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Empirical correlation of heat generation in ball bearings depending on the operational conditions in the supports of aero-engine rotor

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Abstract. Modern gas turbine bearings operate at high rotational speeds, with a speed index of up to $3 \cdot 10^6$ mm·rpm. The methods currently used to calculate heat generation in aero-engine split-inner-ring ball bearings do not always give correct results, because they were developed based on tests at relatively slow speeds. To solve this problem, a number of split inner-ring ball bearings with three-point contact bearings have been tested at various levels of load, rotational speed and oil flow rate. The following bearings are tested: 126114 ($d = 70$ mm), 126126 ($d = 130$ mm) and 126130 ($d = 150$ mm). Tests results allowed authors to develop an empirical correlation for heat generation in split inner-ring ball bearings. Comparison with experimental data, presented in the domestic and foreign literature, shows that the coefficients in the empirical formula are selected correctly and accurately describe the thermal state of the bearing.

1. Introduction.

Modern gas turbine engines' bearings operate at high rotational speeds, reaching the speed parameter $d_m \cdot n = 2.7...3 \cdot 10^6$ mm·rpm. To support the heavy thrust loads in aero-engines, split inner-ring ball bearings are used. The inner ring of these bearings consists of two axial halves. High speeds and load leads to a significant heat generation in ball bearings. As such, correct estimation of heat generation and determination of required amount of oil for heat removal from bearings are vital problems solved at the development stage of a gas turbine engine.

Heat generation in a bearing is related to the friction that occurs between bearing elements. For an oil-lubricated bearing, additional heat is generated due to the balls passing through the lubricant in the bearing's free space. Calculating heat generation in a bearing by accurately accounting all the factors of a complex process of friction is very difficult. In this case, the calculations should take into account the actual distribution of loads among the rolling elements, estimation of elastohydrodynamic lubrication impact and the actual distribution of speeds over the contact area of each rolling element [1, 2].

In this regard, for estimating heat generation in practice, empirical correlations are used, obtained by generalizing a significant amount of experimental data. In article [3], the authors present a review of domestic (such as CIAM [4, 5] and KAI [6]) and foreign (such as FAG [7], SKF [8], MTU [9]) empirical methods for estimating heat generation in split inner-ring ball bearings. A comparison of the calculated values of heat generation, obtained by various methods, with experimental data was carried out, which showed that for aircraft bearings operated at high rotational speeds, the CIAM [5] and MTU [9] methods show the best convergence with the experiment.

The purpose of this study was to develop an empirical correlation for heat generation in split inner-ring ball bearings for high rotational speeds.



2. Bearings testing

In recent years, CIAM has been conducted a number of researches on heat generation in split inner-ring ball bearings at various levels of load, rotational speeds and oil flow rates. The objects tested in the present study are split inner-ring ball bearings with three-point contacts, used in aircraft engines. The rings and rolling elements of the bearings are made of bearing heat-resistant steel EI347Sh (similar to bearing steel M50). The cage is made from bronze with an anti-friction coating. The following bearings are tested:

- 126114 (70x110x18 mm);
- 126126 (130x200x30 mm);
- 126130 (150x225x35 mm).

The bearings test was carried out on a bearing test rig T14-15/2 (figure 1), intended to test heavily loaded rolling bearings installed in the supports of the rotors of gas turbine engines.



Figure 1. Bearing test rig T14–15/2.

Monitoring the test modes, evaluating the technical condition of the examined bearings and the state of the equipment of the test bench was carried out by registering the following parameters:

- rotational speed (n);
- applied axial (F_a) and radial (F_r) loads;
- temperatures of outer (T1-T3) and inner (T4-T6) ring of bearing;
- oil temperature, inlet (T_{oil_in}) and outlet (T_{oil_out});
- oil flow rate (V);
- vibrations.

For lubrication and cooling, synthetic IPM-10 oil was used, with a viscosity of 3.47 cSt at 100°C. Filtration of the circulating oil was carried out with a 10-micron filter.

The tests were carried out at different oil flow rates through the bearing (from 2 to 10 l·min⁻¹). The amount of oil was determined by the number and diameter of the nozzles, as well as by the oil pressure in the oil system. The installation of the nozzles is shown in Figure 2. To ensure a flow rate in range from 2 to 5 l·min⁻¹, one nozzle (N1) was installed; for a flow rate in range from 5 to 7 l·min⁻¹, two nozzles (N1 and N2) were installed; and for a flow rate up to 10 l·min⁻¹, three nozzles (N1, N2 and N3) were installed.

To determine the temperature state of the bearing, thermocouples were installed on the inner and outer rings of the bearing. Thermocouples were also installed to measure the oil temperature at the inlet and outlet of the bearing. The layout of the thermocouples is shown in figure 2. A photograph of the bearing with the installed thermocouples is shown in figure 3.

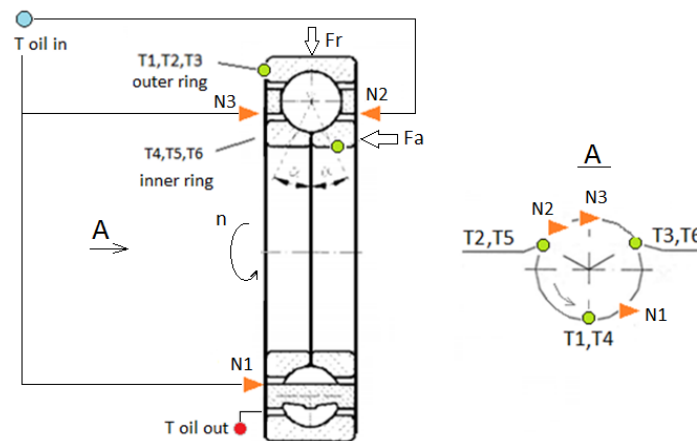


Figure 2. Layout of the thermocouple (T1–T6) and oil supply nozzles (N1–N3).



Figure 3. Installing the outer ring of bearing 126130 ($d = 150$ mm) in the housing with the thermocouples on the outer ring.

3. Empirical correlation of heat generation in the ball bearing.

To plan an experimental study of heat generation in split inner-ring ball bearings it is necessary to choose the sufficient number of experiments to obtain an empirical relationship, by which it is possible to determine the heat generation in a ball bearing under different operational conditions. An analysis of theoretical and experimental researches of ball bearings [5, 6, 9], carried out earlier, showed that heat generation from friction in a ball bearing (Q) depends on the bore diameter of the bearing (d), rotational speed (n), axial load (F_a), oil flow rate through the bearing (V), and viscosity or oil temperature (T).

The goal of the study was to obtain an empirical relationship $Q = f(d, n, F_a, V, T)$, which connects the heat generation in a ball bearing with the magnitude of these factors, kW:

$$Q = 10^{x_0} \cdot d^{x_1} \cdot n^{x_2} \cdot F_a^{x_3} \cdot V^{x_4} \cdot T^{x_5}, \quad (1)$$

where x_0, x_1, x_2, x_3, x_4 and x_5 are empirically determined coefficients.

After taking the logarithm of expression (1), the heat generation value can be represented as a linear

dependence on the parameters d , n , F_a , V and T :

$$\log_{10}(Q) = \log_{10}(10^{x_0} \cdot d^{x_1} \cdot n^{x_2} \cdot F_a^{x_3} \cdot V^{x_4} \cdot T^{x_5}), \quad (2)$$

$$\log_{10}(Q) = x_0 + x_1 \cdot \log_{10} d + x_2 \cdot \log_{10} n + x_3 \cdot \log_{10} F_a + x_4 \cdot \log_{10} V + x_5 \cdot \log_{10} T, \quad (3)$$

This method of obtaining empirical correlation is described in [9]. The authors of the article use in the formula for heat generation determination the oil inlet temperature instead of oil viscosity. Such an approach is convenient for engineering calculations because it connects the level of heat generation with the known operating conditions of the bearing. In addition, the difference between [4], [5] and [9] is that they investigate different ranges of variable parameters. Heat generation studies have been carried out with changes in the following independent factors:

- bore diameter (d), from 70 to 150 mm;
- rotational speed (n), from 1500 to 25000 rpm;
- axial load (F_a), from 2.9 to 34.3 kN;
- oil flow rate (V), from 2 to 10 l·min⁻¹;
- oil inlet temperature (T), from 30 to 115°C.

To develop the empirical dependence of heat generation, the range of the bearing's operating modes included 156 experimental points. For each of the indicated points, the experimental value of heat generation was calculated. Heat generation in a ball bearing was determined by the increase in the heat of the oil from the time it entered to the time it was removed from the bearing, without taking into account heat transfer through the parts associated with the bearing, as follows:

$$Q_{\text{exp}} = Q_{\text{oil}} = \frac{C_p \cdot \rho \cdot V}{60} \cdot (T_{\text{oil_out}} - T_{\text{oil_in}}), \quad (4)$$

where C_p is the heat capacity of the oil in J·(kg·°C)⁻¹; ρ is the density of the oil in g·sm⁻³; V is the oil flow rate in l·min⁻¹; and $T_{\text{oil_out}}$ and $T_{\text{oil_in}}$ are the oil temperatures at the outlet and inlet of the bearing, respectively, in °C.

4. Results and discussion.

The experimental values of heat generation for the bearings with bore diameters of $d = 70$ mm, $d = 130$ mm and $d = 150$ mm are shown in figure 4.

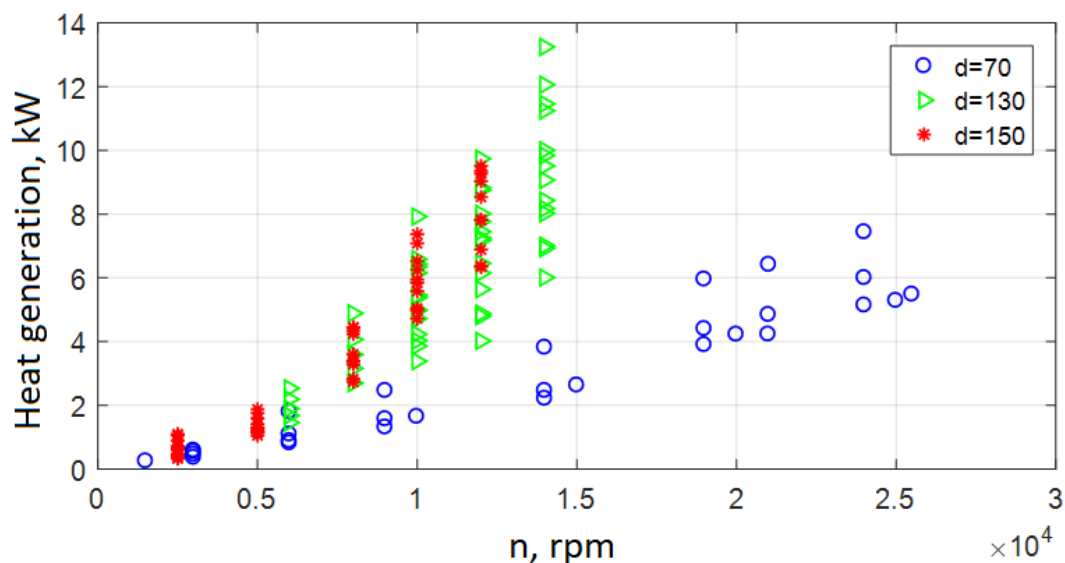


Figure 4. Heat generation in bearings and the experiment results.

The results of the experiment were processed by formula (3) using the least squares method, which allows the regression coefficients determination. These coefficients ensure the minimum of the sum of squares of the experimental data deviations from the values calculated by the usual linear regression equation (that is, the minimum of the following expression):

$$S = \left(\frac{1}{n} \sum_{i=1}^n (Q_{calc_i} - Q_{exp_i})^2 \right)^{1/2}, \quad (5)$$

where Q_{calc_i} and Q_{exp_i} are calculated and experimental values of heat generation, and n is the number of experimental points. According to the regression analysis results, the coefficients x_0, x_1, x_2, x_3, x_4 and x_5 are determined, representing the degrees of the independent parameters d, n, F_a, V and T in formula (1). For each point, the heat generation is calculated from the obtained empirical dependence (formula [1]).

Figure 5 presents a comparison of the obtained calculated values with experimental data. It can be seen from the figure that the calculated values correctly describe the thermal state of the bearings with a diameter of 70 to 150 mm within the studied modes. Some variation may be associated with different ambient temperatures during research. Depending on the air temperature, the intensity of the heat transfer from the body of the test unit to the environment changes.

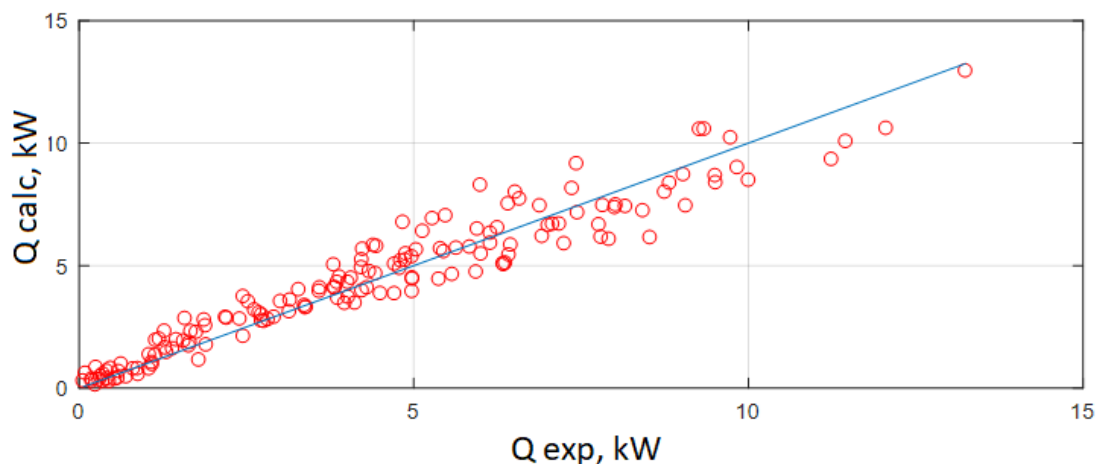


Figure 5. Comparison of calculated values of heat generation with experimental data.

It is worth noting that in order to ensure the performance of the bearing, the temperatures of the bearing parts and oil have primary importance. Heat-resistant bearing steels are designed for operation up to temperatures of 425...500°C. The limiting factor is the maximum allowable temperature of the oil. At high temperatures, the decomposition of oil additives occurs, and the oil begins to coke. In this regard, an important task at the design stage of the oil cooling system is to determine the required oil flow rate to ensure the oil temperature at the outlet of the bearing is below the acceptable level.

According to the obtained empirical dependence (1), the oil temperature at the bearing outlet was calculated (according to formula [4]). Figure 6 compares the calculated and actual oil temperatures at the bearing outlet. The results of the comparison show that, for 94% of the experimental points, the estimated oil outlet temperature differs from the actual temperature by no more than 10°C.

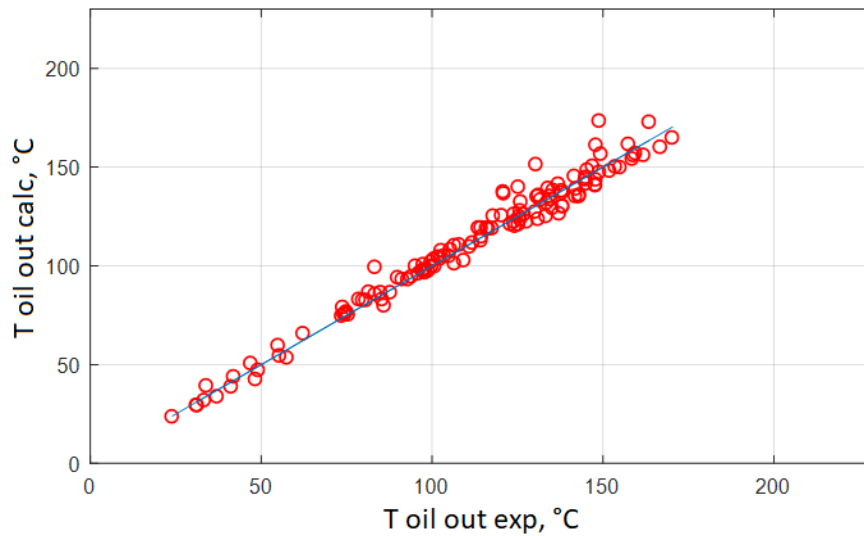


Figure 6. Comparison of the calculated and actual temperatures of the outlet oil.

To verify the correctness of the obtained coefficients from the empirical formula, the calculated values were compared with the experimental data presented in the literature by researchers from KAI (Russia) [6], MTU (Germany) [10] and the National Aeronautics and Space Administration (NASA; USA) [11]. Figure 7 shows graphs comparing the experimental and calculated heat generation values. It can be seen from the figure that the obtained empirical correlation correctly describes the bearing thermal state for these tests. Some deviations may be due to the difference between the viscosity of the oil used in the experiments and the viscosity of the IPM-10 oil.

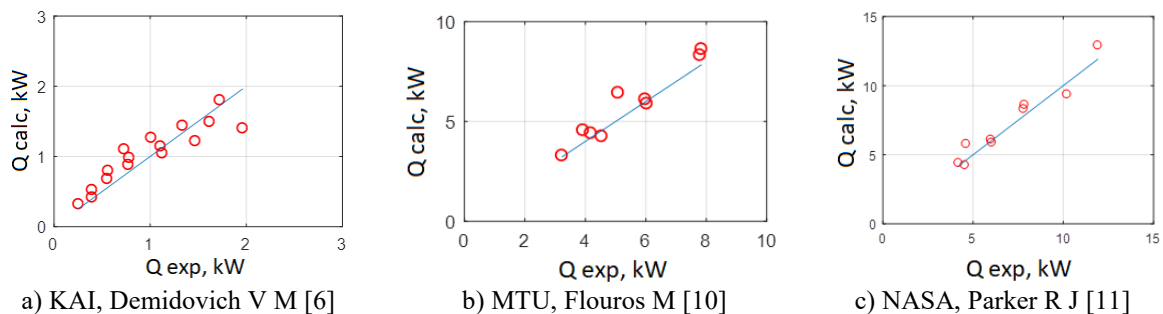


Figure 7. Comparison the experimental data presented in the literature, with the calculated data obtained from the developed empirical correlation.

5. Conclusion.

The authors have developed an empirical relationship for determining heat generation in split inner-ring bearings at a wide range of rotational speeds, loads, oil flow rates and bore diameters, which correctly describes the experimental results in the specified region of the defining parameters:

- bore diameter (d), from 70 to 150 mm;
- rotational speed (n), from 1500 to 25000 rpm;
- axial load (F_a), from 2.9 to 34.3 kN;
- oil flow rate (V), from 2 to 10 l·min⁻¹;
- oil inlet temperature (T), from 30 to 115°C.

A comparison with the experimental data presented in the literature shows that the coefficients in the empirical formula are selected correctly and accurately describe the thermal state of the bearing.

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