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THEORETICAL AND PRACTICAL CONSIDERATIONS ABOUT BEARINGS SUSTAINABILITY

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Abstract: *This paper aims to present a methodology of calculation of sustainability emerged from testing a large number of bearings. This work presents a relationship to calculate bearings reliability, working at low speeds and / or heavy loads.The relationship is based on the general formula, but seven of correction factors adapt it to specific conditions. Conclusions and practical solutions for maintaining unchanged the calculated lifetime of the bearings are presented.*

Key words: *sustainability, factor, calculation, dynamic load, correction factors, bearings, sealing.*

1. INTRODUCTION

Determination of sustainability is the main tool in optimizing the choice of bearings, not only because of the large number of bearings required by the market, but also because of the critical role these components play in multiple applications, including safety.

In the center of all models for calculating the bearings sustainability is contact fatigue, closely related to the tension between running bodies and ways of the bearing.

Bearings are subjected to external factors, leading to a dramatic decrease in their lifetime. The work is based on experimental applications in the field of knitting machines, and 15 years of experience in working with them. This paper promotes sealed bearings, which depend less on external factors as: environment, maintenance, etc. There are presented and motivated some solutions to replacement open bearings with sealed bearings.

2. CALCULATION OF SUSTAINABILITY

According to ISO 281: 1990, nominal sustainability L_{10} is associated with a reliability percentage of 90, of a group of apparently identical bearings, operating under the same conditions, quality of construction material and also operating under conventional conditions, namely the number of revolutions made by rotating the ring until the first signs of material fatigue.At constant speed,durability can be measured in hours.

To take into account in calculating only by material fatigue of the active bearing surfaces, the following conditions must be observed:

- Forces and speeds from the basis of bearing calculation should correspond to actual operating conditions.
- Proper lubrication during the whole operation.
- If the bearing load is low, the output is no longer due to use of the material, but wear.
- Practice has shown that scrapping of a significant number of bearings is due to other causes than fatigue material as: inappropriate choice of the bearing, improper running, improper lubrication, penetration of foreign bodies in the bearing, etc..

The appearance of voids on the functional surfaces of the bearing is the main form of damage for the bearings with $n > 10$ rpm, well lubricated and sealed [5].

The running of rolling bodies causes variable contact stresses in the superficial layers of the rings.

Signs of fatigue appear as cracks that increase over time. The penetration of oil with pressure produces the separation of material particulate. Cracks occur on the rolling paths of the inner ring at the majority of bearings. The voids increases the clearance in bearing. Its function is worsening. To avoid this type of damage, bearings are calculated at sustainability.

Calculation relationships are determined based on a number of experimental determinations, because operating times shows a statistical distribution.

Based on the scattering curve of bearing durability from a lot tested at sustainability (Fig. 1) [1], it finds that a rate of 50 of the bearings exceed approximately five times the basic durability and a rate of 10 about 14 times, although all bearings are apparently identical and are loaded in the same conditions.

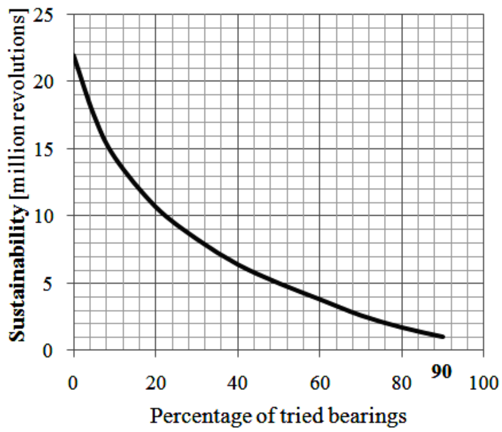

Fig. 1. Sustainability curve for a lot of ball bearing [1].

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Because of the sustainability dispersal, can not be determined accurately whether a bearing touches the theoretic lifetime, but only with a probability of 90. Although this situation seems safe enough, for a machine that contains 10 bearings, almost certainly a bearing is damaged faster than the life for which he was elected.

This means you need stop the machine, move away a greater or lesser part of it and replace the worn bearing, before a scheduled overhaul or capital repair.

Theoretically, the overall relationship was obtained by Lundberg and Palmgren which have extended Weibull's theory to calculate the lifetime. The calculation is based on the fundamental equation [2]:

$$\ln\frac{l}{S} = N^{e}\tau_0^{c} z_0^{l-h} al \,, \qquad (1)$$

where:

S – the probability that failure will not occur;

N – number of load cycles;

τ_0 – symmetric alternating shear efforts, which appears in the substrate of the contact surface to a depth of z_0 [Pa];

a – length of the major semiaxa of the contact ellipse [mm];

l – running path length [mm];

c, e, h – numerical exponents, determined experimentally.

After the substitution of the parameters characterizing the contact voltage, the load and geometry, in equation (1), it obtains the relationship of sustainability, which can be expressed as [2 and 4]:

$$L_{10} = \left(\frac{C}{P}\right)^{p} \text{ [mil. rev.]}, \qquad (2)$$

where p is a constant that depends on bearing geometry and material; p = 3 for ball bearings and p = 10/3 for roller bearings. Note that the sustainability of a ball bearing varies according to the C / P report after a three-degree curve (Fig. 2); C – the basic dynamic load, [N]; is the load for a bearing with a durability of one million revolutions and is calculated in accordance with ISO 281.

Depending on the basic dynamic load of the bearing, time is calculated till the material fatigue, thus it obtains the durability calculated.

Basic static load is defined, according to ISO 76, as the load acting on the stationary bearing and corresponds to a contact pressure calculated in the center of the contact patch of the most loaded raceway, with a value of:

- 4 200 MPa, for the other ball bearings,
- 4 000 MPa, for roller bearings.

This produces a permanent deformation on the raceway and on the rolling elements, of approximately 0.0001 from the roller diameter, being purely radial load for radial bearings, and purely axial load for axial bearings.

P is the dynamic equivalent load, [N], respectively radial and axial load, acting simultaneously, on ball or on roller. It is calculated with formulas [3]:

$$P = F_r \text{, [kN] – for pure radial load,} \qquad (3)$$

$$P = F_a \text{, [kN] – for pure axial load,} \qquad (4)$$

$$P = XF\mathrm{r} + YF\mathrm{a} \text{ [kN] – for the combined load,} \qquad (5)$$

where:

F_r – component of radial load, [kN],

F_a – component of axial load, [kN].

Coefficients X and Y are given in tables.

Sustainability of bearings that have oscillatory movements, instead of rotation, from a central position to an angle γ, is given by the relationship [6]:

$$L_{10OSC} = (180 / 2\,\gamma)\, L_{10}, \qquad (6)$$

where:

$L_{10\,OSC}$ – sustainability, [million cycles],

γ – amplitude of oscillation (angle from the center to the maximum), [degrees].

The very small angles of the oscillatory movements do not count.

Dynamic loads and variable speeds. In many cases the operating speed and load size is variable. For calculate the equivalent dynamic load, it should be determined an average radial load F_{mr}, or an average axial load F_{ma}, thus:

1) At constant speed of the bearing, strength varies linearly between a minimum value, $F_{mr,amin}$, and a maximum value, $F_{mr,amax}$, keeping the direction for a certain period of time. Average strength is obtained from the relationship:

$$F_{mr,a} = (F_{r,amin} + 2\,F_{r,amax}) / 3 \text{ [kN]}, \qquad (7)$$

2) If the radial load acting on a bearing is composed of a force, F_{r1} , constant in size and direction (eg: weight of a rotor) and a rotating force constant F_{r2} (eg: imbalance) average strength resulting from the relationship:

$$F_{r\,m} = f_m\,(F_{r1} + F_{r2}) \text{ [kN]}, \qquad (8)$$

The coefficient f_m are obtained from Fig. 3 [6].

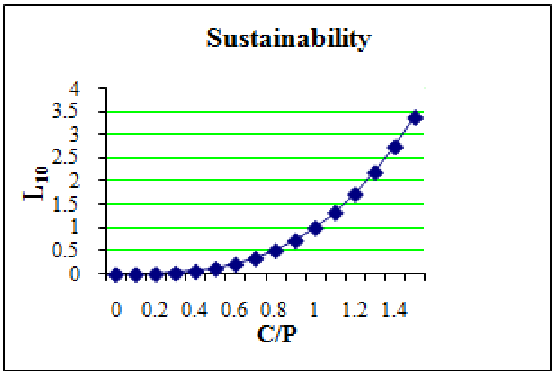

Fig. 2. Curve of sustainability according to the *C/P* report**.**

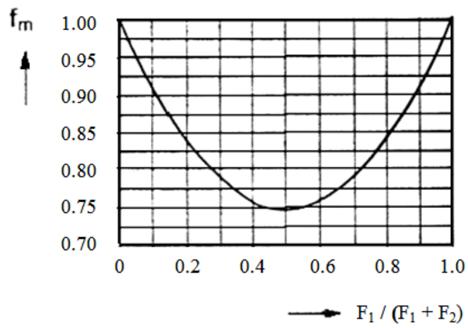

Fig. 3. Coefficient f_m [6].

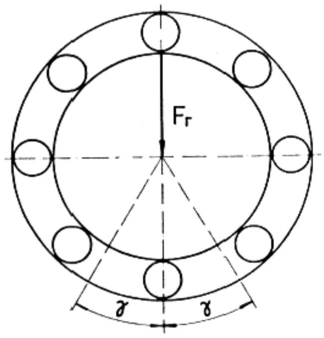

Fig. 4. Angle 2 **γ** [6].

Table 1

Coefficient f_0 [6]

γ^0	F_0	
	p = 3	**p = 10/3**
10	0.47	0.53
20	0.61	0.65
30	0.69	0.72
45	0.79	0.81
60	0.87	0.89
75	0.94	0.95
90	1.00	1.00

3) For radial load Fr , applied to an oscillating motion, in the angle of 2 γ (Fig. 4) [6], the average radial load is calculated with the formula:

$$F_{m\,r} = f_0\, F_r \text{ [kN]}, \tag{9}$$

with values of the coefficient f_0 in Table 1 [6], depending on the oscillation angle, γ and by the exponent of the sustainability formula, p.

For loads varying in time, the average dynamic load is calculated with the formula [3 and 6]:

$$F_{mr,a} = [\sum (F_{ir,a}^{p} ni)/n]^{1/p}\ , \tag{10}$$

where:

$F_{mr,a}$ – average load, in the radial or axial direction [kN];
$F_{ir,a}$ – constant load, applied during the rotations n_i,[kN];
n_i – the number of rotations for corresponding tasks, $n = \Sigma\, n_i$, [rpm].

The basic dynamic load for a bearings group. For two or more identical bearings, the basic dynamic load for a set of i bearings is given by the relationship [6]:

$Cr = i^{0,7}\, C_{r\,i}$, [kN] for bearings with point contact; (11)
$Cr = i^{7/9}\, C_{r\,i}$, [kN] for bearings with linear contact. (12)

This relationship has been accepted by ISO, but it was complemented by introducing correction factors.

2.1. Bearings that works at high speeds

These refers to speeds v > 3.5 m/s, when is reaching some important temperatures. The relationship is [1, 2 and 6]:

$$L_{nh} = a_1 a_2 a_3 f_t \left(\frac{C}{P}\right)^p = a_1 a_2 a_3 f_t L_{10} \quad \text{[mil. rev.]} \tag{13}$$

where:

a_1 – factor of reliability;

Table 2

Factor f_t [6]

Operating temperature [^{0}C]	150	200	250	300
f_t	1	0.73	0.42	0.22

Table 3

Factor a_{23} [6]

v/v_1	0.1	0.2	0.5	1	1.5	2	3	4	5
a_{23}	0.45	0.55	0.75	1	1.3	1.6	2	2.5	2.5

a_2 – factor of material;
a_3 – factor that is function of operating conditions and for lubrification quality;
f_t – correction factor, depending on the operating temperature (Table 2) [6].

The interdependence of the correction factors leads to their unification into a single factor a_{23}, whose value is given in Table 3 [6] and that depends on the ratio between:

- kinematic viscosity of the oil at 40 °C ν, (in cSt or mm^2/s),
- necessary viscosity to correct lubrication: v_1; depending on operating temperature.

2.2. Bearings that operates at low speed and at high contact stress

These refers to speeds v < 3.5 m/s and C / P < 0.6. Formula for sustainability is [2]:

$$L_{nh} = a_1 a_2 a_3 a_v a_r a_c a_a \left(\frac{C}{P}\right)^{p'} \ [mil.\ rev.]. \tag{14}$$

In this relationship we have:

a_1 – reliability is 1.0 for 90 percent, which is a typical level of reliability in use. If reliability is less than 90 percent, sustainability will be noted with L_p, (Table 4 [2]). Index p is 100 minus reliability value, expressed in percentages:

Note that to achieve high reliability of operation, without failure, of the entire lot (reliability 99 percent), the factor a_1 and therefore L_1 will drop dramatically, almost 5 times.

a_2 – correction factor, used for bearings built from special materials.For standard materials has a value equal to unity; for the steels that are produced in vacuum has the value 3.0.

a_3 – the modified factor of lubrication. This factor quantify the effectiveness of the lubricant film between the surfaces in contact through the λ factor (Fig. 5)[2], namely the ratio between the lubricant film thickness, h , and the equivalent roughness of the surfaces .

This roughness is defined by the relation:

Table 4

Values of a_1 [2]

Reliability, %	L_p	a_1
90	L_{10}	1
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

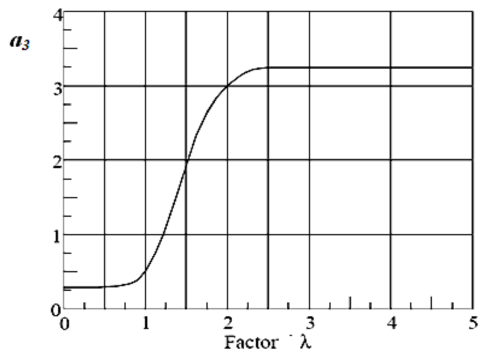

Fig. 5. Variation of $\boldsymbol{a_3}$ factor [2].

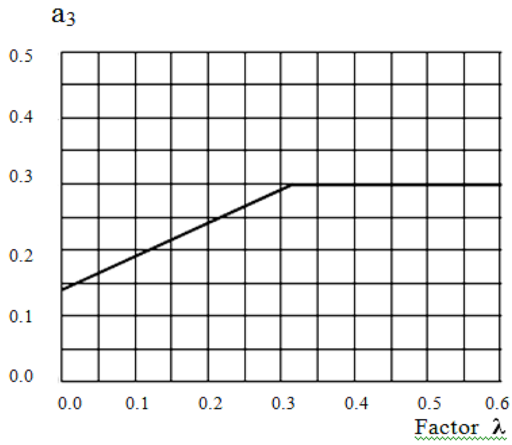

Fig. 6. Variation of the $\boldsymbol{a_3}$ modified factor [2].

$$R_{a,echiv} = \sqrt{R_{a1}^2 + R_{a2}^2}\ [\mu\text{m}], \tag{15}$$

where R_{a1} and R_{a2} are the mean square deviation of the values for the roughness of two surfaces in contact.

If the bearing operates at low speeds, λ factor is usually less than 1.0 and often less than 0.5, which corresponds to the correction factor, $a_3 = 0.3$.

Further researches shown the need to reduce the size of a_3 factor for very small values of λ. Figure 6 [2] presents the variation of the a_3 modified factor, proposed for values of λ lower than 0.6.

***a_v*- speed factor.** After using the modified load exponent in the equation for calculating the sustainability, note that report between real sustainability and calculated sustainability, tends to 1; however, for low speed (0.05 – 1.7 m/s), real sustainability is about 2–3 times higher than calculated sustainability. This led to the proposal for a speed factor, whose change depends by the speed of the inner ring (Fig. 7) [2].

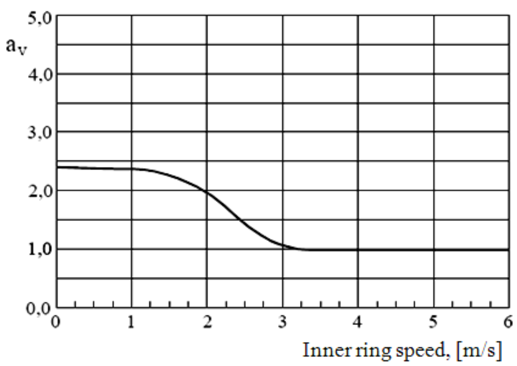

Fig. 7. Variation of speed factor [2].

There are some arguments that pleads for this cofigurative of the proposed speed factor.

The most important argument relates that the steel for bearings has a better fatigue resistance for the low speeds. On the other hand, for such charge cycles, material behaves less brittle, which influences the initiation of cracks.

***a_r* - roughness factor**. Since the proposed method has been developed based on data from experiments conducted in the laboratory, where conditions differ from actual operational situations, a factor was introduced, which takes into account the roughness of two surfaces in relative motion.

It is calculated with:

$$a_r = (1.043 \dots 1.17)\ R_a, \tag{16}$$

where R_a [μm], is the value for the surface roughness, expressed as the mean square deviation.

Deviation from coaxialitate, when is combined with strong loads, may lead to the extra charge of the rolling elements. This situation creates concentrators of tension, leading to premature failure,by spalling. Factor is applied only when the values of the contact surfaces differs substantially as roughness.When surfaces are comparable in terms of the roughness, this factor is equal to 1.0.

***a_c* – contamination factor of the lubricant.** Research has been directed to specific operating conditions for certain types of bearings and for certain operating conditions [12]. Therefore it is difficult to promote a single equation which fully express the influence of the lubricant contamination on the sustainability. However, links were established between the level of contamination and the a_c value (Table 5)[1]:

***a_a*– factor for deviation from coaxialitate.** Values of this factor are presented in Table 6 [1].

***p'* – modified exponent of the load.** After some tests, it was found that the relationship between the tested sustainability and calculated sustainability, increases with contact pressure [1].

Table 5

Values of the contamination factor [1]

Level of contamination	**Values of a_c factor**
Very small (fine filter)	1
Low (coarse filter)	0.9
Moderate (without filtering, particles in small proportion)	0.8
High (without filtering, particles in moderate proportion)	0.7
Very high (without filtering, particles in large proportion)	0.5

Table 6

Deviation a_a factor [1]

Level of necoaxialitate	**a_a**
Very low (< 0.001 mm/mm)	1
Reduced (0.001 ... 0.0019 mm/mm)	0.85
Moderate (0.002 ... 0.0034 mm/mm)	0.7
High (0.0035 ... 0.005 mm/mm)	0.6
Very High (> 0.005 mm/mm)	0.5

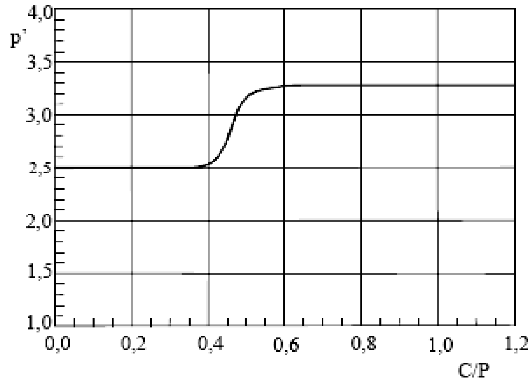

Fig. 8. Variation of the modified exponent, ***p'*** [2].

These results suggest an exponent of the load, which tends to reduce the calculated value of the sustainability, with a smaller percentage, when the load increases.

This can be explained by the fact that when the load applied to the bearing exceeds substantially the capacity of dynamic load, the *C* / *P* report becomes much smaller than 1 and its raising to the 10 / 3 power leads to very low values, which diminishes artificially the calculated value of sustainability.

Exponent 10 / 3 was designed for the situation that actual load is close to the real capacity of the bearing.

Figure 8 [2] presents the curve of the *p*' exponent, according to the *C* / *P* report.

A very high percentage, up to 36, of premature wear of the bearings, is due to an inadequate lubrication (inappropriate use of the lubricant, insufficient or excessive lubrication).

Sealed bearings and anointed "for life" are found in industry in a relatively small proportion, compared with the bearings in open construction; so at least 14 percent of the premature wear are due to the bearing contamination.

At S.C. Emadi Prod SRL, where I worked between 1994–2009, operating conditions of the machinery for knitting socks were : speeds from 60–300 rpm, medium loads (engine power: 0.4–1 kW), dusty environment, temperature above 50^0 C, average level of vibration , daily operating time:16 hours.

Here, the open bearings had a life between 5 and 7 years(21 000–30 000 hours), close to the calculated lifetime. Instead, where I used sealed bearings, I have not changed any bearing in 9 years!

The explanation is that the nominal sustainability of bearing was not influenced by corrosion, or improper lubrication (insufficient or excessive), or contamination due to the improper sealing.

On the other hand, oiling activity was eliminated, lubricant was saved, and the entire bearing was kept clean. Therefore, I strongly recommend the use of sealed bearings where are couplings with not very hard conditions of work.

In the following I will present my proposed solution to maintain the life of bearings.

The knitting machines, 2C Special (are produced in Czech Republic), are driven by a motor reducer with two speeds; engine speeds are 1 400 and 700 rpm, and output speeds from the gearbox: 140 and 70 rpm.

Fig. 9. Reducer and electric motor.

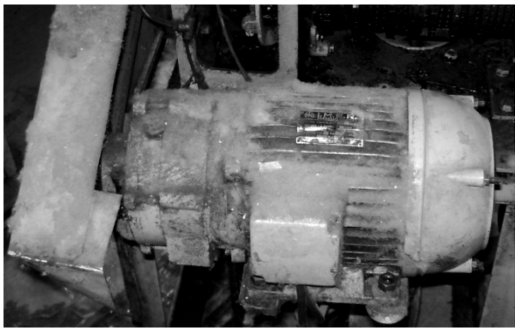

Fig. 10. Lint and puff of cotton.

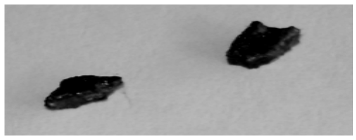

Fig. 11. Metal particles.

Fig. 12. Used oil.

A motor reducer gave signs that needing repairs. The signs were: the increase of noise and of heat.

The reducer has two gears and ball bearings. Anointing is made with oil bath. The reducer is separated from the electric motor (Fig. 9).

The oil had to be replaced, and bearings, too. Note that the main factor for the wear of oil are the gears, and the main factor for the contamination of oil is the environment with dust, with cotton puff (Fig. 10), well as metal particles (Fig. 11). Figure 12 shows degradated oil.

Considering that the life of the bearings was much influenced by the contaminated oil, we decided to use sealed bearings and thus to isolate them by the oil bath.

We know that sealed bearings have higher friction losses than non-sealed bearings. However, after we put the engine in operation, we observed no notable increases of temperature.

3. CONCLUSIONS

The calculation of the bearing durability starts from the contact pressure, that is closely related to the state of stress of the bodies and the ways of running.

These tensions lead to the appearance of the surface cracks, that increases over time.

One cannot determine exactly if a bearing touches the calculated life, only with a probability of 90 percent.

This percentage does not give us a full safety about the achievement by any bearing of the calculated life-time; a bearing from the ten will not meet this requirement.

Therefore, if we need increased safety, will be either oversized the bearings, or will carry out a rigorous selection by subjecting the bearings to some further testing.

For the ideal operating conditions, bearings can have a virtually unlimited sustainability.

In practice, the life of bearings is strongly affected by the certain factors, most important being: inadequate lubrication, overloading, contamination, improper installation.

The bearings have the advantage that "announce" when they are at the end of their life. The main symptoms are the unusual noise and vibration. They can be observed directly, or through various methods and apparatus for testing.

The sealed bearings have obtained a better life than open bearings, because of the significant diminution of the factors influence that reduce their durability.

The using these bearings leads to the simplification of the design, the construction and the maintenance of machinery and so, to the reducing of the costs.

The lubrication in common of the machine parts leads to their mutual contamination.

Therefore, I believe the anointing in individual mod is better than the collective lubrication, especially if the differences of cost are insignificant.

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