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# The Grease Life Factor concept for ball bearings

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ABSTRACT

Bearing fatigue life is calculated using a so called basic rating life equation $L_{10} = \left(\frac{C}{P}\right)^p$ where the operating load is translated into an equivalent load $P$ and divided by the bearing capacity. This makes it possible to test the bearing capacity $C$ under an arbitrary load and therefore on almost any test rig. For grease life such a concept does not exist. Grease life is specified by a number of hours running on standardized test rigs and conditions. A similar concept as for bearing life is introduced in this paper for lubricating grease, called the 'Grease Life Factor' (GLF) concept. It is based on a grease life model for axially loaded grease lubricated ball bearings and makes it possible, similar to bearing life, to quantify the life properties of a grease using 'any' grease life test rig. It also implies that there is no longer a need to specify grease life for standardized test rigs. The validity is illustrated with test data obtained from the most widely used test methodologies: FE9 and R0F.

## 1. Introduction

The main performance parameter for lubricating greases is grease life. For sealed-for-life bearings grease life usually determines bearing life. In the case that relubrication can be applied, grease life is used to calculate the relubrication intervals [13]. Grease life performance is measured using test rigs where a multiple of greased bearings are run to failure. Weibull statistic is then used to calculate a percentile life, usually the life where the probability of failure is 10%, $L_{10}$ or 50%, $L_{50}$.

Unfortunately, since there are so many lubricating greases, grease manufacturers and bearing manufacturers cannot afford to use a very large test population. There is a consensus that a grease life test should have led to at least 5 failed bearings.

Grease life is always specified as the number of hours to grease failure (with a specified probability) where bearings have been running under specified conditions such as described in standards such as DIN [3, 4] using FE9 test rigs or ROF [15]. Having such a number is good for quality checks but does not quantify the grease life properties in general. Moreover, testing on other rigs and other conditions will give different numbers for the same grease. In this paper we will use a model to show that the grease life capability can be captured into an 'Arrhenius Temperature', which denotes the impact of temperature on grease life and a 'Grease Life Factor", which is a measure for the capability of a grease to give a long life at a particular temperature. These can be measured on any test rig. It will be shown how the results from any test can be used to generally quantify the life properties, which should be part of the grease specifications. The approach shows similarities with to what is used for quantifying the bearing life capacity of a bearing.

Bearing fatigue life is quantified by the bearing capacity $C$ and bearing life can then be calculated by using the $C/P$ concept where the axial and radial loads are transfered into an equivalent load, which is a measure of the stress condition in the bearing. This concept was originally developed by Lundberg and Palmgren [18,19] and is still applied today and standardized in ISO 281 [10]. A bearing that gives a long life will have a high capacity, i.e., a high $C$ value. According to this standard, bearing rating life for ball bearings reads $L_{10} = \left(\frac{C}{P}\right)^3$, expressed in number of rotations. Hence, the bearing capacity for a specific bearing can be measured under any (reasonable) load and speed. Such a model does not exist for lubricating greases. Other than for bearing life, grease life can so far only be expressed as a life obtained in a specific test. In this paper the 'Grease Life Factor' (GLF) concept is introduced, a concept that is essentially the same as for bearing life. It will be shown that the same number can be measured on any of the commonly used test rigs for

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grease life such as R0F, R0F +, FE9 or the ASTM wheel bearing test. It is important though that the bearings in the various tests are operating under the conditions for which the grease was designed. Therefore we will first give a description of these. Next, we will go deeper into the analysis of grease life tests. We will then describe the most commonly used test methods. A major part of this paper is dedicated to a description of the grease life model and how it is derived. We will then illustrate the GLF concept by using results from the two most widely used test methods that are using ball bearings with different size, load and temperatures. Here two greases are used, one with a lithium soap thickener and mineral base oil, denoted by Li/M and the other with a polyurea thickener and mineral base oil, denoted by PU/M.

## 2. Lubricating mechanism

A properly lubricated bearing will have a long life because the rolling elements (balls or rollers) are (at least partly) separated by a lubricant film. This film is generated by hydrodynamic action between the elastically deforming bodies and the mechanism is therefore called Elasto-Hydrodynamic Lubrication (EHL). Initially macroscopic grease flow will occur inside the bearing (churning). Grease is a semi-solid material and, depending on the bearing speed, a channel of grease is formed within a very short time (channeling) followed by a longer period (several hours) where the raceway is cleared from grease (clearing) [2,7,9]. During this "churning phase" [13] fully flooded conditions prevail, meaning that the inlet of the contacts are full with lubricant and that the film thickness is equal or higher than values predicted with equations as given by Dowson and Hamrock [6] or Nijenbanning et al. [20]. However, after this short churning phase starvation will occur, where the film thickness is not only determined by the rheology of the lubricant but also by the availability of it. Thickener material may well still be in the contact but the feed after longer times is primarily given by oil bleed from the relatively stationary grease (also called 'reservoirs for oil'). Oil is lost by evaporation, oxidation and leakage. The availability of (primarily) bled oil on the running tracks of the bearing will slowly decrease leading to thinner films and ultimately seizure of the bearings, an effect that is accelerated by a potential loss of "lubricity" by oxidation. Shear will lead to mechanical degradation leading again to a change in bleed properties and rheology. When bearings are running under these conditions it is possible to predict grease life. There can be various factors that will reduce grease life such as contamination by water, air flow through the bearing, vibration and many more but this complexity is not addressed here because these will not occur in grease life tests.

## 3. Grease life

The end of grease life in a rolling bearing is determined by the point in time where the grease is no longer able to lubricate the bearing because of the above mentioned mechanisms. If oxidation would be the only degradation mechanism then grease life would be deterministic. The process becomes more random in the case that also other degradation mechanisms are significant. This is ascribed to the non-uniformity of grease flow caused by the non-linear rheology and flow in 3 phases (air, grease and bled oil), leading to a non-uniform shear distribution onto the grease and therefore non-uniform degradation. Particularly differences in starting conditions such as minute variations in initial plays a role [17]. To properly measure grease life one would therefore need to run multiple bearings to grease failure. From this a certain life percentile is calculated, such as $L_{50}$ [13].

A grease/bearing should operate for a very long time under the application conditions for which it was designed or selected. This would lead to unacceptable testing times and therefore, almost always, accelerated testing is applied. This means that testing is done at very high temperatures (but not too high as will be explained later) after which the results are extrapolated to 'real life'. This can obviously only be done if the test conditions lead to the same failure mode as expected in real life.

If grease failures are caused by oxidation only, which is deterministic, than this would lead to a very small spread in life (high Weibull slope). In that case $L_{10}$ is reasonable accurate. At realistic temperatures the spread is larger because the failure is also caused by other 'degradation mechanisms'. The test accuracy is higher for $L_{50}$ than for $L_{10}$. It also increases with an increase of the test population. Since a test usually consist of only 5 failures, $L_{50}$ is used as a measure of grease life in a test. In the figures in this paper $L_{50}$ is plotted including its 90% confidence intervals. For more details on the evaluation of grease life test data the reader is referred to [13]. Other life percentiles can subsequently be calculated by assuming a particular Weibull slope. For example Huiskamp [8] used a value of $\beta = 2.3$, giving for the most commonly used life percentile $L_{10}$: $L_{10} = L_{50}/2.7$.

## 4. Various test rigs and methods

The FE9 test rig has been standardized in DIN 51 821 [3,4]. Following [4], in this test angular contact ball bearings 7206 are used, running at 6000 rpm under a pure axial load $F_a = 6000$ N and temperature $T = 140\ ^\circ$C. However, very often higher temperatures are applied. There are two variants of this test: open bearings (variant A) and shielded bearings (variant B). In this paper only shielded bearings are considered because this is the most widely used execution but also because it is not common to use open bearings in grease life testing. For an excellent paper on this test methodology the reader is referred to Kleinlein [12].

The R0F test rig has not been standardized but is the second most used grease life test rig. This test rig has been upgraded into the R0F + [14,15]. In the R0F, standard deep groove ball bearings, 6204-2Z/C3, are used, loaded under an axial load $F_a = 0.1$ kN and radial load $F_r = 0.05$ kN. In the R0F + configuration the loads are not fixed. They can be selected up to a radial load of 0.9 kN and axial load 1.1 kN. The R0F + can also run under R0F conditions and these are therefore "standard" in this rig as well. Similar to FE9, the rig can run under different (controlled) temperatures. In both R0F(+) and FE9, in a single test 5 bearings are run to failure. In the R0F(+) there will be 5 additional suspended bearings.

The ASTM D3336 [1], also called the POPE test, is quite similar to the R0F. A similar bearing as in the R0F is used: 6204 and also the speed is similar: 10,000 rpm. The axial load is $22 < F_a < 67$ N. The radial load is $F_r = 67$ N. Unfortunately in the ASTM test protocol the conditions are varied in time. Moreover, a very large quantity of grease is used (3.2 cm$^3$ versus about 1.4 cm$^3$ in the R0F). This test is therefore not considered as applicable for this paper.

Running-in will have an effect on grease life. Some greases are more sensitive to that than others. This is related to the churning/channeling/clearing properties of a grease. The running-in processes are different for the different test methods. In all test methods the test bearings are brought up to the test speed very quickly. There is only a difference in heating up the bearings during running in. The test speeds are moderate and it is therefore not expected that these differences in running-in procedures will lead to a significant difference in grease life.

## 5. Impact of grease volume

Grease life is proportional to the volume of grease that is available [5,11,13]. A rule of thumb is to fill a bearing with about 30% of it free volume with grease. The free volume is the volume of the bearing minus that of its steel components [21] (with density $\rho = 0.0078$ kg/m$^3$):

$$V_b = \frac{\pi}{4} B\left(D^2 - d^2\right) \times 10^{-3} - \frac{M}{0.0078} \tag{1}$$

where $D$, $d$ and $B$ are the outer diameter, inner diameter and width of the bearing (in mm). $M$ is the mass (kg). Filling the bearing with a grease volume $V_g$, gives then for the filled fraction:

$$f = 1 - \frac{V_b - V_g}{V_b} \quad (2)$$

In this equation, for bearings with polymer cages, the volume of the cage is neglected. However, this is considered to be not only small but also similar for all bearings. The model was developed using R0F + tests using 6204-2Z deep groove ball bearings filled with 1.5 ml of grease corresponding to $f = 0.27$ and is therefore the reference for obtaining a GLF = 1 with the reference grease. The FE9 test bearings (7206, D = 62, d=30, B = 16 mm) are filled with 2 ml, which corresponds to $f = 0.18$. The test result should therefore be multiplied with a factor 0.65. The angular contact bearings 7204 that were used in the R0F + have similar outer dimensions as the 6204 deep groove ball bearing. However, more grease was used here: 2.3 ml. This corresponds to $f = 0.4$ filling and it can therefore be expected that the life with the reference grease will be a factor 1.7 longer. So here the tested life will be multiplied with a factor 1.7.

## 6. Grease life model

### 6.1. The green temperature window

Every lubricating grease has been designed to operate safely in a certain temperature window. The Low Temperature Limit (LTL) is given by the temperature where the stiffness of the grease is so high that there is a risk that when starting up the bearing the rolling elements will not rotate leading to skidding and bearing damage. The High Temperature Limit (HTL) is given by the point where the thickener will irreversibly loose its structure. The window in which grease life can be predicted is narrower thought. Grease life can only be predicted if the lubrication mechanism follows the physics for which the models have been developed (the grease should not be 'overheated'). The generally accepted mechanism for continuously rotating bearings was described above. This means that there should be oil separation from a grease matrix that is formed by a relatively stable reservoir (on the bearing shoulders and/or under the bearing cage). The Low Temperature Performance Limit (LTPL) is then given by the temperature where oil separation practically stops and the High Temperature Performance Limit (HTPL) by the point where the grease matrix looses it ability to keep oil in its structure or where the grease becomes so soft that excessive leakage occurs. This temperature window is typically 80˚ C wide for Lithium soap thickened greases. Note that bearing friction is relatively high at very low temperatures due to the increase in base oil viscosities with decreasing temperature and that the self-induced temperature is easily 50 °C. Hence, a bearing operating at very low temperatures will usually run only for short times under these conditions.

The temperature window between LTPL and HTPL is generally denoted as the "green zone" where grease life can be predicted based on mechanical and chemical degradation of the grease. In this zone grease life follows the Van 't Hoff or Arrhenius behavior, conveniently written as:

$$L_{50} = L_{50,r} \left(\frac{1}{2}\right)^{\frac{T-T_r}{T_a}}, \quad (3)$$

where $L_{50}$ is the time where the probability of failure is 50%, $L_{50,r}$ is $L_{50}$ at temperature $T = T_r$ and $T_a$ the Arrhenius temperature. $T_a$ is typically $T_a \approx 15$ ˚C [13,21], which means that grease life is reduced by a factor 2 with every 15 °C temperature increase (in the green zone).

This "green zone" temperature window should be considered when testing a grease. Testing is almost always done under accelerated conditions, which should be not so extreme that the lubrication mechanisms are different from those under which the grease operates in "normal" conditions. Hence, grease life testing should never be done outside the green zone. There are no methods available to determine the HTPL other than by doing grease life tests where the temperature in successive tests is increased until grease life deviates from Eq. (3). This is very unpractical/expensive because it will require multiple grease life tests with a large number of test bearings. Therefore in practice "rules of thumb" are used [13].

### 6.2. Varying temperature

The Arrhenius temperature from Eq. (3) can be calculated from two R0F tests with lives $L_{50,1}$ at $T = T_1$ and $L_{50,2}$ at $T = T_2$. Then

$$T_a = -(T_1 - T_2)\frac{\ln 2}{\ln L_{50,1} - \ln L_{50,2}} \quad (4)$$

Fig. 1 shows the impact of temperature on grease life for the Li/M grease. Here a best fit gives $T_a = 12$ °C. For the other grease PU/M, applying Eq. (4) to the FE9 test results in Table 1, gives $T_a = 19$ °C. The figure also show the bandwidth of the test results: $0.7 < L_{50} < 1.4$. The 90% confidence intervals are skewed towards longer lifes. This is caused by including the suspended bearings in the calculation of the expected $L_{50}$.

This figure also gives the 90% confidence intervals in the tests. It illustrates the accuracy of the tests.

### 6.3. Varying load

Fig. 2 shows the impact of load as measured on the R0F + test rig. These measurements show that there is an exponential relation between grease life and the dimensionless load:

$$L_{50} \propto \left(\frac{C}{F_a}\right)^{0.73} \quad (5)$$

### 6.4. Varying speed

To show the impact of speed, grease life was measured for R0F conditions and plotted in Fig. 3 for $T = 130$ °C. The figure shows that grease life is almost inversely proportional to speed. A best fit gives

$$L_{50} \propto \frac{1}{n^{1.09}} \quad (6)$$

It is common practice to express speed in rolling bearings as the circumferential speed $nd_m$, i.e., the product of rotational speed and the mean diameter of the bearing (the mean of bore and outer diameter). After normalizing the speed to a typical speed $nd_m = 33{,}500$, giving a

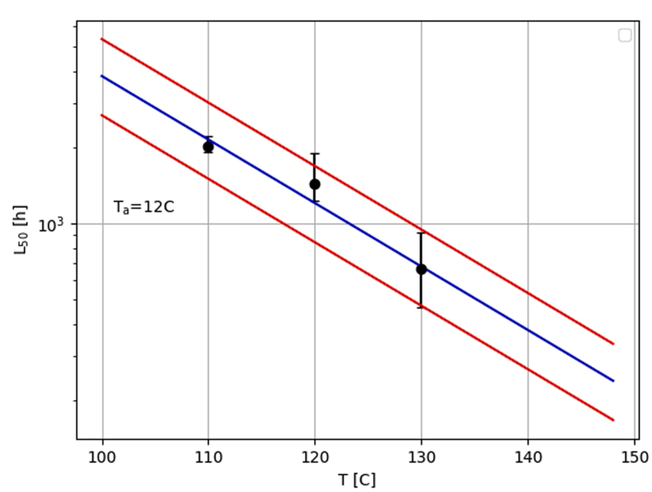


**Fig. 1.** Grease life versus temperature for the Li/M model grease. Test data shows the 90% confidence intervals. The blue line is a best fit through the test data. The red lines represent a bandwidth $0.7 < L_{50} < 1.4$, showing the accuracy of the tests.

**Table 1**
FE9 test result (variant B), ndm = 276,000, C = 22.5 kN. Correction factor for filling is 0.7. So average corrected GLF for Li/M grease is 1. Average corrected GLF for PU/M grease is 5.

| Grease type | Temp, °C | $L_{50}$ | GLF |
|---|---|---|---|
| Li/M | 120 | 395 | 0.9 |
| Li/M | 120 | 474 | 1.0 |
| PU/M | 160 | 384.8 | 5.3 |
| PU/M | 180 | 185.7 | 6.4 |

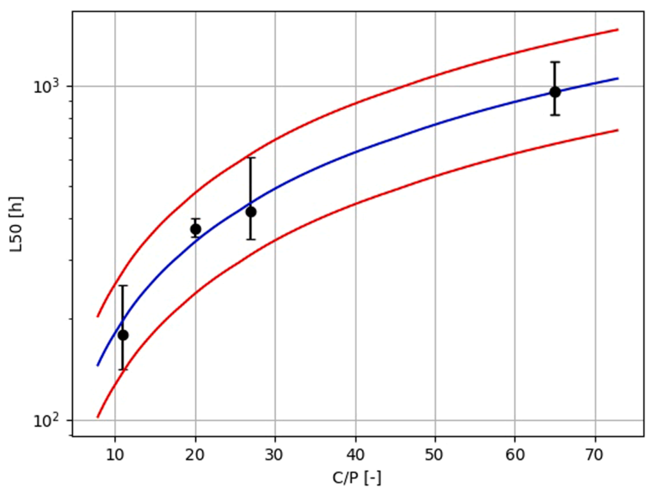


**Fig. 2.** Grease life versus pure axial load for the Li/M grease. Temperature $T = 130$ °C. The red lines represent a bandwidth $0.7 < L_{50} < 1.4$, showing the accuracy of the tests.

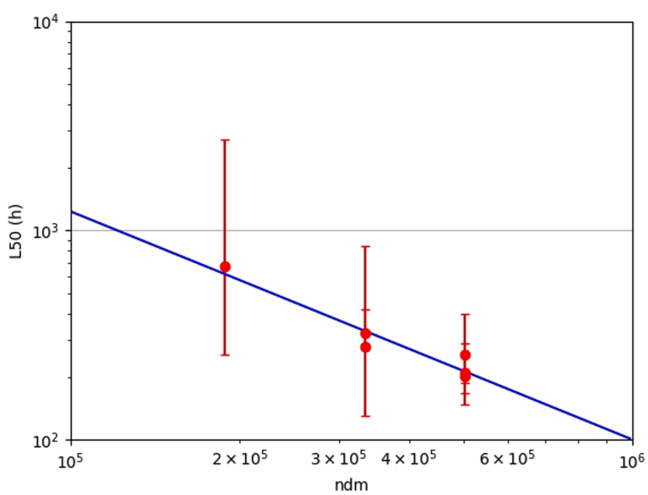


**Fig. 3.** Grease life versus speed for R0F conditions with the Li/M grease, T = 130 °C. The blue line is a best fit.

speed parameter in the order of 1

$$L_{50} \propto \frac{1}{n_R^{1.09}} \quad \text{with} \quad n_R = \frac{nd_m}{33500} \tag{7}$$

### 6.5. Grease type

The impact of grease type on grease life is given by the Grease Life Factor *GLF*. A standard good quality lithium grease is the reference and has a $GLF = 1$. A grease that will give double life (in the "green temperature window" at the same temperature, speed and load) has a $GLF = 2$, etc.

### 6.6. Grease life equation

Combining the effect of load, speed, filling and temperature then gives a grease life equation *for axial load*:

$$L_{50} = GLF \times 52f \left(\frac{C}{F_a}\right)^{0.73} n_R^{-1.09} \left(\frac{1}{2}\right)^{\frac{T-120}{T_a}} \tag{8}$$

## 7. Test results for the reference greases

The reference grease has been tested on both the R0F and the FE9 test rig. The results are shown in Tables 2 and 1.

The results from Tables 2 and 1 show that the Grease Life Factor (GLF) is similar for the test results obtained from R0F and FE9. The Li/M grease has also been tested on R0F + with pure axial load. These results were already shown in Fig. 2 and also here the same GLF was obtained. Tables 3 and 4 shows the average GLF for both greases. The GPF for Li/M grease obtained from both the R0F test and FE9 are GLF = 1. For the PU/M this is 5 for both tests methods.

## 8. Discussion

The performance of a lubricating grease for example an electric motor bearing is measured by its ability to give a long service life or high durability. This is generally specified as a certain percentile life obtained from a specific test rig under specific conditions. However, the conditions under which the grease was tested is never the same as in the application. Different test methods and rigs have been developed over the last 50 years which have even led to different standards. There has been much discussion on the reliability of the various test methods since the results do not seemed to be consistent. This paper shows that the grease life results from different methods should not be presented in hours but in terms of performance compared to a reference. This performance parameter should obviously be a true grease property and no depend on bearing size, temperature and speed under which it has been measured. Moreover, this performance factor should be such that it can be used to predict the performance of the grease in the application, such as an electric motor as well. This is done here by using a grease life model for pure axial load. In this paper we have shown that the grease life performance can be expressed by a 'Grease Life Factor', GLF, which applies to any test rig or test condition. The parameter is defined as:

$$GLF = \frac{1}{52f} L_{50} n_R^{1.09} \left(\frac{C}{F_a}\right)^{-0.73} 2^{\frac{T-120}{T_a}}, \tag{9}$$

where $C/F_a$ is the normalized load, $n_R$ the normalized speed and $T_a$ an Arrhenius temperature. This equation applies to a ball bearing that is filled for 27% of the free volume calculated using Eq. (1). There may be some deviation due to the fact that the volume of the cage was not included and that different test methods use different bearings with different cage types with different volumes. However, this effect is

**Table 2**
R0F test result. Average GLF for Li/M grease is 1. Average GLF for Pu/M grease is 5.

| Grease type | ndm | T | $L_{50}$ | GLF |
|---|---|---|---|---|
| Li/M | 502,500 | 130 | 889 | 1.3 |
| Li/M | 502,500 | 130 | 665 | 1.0 |
| Li/M | 502,500 | 130 | 549 | 0.8 |
| Li/M | 335,000 | 130 | 1030 | 1.0 |
| PU/M | 335,000 | 150 | 2270 | 4.9 |
| PU/M | 335,000 | 150 | 3624 | 7.8 |
| PU/M | 502,500 | 150 | 1387 | 4.6 |
| PU/M | 335,000 | 150 | 3184 | 6.8 |
| PU/M | 335,000 | 150 | 2916 | 6.2 |

**Table 3**
ACBB test result for R0F +, C= 13.3 kN. Correction factor for filling is 0.6. So average corrected GLF for Li/M grease is 1. Average corrected GLF for Pu/M grease is 3.

| Grease type | ndm | T | $L_{50}$ | GLF |
|---|---|---|---|---|
| Li/M | 502,500 | 130 | 701 | 1.2 |
| Li/M | 502,500 | 120 | 1113 | 2.4 |
| Li/M | 502,500 | 110 | 1767 | 2.1 |
| Li/M | 335,000 | 130 | 912 | 2.8 |
| PU/M | 502,500 | 120 | 1976 | 5.4 |

**Table 4**
Final Grease Life Factors obtained on different test rigs.

| Grease type | R0F +/DGBB | R0F +/ACBB | FE9/ACBB |
|---|---|---|---|
| Li/M | 1 | 1 | 1 |
| PU/M | 5 | 3 | 5 |

assumed to be small compared to the accuracy of the tests. Of course it is important that tests are executed under conditions that the grease is not ‘overheated’. Hence, at temperatures that do not exceed the HTPL, speeds that do not exceed the reference speed of the bearings, loads that are not so high that failures are not caused by grease life but by bearing fatigue life.

The accuracy of the GLF depends on the test conditions and the test rig that is used. If the tests are done at a temperature of 120 °C then the selected value for the Arrhenius temperature $T_a$ is not relevant and the GLF will be ± 0.5 for the R0F + and FE9 test rigs. For test temperatures deviating to this the accuracy may be estimated on ± 1. The accuracy can be increased by increasing the number of tests. Remarkable is the deviation that we observed with testing the PU/M for the ACBB tests on the R0F + rig. We do not have an explanation for this because the results with Li/M grease on ACBB in R0F + are similar to what we measured on DGBB in R0F(+) and FE9. This may be due to a variation in batch quality. The model is an engineering model based on DGBB tests and the reliability for applying it to ACBB is therefore somewhat uncertain, although it works well for the FE9 bearings/test. It is therefore recommended to develop a grease life equation, similar to Equation (4), for ACBB.

A similar analysis can be made for test rigs that are primarily using radial load [16]. The concept can also be used to determine the high temperature performance limit of a lubricating grease. If the temperature exceeds this limit then the GLF as measured according to Eq. (9) will start deviating from the value found at lower temperatures. It is expected that the lubrication mechanisms will be different for very low and/or high speeds and for large bearings. However, this uncertainty does not apply to grease life testing since this always done at moderate speeds and medium or small bearings.

## 9. Conclusions

The work in this paper shows that grease life specifications can be captured in only two parameters: a grease life factor ‘GLF’ and Arrhenius temperature $T_a$. The Arrhenius temperature is typically $10 \leq T_a \leq 20$. The grease life factor is the main factor that determines the quality of a lubricating grease, where $GLF = 1$ for a good quality Lithium grease. Grease life is proportional to this factor. This concept is applicable to any test rig or condition provided that the test is done in the temperature range for which the grease was designed, i.e., $LTPL \leq T \leq HTPL$, speeds that are lower than the reference speed, bearing sizes comparable to those that have used in this paper and loads that are such that failures are related to grease failures rather than bearing fatigue life failures. This factor should replace the grease life specifications, which are generally expressed in a number of hours in a specific test rig. This factor can be measured on any test rig with ball bearings, under primarily axial load, or even any electric motor, where the load (only axial load), speed and temperature can be fixed.

## Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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