30th Congress of the International Council of the Aeronautical Sciences
DCC, Daejeon, Korea ; September 25-30, 2016

CONTROL OF CONDITION OF GAS TURBINE ENGINE ROTOR BEARINGS IN DIAGNOSTIC PARAMETERS IN A DIFFERENT PHYSICAL NATURE

Kiselev D.Y., Kiselev Y.V., Birukov R.V.
Samara State Aerospace University

Keywords: *vibration parameter, temperature, rotor bearing, power consumption*

Abstract

The article proposes the use of a corrected oil temperature values for the diagnosis of gas turbine engines bearings. In the case where the oil temperature measurement is not possible, to determine the technical condition are invited to use the vibration spectrum on rotor rundown.

General Introduction

To 25% cases of failure of gas turbine engines, it is connected with destruction of rotor bearings that refusals lead to serious effects. Therefore, the problem of providing effective control of gas turbine engine bearings is actual.

Development bearing fault is accompanied by a change in vibration, increase of heat release and concentration of metals in oil.

Practice shows that most effectively malfunctions of bearings become known in a temperature and vibration parameters [1]. As a rule, control of a specific rotor bearing is implemented either on temperature, or in vibration parameters. Therefore, for coverage by a monitoring system of all rotor bearings it is necessary to combine methods of temperature and vibration diagnostics.

Temperature Condition of Bearing

In the majority of monitoring systems, access control of temperatures is implemented. This method allows revealing change of a state only at a late stage of fault development. For earlier detection of bearings malfunctions, it is offered to use algorithms of a temperature trend analysis. In too time, application of algorithms of the trend analysis enough essential fluctuations of bearing temperature at changes of a mode and operating conditions of a gas turbine engine are complicated in a look. It is offered to use temperature corrected taking into account the mode and operating conditions of a gas turbine engine in algorithms of the trend analysis.

Most fully the question of temperature condition of gas turbine engines bearings is considered in work [2]. Temperature of bearings depends on several major factors:

- power consumption, in other words, thermal emissions in a bearing;
- external heat input (through configuration items of support or because of bearing blowing hot air);
- pumping of oil and temperature of oil on input.

In work [2] division of power consumed by the ball bearing Q_Σ on two components is accepted: hydrodynamic losses Q_{hydr} (on lubricant hashing) and friction losses Q_{fric}:

$$Q_\Sigma = Q_{hydr} + Q_{fric}$$

Hydrodynamic losses in work [2] are calculated with use the equation:

$$Q_{hydr} = 16.6 \cdot 105 \cdot Re^{-1.25} \cdot Pr^{-1} \cdot z_b \cdot d_b \cdot \rho \cdot u^3 \quad (1)$$

where u - the circumferential speed of separator

$$u = \frac{\pi \cdot (d_m - d_b \cdot cos\gamma)}{120} \cdot n,$$

$Re = \frac{u \cdot d_b}{\nu}$ - Reynolds number,
$Pr = \frac{\nu}{a}$ - Prandtl number,
z_b- number of rolling element,
d_b - diameter of rolling element,
ρ - lubricant density,
n – rotor speed.

In expression (1) as variables only rotor speed n and entry temperature of oil $t_{o\ en}$ (through viscosity Prandtl number).

After making all the values of constants in coefficient a_1, exponent of rotor speed b_1 and at oil temperature c_1 we obtain an expression for hydrodynamic losses:

$$Q_{hydr} = a_1 \cdot t_{o\ en}^{c_1} \cdot n^{b_1} \quad (2)$$

Friction losses are defined as follows [2]:

$$Q_{fric} = C_1 \cdot z_b \cdot d_b^2 \cdot \rho \cdot u^3$$

$$C_1 = \varphi\ (Re, Eu, Pr) = 14{,}7 \cdot 10^{-5} \cdot Re^{0{,}214} \cdot Eu^{0{,}287}\ Pr^{0{,}44},$$

Where

$Eu = \frac{P_{al}}{\rho \cdot (u \cdot d_b)^2}$ - Euler number

$P_{al} = \frac{A}{z_b}$ – average load on rolling element

A - an axial force operating on the bearing.

The last expression can be transformed like expression for hydrodynamic losses:

$$Q_{fric} = a_2 \cdot t_{o\ en}^{c_2} \cdot n^2$$

As a result, expression for dissipated power by a bearing:

$$Q_\Sigma = a_1 \cdot t_{o\ en}^{c_1} \cdot n^{b_1} + a_2 \cdot t_{o\ en}^{c_2} \cdot n^2.$$

The bearing temperature t_n and flow q_{oil} based on entry temperature of oil $t_{o\ en}$ is defined by

$$t_n = t_{o\ en} + \frac{3600 \cdot Q_\Sigma}{C_p q_{oil}}.$$

For accounting of external heat input to the bearing on work [2] the introduced additive Δt_n:

$$\Delta t_n = \chi \cdot Q_{ext} \cdot l,$$

where χ - experimentally define coefficient,

Q_{ext} - heat flow of external heat additive,

l - characteristic dimension.

Taking into account earlier made transformations the model for calculation of bearing temperature in look is offered

$$t_n = n_{rel}^a \cdot t_{oil\_in\_rel}^b \cdot G_{oil\_rel}^d \cdot c \qquad (3)$$

where: t_n - the estimated temperature of the corresponding bearing;

n_{rel} - the physical turns of the corresponding rotor carried to some chosen characteristic operational mode;

$t_{oil\_in\_rel}$-the oil temperature on an entrance carried to some chosen characteristic operational mode;

G_{oil_rel}-relative pumping of oil;

- a, - b, - d, c - empirically received coefficients.

It is model allows to enter corrections to the taken temperature for taking note of the mode and working conditions.

With use of expression (3) comparison of the values of thrust bearing temperature received on model with measured is carried out. Results of comparison are given in figure 1.

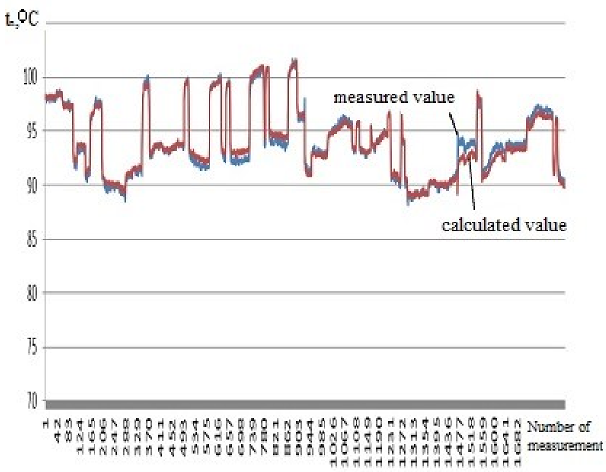

Fig.1. The measured and model value of temperature of rotor thrust bearing

The figure shows the coincidence of the model and the measured value.

Comparison of temperature calculations results for this dependence with experimental data, showed discrepancy no more than 5%.

For the solution of diagnostics problems, it is necessary to use the received model for "adjustment", or in other words - reductions of the measured values of temperature according to the actual working conditions. In this case the corrected (given) value ideally at steady-state technical condition has to remain constant.

Correction of values of temperatures of bearings is carried out on the formula received by transformation of formula (3):

$$t_n = t_{n\_det} \cdot n_{rel}^{-a} \cdot t_{oil\_in\_rel}^{-b} \cdot G_{oil\_rel}^{-d} \cdot c \qquad (4)$$

where: t_{n_det} - the calculated temperature of the corresponding bearing.

In figure 2 the example of formula application for correction of gas turbine engine bearings temperatures values is given.

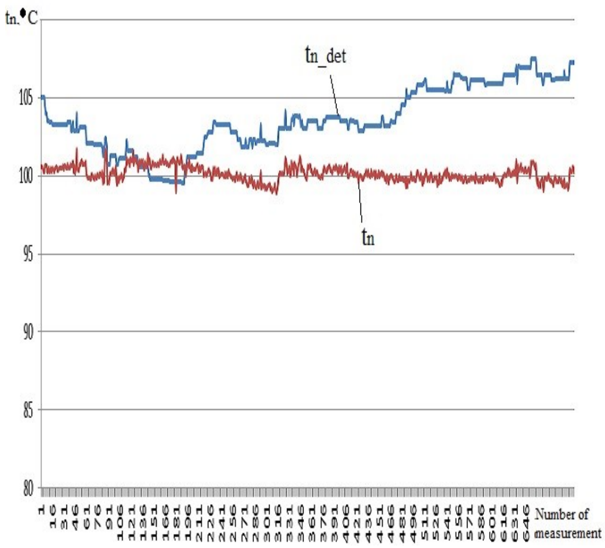

Fig. 2. The measured and corrected value of bearing temperature

When using the formula (4) eliminates the disadvantage of the model associated with the lack of taking into account individual measurements of the bearings temperature for different engines, and used part of the formula, reflecting the impact of the regime on the temperature and operating conditions.

Apparently from drawing, corrected on external conditions and operational mode of temperatures in good repair the smaller dispersion, in comparison with the taken temperature matters significantly. It gives the grounds for use of the corrected temperature for the trend analysis. Implementation of a trend analysis will allow to reveal malfunctions of rotor bearings at earlier stage.

Vibration Monitoring of Bearing

It is constructive it is not always possible to implement measurement of temperature parameters on specific gas turbine engine. Therefore, ensuring control of all bearings requires measurement of diagnostic parameters of other physical nature.

In literature [3] it is specified essential opportunities of vibration diagnostic methods in relation to diagnosing of gas turbine engine including their bearings. It is recommended to trace for assessment of bearings condition in vibration range the following characteristic components generated by a bearing:

- vibration with frequency of swing bodies rolling on outside bearing body;
- vibration with frequency of swing bodies rolling on internal bearing body.

Frequencies of these components are connected strictly with rotating speed of rotor $K \cdot f_{rot}$. Specific values of the bearing frequencies for a given geometry of a bearing is calculated by well-known formulas. At emergence on bearing tracks, rolling elements or retainer of defects there is increase in amplitude vibrations at these frequencies (concerning range in good repair). However, detection of these components in range of gas turbine engine vibration for a number of reasons represents difficult task.

Measurement of vibrations is in most cases carried on outer surfaces of an engine bearing support housing. Vibration of gas turbine engine is defined by set of different origin vibrations - first of all rotor, and also aerodynamic, chamber, bearing, etc. When passing signal of vibration from bearings to sensing point there is its attenuation, damping in design joints, especially taking into account that bearing support are carried out by elastic and damping. As a result, the size of bearing vibration on the body can become so small in comparison with rotor and aerodynamic that its impossibility to find in range.

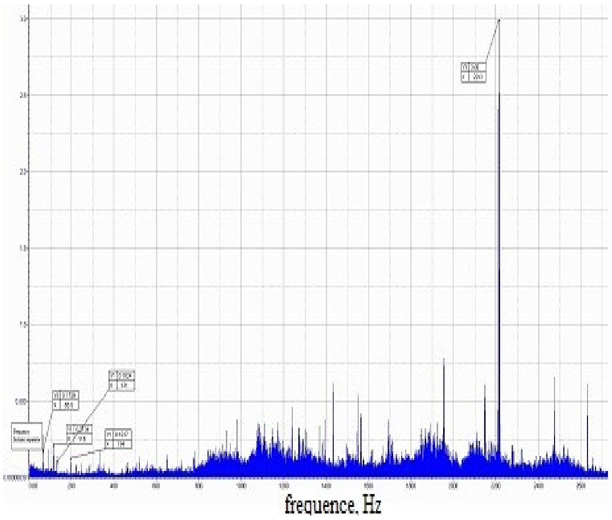

Fig. 3. Range of gas turbine engine vibration

Figure 3 shows the spectrum of three-rotor gas turbine engine vibration. The spectrum is uniquely identified by the rotor harmonics and aerodynamic vibrations generated by an engine fan. Identification of most of the other components is difficult and ambiguous.

In work [3], it is offered to spend for reliable components identification to carry out the vibration analysis of at engine rotors runout with use of the cascade mode (three-dimensional) spectral analysis (in coordinates "frequency-amplitude-time"). In work [4] emergence of resonance for low-power components of vibration, at certain moments of runout is noted.

For the solution of problem of control of three-rotor gas turbine engine bearings it has been decided to carry out the three-dimensional spectral analysis of vibration with to use the mode of cold scrolling including site of forced promotion of high pressure rotor starter and then after starter disengagement - section of engine rundown.

Such approach gives number of advantages:

- unambiguous definition of interrelation of vibration components and rotor speed (on cascade ranges most vibration components represent as if "trajectories", equidistant to change of rotating speed on transient regimes of work - thus, the components which are rigidly connected with rotor speed are obviously visible);

- elimination of accidental and noise components;

- when using the mode of dry motoring for the analysis of vibration a range turns out significantly "simpler", than on operational modes of work that is connected, in particular, with lack of chamber origin vibrations.

By results of the cascade spectral analysis it was succeeded to identify two components of the intershaft bearing between the high pressure and medium pressure rotors rolling bodies roll on internal and outside bearing rings (figure 4).

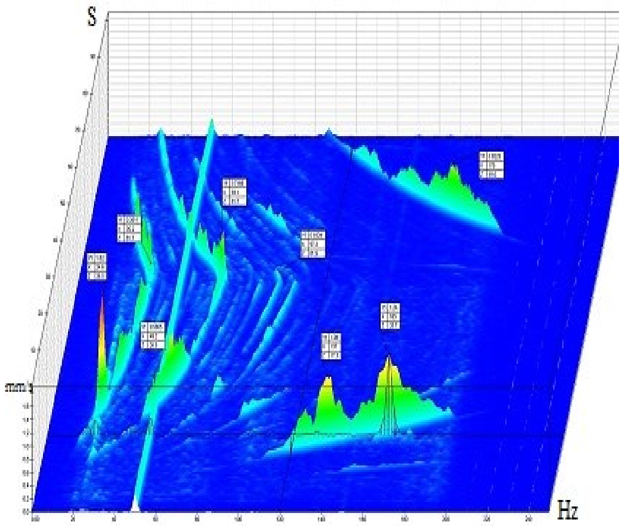

Fig. 4. The cascade range received on a dry motoring

After three-dimensional analysis has been carried out the watching spectral analysis of the identified components of vibration (figure 5).

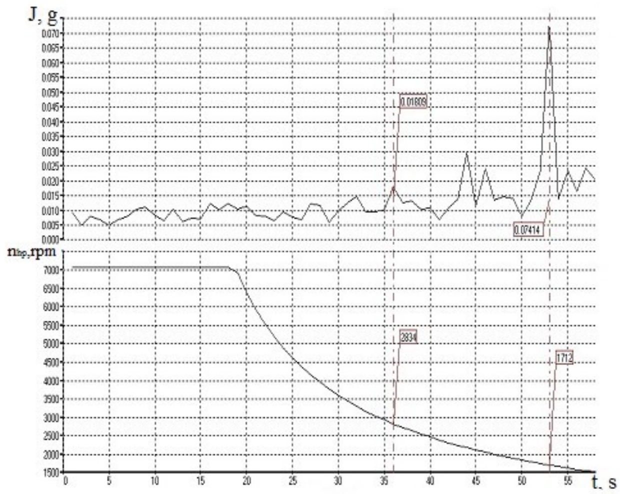

Fig. 5. Results of the watching analysis behind the frequency of rolling of bodies of swing on outside holder on runout

Let's notice that in drawing the moments of resonant strengthening of signal are well visible. The difference in vibration size in comparison with initial state is fault development sign.

Thus, operational use of the developed technique has allowed to identify unambiguously components of the of the intershaft bearing between the high pressure and medium pressure rotors that does possible fault detection of the bearing at early stage of development.

The provided results show that joint carrying out the trend analysis on the corrected temperature and the watching analysis of vibration on bearing frequencies on the mode of "runout" significantly increases reliability of fault detection of the gas turbine engines rotor bearings at early stage that will allow passing to operation of gas turbine engine on condition.

References

[1] Lulko V I. *Operation of aircraft engines on technical condition (the theory and practice).* MSU, 2002.

[2] Demidovich V M. *Research of the thermal mode of gas turbine engines bearings.* Mechanical engineering, 1978.

[3] Egorov I V, Karasyov V A and Skibin V A *Information technologies in diagnostics of technical condition of gas turbine engine.* TORUS PRESS, 2011.

[4] Nemkov V, Shabayev V, Leontyev M, Vinogradov S, Potapova O, Garanin I. Use of the mode of runout of rotors for definition of resonant characteristics of gas turbine engines. *The engine*, Vol. 6, No. 36, pp 114-117, 2004.

Contact Author Email Address

mailto:eat@inbox.ru

Copyright Statement