run long duration tests till failure to judge the fatigue behaviour of rolling element bearings and the effect of the grease on it.

2.0 TEST RIG DESCRIPTION:

The FE8 test rig as depicted in Figure 1 consists of a drive unit, drive unit, gears test head, test head cooling, measuring electronics, switch cabinet. The basic machine of the test rig consists of a test head connected to the drive unit. The drive shaft of the drive unit is supported by two auxiliary bearings which are preloaded by Belleville Springs. The shaft is driven by a triple pole changing DC motor directly or via a gear (spur gear, belt drive). With this arrangement, the speeds can be varied from 7.5 rpm to 12000 rpm. The test head shaft is tapered at the drive end so that the entire test head module can be removed from the drive unit. The test head contains the test bearings. The bearings are loaded axially by means of Belleville Springs. The magnitude of the load can be set by a washer. The axial load direction in which all the test bearings are loaded due to the axial load structure is advantageous since all the rolling elements are subjected to the same load and the testing is tighter than under radial or combined loading. The electric motor, test head temperature and running time are controlled centrally from a switch cabinet. There are different heads to accommodate different geometries of rolling element bearings as listed below:

(1) Angular Contact ball bearings
(2) Tapered Roller Bearings
(3) Cylindrical roller Thrust Bearings

In the case of grease, the heads with the Angular contact ball bearings and the Tapered Roller Bearings are used, while in case of oil lubrication, the test head with the Cylindrical Roller Thrust bearings type 81212 with brass cages are used. To properly simulate the lubrication of rolling element bearings with oil as that occurring in the field, it is therefore necessary to install an oil circulation system comprising an oil container, pump, distributor elements, a filter and an oil preheating unit. The latter contains heated guide plates over which the oil flows and by which it is brought to almost the same temperature as the oil as in the field condition. The inter-action between the oil and the oxygen in the air is also ensured in this way.

The machine is automatically switched off at a certain set level of bearing friction, thus preventing damage to the bearings and the test head. The temperature if exceeding 160 Deg C for the thrust bearings or 200 deg C for the angular contact or tapered roller bearings during any of the stages of test will shut of the test rig.

A photograph of the FE8 rig connected to the oil recirculation system is shown in Figure 2. The schematic of the arrangement of the oil circulation system connected to the test rig is shown in Figure 3.

The following criteria are used for the assessment of the Frictional performance of the lubricating oil in the bearings:

(1) The temperature of the outer race of each of the test bearings
(2) Friction Moment of the bearings (the product of the force required for holding the housing structure and the distance between the holding point and the shaft axis. A force transducer mounted near the bearings with a loading arm measures the holding force at the housing and sends it to the amplifier.

These bearings have a large friction area between the roller surface and the raceways and therefore require an oil which provides good protection against wear. Generally the specifications require a test to be carried out at low speeds so that the bearing run in the boundary regime of lubrication where the frictional effects are maximized.

3.0 EXPERIMENT:

In this study, various gear oils were evaluated using a set of two single row Cylindrical roller thrust Bearing Configuration FAG 81212 (OD 95 mm, ID 60 mm width 26 mm) mounted on the shaft as shown in Figure 4 and 5 . This bearing is equipped with brass cages.

The test conditions selected for stage wise testing were as under:

Table 1:

Sr No	Load	Speed	Initial starting Temperature	Duration
1	80KN	7.5 rpm	60 Deg C	6 Hours
2	20KN	750 rpm	60 Deg C	6 hours
3	10 KN	1500 rpm	60 Deg C	6 Hours

The test conditions were selected to be able to assess the frictional characteristics and the resulting energy consumption characteristics over a range of conditions ranging from **High Load, Low speeds at Sr No 1** to **Medium Load, Medium Speed at Sr No 2** to **Low Load, High speeds at Sr No 3**, in which the gear boxes in the different field applications generally operate.

The following lubricating oils were evaluated on the FAG FE8 test rig in the above test stages to assess the reasons for the energy efficiency of oils in bearings:

Table 2:

Sr No	Oil	Base oil Viscosity	Viscosity at 60 Deg C
1	**OIL A**	VG 320 Mineral	107 cSt
2	**Oil B**	VG 220 Mineral	79 cSt
3	**OIL C**	VG 220 Fully synthetic	94.83 cSt
4	**OIL D**	VG 320 Part synthetic and Friction modified	103.4 cSt
5	**OIL E**	VG 320 Part synthetic and Friction Modified	102.0 cSt

The criteria for the selection of the oils was to assess the factors responsible for the Energy efficiency characteristics of the lubricating oils as follows

(1) Viscosity
(2) FMs
(3) Effect of synthetic base stocks

Besides the selection of the operating conditions led to the lubricating oils operating under different regimes of lubrication, with the Condition at Sr No 1 predominantly in the Boundary to Mixed Regime of lubrication, the Condition at Sr No 2 predominantly operating in the Mixed to

EHD regime of Lubrication, and the Condition at Sr No 3 in the EHD regime of lubrication as illustrated in the Stribeck Curve.

The following parameters were observations during the study.

(1) Starting friction moment(N.m): This was taken at the ambient temperatures in the range of 18-20 deg C and was indicative of the resistance encountered due to the starting in the morning cold conditions. It depends upon the viscous drag encountered due to the viscosity effect and the boundary friction encountered on starting the rotation of the rolling elements.

(2) Stabilized running moment (N.m): The running Friction moment stabilized after a few hours of running and remained steady at that condition. It is representative of the steady state condition, and was indicative of the energy consumed in the steady state conditions prevalent in the operation. At the low speed-high load condition of 80KN, 7.5 rpm, the regime of lubrication was in the mixed to EHL regime, so the steady state running Friction moment did not reduce considerably for all the oils from the starting value.

(3) Stabilized mean temperature of bearings: The temperature stabilized at a particular value after a few hours and was the net effect of the energy consumed at the bearing contact less the dissipated heat to the environment. Since the ambient temperature remained in the range of 18-22 deg C, the net effect of increase of temperature was dependent on the frictional characteristics due to the oil.

4.0 OBSERVATIONS:

1) The oils were screened for their film thickness on the EHD apparatus over the different regimes of lubrication (in the speed range from 20 mm/sec to 4.5 m/sec) at 60 deg C, 20 N(0.48GPa) test conditions under a point contact as shown in Figure 7. Oil A shows the highest film thickness followed by Oil B and C and Oil D and Oil E in that order.
2) The screening for the friction coefficient was done on the SRV apparatus at conditions of 200N, 50deg C, 1mm and 1 hour as shown in Figure 8. Oils D and E being friction modified exhibit the lowest friction coefficient.
3) The observations for the various oils at different conditions are given in Table 3. The following are the main observations that can be made from the readings.
    a) Starting Friction moment values are affected by the running speeds/loads. At conditions of low speed/high loads, these values were significantly higher compared to the conditions of medium load/Medium speeds and high speeds/low load in that order.
    b) For all the oils, the operation in the low speed high load conditions, the running Friction did not reduce to the same extent from the starting Friction moment in the cold conditions. This indicates that it operates predominantly in the mixed regime of lubrication, and there is no EHL effect as experienced at higher speeds where there

is a buildup of EHL film. Under these conditions, due to the predominant mixed/boundary regime, it would be useful to assess the wear and endurance characteristics of the lubricant, as per the standard DIN 51819 conditions.

c) The temperature rise values for three of the oils Oil A, B, C without any friction modifiers were much higher than that for Oil D incorporating Friction modifiers, which indicates that the FMs in the oil reduce the frictional heating to some extent. However there is not much discernible change in the Friction moment between the starting and steady state values, which could be due to inertial effects in its measurement as well as the higher viscous drag.

d) In regard to the studies done at Medium Speed-Medium Load conditions and the High speed-low load condition, the starting friction moment dropped considerably with the increase in speeds and reduction in applied load for all the oils. The same trend followed for the steady state running friction moment to a much lower value than the starting friction moment due to be buildup of the EHL film in the contact zone.

e) The friction moment exhibited for the mineral oils under these two set of conditions were higher than those for the fully synthetic and the part-synthetic-friction modified oils. The temperature rise trend also exhibited the same characteristics. The synthetic oils show a similar trend in friction Friction moment and temperature rise as the friction modified part synthetic gear oils.

5.0 CONCLUSIONS:

(1) The FE8 rig can be used to screen the frictional properties of the gear oils under various set of operating conditions.

(2) Friction moment at low speed, high load conditions is the highest compared to Medium Speed-Medium Load and High Speed-Low Load conditions for all the oils in that order. This is due to the operation of the bearings in the boundary-mixed lubrication regime at that condition.

(3) Mineral oils exhibit higher friction moment compared to the synthetic and part-synthetic/Friction modified oils.

6.0 REFERENCES:

(1) Kleinlein E., "Using the FE8 System as a testing method for ball and Roller Bearing Greases", NLGI Spokesman, Vol 59, No 2, September 1995.

(2) Grasshoff H D, Maak Herbert, "Improved European Techniques for Grease Testing", NLGI Spokesman, Vol 49, April 1985.

(3) DIN 51819-1:1999, Testing rolling bearing lubricants using the FE 8 wear test machine – Principles.

(4) DIN 51819-2:1999 Testing of lubricants - Mechanical-dynamic testing in the roller bearing test apparatus FE8 - Part 2: Test method for lubricating greases, oblique ball bearing or tapered roller bearing.

(5) DIN 51819-3 Testing of lubricants - Mechanical-dynamic testing in the roller bearing test apparatus FE8 - Part 3: Test method for lubricating oils.

(6) Ajay Kumar, A H Zaidi, V Martin, Anoop Kumar, G K Sharma and A K Mehta, "Study of Frictional Characteristics of Greases in Rolling Element Bearings, presented at 5th Lubricating Grease Conference NLGI India Chapter, 2004.

Fig. 1

FAG Test Rig

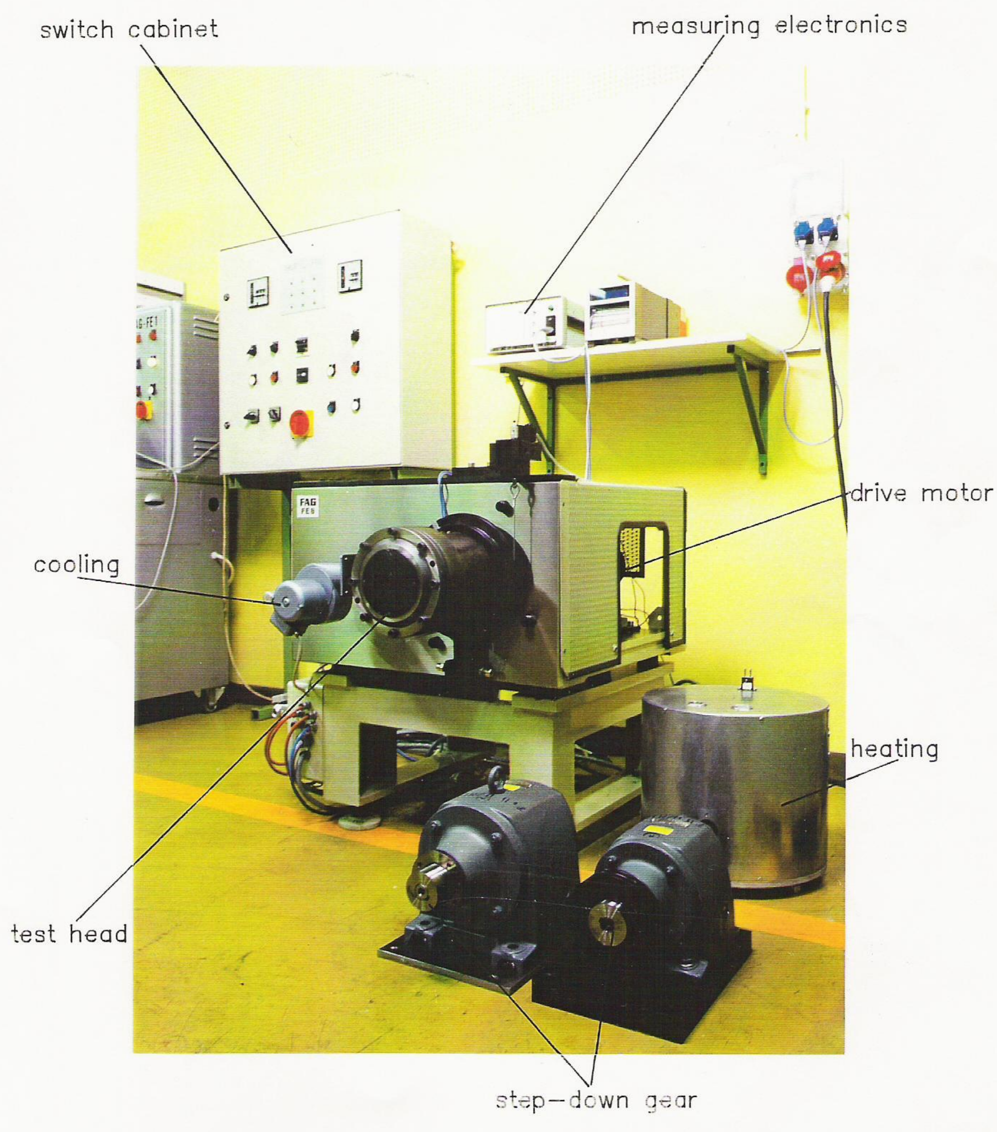

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Figure 1 : FAG FE8 Test rig (picture courtesy Schaeffler AG)

Figure 2: FE8 Test rig connected to an oil recirculation system

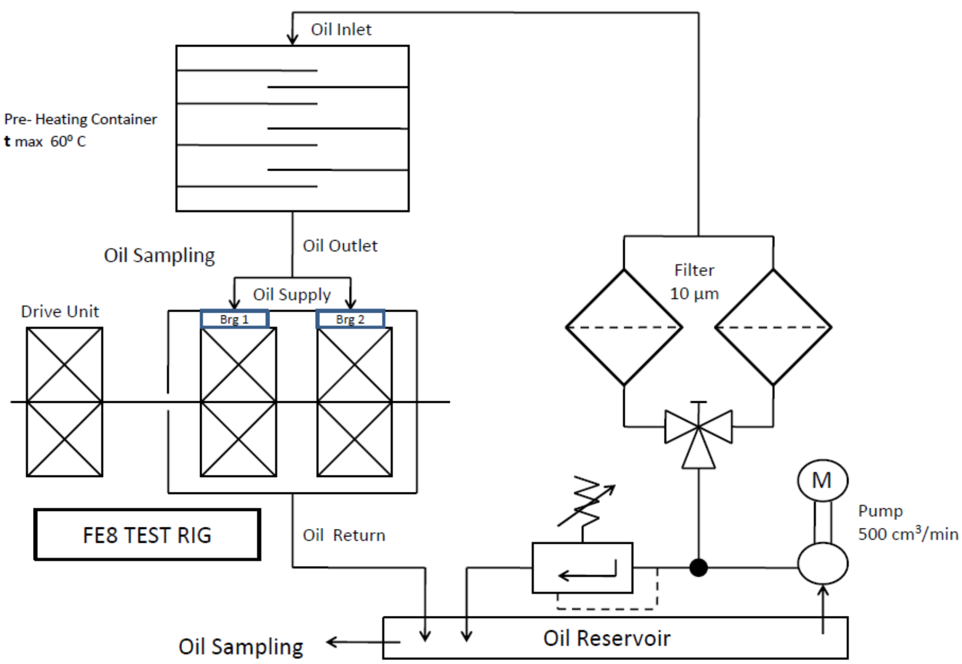

Figure 3: Oil circulation system circuit connected to FAG FE8 test rig.


Figure 4: Sectional view of the Thrust bearing 81212(picture courtesy: Schaeffler AG, Germany)


Figure 5 : Another view of the Thrust bearing mounted on the shaft (picture courtesy: Schaeffler AG, Germany)

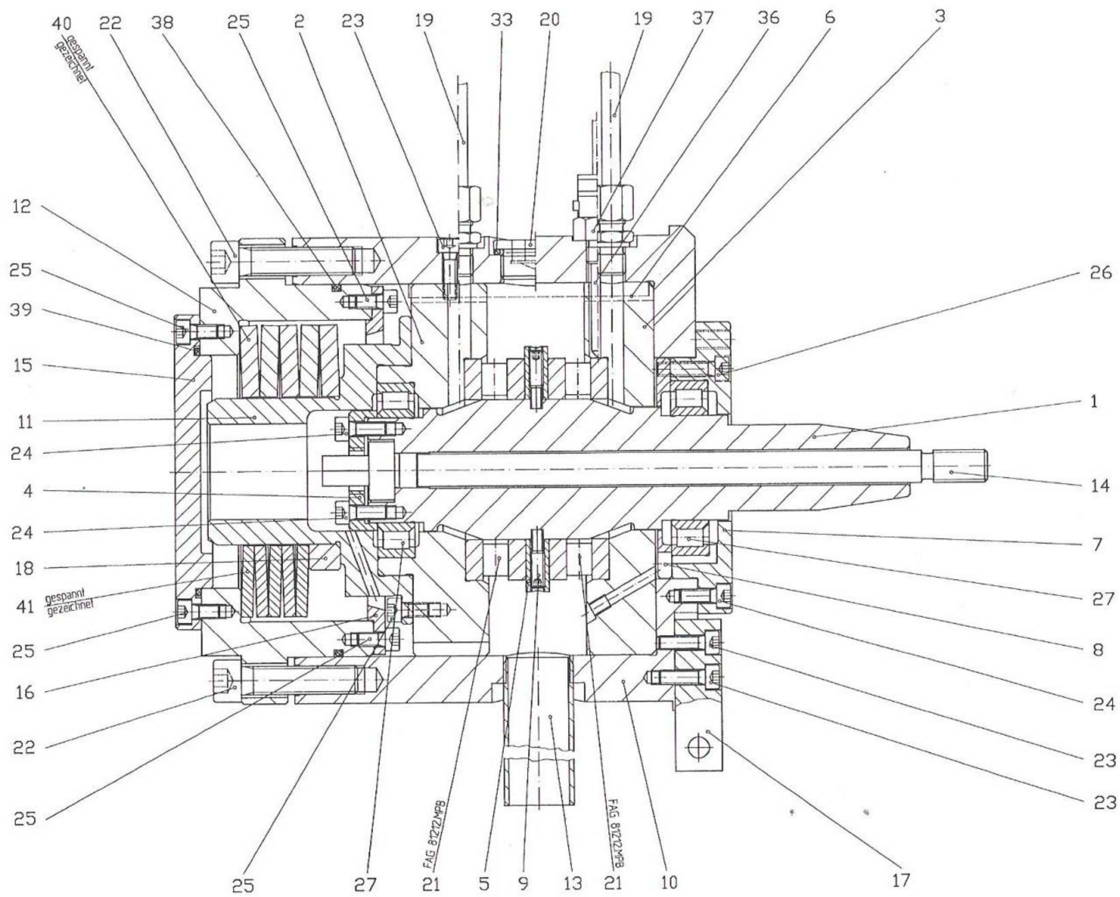

Figure 6: **FAG FE8 Thrust bearing Head (picture courtesy: Schaeffler AG, Germany)**

Table 3 : FE8 Studies on Different Oils at Different test conditions:

Oil	Load, speed, Temperature	Ambient temperature deg C	Starting Friction moment, N.m	Steady state Temperature Deg C	Running Friction moment N.m
Oil A	High load, low speed 80 KN,7.5 rpm,60 deg C	22	24.4	69	21.4
	Medium Load, Medium speed 20 KN,750 rpm,60 deg C	21	13.1	70	2.8
	Low load, high speed 10 KN,1500 rpm,60 deg C	20	8.2	77	2.1
Oil B	High load, low speed 80 KN,7.5 rpm,60 deg C	22	25	68	20.3
	Medium Load, Medium speed 20 KN,750 rpm,60 deg C	20	11.5	71	2.39
	Low load, high speed 10 KN,1500 rpm,60 deg C	20	9.8	72	1.7
Oil C	High load, low speed 80 KN,7.5 rpm,60 deg C	18	24.8	67	20.4
	Medium Load, Medium speed 20 KN,750 rpm,60 deg C	18	10.1	69	2.4
	Low load, high speed 10 KN,1500 rpm,60 deg C	20	9.3	76	1.4
Oil D	High load, low speed 80 KN,7.5 rpm,60 deg C	21	23.2	62	21.5
	Medium Load, Medium speed 20 KN, 750 rpm	20	9.5	63	2.76
	Low load, high speed 10 KN, 1500 rpm	20	7.5	72	1.4
Oil E	High load, low speed 80 KN,7.5 rpm,60 deg C	19	26	63	21.9
	Medium Load, Medium speed 20 KN, 750 rpm	19	10.1	67	2.54
	Low load, high speed 10 KN, 1500 rpm	20	7.5	75	1.3

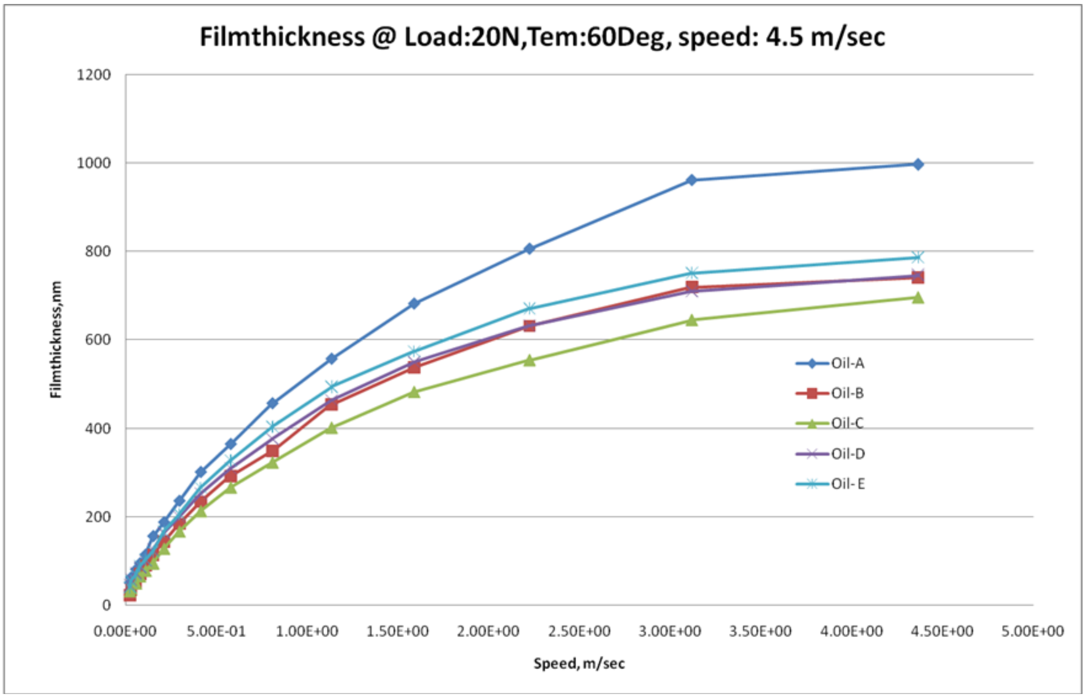

Figure 7: EHD Film thickness studies on the Gear oils

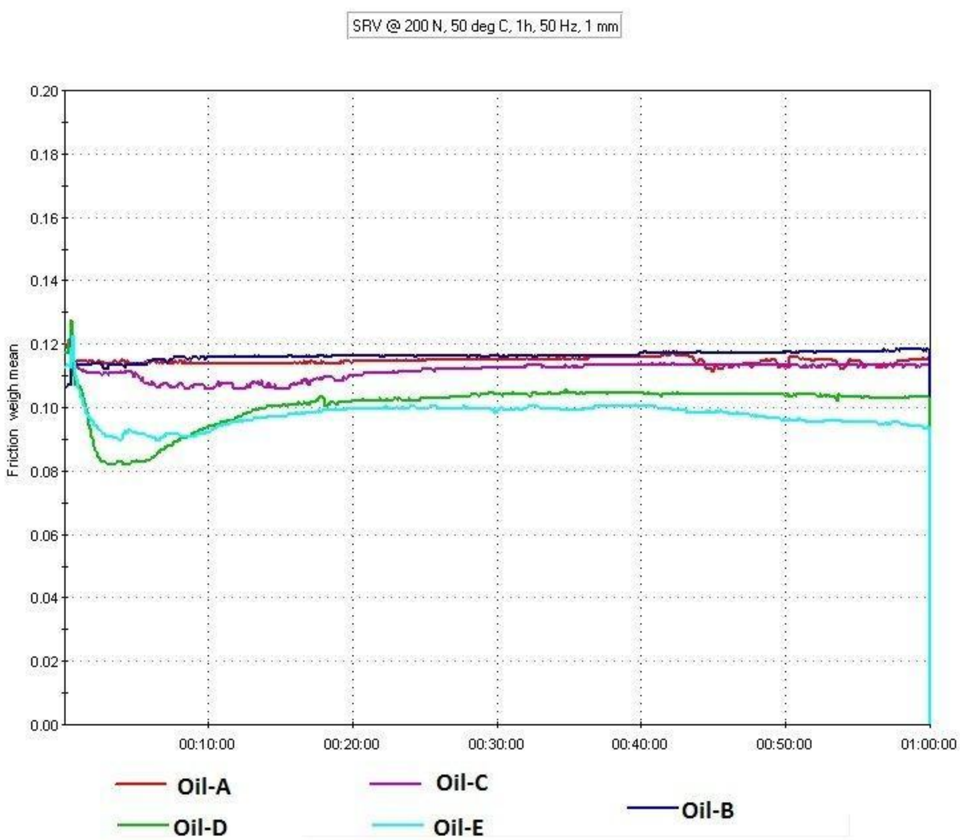

Figure 8: SRV Studies on the Gear oils

DEVELOPMENT OF A TEST RIG FOR GREASE LUBRICATED HUB BEARING ASSEMBLY

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ABSTRACT

Hub bearing assembly of an automotive vehicle suffers severe loading under bad road conditions. It is subjected to dynamic vehicle weight and cornering forces. It operates in the presence of dust and mud. Entrapment of these in the lubricating grease of bearing due to ingress leads to rapid failure. Extensive field testing is required to validate its design for reliability.

This paper describes development of a hub bearing test rig which is capable of simulating operating conditions in realistic manner. Once assembly is proven on this rig, it is likely to pass expensive and time consuming field testing. Test results illustrating four modes of failure observed during testing are presented.

KEY WORDS

Automotive tribology, bearing tester, hub bearing tester, dust ingress, mud ingress.

INTRODUCTION

Hub bearing or wheel bearing assembly of an automotive vehicle is a critical component. Its failure leads to accident which may result in loss of life and property. The weight of vehicle acts vertically downwards as radial load on bearing assembly. Turning generates centripetal force which is horizontal. It results in a moment on bearing with tire-road contact radius (dynamic tyre radius – DTR) as arm. This moment is bi-directional, depending on the direction of turn. Radial load is dynamic. Its profile depends on road undulations, speed and suspension characteristics.

It is possible to estimate fatigue life of a bearing under ideal conditions. Bearing manufacturers provide extensive information on various life adjustment factors [1] for improved estimation. These factors take into account aspects of reliability, material and operating conditions [2]. This model of life calculation is based on assumption that all avoidable causes of failure are absent. They include bearing damage during handling, material in-homogeneities, corrosion, wear debris and external contaminants. Life of bearing is drastically affected if any one of these is present.

An automotive wheel bearing operates in an environment far from perfect. Ingress of dust, water and mud in bearing is found to occur on road despite of seals. Ingress of contaminants causes excessive wear, corrosion and pitting. Pitting leads to rapid failure, much before the predicted theoretical life under ideal conditions. Lack of a reliable computational model of bearing life in the presence of contaminants makes it imperative that life is determined experimentally in simulated bench test.

A hub bearing test rig was designed to conduct endurance tests which meets and exceeds requirements of test procedure SAE J1095 revised March 2003.

BEARING LIFE EQUATIONS

According to DIN ISO 281, "adjusted rating life equation" of a rolling element bearing under constant load is given by:

$$L_{na} = a_1 \, a_2 \, a_3 \, (C/P)^p \qquad - (1)$$

Or simply

$$L_{na} = a_1 \, a_2 \, a_3 \, L_{10} \qquad - (2)$$

Where

L_{na} = adjusted rating life, millions of revolutions (the index n represents the difference between the requisite reliability[1)] and 100%

a_1 = life adjustment factor for failure probability
a_2 = life adjustment factor for material
a_3 = life adjustment factor for operating conditions
C = basic dynamic load rating
P = equivalent dynamic bearing load
p = exponent of the life equation
 p = 3 for ball bearings
 p = 10/3 for roller bearings
L_{10} = basic rating life, millions of revolutions

Hub bearing assembly during operation is subjected to dynamic loading at varied speed. Under these conditions, life can be estimated using linear cumulative damage rule [2] to determine equivalent dynamic load P. It is given by:

$$\mathbf{P} = \left\{ \sum_{r=1}^{N} (\mathbf{P_r})^{\mathbf{p}} \left(\frac{\mathbf{n_r}}{\mathbf{n_M}} \right) (\mathbf{q_r}) \right\}^{\frac{1}{p}} \quad - (3)$$

where,

P_r = Load
p = Exponent of life calculation
n_r = Rotational speed
n_M = Mean rotational speed
q_r = Duty cycle of duration at load P_r

DESCRIPTION

Figure 1 shows a typical hub assembly mounted on test rig and figure 2, general arrangement of the rig. The rig consists of following sub-assemblies:

1. Drive
2. Hydraulic loading
3. Lubrication
4. Sensors and instrumentation
5. Machine control
6. Data acquisition
7. Safety

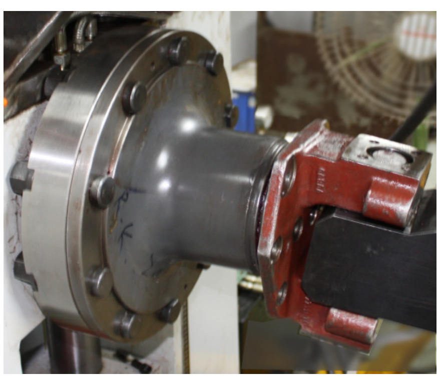

Figure 1. Front wheel hub assembly of a truck mounted on rig.

Hub assembly under test is mounted on the rig as illustrated in figure 3. It is driven at desired test speed. Radial and axial test loads are applied independently with hydraulic cylinders. Note that the location of axial load line is at dynamic tyre radius. Another method of application of load is "resultant load technique". In this method a single loading cylinder is used. Its angle of inclination with respect to axis of rotation is such that the radial and axial components of resultant force are equal to test loads in these directions. Location of resultant load line is such that radial and axial load lines pass through wheel center line and dynamic tyre radius respectively.

The spindle of the test rig is a critical sub-assembly. Distance of spindle bearings from load line is more than that of sample. This applies a large moment loading of these bearing. By the selection of spindle bearings with ample load ratings, it was ensured that spindle life was longer than 50 full load tests. A lubricant re-circulation system was used to lubricate and cool spindle bearings.

The hydraulic power pack has a variable displacement piston pump and an accumulator. With these two components, average power consumption could be minimized to 0.5 Kilowatt for a peak hydraulic power of 12 Kilowatts. This makes the power pack energy efficient. Solenoid operated electro-hydraulic valves were used for cylinder pressure control with feedback from load cells.

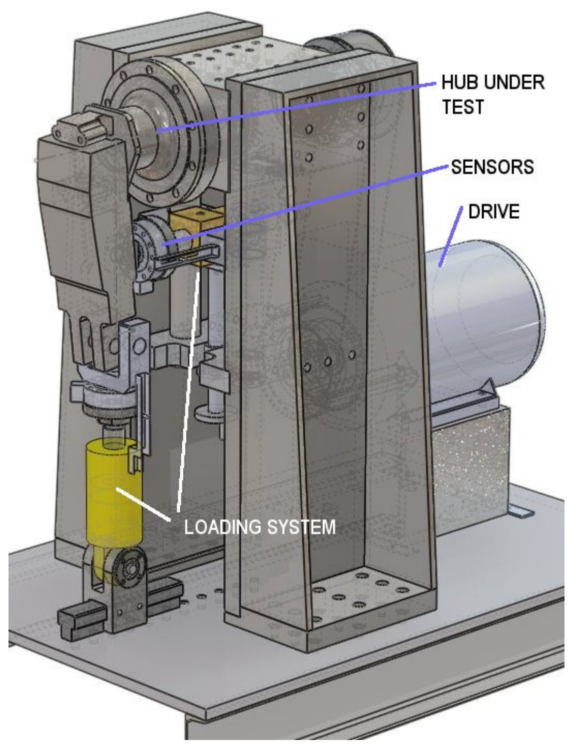

Figure 2. Sub Systems of a Ducom Automotive Hub Bearing Test Rig.

Fatigue rated load cells were used to measure horizontal and vertical loads. Output of each load cell was used in servo loop for precision control of hydraulic loading.

Test load could be varied from 2kN to 100kN. Response of the loading loop was a slightly over-damped to eliminate possibility of overshoot. It is necessary to ensure loading is maintained within tight limits. This requirement arises out of the fact that the sensitivity of life of a bearing to change in load high as it is proportional to P^p (3). A mere 10% change in load results in more than 33% change in life.

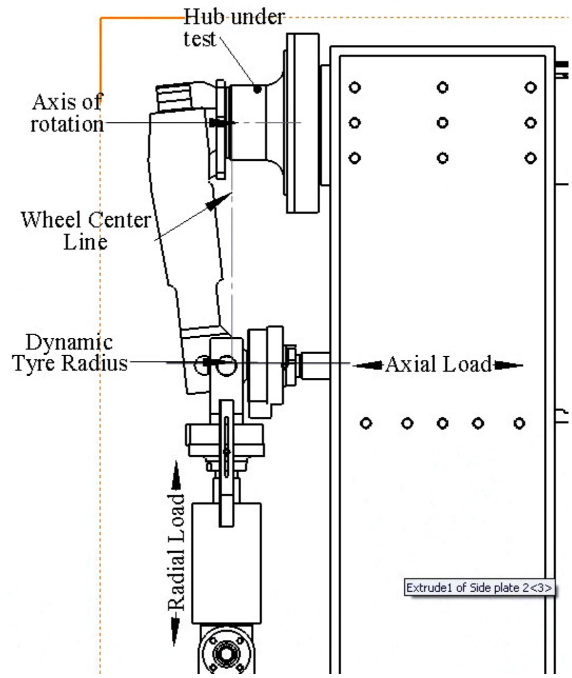

Figure 3. Schematic view of test rig.

Vibration level of the assembly under test was measured with a piezoelectric accelerometer. Its value after initial run-in is noted. Failure of the hub bearing due to fatigue results in rapid increase in vibration level. A user settable trip level above initial value terminates the test in the event of vibration exceeding this threshold. Vibration trip works only when its level stays continuously above threshold for a specified duration. This logic eliminates false tripping due to momentary increase due to reasons other than bearing failure.

Another important test result is hub temperature. When test starts, temperature rises rapidly and stabilizes after run-in period. In some cases continuous running may cause progressive loss of lubrication. It is easily identified by corresponding rise in temperature. A temperature trip terminates test in case temperature exceeds preset value (Fig. 10).

Machine software has three parts – machine control, data acquisition and post processing. Test schedule and acquired data are displayed graphically online. Test results of multiple tests can be viewed for comparison on same screen. For unattended operation, provision to auto-restart after a power outage is provided.

Test plan is programmed in the machine control software opening screen by assigning horizontal and vertical test load cycles, speed and duration. Dust and mud application cycles can also be programmed. Load cycle could either be a simple on/off pattern with specified durations or a complex road load data (RLD) profile (Figure 4). These features facilitate the realistic testing, minimizing requirement of expensive field tests.

Figure 4. Typical road load data profile (RLD) with components of radial load (blue), cornering force (red) and speed (green).

Road load profile is recorded by a test vehicle fitted with wheel force dynamometer (Figure 5) which is driven on proving ground.

Figure 5. A passenger car fitted with a wheel force dynamometer.

A chamber surrounding hub assembly facilitates testing under controlled dust conditions (Figure 6). Measured quantity of test dust of specified grade is put in the chamber. Puffs of compressed air keep dust afloat in it. Duty cycle of dust application can also be specified in the test plan. Similarly mud chamber is used in place of dust chamber to conduct testing with mud jet directed at test sample. Mud of controlled composition is re-circulated. Agitators in the mud tank do not allow its settlement. Dust chamber and mud chamber are detachable accessories which can be fitted when required.

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Figure 6. A hub bearing tester with dust attachment.

TEST RESULTS.

Tests on a variety of hub bearing assemblies were conducted. Samples included were of a small passenger car, light commercial vehicle and a large truck. Tests were conducted in dry, in dust and with mud. At the end of the test assembly was carefully dismantled and examined for failure analysis. Temperature and vibration data related well with observations. There were four modes of failures observed which are illustrated in the following section:

1. Failure due to loss of lubrication

Hub assembly of a truck was subjected to dry dust test. Record of hub temperature is shown in Figure 7. Temperature rose rapidly and when it reached 60^0C, an air blast directed towards the test assembly was switched on. Notice quick drop in temperature due to forced cooling. Temperature gradually dropped and stabilized after run-in. In this sample with a particular grease it was noticed that temperatures started rising as testing continued further. Test terminated when the temperature exceeded preset trip value.

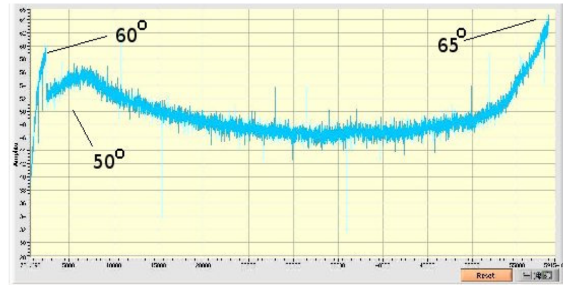

Figure 7. Temperature record indicating lubrication failure.

Figure 8 shows the assembly immediately after test. Notice excessive leakage of grease from the bearing which led to a gradual loss of lubrication.

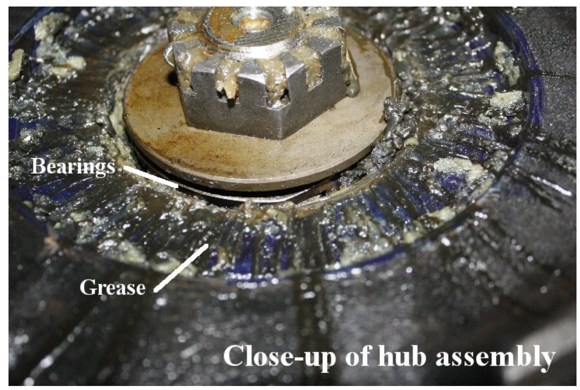

Figure 8. Excessive spillage of grease leading to lubrication starvation.

Figure 9. Oxidation due to overheating.

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Grease leakage resulted in increased temperature due to higher friction, initiating a self supporting cycle of progressive heating, leading to failure. Oxidation due to frictional heating can be clearly seen in the bearing in Figure 9.

2. Failure due to pitting

Figure 10 shows vibration record of a passenger car which was tested with mud jets directed towards it. Notice rapid rise of vibration level towards the end of the test.

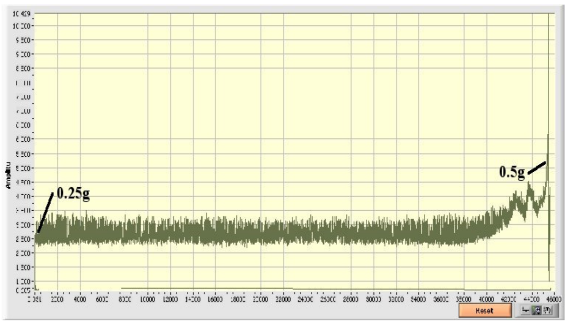

Figure 10. Vibration level increased two folds indicating bearing failure.

The test terminated due to vibration level trip. Examination of failed bearing indicated ingress of mud which caused pitting and abrasive wear (Figure 11) leading to increase in vibration.

Figure11. Pitting and corrosion due to mud ingress.

3. Cage failure due to bending moment

Bending moment on the bearing leads to cage failure. It is a catastrophic failure characterized by sudden increase in vibration followed by seizure. Figure 12 shows cage failure which led to seizure.

Figure 12. Entrapment of failed cage.

4. Failure of housing

It was not always the bearing which failed first. Fracture of housing was the first failure in certain samples due to improper material properties. Figure 13 shows one such failed housing.

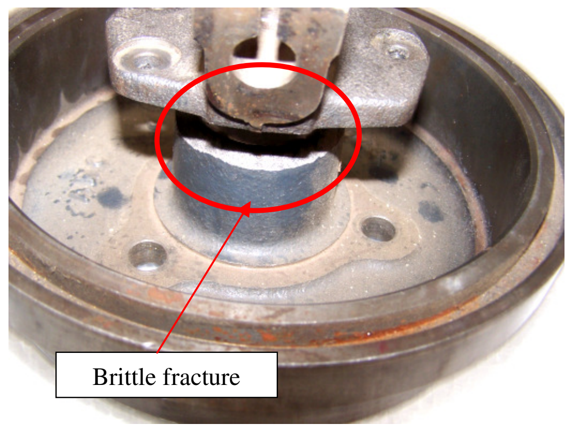

Figure 13. Failure of hub bearing housing.

CONCULSION

This paper describes development of a bench test rig for hub bearing assembly testing. The unique feature of this rig was the ability to conduct tests under realistic conditions of load and environment. Hence, results closely matched those of field testing. Once an assembly passes simulated tests, it is unlikely to fail during field tests. This minimizes time and cost of repeated expensive field testing.

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STUDY ON FLOW CHARACTERISTICS OF GREASES

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Centralized lubrication systems are widely used in industries and in heavy duty mobile equipment due to many advantages over conventional grease lubrication methods. The efficient operation of centralized lubrication systems ensures continuous and effective lubrication of machineries to achieve peak performance for a long time. Information of grease shear properties is essential for proper selection of greases and appropriate system engineering. NLGI grade provides grease consistency, but it does not provide information on grease's suitability for use in a centralized lubrication system. Many factors such as nature and content of soap, oil viscosity, polymer, solid additive contents, etc., have major determining role on flow characteristics. Compared to Lincoln ventmeter, MINITEST FFK is an easy to operate equipment and quick test method for evaluation of flow properties of grease over wide temperatures. Flow properties of lithium and lithium complex greases were studied using MINITEST FFK and the effect of different complexing agents on the rheological behavior of greases is discussed in this paper.

Key words: Complex grease, complexing agent, flow property

1.0 INTRODUCTION

The function of any lubricant is to prolong life and increase the efficiency of mechanical devices by reducing friction and wear. As compared to oil, grease has different physical and performance characteristics which are derived from the multiphase structured matrix formed out of oil, thickener and various additives[1]. Dispersed in base fluid, thickener gives grease its main physical characteristics, a gel structure and associated rigidity commonly referred to as consistency. Grease can demonstrate the properties of a solid or a liquid, depending on the conditions to which it is subjected. As a non-newtonian pseudo plastic material, flow of grease is not initiated until stress is applied [2,3]. Increase in stress or pressure produces disproportionate increase in flow; hence grease is not considered to have viscosity. The term apparent viscosity is "termed" to describe the observed viscosities of greases. Apparent viscosity is a major parameter determining flowability or pumpability of greases. Since apparent viscosity varies with both temperature and shear rate, the specific temperature and shear rate are reported along with the measured viscosity. The consistency as measured by cone penetration test also provides an indication of flow behavior of greases, but it is not sufficient to predict the flow performance across a temperature gradient.

2.0 GREASE APPLICATION METHODS

Maintaining properly lubricated machinery is an important part of any lubrication program. After the initial grease fill during installation, periodic re-lubrication is essential to realize the desired service life. Without administering lubricant at correct intervals with proper amount, a machine can experience costly failures. Theoretically, it is preferable to apply small amounts of grease at short intervals rather than large amounts of grease at long intervals. It is reported that over 50 % of bearing failures were the result of improper lubrication [4]. Inadequate lubrication and bearing contamination were the biggest contributors to improper lubrication. Type of application of grease is very important factor affecting effectiveness of lubrication.

Depending on type of industry, nature of machinery, safety and environmental consideration, grease application methods followed in industries may vary. Conventionally grease lubrication is done manually, either by hand packing or man grease gun. Hand application is typically performed infrequently and may result in uneven amount of grease being applied, which can lead to over greasing resulting in damaged seals arising elevated bearing temperatures caused by grease churn. There are many industries like steel, mining, cement, etc., where it is rather impossible to lubricate bearings manually while machineries are in operation due to high temperatures. Moreover, the periodic re-lubrication requirement can be daunting in such industries typically upwards of 7500 industrial points for a paper mill, 5500 for an automotive assembly plant, 4000 for steel, 3500 for refinery, 2000 for a cement mill and 1500 for a plastic plant [5]. Miniaturization trend in modern machinery design and construction has further increased the complexity of re-lubrication process. To make grease re-lubrication more effective and systematic centralized lubrication systems were introduced in 1930s [6]. Modern centralized lubrication system also known as automatic lubrication system (ALS) generally comprises of a controller or timer, a pump and reservoir, metering valves and fittings and supply and feed lines [7]. It typically delivers a controlled amount of lubricant (either grease or oil) to multiple, specific locations on a machine while machine is operating, at specific times from a central location.

2.1 Why Centralized Lubrication System?

The conventional old method of employing personnel to lubricate different parts of machines has its disadvantages. Few of them are listed below:

- ➢ Need to employ special trained personnel
- ➢ There is wastage of precious petroleum products due to spillage
- ➢ Machine has to be stopped for lubrication

In contrast centralized lubrication system comes with many advantages like:

- ✓ Easy to operate and user friendly, as all points can be lubricated by a single point located at convenient, accessible position

- ✓ Lubrication is achieved equally to all points, including hard reaching, irrespective of its piping and backing pressure due to its high pressure rating
- ✓ The device distributes lubricant in optioned quantity. Under or excessive lubrication is eliminated there by saving precious lubricant

The centralized lubrication systems are commonly used in following sectors: [8]

- Steel Industry
- Packaging machinery
- Cement industry
- Paper industry
- Sugar industry
- Tyre machinery
- Milling machinery
- Earth moving equipment
- Overhead cranes
- Chassis of a vehicle

2.2 Pumpability and Significance of Flow Property of Greases

There are many applications in industry where grease is required to be pumped to long distances through centralized pumps where pumpability of grease play major factor in system design. Despite a grease has all good characteristics to perform satisfactorily in applications, many instances are known where the grease is not considered due to poor pumpability. In order for any grease to perform effectively in centralized lubrication systems, it is essential that grease has good slumpability and pumpability. Grease that permit satisfactory delivery from lines, nozzles and fittings of dispensing system and subsequent lubrication on moving components is considered to have good pumpability and one which is easy to be drawn into inlet of pump (sucked in) to posses good slumpability [9]. Fibrous grease tends to have good feedability but poor pumpability, whereas buttery textured greases behave the other way.

The normal design parameters of centralized grease systems include the volume and frequency of grease required at each point, number of points requiring grease, operating conditions, pump pressure, line diameter and distance to the grease points**.** Flow properties of greases, like apparent viscosity are critical and essential input required for system design. Most common methods to study the flow characteristics of greases are US Steel mobility test, Lincoln ventmeter, Apparent viscosity etc. Lincoln ventmeter provides information about pumpability behavior of greases, which helps in correlating to field conditions. Many test methods are employed for determining viscosity and related parameters in relevance to flow properties and the lubrication applications intended.

3.0 TEST METHODS FOR STUDYING FLOW PROPERTY OF GREASES

Brief description of most commonly used test methods for measuring flow properties and viscosity of greases are given below:

3.1 Cone Penetration: ASTM D 217

In grease industry, cone penetration of grease and NLGI classification are considered as flow measurement**.**

3.2 Apparent Viscosity: ASTM D 1092

In this test, a sample of grease is forced through a capillary tube by a floating piston actuated by a hydraulic system using a two speed gear pump. From the flow rate and the force developed in the system, apparent viscosity is calculated. A series of eight capillaries and two pump speeds provide 16 shear rates for determination of apparent viscosity and measure the pressure drop of greases under steady-flow rate conditions at constant temperature. The information can be used to estimate the pressure-drop or required pipe diameters in distribution system. Also apparent viscosity datas are useful for evaluating the ease of handling or pumping at specified temperature of dispensing system. It is often used to evaluate pumpability at low temperatures.

3.3 Flow Property: ASTM D 3232

This test is used for evaluating flow property of greases under high temperatures and low shear conditions. In this method grease is packed in annular channel in an aluminum block which can be programmed to temperature in excess of 315°C. Using special trident spindle attached to Brookfield Viscometer, torque is measured at constant spindle speed of 20 rpm at various temperatures. The data is used for generating apparent viscosity verses temperature plot, which provide an indication of the flow properties of greases between room and elevated temperature. Although the test does not give actual flow rates, as in pipe line, it provides a means for obtaining some indication of this property.

3.4 Lincoln Ventmeter

Lincoln ventmeter is an established method for studying pumpability characteristics of greases by measuring the shear stress of lubricant at a test temperature to arrive at the minimum stress to produce a flow. The apparatus consists of 25 feet of ¼ inch diameter copper tubing, a pressure gauge, pump and two open/close valves. In this test, the grease is pumped into the ventmeter by a grease lever gun until the pressure has developed to 1800 psig. The pressure is then vented and the venting is timed and the pressure on the gauge is recorded at the end of 30 seconds. As per the literature, ventmeter reading above 600 psig (41bar) at the end of 30 seconds indicates that the test lubricant is not suitable for use in a centralized lubrication system. The test is used for arriving at pressure drop in long run pipes, effect of pipe diameter on dispensing characteristics and suitability of NLGI consistency in centralized system.

Literature provides exhaustive and numerous reports on studies carried out on pumpability of grease using Lincoln ventmeter, with thrust being on correlating the data to field conditions. However, studies on aspects related to the effect of type of base oil, base oil viscometrics, type of thickener, consistency of grease, processing conditions and correlation thereof have not been reported to a significant extent. Better understanding of the factors affecting flow characteristics and apparent viscosity will help formulators to design custom made greases meeting intended application requirements including flow property.

3.5 Flow-Property: MINITEST FFK by Kesternich Method (DIN 51805)

MINITEST FFK is the automated version of Kesternich method. The test is used for measuring flow pressure of greases over wide temperature from 30 to -60°C. In this test, sample grease is taken in a standardized nozzle by pressing on thin layer of grease put on a glass plate without any air bubbles. The full nozzle is inserted in the thermostatic block, closed at the bottom with a small air tight flask to protect the test nozzle against condensation of water and sealed air tight with seal stopper. (The flask also serves as collector of the sample after the test).The sample is subjected to test pressure generated by a motor-driven piston and measured with a precision pressure transducer. The test is programmed for the test temperature, the equilibrium time, the expected flow pressure and the pressure increase for each step. Cooling is achieved/regulated by a cascade block with Peltier elements down to -30°C where heat is dissipated over a heat exchanger and for temperatures lower than -30°C heat exchanger is required to be cooled with cold water. The test is performed automatically. When the test temperature is reached, the equilibrium time of two hours is started. After the equilibrium time, the pressure above the sample is increased in steps until a sudden pressure decrease, indicating that the grease has passed through the nozzle. The test is stopped automatically and the last pressure value is displayed as flow pressure. The MINITEST FFK equipment and the schematic flow diagram of the test method are given in Figure 1 and 2 respectively.

This test method is very simple and less time consuming compared to other tests like low temperature torque tests, flow properties by Lincoln Ventmeter and Apparent viscosity by ASTM D1092 etc. The temperature and pressure control of this test method is much better and average test time is $2\frac{1}{2}$ hours compared to the other tests which require minimum of 6 hours.

Fig-1: MINITEST FFK

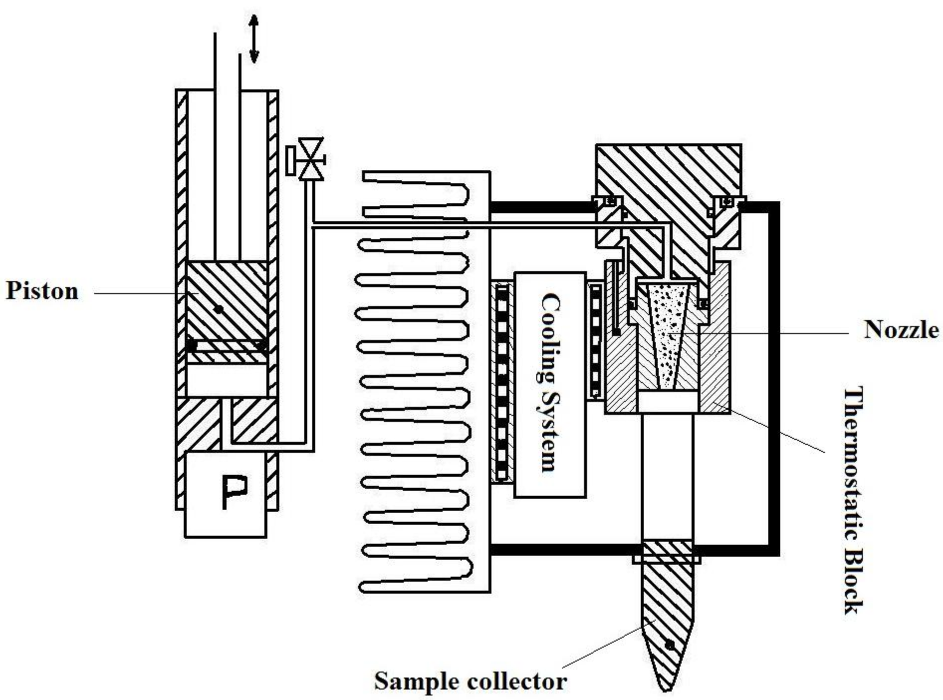

Fig-2: Diagram of MINITEST FFK

4.0 FACTORS AFFECTING FLOW PROPERTY OF GREASES

Base oil is a major component of any grease and it constitutes about 80-90% of a typical grease. Load support at moderate to high shear rates is mainly attributed to the viscosity of the base oil particularly in the absence of EP additives. Being the main constituent, the base oil and its characteristics like viscosity, viscosity index and pour point have significant effect on grease's viscosity and pumpability [11]. Generally grease formulated with low viscosity oil and low pour point has good flow property, whereas one with high viscosity and high pour point perform poorly in pumping.

Despite a minor constituent in grease, thickener is the one which gives grease its structure and consistency. Thickeners are of two types: organic thickeners and inorganic thickeners. Organic thickeners can be either soap based or non-soap based, while inorganic thickeners are non-soap based. The thickener in the grease immobilizes the lubricating oil, much like a sponge and releases oil, as it is needed at a controlled rate on applying shear. The higher the thickener amount the harder the grease, which means less mobility. Type of fibers of thickeners influences the pumping behavior of grease. Long fibers exhibit poor pumpability like soda base grease and short fibers exhibit better pumpability, such as Al complex grease. The role of thickener in grease is mixed in the sense that it also affects flow properties to a limited extent. Grease being a multiphase system, entrainment of base oil by thickener favorably affects the pour point of the oil enabling grease to perform satisfactorily at temperature much lower than the pour of the oil used in the grease. Similar to oil, flow properties of greases are also affected by temperature. Since most of the oils used in formulating grease have pour point not exceeding 10°C, pumping problems are rarely encountered for commonly used grades such as NLGI 1& 2 at temperatures above 30°C.

Literature survey indicates that eventhough there is a limited report on factors such as nature of thickener, base oil properties and temperature affecting flow properties of grease, hardly any reported work is available on systematic and comparative study on the effect of complexing agent on apparent viscosity and consequent flow properties of grease. Since the nature and extent the complexing agent affect the flow pattern is not known and also it is expected that the variation may not be very wide ranging, study need to be carried out under controlled conditions, minimizing manual operational variables to obtain useful information which can be correlated with other properties of grease.

As per NLGI survey volume of Li-complex greases has an increasing trend and these greases are widely used in various heavy industries where centralized grease lubrication systems are employed. Formulators have options to design Li-complex greases using different complexing agents, but information is not available regarding the selection criteria of complexing agents especially in relevance to pumpability / flow characteristics of the grease.

5.0 EXPERIMENTAL

In this study one Lithium base grease and three Li-complex greases based on different complexing agents viz boric acid, sebacic acid and azelaic acid were processed at bench scale in NLGI grade 2 using 10 -11% thickener. The base oil used in processing candidate greases was group I mineral oil of ISO VG-220. The greases were formulated in such a way that penetration of the products is more or less same to minimize its effect on flow properties. Few important properties of the selected greases are given in Table -1. Standard ASTM test methods were followed in determination of the above properties. The flow behavior of these greases were studied using MINITEST FFK at six different temperatures viz 25^0C,10^0C, 0^0C, -10^0C, -20^0C and -30^0C as per DIN 51805. Starting torque was also determined as per ASTM D 1478 at the same temperatures. Flow behavior and starting torque pattern of the four greases at various temperatures are shown in Table -2, Figures 3 and 4 respectively. The study was extended to correlate flow behavior with individual soap structure using SEM technique.

Table -1: Properties of Candidate Greases

Property	Test method	LC2 (Azelaic)	LC2 (Sebacic)	LC2 (Boric)	Li base grease
Appearance	Visual	Homogeneous	Homogeneous	Homogeneous	Homogeneous
Colour	Visual	Light brown	Light brown	Light brown	Light brown
Penetration at 25°C, worked	ASTM D 217	282	276	277	280
Penetration after 10^5 strokes, difference	ASTM D 217	27	28	22	20
Dropping point °C	ASTM D 566	>250	>250	>250	198
Base oil viscosity grade	ASTM D 445	VG-220	VG-220	VG-220	VG-220
Oil separation, %	ASTM D 6184	1.5	1.4	1.8	2.2

Table-2:Test Data of Flow Pressure and Starting Torque

Flow Pressure as per DIN 51805 Starting Torque as per ASTM D 1478

	LC2 (Aze)		LC2 (Seb)		LC2 (Boric)		Li Base grease	
Temp ^{0}C	Flow Pressure hPa	Torque g cm	Flow Pressure hPa	Torque g cm	Flow Pressure hPa	Torque g cm	Flow Pressure hPa	Torque g cm
25	92	295	86	295	96	295	72	295
10	134	295	134	295	116	295	98	295
0	164	590	178	590	144	590	134	295
-10	284	1475	254	1180	222	1180	210	1180
-20	780	4720	640	3540	586	3540	532	3540
-30	1544	17405	1204	15930	1180	15045	1132	12390

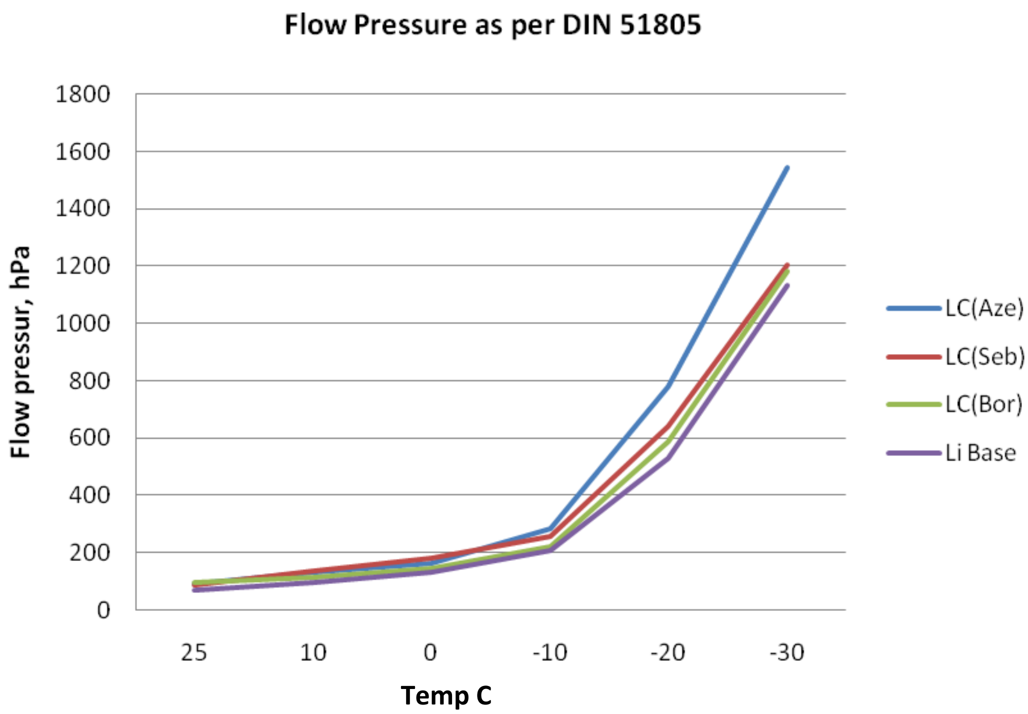
Flow Pressure as per DIN 51805
Flow pressur, hPa
1800
1600
1400
1200
1000
800
600
400
200
0
25
10
0
-10
-20
-30
Temp C
LC(Aze)
LC(Seb)
LC(Bor)
Li Base

Fig-3

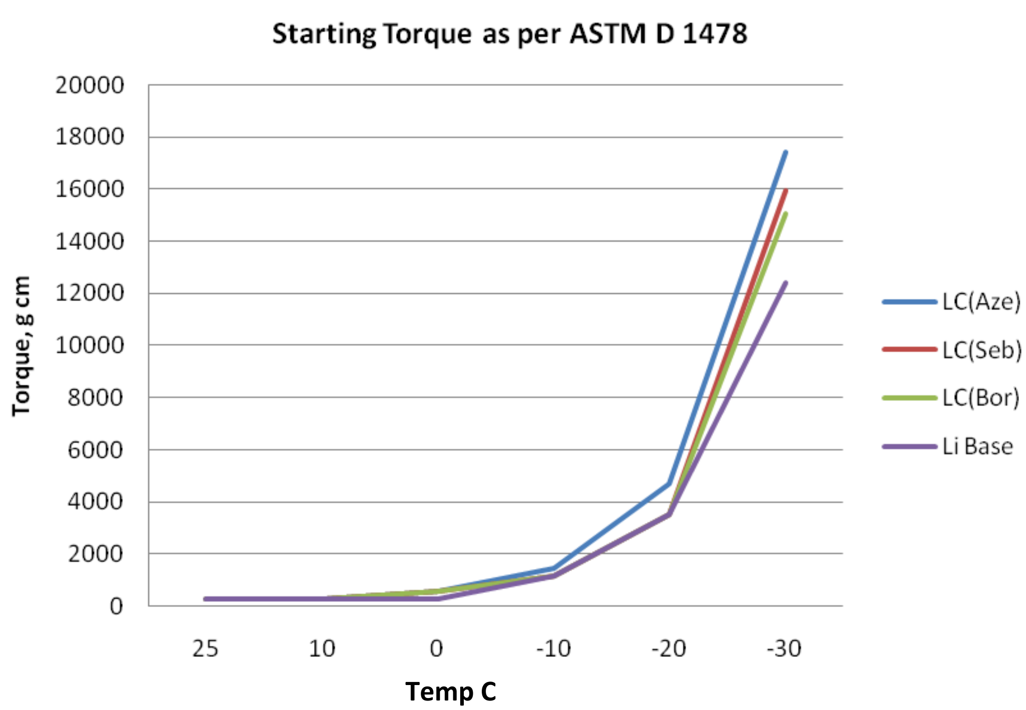
Starting Torque as per ASTM D 1478
Torque, g cm
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
25
10
0
-10
-20
-30
Temp C
LC(Aze)
LC(Seb)
LC(Bor)
Li Base

Fig-4

5.1 Scanning Electron Microscope (SEM) Study

To investigate the grease soap fiber structure, size and morphology, SEM studies were performed on these grease samples using HITACHI S-3400N Scanning Electron Microscope at room temperature. Film of a speck of grease sample was made on a micro-slide by means of a spread rod and this film was rinsed with 70:30 v/v, hexane/toluene mixture to remove the oil completely from the soap fiber matrix. The slide was then dried by gently sweeping dry air with the aid of air blower and observed the structure under scanning electron microscope.

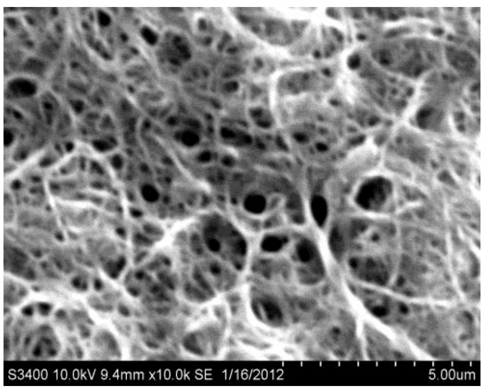

Fig-5 Li Base grease

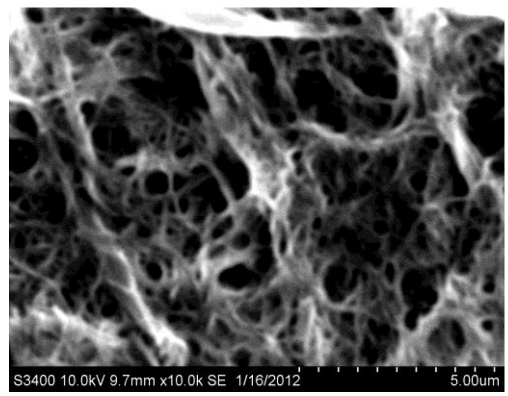

Fig-6 Li Complex with Boric acid

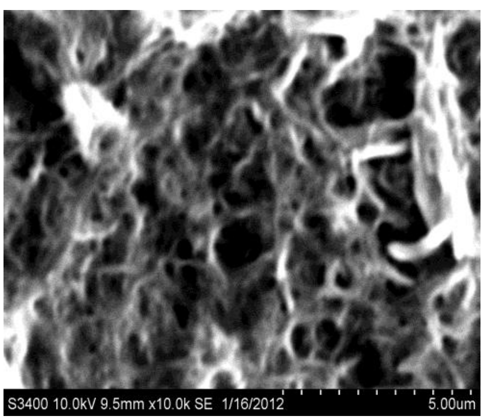

Fig-7 Li Complex with Azelaic acid

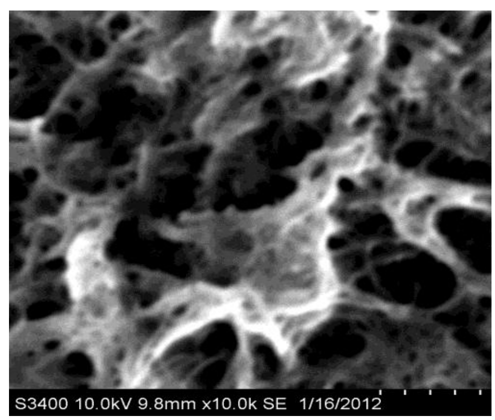

Fig-8 Li Complex with Sebacic acid

6.0 RESULTS & DISCUSSION

The flow pressure studies of lithium base and lithium complex greases show some interesting trend. The flow pressure behavior of the four greases at subzero temperatures is Li < LC (Boric acid) < LC (Sebacic acid) < LC (Azelaic acid). Same trend is observed in the starting torque also as determined by ASTM D 1478. More or less, same trend is also seen in oil separation study conducted as per ASTM D 6184 at

100°C. The flow pressure values of all except lithium complex with azelaic acid are below 1400 hPa even at -30°C, indicating that these greases posses adequate pumpability and can be safely used in centralized lubrication system.

In the SEM study structure of Li-base grease was found to have short to long fiber, twisted and tangled [12]. The fiber characteristics of boric acid based thickener appear similar to that of simple lithium grease to some extent but is finer compared to sebacic & azelaic acid based greases. The three dimensional network of all three complex greases are seen superior in comparison to lithium grease. But boric acid based complex grease is comparatively more loosely packed than sebacic and azelaic acid based greases.

Structure of sebacic and azelaic acid base Li-complex thickeners appear to have almost similar network having cross linkage mechanism. The fibre network of sebacic and azelaic acid based complex greases are seen aligned and grouped less orderly giving more complex packing, which could be the probable reason for their better oil retention and high dropping point. Irregular fibre networking is expected to negatively affect the flow characteristics and pumpability of greases. Among sebacic and azelaic acids, networking is more closed with less cavities in case of azelaic acid based thickener favouring better oil holding and rigid network affecting flow properties.

7.0 SUMMARY

- Upto 0^0C flow pressure and low temperature values were found almost similar, indicating that complexing agents have little significant role on flow properties of greases.
- In case of azelaic acid, flow pressure was found increasing significantly compared to others, when temperature was further lowered.
- At -30^0C the flow pressure appears to be almost similar in case of sebacic acid and boric acid based greases.
- Huge jump in both flow pressure and torque was seen when temperature was lowered from -20 to -30^0C.
- Test datas indicate that the greases may be safely handled ~ 15^0C below the pour point of base oils used.
- The better confinement of oil in closed fibre net work retarding wax precipitation or lowering of pour point, probably is the reason why grease gives better flow properties than the oil used in processing the greases.
- Structure of Li-soap indicates finer networking in SEM photograph. The finer and less orderly fibre net work may be reason for lithium grease to perform better in flow property compared to complex greases, even though same oil was used in all the four greases.

8.0 CONCLUSIONS

- Li and Li-Complex grease with boric acid behave better in low temperature properties compared to other complex based greases.
- Sebacic and azelaic acid base thickeners behave similar in their structure pattern because of their close C8 & C9 chemistry, but sebacic acid based grease exhibited low flow pressure and better low temperature torque behavior compared to azelaic based grease.
- Studies indicate that greases may be safely handled ~15 ^{0}C below the pour point of base oils used.

Acknowledgement

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