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A New Theory of Damage Estimation and Fatigue Life Prediction

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Abstract: There are a considerable number of fatigue damage estimation theories and fatigue life prediction of mechanical components. The most popular one is Palmgren-Minor (P-M) theory. This theory has been used in the standards for selecting the bearing –as a component subject to fatigue loading- and for expecting the bearings lives. In Wind Turbine Gearboxes (WTGs), the bearings were selected to be without maintenance for 20 to 25 years; however, in real service life, the bearing suffer from premature failure within a life span of quite less than the design life (1 to 5 years). A new applicable methodology and a procedure of calculation for damage estimation due to fatigue loading and predicting the life has been suggested and tested. Results of 20 rolling and sliding tests which conducted under severe contact loading are used to test this method. The suggested method depends on calculating the number of operating cycles under a specific contact loading level to an equivalent number of loading cycles under the average loading level. This method depends on the area under the S-N curve without any correction or loading factors and can be used to predict the WTG bearings failure to manage the maintenance because the current life prediction standards have very high percentages of error (> 400%). The reliability of this approach can be further verified by utilizing actual operational data from Supervisory Control and Data Acquisition (SCADA), used for overseeing wind turbine operations. Additional examinations are necessary to confirm the dependability of this novel method.

Keywords: Fatigue life prediction, Damage estimation, Fatigue test, bearing, Premature failure

1. Introduction

Over the past century, numerous models for predicting fatigue life have been proposed. However, no model has been able to accurately expect the operating life of bearings, particularly for those like Wind Turbine Gearbox Bearings (WTGBs) which experience varying load conditions throughout their use. As a result, unexpected failures cause disruptions due to the immediate need for specialized, expensive maintenance equipment. This significantly hikes up the cost of wind energy and discourages investment in the wind energy field. The standard ISO/ICE 8400-4:2005 [1], presents in Appendix H a gear contact life prediction as a fatigue issue. This standard relies on a methodology that estimates damage using the linear Palmgren-Minor (P-M) theory. This theory is solely reliant on the proportion of cyclic loading to the fatigue failure cycles (n/N) [2]. The life estimation method described in the standard presumes the contact pressures, and the number of rotating cycles they undergo at all loading stages until failure, are known. However, it is unknown what the future loads and their associated cycles will be in actual scenarios. Also, it's necessary to utilize the "application factor," which embodies the proportion of equivalent to nominal torques. This, too, remains unknown given the unpredictable nature of future loadings. Besides, the slope of the S-N curve is employed in the process of damage estimation [2]. This incline indicates the changing pattern in the number of cycles to failure under various loading stages [1]. Hence, utilizing this slope in conjunction with the number of cycles prior to failure does not carry any physical significance and appears irrational. The influence of transient operating events in a Wind Turbine (WT) on bearing life contributes significantly to fatigue failure due to elevated loading levels, exceeding the design stress. Consequently, the impact of these operating episodes should be accurately assessed and deliberated upon in order to precisely anticipate the fatigue lifespan. The margin of error in predicting the lifespan of WTG bearings using standard methods can result in approximately a 400% error, leading to premature failure [3][4].

A review of relevant topics in theories predicting fatigue life based on stress might be crucial to underscore disparities between fatigue damage assessments and life prediction. Various models and theories are available that are designed to estimate the fatigue lifespan of mechanical components and bearings. **Table 1** depicts some of the theories based on stress that are used to calculate fatigue damage (D) [5][6]. **Figure 1** presents cases the procedure of damage accumulation by using five damage estimation methods, which are influenced by the cyclic ratio. This illustration is derived from proposing values for the needed variables and assuming various cyclic ratios (n/N). Starkey's nonlinear method gives the lowest fatigue life with (x_k=0.6) compared with the other models. This technique showcases a steady escalation of damage, and the rate of damage decreases as the component degrades - that is, the curve's slope decreases as the n/N ratio increases. This is starkly different from the actual scenario where the majority of components gradually worsen before abruptly failing.

Table 1: Damage estimation using stress-based formulas.

Theory or method	Damage calculation formula
Palmgren – Minor linear Theory (P-M)	$D = \sum_{1}^{k} \frac{n_k}{N_k}$
Marco-Starkey Method	$D = \sum_{1}^{k} \left(\frac{n_k}{N_k}\right)^{x_k}$
Owen and Howe hypothesis [7]	$D = A\frac{n_k}{N_k} + B\left(\frac{n_k}{N_k}\right)^2$
Lamaitre hypothesis	$\mathrm{D} = 1 - (1 - \mathrm{n/N})^{b}$
Datoma et al. theory	$\mathrm{D} = 1 - (1 - (\mathrm{n/N})^{1/(1-\alpha)})^{1/(1+\beta)}$

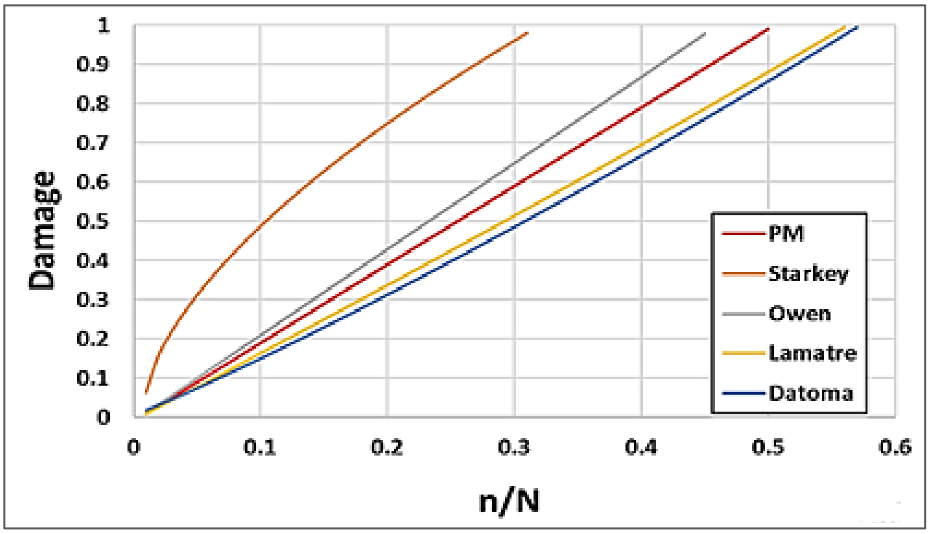

Figure 1: Accumulation damage curves of five stress-based theories and hypotheses.

The referred models may be suitable for particular instances, yet their outcomes do not align with the real-life scenarios of bearings. For instance, the error percentage in the life expectancy prediction of WTG bearings exceeds 400%.

2. Methodology

Fatigue trials involving two cylindrical contact discs were structured to mimic the interaction between the roller and the inner race in bearings. These two discs exert vertical pressure on each other through a specially designed testing rig and the contact force can be gauged with a load cell. The Hertz contact theory was applied to compute the contact stresses, and the superposition method was similarly employed to append 15% of the contact force as a traction force towards the direction of roll. Impact cycles, making up 7% of the rotation cycles, were implemented to depict the intermittent loadings during the operational duration of the bearing. This percentage was drawn from the activities during the actual operation of WTG bearings [8][9]. The service cycle can be determined by the number of cycles it took to inflict damage on the contacting surfaces. Though the methods of calculating contact stress differ between line and elliptical contacts, the life expectancy (measured by the number of cycles until failure) relies solely on the contact stress. Therefore, both elliptical and line contacts utilized curved and flat test discs. Four experiments, labeled as T1P, T2P, T3P, and T4P, were performed utilizing curved contact discs placed on a straight one, (see **Table 2**). The structure of the test rig and the geometry of the test disc are illustrated in **Figure 2**. As fatigue tests tend to be lengthy, accelerated tests can be employed under high contact stress that exceeds σ_{yeild}. These levels of loading are deemed feasible given that the bearings endure elevated stress levels through the unavoidable issue of transient loading.

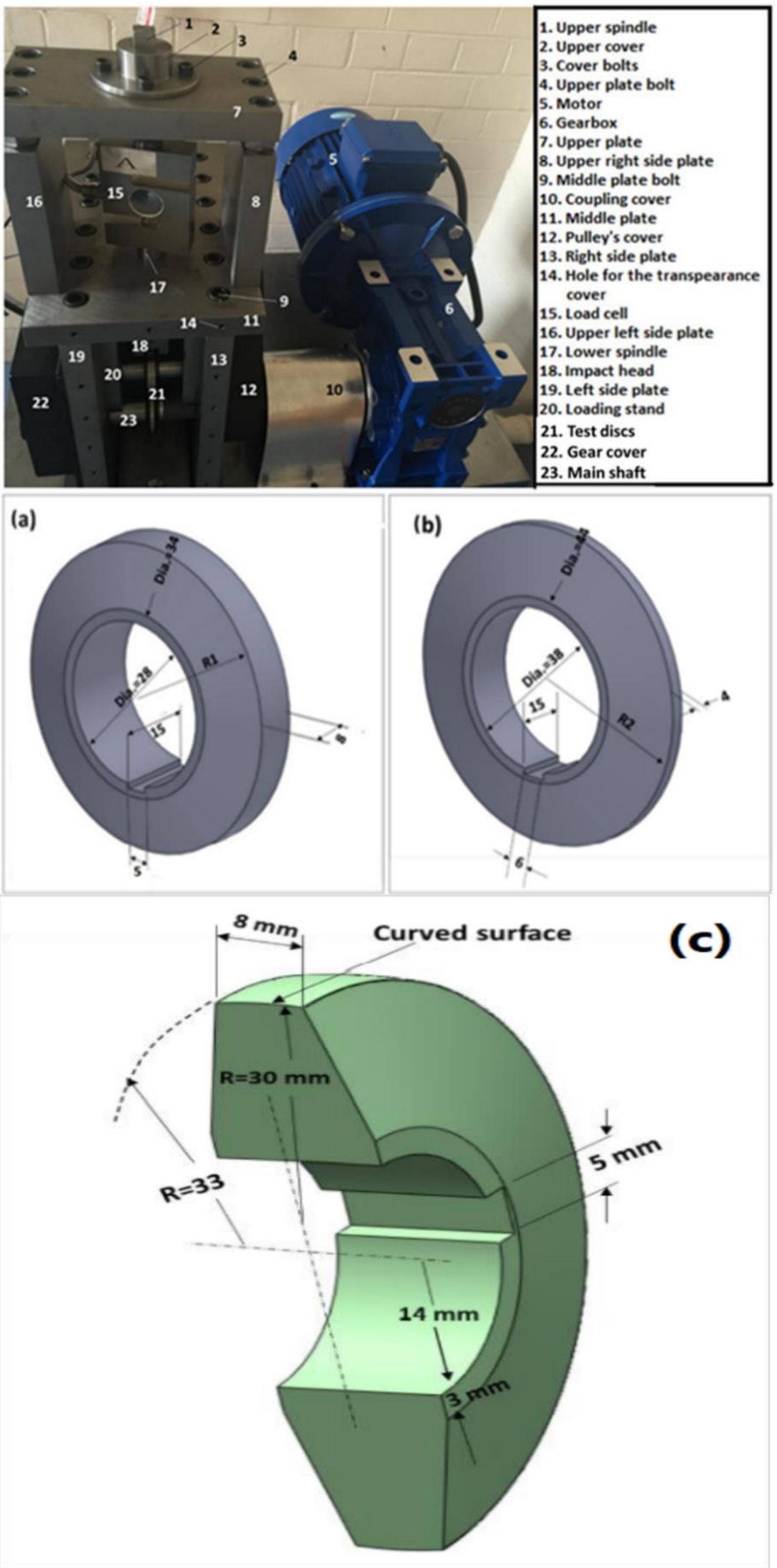

Figure 2: Test rig used (upper) and the test discs (Lower), (a) Lower test disc; (b) flat upper test disc and (c) a cross-section of a curved upper test disc.

3. Results

The findings from the executed fatigue tests are depicted in **Figure3** which presents the S-N curve for the material of the test disc. The precision of this curve may be compromised due to the utilization of the lifespan results of the elliptical and line contacts to compose it; given the differing initiation of damage and its propagation mechanisms these two contact geometries might employ. The S-N curve can be leveraged to estimate the lifespan of the disc based on the proposed theories and hypotheses. In addition, the influence of impact loading, which constitutes 7% of the rotational cycles, has not been accounted for in prior methods of life prediction. The influence of impact can be accounted for by multiplying the stresses due to impact by their cycle count, and adding this to the product of each stress test and its respective failure cycle count. This forms the foundation of a new method of life prediction, which will be elaborated in the subsequent section. By incorporating the impact effect in the administered tests, it is apparent that a linear fit offers slightly higher accuracy in describing the S-N curve compared to curve fitting. It is crucial to note that the S-N curve is pertinent to the specific level of material cleanliness employed (particular dimensions of impurities in the cast material) and should not be generalized for other levels of material cleanliness.

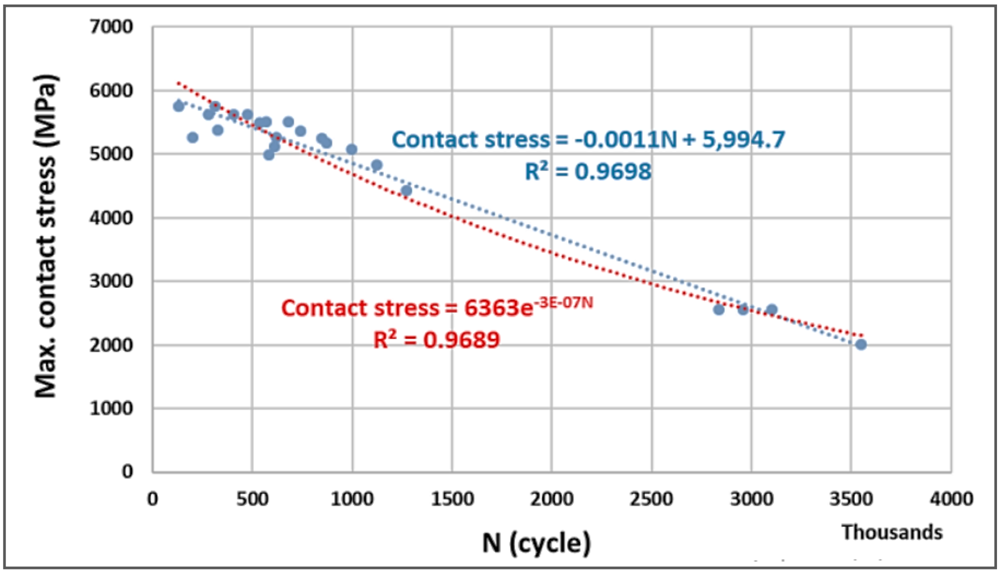

Figure 3: Results of the conducted tests.

4. The New method

A notable discrepancy emerged between the predicted and actual bearing life, particularly when the bearings were exposed to inevitable variable loading (transient). In any mechanical setup undergoing fatigue loading, a portion of the differential between output and input energies is converted into internal energy. This energy is utilized to create microstructural damage, or plastic deformation (provided the stress level is sufficiently high), heat the lubricant or the mechanical parts themselves, and to absorb elastic energy via damping, as depicted in Figure 4. It may be postulated that a certain portion of the energy of loading will be progressively turned into damage through a cumulative process. This type of damage correlates with the stress from loading and its number of loading cycles (rotating cycles) in a gradual accumulation process. Hence, this technique can be termed Energy Fraction of Damage Accumulation (EFDA). It's based on the assumption that with each loading cycle, damage accrues proportionally to its stress level.

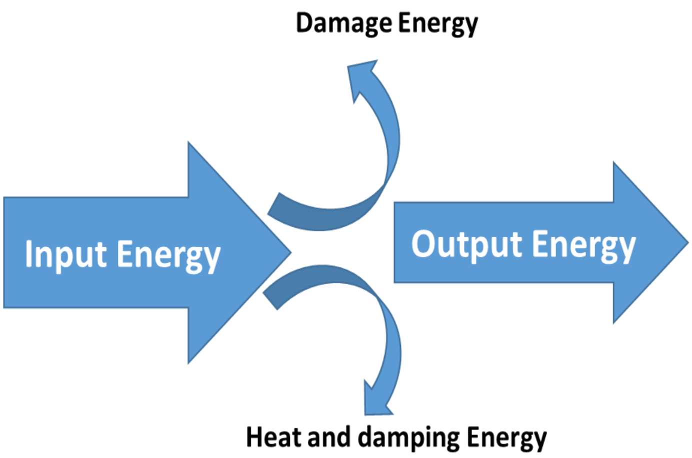

Figure 4: Energy of damage in a mechanical system.

Due to the near-linear association of the S-N curve for the majority of engineering materials, it can be hypothesized that the fraction of damage energy bears a linear relationship with the loading stress. That is, the Damage Energy (DE) is presumed to directly correlate with the loading stress level (σ) and the cycle count (n);

$$DE \propto \sigma.\, n \qquad (1)$$

By assuming the energy fraction transforms into damage throughout the testing is (F1) and the service load is (F2); the energy of damage can be calculated as:

$$DE = F1.\, \sigma.\, n \qquad \text{for the test} \qquad (2)$$

$$DE = F2.\, \sigma.\, n \qquad \text{for the service} \qquad (3)$$

By assuming the conditions of the service and the test are identical, Eqn. 2 matches Eqn. 3, thus F1 is equal to F2. To elucidate this fundamental concept, consider a mechanical component put under a stress level $\boldsymbol{\sigma 1}$ for N1 cycles until failure; the rectangle A1's area depicted in **Figure 5** can be easily computed, and this is proportionate to the damage triggered by this stress level, meaning complete, or 100%, damage. The given number of rotating cycles, N1, can be converted into an equivalent number of cycles (N2) at a different stress level, capable of creating the same degree of damage (100%). If a significant variance exists between the two stress levels, the damage mechanism might differ, thus, the average stress applied can be utilized in the recalculations. By dividing A1 by the new introduced stress level, the equivalent number of cycles (N2) can be determined. The renewed methodology presumes that; the damage induced by these two levels of stress ($\sigma 1$ and $\sigma 2$) and their cycles of rotation is identical since both induce the same degree of damage (100%). Similarly, any contact stress level and amount of rotating cycles that do not yield 100% damage could be converted to a different stress level with an equivalent cycles, adhering to the same procedure as explained. The life expectancy methodology outlined in ISO/ICE 8400-4:2005 standard [1] stipulates that all the loading levels and their cycle counts must be known until failure for the Loading Factor (LF). In context of bearings, where it's necessary to anticipate the remaining lifespan and the load is unknown, it becomes unnecessary to apply this factor.

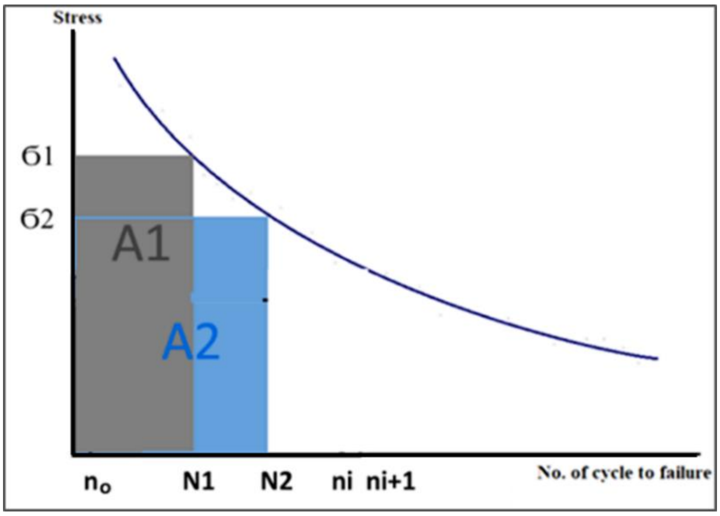

Figure 5: Basic of the new EFDA method.

In the P-M theory (Refer to **Table 1**), if both the numerators and the denominators are multiplied by their respective stresses, the damage hence becomes the ratio of the area's segment beneath the S-N curve (in the numerator) to the complete area under the S-N curve in the denominator. The subsequent process of resolving an instance demonstrates the step-wise nature of the proposed technique for easier understanding. Let's presume a bearing endures three levels of loading stress S1, S2 and S3 and rotates for n1, n2 and n3 cycles respectively. The outlined procedure can be applied:

1) Compute the area beneath the S-N curve for the initial stress level — that is, a_1=S_1×n1— and the ensuing damage can be evaluated by dividing this area by N_1, which represents the entire area under the S-N curve at this stress level (A_1=S_1×N_1). Here, N_1 signifies the cycle count till failure under S1 stress. The damage inflicted at this loading phase equates to that determined by the P-M theory, i.e., $D = S_1 n_1/S_1 N_1$ (D = a_1/A_1 or D= n_1/N_1.
2) Upon the application of a new stress value (S_2), the initial stress level needs to be converted to an equivalent cycle counted under the other stress level. This is because the continuous extension of the second stress level until failure is anticipated, given that future loading is uncertain. This adjustment can be achieved by dividing a_1 by the new induced stress level, i.e.

   $$n_{eq1.} = a_1/S_2$$

3) The updated equivalent number of rotation cycles, referred to as $n_{eq.2}$, is the sum of n_2 and $n_{eq.1}$. Damage (D) is calculated by dividing the new area (determined by multiplying S_2 with $n_{eq.2}$) by the total area under the S-N curve under the average stress level, i.e., $(S_1+S_2)/2$. The average stress level can be viewed as the selected stress level for the bearing, which is the stress level under which bearing manufacturers repeatedly test the produced bearings to determine their failure cycle number and the results of these tests are known to be highly reliable. Consequently, the damage becomes the area resulting from the two stresses ($n_{eq2}\sigma_2$), divided by the entire area beneath the S-N curve relative to the design stress and its corresponding number of cycles to failure.

4) If a new level of stress S_3 and a corresponding number of cycles n3 are introduced, only the second level of stress will be converted into an equivalent cycle count, i.e., $n_{eq.3} = n_3 + (S_2 \times n_2 / S_3)$. The resultant damage is $D_3 = S_3 \times n_{eq.3} / (S_d \times N_d)$, where S_d and N_d represent the stress of bearing selection (or design stress) and the number of cycles that lead to failure under the design stress, respectively. This design stress and corresponding cycle count data is supplied by the manufacturers of the bearings.
5) The damage determined at each stress level and the count of rotation cycles are employed twice throughout this process, so it's necessary to divide the calculated damage by 2. Broadly speaking, the formulas to calculate the equivalent cycles and damage can be written as follows:

$$n_{eqi} = \left(n_i + \frac{S_{i-1} \ * \ n_{i-1}}{S_i}\right) \tag{4}$$

$$D_i = \left(\frac{1}{S_d * N_d} \sum 0.5 * n_{eq\,i} * \ S_i\right) \text{, or} \tag{5}$$

$$D_{i+1} = \left[\sum_i \frac{0.5 * \left(\frac{n_i S_i}{S_{i+1}} + n_{i+1}\right) S_{i+1}}{(S_d N_d)}\right] \tag{6}$$

Where, D is the damage, i the loading counter, S the stress, n the number of rotating cycles and N is the cycles number to failure and the subscript d refers to the designed values. Bearing design stress and the design number of cycles to failure is independent on the counter i, thus, the damage equation can be rewritten as:

$$D_{i+1} = \frac{0.5}{(S_d N_d)} * \left[\sum_{\boldsymbol{i}} \left\{\left(\frac{\boldsymbol{n_i S_i}}{\boldsymbol{S_{i+1}}} + \boldsymbol{n_{i+1}}\right) \boldsymbol{S_{i+1}}\right\}\right] \tag{7}$$

$$\text{Or simply, } D_i = \frac{\sum S_i * n_i}{S_d * N_d} \tag{8}$$

In the case of Wind Turbine Gearboxes (WTGBs), the coefficient of friction is not explicitly defined, and their lifespan generally exceeds that of the test discs. Therefore, AL-Bedhany [8] applied this method using actual wind turbine operating data, gathered by the Supervisory Control And Data Acquisition (SCADA) system to identify contact stresses and the corresponding cycles under various wind speed conditions. Some wind turbines feature a SCADA system to capture and average operational conditions (such as generator power, rotation speed, and speed of the wind) every ten minutes. Consequently, this method will be validated using the experimental data derived from these tests. The percentage error for all test results can be computed using the following formula:

$$Error\ \% = \frac{life\ calculated\ by\ the\ considered\ fitting - Experimental\ life}{Experimental\ life} X\ 100\% \tag{9}$$

The average contact stress applied during testing was 4810.5 MPa and resulted in a failure after 1,065,243 cycles. The estimated disc longevity of each test can be obtained by multiplying these two quantities and dividing the result by the stress of contact at each test level. After factoring in the impact of the coefficient of friction, i.e., multiplying the contact stress by 1.07, the computed values for the average stress of contact and its number of failure cycles are respectively 5147.212 MPa and 757,129 cycles. Following the same methodology to estimate the disk life, the percentage of error of the calculated results using Eqn. 9 is outlined in **Table 2**. Including the effect of friction significantly diminishes the error percentages, suggesting that even if it was not taken into account during the S-N curve sketching, friction likely reduced the error. However, some discs still have a sizeable percentage of error, especially those with relatively shorter life spans (less than the average disk life of 611,340 cycles). Ideally, in any engineering design, the cycle count to failure should be maximized to extend the service life. Hence, it's realistic to anticipate that in real-life application scenarios, the error percentage in this life estimate could be considerably smaller, leading to a more precise life prediction. The high error percentages for several tests can likely be attributed to the following factors:

1. The tests' reliability is low the tests have not repeated for at least three times and calculating the average life. This was due to time facilities.
2. The Stress-Number of cycles (S-N) curve of the test discs was drawn independently on the contact type, i.e. using the line and elliptical tests' results. These two types of contact have different stress states (plane stress and plane strain respectively).
3. The effect of stress concentration makes the initiation of damage being from the contact surface however, the mechanism of subsurface damage initiation probably taking place throughout the elliptical contact.
4. When the life of fatigue is relatively small, the error percentage increases significantly due to considerable fraction of dividing the difference between the estimated and the experimental lives by a relatively small value (See Eqn. 9).
5. Increasing the contact stress by 0.07% by introducing the friction effect will considerably reduce the error percentage. This probably due to the RCF life which already affected by the friction and traction. Therefore, for the S-N curve presented by the stress of fatigue test, should take the friction on consideration for life prediction even when the S-N curve has been drawn by ignoring the friction.

Table 2: Life prediction of the conducted tests and the percentage of error.

Test code	Experimental No. of cycles to failure	Life prediction by EFDA method	% error of EFDA by considering the friction	% error of EFDA without friction
T14	1,120,826	752985	-32.8	-5.5
T13	997,040	717503	-28.0	1.2
T12	871,830	704612	-19.2	13.7
T16	620,430	691423	11.4	56.8
T1	849,893	694656	-18.3	15.0
T10	741,240	678664	-8.4	28.8
T15	567,836	662146	16.6	64.1
T2	584,617	731042	25.0	75.9
T6	612,759	711641	16.1	63.4
T4	537,907	662762	23.2	73.4
T8	406,317	647355	59.3	124.2
T3	325,400	677175	108.1	192.8
T7	312,681	633781	102.7	185.2
T9	279,840	647928	131.5	225.8
T11	680,241	661570	-2.7	36.8
T5	202,700	693037	241.9	381.0
T6B	1,272,750	823062	-35.3	-9.0
T1P	2,837,250	1429198	-49.6	-29.1
T2P	2,959,320	1429198	-51.7	-32.1
T3P	3,548,839	1814762	-48.9	-28.1
T4P	3,102,436	1429198	-53.9	-35.2

5. Conclusions and key findings

A new method of damage accumulation has been proposed and tested by conducting a 20 fatigue tests. The new method suggested a fraction of the loading energy, which is proportional to the contact stress and number of loading cycles are responsible for introducing the damage. The following conclusion can be drawn:

- The stress used in the procedure of calculation could be shear, principal or equivalent stress depending on the type of stress used in introducing the S-N curve of the bearing material or that provided by the bearing manufacturing companies. It is recommended to introduced the S-N curve by conducting the tests under real loading levels which should be close to the average bearing service contact stress. This will increase the results' reliability and also increase accuracy of the damage estimation.
- The new (EFDA) method is simple and applicable. There is no requirement for any exponent or factor to expecting the damage, however, further tests are still required to increase the reliability of this method.
- The bearing material S-N curve is used for comparing the P-M theory with the new method, however, in the suggested method, the S-N curve of the bearing material and the behavior under different conditions of loading, i.e. S-N curve is not important if we know the design stress and its cycles to failure are

known. The required is only the level of contact stress and the cycles' number to failure under this specific level. The other conditions of operating and their cycles to failure can be introduced by dividing the result of multiplying the design stress and its cycles to failure.

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