The process of optimizing the radial sliding bearing

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Abstract. The problem of optimizing the bearing choice process, caused by the failures due to the short life of the bearings, caused by the difficult working conditions, is becoming more and more frequent. By further optimizing the type of bearings used, we aim to both improve efficiency and productivity and reduce costs. For the choice of a radial sliding bearing, the three basic steps are followed: calculation of the diameter of the shaft, verification of the bearing pressure, thermal calculation. If at least one of the last two is not appropriate, the designer must change the material, the cooling conditions or the length / diameter ratio and then recalculate the bearing. The methodology of optimizing the choice of bearings involves the establishment of a set of initial constraints to reduce arbitrary recalculations. In optimal design, we eliminate the drawbacks presented in usual design and we impose at the beginning some restriction for shaft material, deflection, cooling capacity, we determine the type of fit and tolerances, that is neglected in usual design.

1. Introduction
Bearings are defined as machine parts used to support shafts or other rotating parts, used to take over the acting tasks on them. After the frequency inside the camp it is rubbed with rollers, the bearings they are called roller bearings. The bearing is the main element of the bearing with rolling. In addition to the bearing, the shaft spindle, housing, axial fasteners, lubrication and sealing systems are included in the roller bearing. [1]

The component parts of the bearings are essential to reduce friction and to allow machines where they are incorporated to operate efficiently. In addition, a concrete example of the application of large-scale bearings long ago is the case of bicycles that have become extremely popular nowadays.[2]

The new technologies used in the case of bearings over time have also led to real revolutions in the manufacturing industry, which facilitated the processes of designing and manufacturing machinery and equipment with extremely efficient operating systems and for decent prices. Both the large-scale use of the various models of bearings available and the machineries that have been put into motion with their help over time have made it possible to rapidly increase productivity in various fields.[3,4]

This paper proposes a method of optimizing the radial sliding bearings using dry and boundary friction.
2. Usual design of the radial sliding bearing

Calculation of sliding radial bearings usually follows in three steps:

- Shaft diameter calculation
- Bearing pressure checking
- Thermal calculation

If the second and third steps are inadequate, the designer change the material for bearing, the value of length to diameter ratio or the cooling conditions. This methodology supposes to recalculate the bearing, after the initial calculation is ended. Optimal design suppose to establish, initially, a set of restrictions, permitting to avoid the arbitrary material selection and arbitrary recalculations

a) Shaft diameter calculation

Based on bending stress, the diameter of the shaft is \( d_1 \) and based on contact pressure, \( d_2 \)

\[
d_1 = \sqrt{\frac{M_i}{0.1 \cdot \sigma_{ai}}} \quad [\text{mm}]
\]

\[
d_2 = \sqrt{\frac{F \cdot D}{p_a \cdot B}} \quad [\text{mm}]
\]

where

\( \sigma_{ai} \) – allowable fatigue stress [MPa]

\( B, D \) – bearing dimensions [mm]

\( F \) – the radial force in the bearing [N]

\( p_a \) – the allowable pressure of sliding contact [N/mm²]

The recommendation is to retain the utmost value of diameters \( d_1 \) and \( d_2 \). [5]

b) Bearing pressure checking

The heat generated in the bearing is as "specific less power by friction"

\[
P_f = \frac{\mu \cdot F \cdot v}{A_f} = \mu \cdot p \cdot v
\]

where

\( \mu \) – the friction coefficient

\( v \) – rubbing velocity [m/s]

\( p \) – the average pressure [N/mm²]

\[
p = \frac{F}{B \cdot D} = \frac{F}{B \cdot d} \quad [N/mm^2]
\]

It is must be necessary to observe the condition

\[
(pv)_c \leq (pv)_a
\]

where
"c" – the index for “calculated value"
"a" – the index for “allowable value"

c) **Thermal calculation**

In order to realize the calculation of heat dissipated heat amount \( Q \) is calculated using the equation (6):

\[
Q = K \cdot A(t - t_0); \quad Q = P_f \quad [W]
\]

where

- \( A \) – dissipation heat area \([m^2]\)
- \( K \) – heat dissipation coefficient \([W/m^2\cdot ^\circ C]\)
- \( t \) – operating temperature \([^\circ C]\)
- \( t_0 \) – ambient temperature \([^\circ C]\)

If the temperature \( t \leq t_0 \) – condition for well, cooling.

The following aspects must be taken into account:

1. The \( d_2 \) determination suppose to select an arbitrary material for bearing, without knowing if this is the best selection
2. If the relation (4) is not fulfilled, the solution is to change the material of bearing and recalculate \( d_2 \)
3. If the relation (5) is not fulfilled and \( t > t_0 \), the solution are:
   - Modifying the \((B/D)\) ratio, for decrease of \( P_f \), and decrease \( Pf \)
   - Modifying the diameter \( d \), for the decrease of \( v \) and decrease \( P_f \)
   - Recalculate the \( d_1, d_2 \) diameters for the new value of \((B/D)\) and remake the entire design
   - Increasing the value \( A \)
   - Modify \( K \), by well ventilate the bearing.

3. **Optimization criteria**

3.1. **Criteria for achieving the optimal design**

In above section, we notice the following:

→ \((A/B)\) ratio is selected arbitrary: few empirical recommendations exist, especially for types of machines
→ The material selection is arbitrary, frequently it is necessary to allow it
→ The stiffness of the shift is not considered; however practical recommendations exists for the angular deviation of the shaft into the bearing
→ The clearance it not considered, however, for the shaft and the bearing we must indicate the values of diametrical deviations and fittings.

To eliminate those design imperfections and the multiple recalculations, the optimum design propose to establish some criteria to ensure a reliable design algorithm

3.2. **Mechanical strength**

Mechanical strength is based on the bending stress and results by the equation (7):

\[
M_t = F \frac{B}{2} \leq \frac{\pi \cdot d_3}{32} \cdot \sigma_{ai}
\]

Or
\[ F \cdot B - \frac{\pi \cdot d^3}{16} \leq 0 \]  
(8)

and considering \( \frac{16}{\pi} \approx 5 \)

\( B \) – bearing dimension [mm]

\( F \) – the radial force in the bearing [N]

results

\[ B = \alpha \cdot d^3 \ [mm] \]  
(9)

where

\( \alpha \) – factor depending on the length of the shaft and by the material

\( \sigma_{al} \) – can take several values depending on material

- For carbon laminated steels (OL) and cast iron: \( \sigma_{al} = 45 \ldots 60 \text{ MPa} \)
- For high quality laminated steels (OLC): \( \sigma_{al} = 60 \ldots 75 \text{ MPa} \)
- For alloyed steels: \( \sigma_{al} = 75 \ldots 100 \text{ MPa} \)

\[ \alpha = \frac{\sigma_{al}}{5F} \]  
(10)

3.3. Resistance to sliding contact pressure of the bearing

The average value of the pressure can be determined with relation (11), as follows:

\[ p = \frac{F}{B \cdot D} \leq p_a \ [N/mm^2] \]  
(11)

and considering \( D \approx d \)

where \( p_a \) is allowable pressure in contact zone

The condition (11) can be written

\[ B \geq \frac{\beta}{p_a} \]  
(12)

where

\( \beta = \frac{F}{B} \) – the force acting to unit length of bearing [4,6]

3.4. Stiffness of bearing figure 1

The shaft deflection produces an inclination of the shaft axis. This deviation is limited by \( \theta \) value, prescribed at the beginning of the design.
Let’s consider that \( j \) is the clearance of the bearing and \( j_p \) the probable clearance of the bearing. The condition is:

\[
i_p \geq j = B \cdot \theta
\]  

must be observed, where \( \theta \) [rad] is the inclination angle of the shaft into bearing, and

\[
j_p = \frac{j_{\text{max}} + 2 \cdot j_{\text{min}}}{3}
\]

where

- \( j_{\text{max}} \) – maximum limit of the diameter
- \( j_{\text{min}} \) – minimum limit of the diameter

The value of clearance of the bearing (from the hole basis system) are the following:

- \( H7/g6 \) – for the close running fit
- \( H7/f7 \) – for the normal running
- \( H7/e8 \) – for the easy running fit
- \( H8/d10 \) – for the loose running fit

This tolerance zones have a specific upper and lower standard deviations, offered by technical textbooks.[4,7]

These values correspond, to empiric relation:

\[
\lambda^{2.5} \cdot d^{1.5} = A
\]

with

- for the close running fit \( H7/g6 \) - \( A = 70 \)
- for the loose running fit \( H8/d10 \) - \( A = 3600 \)

\( \lambda \) is defines by the diametral clearance ratio \( \psi \):
\[ \psi = \frac{d_p \, [\text{mm}]}{d \, [\text{mm}]} \]  

(16)

\[ \lambda = 10^3 \cdot \psi \, [\mu m/mm] \]  

(17)

From relations (16), (17) and (13) results:

\[ \lambda = 10^3 \cdot \psi = 10^3 \cdot \frac{d_p}{d} \geq 10^3 \cdot \frac{B \cdot \theta}{d} \]  

(18)

The fit is into the limits provided by A if

\[ B \geq \frac{25 \sqrt{70 \cdot d}}{10^3 \cdot \theta} \]  

(19)

and

\[ B \leq \frac{25 \sqrt{3600 \cdot d}}{10^3 \cdot \theta} \]  

(20)

3.5. Heating and cooling of the bearing

The relations (3) and (5) are valid, but we notice that \((p \cdot v)_a\) has two values:

- for various types of machines: \((p \cdot v)_{am}\)
- for various types of bearing materials: \((p \cdot v)_{ai}\)

The “calculated value” is:

\[ (p \cdot v)_c = \frac{F}{B \cdot D} \cdot \omega \cdot r = \frac{F}{B \cdot D} \cdot \frac{\pi \cdot n \cdot d}{30} \cdot \frac{1}{2} = \frac{\pi \cdot n \cdot F}{60 \cdot B} \leq (p \cdot v)_a \]  

(21)

\[ B \geq \frac{\pi \cdot n \cdot F}{60 \cdot (p \cdot v)_a} \]  

(22)

Where

\[ (p \cdot v)_a = \min [(p \cdot v)_{am}; (p \cdot v)_{ai}] \]  

(23)

3.6. Other restrictions

- Sliding speed of the shaft must be taken into account to decrease the wear of the shaft and the operating temperature. The condition is:

\[ v = \frac{\pi \cdot d \, [\text{mm}] \cdot n \, [r.p.m]}{60 \cdot 1000} \approx \frac{d \cdot n}{20000 \, [\text{m/s}]} \leq v_a \]  

(24)

where

\(v_a\) – the allowable speed, specific for different materials
Available pressure $p_a$, has two specific values:
→ for each type of bearing material: $i$ values
→ other values proceeding from: $(p \cdot v)_a$ values
From figure 2, results that the couple of values $p_i, v_i$ must be limited of to two limits: $p_{\text{max}}$ and $v_{\text{max}}$.

4. Optimal design model

4.1. “A” alternative of calculation
From the (7) and (9) conditions, results:

$$B = \alpha \cdot d^3 = \frac{2.5 \sqrt{70} \cdot d}{10^3 \cdot \theta}$$

(25)

and

$$d_A = \frac{1.92}{(10^3 \cdot \theta \cdot \alpha) \cdot 0.385}$$

(26)

$$B_A = \alpha \cdot d_A^3$$

(27)

4.2. “B” alternative of calculation
From the (9) and (20) conditions, results:

$$B = \alpha \cdot d^3 = \frac{2.5 \sqrt{3600} \cdot d}{10^3 \cdot \theta}$$

(28)

And

$$d_B = 1.84 \cdot d_A$$

(29)

$$B_B = \alpha \cdot d_B^3$$

(30)
4.3. “C” alternative of calculation

From the (9) and (22) conditions, results:

$$\frac{\pi \cdot n \cdot F}{60 \cdot (p \cdot v)_a} \leq \alpha \cdot d^3$$

(31)

and

$$d_c = \frac{3 \sqrt{\frac{\pi \cdot n \cdot F}{60 \cdot \alpha \cdot (p \cdot v)_c}}}{3}$$

(32)

$$B_c = \alpha \cdot d_c^3$$

(33)

4.4. “D” alternative of calculation

From the (20) and (21) conditions, results:

$$\frac{1}{(10^3 \cdot \theta)} \cdot \sqrt{3600 \cdot \frac{\pi \cdot n \cdot F}{60 \cdot (p \cdot v)_a}}$$

(34)

$$d_P = \left[\frac{10^3 \cdot \theta \cdot \pi \cdot n \cdot F}{60 \cdot (p \cdot v)_a}\right]^{2.5}$$

(35)

$$B_P = \alpha \cdot d_P^3$$

(36)

This model of optimal design presents 3 cases discussed in the following section. [1,8]

5. Calculation steps for optimal design algorithm

5.1. Given data

- $\sigma_{al} [N/mm^2]$ – allowable stress for fatigue bending
- $F [N]$ – resultant force in bearing
- $\theta [rad]$ – rotation angle of the shaft
- $n [r.p.m]$ – rotational speed of the shaft
- $(p \cdot v)_m$ – cooling capacity of the machine
- Type of friction (dry, boundary).

5.2. Calculation coefficient determination:

The relationship (37) is used to determine the calculation coefficient

$$\alpha = \frac{\sigma_{al}}{5 \cdot F} [mm^{-2}]; \ 10^3 \cdot \theta \cdot \alpha [rad/mm^2]$$

(37)

5.3. Optimal solution for geometrical dimensions of the shaft

- If $d_c \leq d_A, d = d_A = B = \alpha \cdot d_A^3$

(38)

- If $d_c \in [d_A; d_B], d = d_c = B = \alpha \cdot d_c^3$

(39)
If \( d_c \geq d_B, d = d_D \Rightarrow B = \alpha \cdot d^3 \) \hspace{1cm} (40)

5.4. Final selection for bearing material

For the materials table, we select those ones that fulfill the followings conditions:

\[
(p_a)_{tab} \geq \frac{F}{B \cdot d}; \quad (v_i)_{tab} \geq \frac{n \cdot d}{20000}; \quad (p \cdot v)_c \geq (p \cdot v)_a
\] \hspace{1cm} (41)

5.5. Fit and tolerance calculation

- From standard (STAS), we determine
  - the upper deviation \( A_s \) – for the bearing and \( a_s \) – for the shaft
  - the lower deviation \( A_l \) – for the bearing and \( a_l \) – for the shaft

- Determine maximum and minimum clearance
  \[
  j_{\text{max}} = A_s - a_l
  \] \hspace{1cm} (42)
  \[
  j_{\text{min}} = A_l - a_s
  \] \hspace{1cm} (43)

- Determine probable clearance (14) and verify (13) relation
- Eliminate the fits non corresponding to (13) relation

6. Conclusions

a) In usual design:

In the beginning, we select the value \((B / D)\), no taking into account the type of material, but only the type of machine and then we determine the diameter of the shaft \((d_1 \text{ and } d_2)\), based on the bending of the shaft and the contact pressure and choose the highest value, which involves selecting the bearing material before calculating \(d_2\).

If the condition is not met, we select another material with modified ratio \((B / D)\) and repeat all the calculations, and if the condition is not met even in this situation, we change the diameter \(d\), the ratio \((B / D)\) and the bearing material and we repeat all the calculations.

b) In optimal design, we eliminate the drawbacks presented in section 1 and realize the following:

We impose at the beginning, restriction for:
- shaft material selection \((a_{sl})\)
- shaft, deflection (figure 1)
- cooling capacity by \((p \cdot v)_a\)

From equal strength condition of bearing and shaft, we determine four values of \(d(d_A, d_B, d_C, d_B)\) and we select a single value.

Material selection of bearing is performed after \( \theta \) and \( d \) determination but not arbitrary (as in usual design) and after we determine the type of fit and the tolerances, that is neglected in usual design.

7. References

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