CHALLENGES OF THE SIZING OF BALLSCREWS FOR THEIR USE IN THE PRIMARY FLIGHT CONTROL SYSTEM OF A HELICOPTER

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ABSTRACT

Aligned with the advancing electrification in aircrafts and helicopters, the substitution of hydraulic pistons by electromechanical actuators is gathering increased attention. Depending on the application, current sizing approaches for the intended rolling contact drivetrain elements are not able to consider all aspects of the present operation conditions. These are in particular high frequent dynamic axial loads in combination with small motions. Aerospace applications furthermore are faced with wide ranges of operating temperatures and strict requirements regarding failure safety.

State of the art sizing approaches only allow a service life calculation regarding the failure mode of fatigue. Since other failure mechanisms, especially tribo-chemical ones, become more likely during the presented operating conditions a new sizing method must be determined that allows the avoidance of wear and therefor enables a modified fatigue life calculation.

Index Terms – helicopter, swash plate, actuation, bearing, ball screw, sizing method

1. INTRODUCTION

The sizing of rolling contact drivetrain elements (RCDEs), like (ball-) bearings or ball screws, presupposes good lubrication conditions and a continuous rotating system. If these conditions are ensured, the RCDE will fail due to the failure mode of fatigue since the surface of the contact partners (here rolling element and raceway) can only withstand a certain number of revolutions before first damages within the subsurface occur.

The relationship between applied load and service life was determined by Palmgren and Lundberg [1], [2] in the middle of the last century and is the basis of today's standard nominal life calculation approaches for both bearings [3] and ball screws [4]. One of the greatest contributions to the nominal life calculation is the introduction of the a_{iso}-factor, which allows the consideration of different lubrication conditions and a fatigue limit within the S-N-curve. However, all approaches are based on the introduced assumptions.

The assumption of a continuously rotating system cannot always be fulfilled. Stops and direction changes cannot be considered in today's sizing approaches. Furthermore, RCDEs are used in positioning systems where they have to face small or almost zero oscillating motions with superposed high frequent dynamic loads.

Since the replacement of conventional hydraulic actuators by electromechanical actuators (EMAs) promises several benefits in terms of efficiency and safety, their market share in industrial products is still increasing. Typical examples for these substitutions are feed axis of milling machines or closing units of (small and medium) presses. In these applications fatigue is not the critical failure mechanism but for example abrasive wear and its resulting loss of

preload, stiffness and accuracy. Because of the oscillating instead of rotating motion, a fatigue life calculation according to [3] and [4] is not sufficient.

The benefits of EMAs do not only support their usage in mechanical engineering applications, but also in aerospace applications. They are already used in landing flaps, stabilizers or ailerons. Further areas of applications are yet limited due to the insufficient sizing approaches delivered by the standards. An extremely challenging scenario for a RCDE is the application on the primary flight control system of a helicopter. On the one hand, the expected ball screw and its bearing has to face high frequent hammering loads applied by the rotor of the helicopter. On the other hand, the system fulfils only small motions during constant flight conditions. Furthermore, the system is used in a wide range of temperature und finally has to fulfil the requirements regarding failure safety.

2. MOTIVATION

The main mechanical component of the steering system of a helicopter is the so-called swash plate (Figure 2.1). Acting as a thrust bearing it connects the rotating rotor blades with the stationary actuation system. It can be moved axially along the rotor shaft (collective pitch adjustment) and tilted (cyclic adjustment). The changed adjustments results in a modified uplift and are the basis to fulfil manoeuvres and change flight conditions.

The positioning of the swash plate is hitherto realized by three hydraulic actuators (one translatory, two rotatory degrees of freedom). They fulfil the positioning demand of both pilot and control system on the one hand and withstand the resulting forces of the rotor on the other one.

An exemplary flight profile for one actuator is shown in Figure 2.2. The applied load can be interpreted as a sinusoidal signal and is primarily depending on the aerodynamic loads of the rotor and hence depending on the swash plate position. Its frequency is a function of rotor speed and number of rotor blades and therefore typically constant. During one flight condition the position of the actuator is defined by the control system of the helicopter. Depending on the difference between intended and actual position, frequency and amplitude of the control system can vary. However, the position signal will not exceed a certain value.
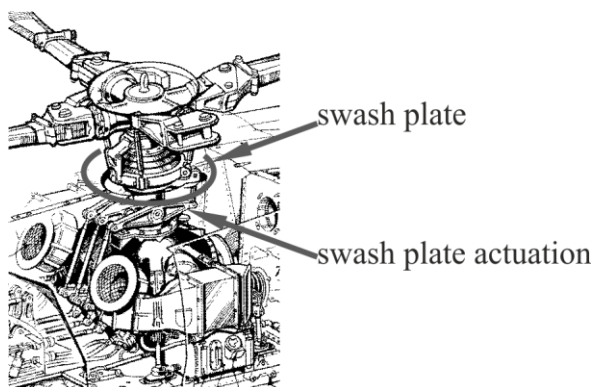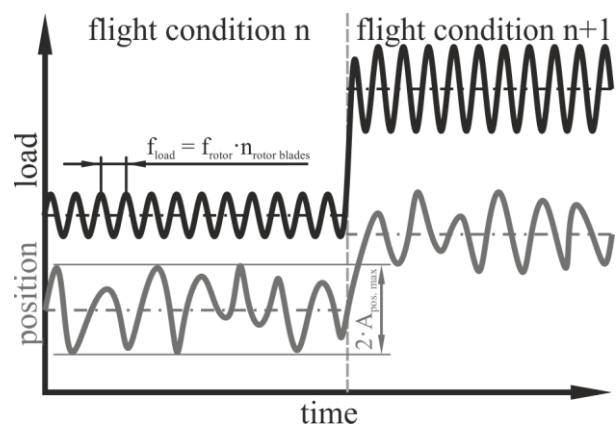

Figure 2.1: Present application [5]

Figure 2.2: Swash plate load and position profile according to [6]

According to [6] the applied load on such an actuator varies in the range of -20 up to 50 kN (over all flight conditions) at oscillating frequencies in the range of 20 to 30 Hz. In contrast, the control system signal varies between 0.6 and 10 Hz. The signal results in a maximum movement of 3 mm (peak to peak). Due to the application additional requirements like wide

ambient temperature ranges (-45 up to 110 °C, in case of emergency like a burning main gearbox 300 °C) and failure rates must be fulfilled.

The actuation of the swash plate is currently realized by a hydraulic system. Current research projects are focusing on a more electric aircraft, where hydraulic actuators are replaced by electromechanical ones. The spindle of a suggested ball screw will fulfil an oscillating motion of about 100° to realize the expected control signal within one flight condition. Due to system compliance the macroscopic motion will be superposed by a microscopic motion resulting from the compliance followed by the applied load.

Since state of the art sizing methods disregard operating conditions as described above, a new sizing method has to be developed. This should ideally be based on existing approaches.

3.   AVAILABLE SIZING APPROACHES

For the new sizing method the available approaches given by standards and publications have to be checked and, if applicable, be adopted. If approaches are only valid in a certain operating condition, design guidelines can deliver ranges for a safe calculation.

3.1   Sizing according to ISO-standard

The sizing of RCDEs can be separated into a static and a dynamic sizing. For static sizing, the static load rating C_0 is defined as the load that leads to a plastic deformation of the raceway of 0.0001 times the ball diameter [7]. At this point the bearing or ball screw has not necessarily failed yet, but during operation the plastic deformations can lead to unacceptable high vibrations or noise [8].

Dynamic sizing assumes the failure mode of fatigue. With the definition of the dynamic load rating C (load, that can be withstand one million revolutions) and the life exponent p, determined by [1] and [2], the nominal rating life for at least 90 % of a huge number of RCDEs can be calculated by Eq. (3.1), (Figure 3.1). Furthermore, Miner's Rule enables the consideration of different operation conditions.

$$\left(- \right)^{p} \qquad \text{Eq. (3.1)}$$

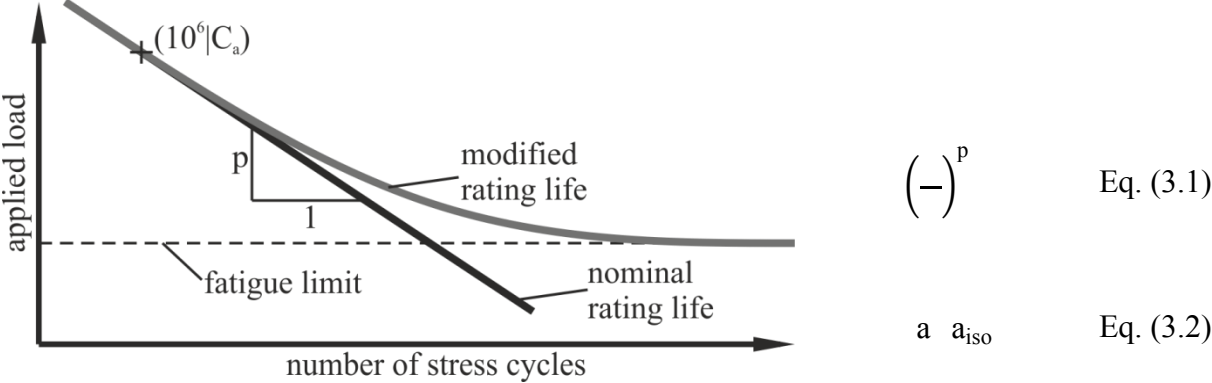$$a \quad a_{iso} \qquad \text{Eq. (3.2)}$$

Figure 3.1: S-N-curve for nominal and modified rating life calculation

The modified rating life calculation (Eq. (3.2)) allows for both different probability factors (a_1-factor) and the determination of lubrication conditions and low loads – which are highly conservative in the nominal life calculation – by the a_{iso}-factor.
However, these approaches are only valid for continuously rotating bearings or ball screws.

3.2 Modifications on ISO-approaches

As long as lubrication in the contact zone can be ensured, oscillating motions can be calculated by ISO approaches by modifying the equivalent load rating or the equivalent load. Depending on the amount of the pivoting angle θ, different approaches are available.
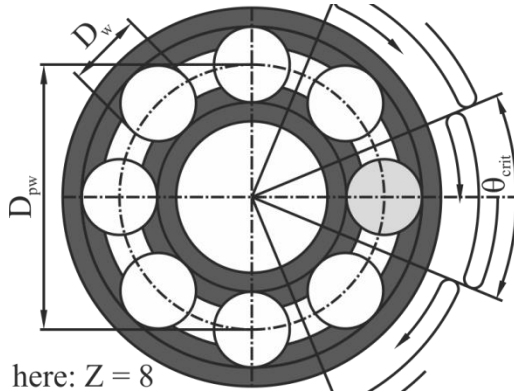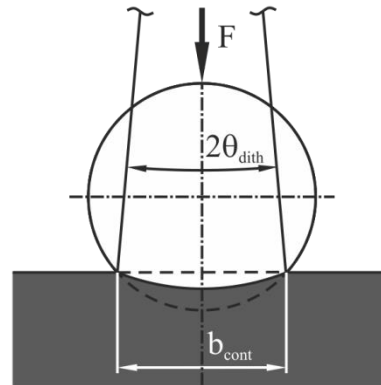

Figure 3.2: Definition of critical angle θ_{crit}   *Figure 3.3: Definition of dither angle θ_{dith}*

In this context, two different angles must be considered. The critical angle θ_{crit} (*Figure 3.2*, Eq. (3.3)) is the angle, of the inner raceway relative to the outer raceway, when one rolling element touches, but not overlaps the raceway portion of an adjacent rolling element [9]. Furthermore, the dither angle θ_{dith} (*Figure 3.3*, Eq. (3.4)) represents the relative movement of inner and outer raceway that results in such a small motion of the rolling element, that parts of its contact ellipse will always stay in contact [9].

While the definition of θ_{crit} is only depending on geometric parameters, θ_{dith} is a function of the size of the Hertzian contact ellipse and therefore depending on the applied load. Assuming the definition of the static load rating C_0, a conservative estimation of b_{cont} and thus θ_{dith} can be realized by Eq. (3.5).

$$\theta_{crit} \quad \overline{\quad (\mp) \quad} \quad \overline{\quad} \quad \text{with} \quad \overline{\quad}_{p} \cos \qquad \text{[8], [9]} \qquad \text{Eq. (3.3)}$$

$$\theta_{dith} \quad \overline{\quad \dfrac{b_{cont}}{\quad_{p} \;(\mp)\quad} \quad} \qquad \text{[9]} \qquad \text{Eq. (3.4)}$$

$$b_{cont} \quad . \qquad \text{Eq. (3.5)}$$

Oscillating motions larger than 180° can be treated the same way as rotating motions [8].

For pivoting angles θ larger than the critical angle θ_{crit} follows Eq. (3.6) i. a. Eq. (3.7) where only one approach is necessary. Both equations are used in different sizing methods and increase the rating life L_{10} in the same way. A modified oscillating load P_{osc} regards the load at one certain raceway part while a modified oscillating load rating C_{osc} changes the definition of the dynamic capacity (one million revolutions). Since rating life is getting increased by these approaches, the most conservative way is to leave them out.

$$_{osc} \quad \left(\overline{\quad\quad}\right)^{\overline{p}} \qquad \text{for } \theta_{crit} \le \theta < 180° \qquad \text{[9], [10]} \qquad \text{Eq. (3.6)}$$

$$\text{osc} \quad \left(\frac{\quad}{\quad} \right)^{\bar{p}} \qquad \text{for } \theta_{crit} \leq \theta < 180° \qquad [8] \qquad \text{Eq. (3.7)}$$

At pivoting angles smaller than θ_{crit} the probability of fatigue as failure decreases compared to the probability of the occurrence of wear mechanism. Assuming sufficient lubrication, [9] proposes Eq. (3.8) for a modified load rating. Transformed you can derive Eq. (3.9) for a modified applied load which can be used for the standard nominal life calculation.

$$\text{osc} \quad \left(\frac{\quad}{\quad} \right)^{\bar{\quad}} \cdot \qquad \text{for } \theta_{dith} \leq \theta < \theta_{crit} \qquad [9] \qquad \text{Eq. (3.8)}$$

$$\text{osc} \quad \left(\frac{\quad}{\quad} \right)^{\bar{\quad}} \cdot \qquad \text{for } \theta_{dith} \leq \theta < \theta_{crit} \qquad \qquad \text{Eq. (3.9)}$$

Figure 3.4 illustrates the influence of the pivoting angle on the rating life considering Eq. (3.7) and Eq. (3.9). The diagram assumes a bearing with 18 balls (typical number of balls in one ball screw threat). The dither angle θ_{dith} is a result of an applied load smaller than C_0. In this general case, an oscillating motion in the range of the critical angle increases the rating life about 65 %. Smaller angles increase rating life multiple times. Even if these approaches are correct regarding fatigue, it does not seem that they can represent the operational life of an RCDE correctly, since other failure mechanism might be more dominating. For this reason comparative analyses (for example [11]) recommends to disregard the presented approaches as long as no additional internal knowledge is available.
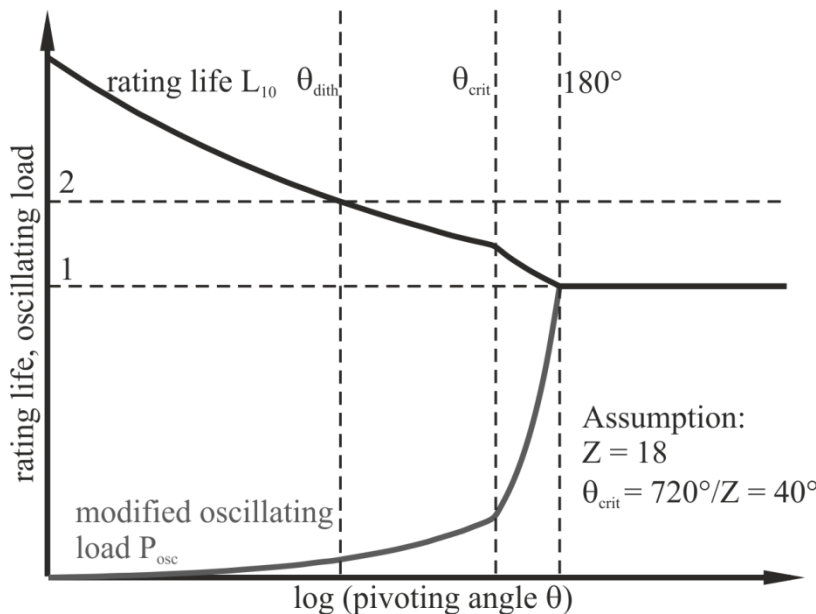

Figure 3.4: Oscillating load P_{osc} and rating life L_{10} depending on the oscillating angle θ (normalized to 1)

3.3 Further failure mechanisms and methods to avoid them

As described in the previous sections, ISO-calculation and its modifications assumes a failure due to fatigue. With degrading lubrication, further mechanisms, based on wear, are getting

possible and more likely. In a rolling contact, pure rolling occurs only in the centre of the Hertzian contact ellipse (*Figure 3.5*). Depending on the contact pressure, the contact ellipse can be divided into a region of a sticking contact and a slipping contact zone. At insufficient lubrication, friction forces will activate the raceway surface. This can be the initial situation for mechanisms like fretting, abrasive wear or false brinelling.
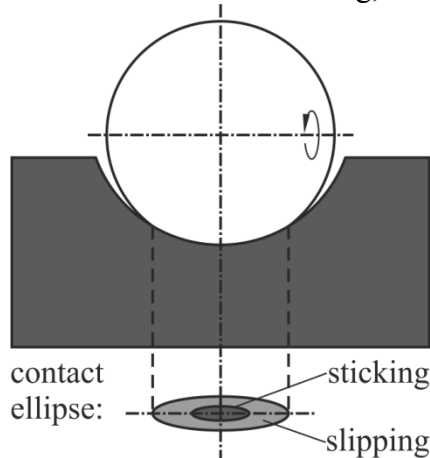

Figure 3.5: Hertzian contact ellipse

3.3.1 Abrasive wear

Wear models are typically based on the works of Archard, Kragelski, Ikramov and Fleischer (e. g. [12], [13], [14], [15]).
Abrasive wear leads to

- an increased number of particles in the lubricant,
- a mass loss of the single components (e. g. balls) which results in
- a loss of preloading and/or rigidity.

Haberkern [16] extends existing approaches to determine the wear behaviour of ball screws. He focuses on a comparison of the benefits of different coatings. Therefore, the derived approaches do not need to deliver quantitative results but only qualitative ones to evaluate the benefits of different coatings. The wear behaviour after the first run-in effects is regarded and afterwards a linear wear amount W_l (Eq. (3.10)) and a linear wear speed W_l^* (Eq. (3.11)) are introduced.

$$_l \qquad s_{rel}\, p \qquad \qquad \text{Eq. (3.10)}$$

$$_l \qquad _{rel}\, p \qquad \qquad \text{Eq. (3.11)}$$

This approach is adopted by [17] and [18]. They extend the determination of the factor K and formulate an approach for a life prediction model.
All approaches have two main deficits:

- The wear approaches can-not reflect running-in effects. A starting point for the wear calculation must be assumed.
- Hitherto there is no approach for a quantitative definition of a failure criterion W_{crit}.

Therefore the shown approaches are not considered suitable. However, an abrasive wear model would greatly contribute to RCDE calculation. As long as a calculation is not possible, the occurrence of wear must be avoided or at least minimized by the design of the RCDE and indirect, lubrication like surface treatments or lubricants with good dry-running behaviour.

3.3.2 False Brinelling

False brinelling describes the formation of hollows at rolling element contacts at oscillating motions of little amplitude ($\theta < 1°...3°$). The resulting marks look similar to marks resulting from standardized brinell testings. Because of the different mechanisms the phenomenon is called false brinelling.

False brinelling is a tribo-chemical process. After the lubricant is pushed out of the contact zone micromovements activate the surface followed by microcracks due to adhesive mechanism. Since no lubricant gets into the contact zone to carry the debris out of the contact area, some kind of lapping compound is formed that accelerates the damaging progress. [19]

Deters [20] shows, that the service life of a false brinelling pre-damaged bearing decreases up to 95 % compared to a bearing without pre-damage.

Since false brinelling can occur at any operation condition during standstill the following suggestions can be made:

- Ensure relubrication by sufficient often, large motions of the RCDE system.
- Since the critical oscillating angle for false brinelling seems to correlate with the dither angle, a smaller contact ellipse leads to a smaller dither angle and so increases the range of safe operation conditions. This can be achieved by materials of higher strength, like ceramic balls.
- Ceramic balls or DLC-coatings can furthermore prevent adhesive mechanisms that are part of the false brinelling formation.
- For ball screws, a larger lead angle (larger system compliance) can result in motions greater than the critical false brinelling angle.

3.3.3 Jerk loads

The rotating part of a ball screw is exposed to jerk loads once due to acceleration effects of the driving motor and twice as a result of system compliance resulting from external loads. These jerk loads temporally increase the Hertzian contact pressure and therefor decrease the estimated service life.

Yagmur [21] shows for a certain ball screw assembly that jerk values about 2000 m/s³ decrease the estimated service life more than 15 %. Ball screw manufacturers consider higher contact pressures by equations like Eq. (3.12), however, these approaches vary within different manufacturers and are not standardized yet.

$$\text{ith} \begin{cases} i \quad p_a \qquad M \quad a \\[2em] \left(\dfrac{p_a}{M \, a} \right) i \ p_a \qquad M \quad a \end{cases} \qquad [22] \qquad \text{Eq. (3.12)}$$

If the compliance behaviour is known, the jerk and the resulting contact pressure can be determined.

4. CONCLUSION

The actuation of the primary flight control system of a helicopter is faced by dynamic loads and small displacements at different ambient temperatures during one flight condition. The usage of ball screws and bearings in this actuation system is yet limited since there is no sufficient sizing approach available, covering all relevant aspects.

Rolling contact drivetrain elements, like ball screws and thrust bearings, can fail due to several failure mechanisms. Currently, only approaches for a fatigue life calculation enable a

determination of an operating life and hence a sizing of RCDEs. Therefore, a new sizing approach must ensure a design that avoids an early failure as a result of wear mechanisms like fretting or false brinelling and must regard the influence of dynamic applied loads, like jerk loads (*Figure 4.1*). This will be on the focus of further studies.

Figure 4.1: preliminary sizing approach

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