The rotating system vibration and diagnostics

M. Vasylius*, R. Didžiokas**, P. Mažeika***, V. Barzdaitis****

*Klaipėda University, Bijužnų str. 17, 91225 Klaipėda, Lithuania, E-mail: mvasylius@yahoo.com
**Klaipėda University, Bijužnų str. 17, 91225 Klaipėda, Lithuania, E-mail: rimantas.didziokas@ku.lt
***Kaunas University of Technology, A. Mickevičiaus str. 37, 44244 Kaunas, Lithuania, E-mail: pranasmazeika@centras.lt
****Kaunas University of Technology, A. Mickevičiaus str. 37, 44244 Kaunas, Lithuania, E-mail: vytautas.barzdaitis@ktu.lt

1. Introduction

The stationary condition monitoring, safety and diagnostic systems are mainly applied to technologically critical, expensive machines and based on vibration and technological parameters measurements [1 - 6]. The air blower machine SF01-18 is the high efficiency technological equipment operating in continuous long term running mode in the chemical plant JSC Lifosa, Fig. 1. The machine’s induction electric motor EM vibration sources were identified and eliminated [2]. The motor experimental testing, modeling and simulation explained the main reason of high vibration level - insufficient dynamic stiffness of rotor with journal bearings. In this article the condition and failure diagnostics of the radial tilting-pad journal bearings of the blower rotor BR is monitored, experimentally tested modeled and results implemented in industry.

2. Air blower condition monitoring and diagnostics

The air blower machine SF01-18 comprises electric motor EM, gear box GB and blower rotor BR, as shown in Fig. 1. The 5.6 MW power induction electric motor runs at 1500 rpm and rotates an air blower rotor BR through flexible 8 pin type coupling C1, gear box GB and flexible coupling C2 at 3119 r/min.

For experimental measurements the following test and diagnostic equipment has been used: Dynamic Machine Analyser DMA04 (Epro, Germany, Profess s.r.o., Czech Republic), portable high technology vibration signal analyzer A4300, DDS2000 Database (Adash s.r.o. Czech Republic). The technical condition is evaluated by monitoring bearings housings vibration periodically and BR rotor shaft vibration displacements permanently.

This article concerns dynamic behavior of the blower rotor BR tilting-pad journal bearings. The task is to determine the main reason of the journal bearings failure. The blower rotor shaft vibration displacement $s_{p-p}$ and $s_{\text{max}}$ values at 7th bearing and maximum vibration displacement $s_{\text{max}}$ values of the shaft at the middle location point between 7th and 8th bearings are monitored by means of 7X, 7Y proximity probes, Fig. 2. Additionally, the vibration displacement of EM 2nd bearing was monitored periodically with proximity probes 2X, 2Y and evaluates technical condition not only of 2nd bearing but flexible coupling C1 too.

The failure of 7th (Fig. 3) and 8th tilting-pad journal bearings caused additional experimental testing and theoretical modeling of blower rotor dynamics.

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The failure of 7th (Fig. 3) and 8th tilting-pad journal bearings caused additional experimental testing and theoretical modeling of blower rotor dynamics.
Fig. 4 The blower rotor 7th bearing shaft kinetic orbits: a) at full loading: $s_{\text{max}} = 21 \mu m$ at 3132 rpm (52.2 Hz) steady-state rotational speed; b) at coast down running mode: $s_{\text{max}} = 112 \mu m$ at 1458 rpm (24.3 Hz) resonance speed; c) at coast down running mode: $s_{\text{max}} = 30 \mu m$ at 1038 rpm (17.3 Hz) low rotational speed.
Fig. 5 The gapB values (a-vertical direction, sensor 7Y) trend plot (vertical axes–gap values in µm) versus real time (horizontal axes – time in minutes:seconds) at coast down running mode at the 7th bearing shaft: vertical direction plot - gap 805 µm at 3132 rpm, 799 µm at resonance 1450 rpm and 607 µm at 0 rpm

3. Modeling and simulation of blower rotor

The main task of theoretical modeling is to evaluate forces acting on journal tilting-pad bearings.

The second type of Lagrange equations to construct the equations of system vibration for complicated vibration system is recommended to use. These equations are applied for air blower vibration system:

\[
\frac{d}{dt}\left(\frac{\partial T}{\partial q_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial R}{\partial q_i} + \frac{\partial V}{\partial q_i} = F_i(t)
\]

\(i = 1, 2, \ldots, n\)

(1)

here \(T, V\) are kinetic and potential energies respectively, \(R\) is Rayleigh dissipative function, \(q_i\) are system generalized coordinates, \(F_i(t)\) are external forces.

The dynamic model of all system in quiet state is showed in Fig. 6.

There are used three types of coordinate systems of axes to localize rotor position (Fig. 7).

1) Fixed coordinate system \(O_1 X_1 Y_1 Z_1\). Point \(O_1\) in steady state coincident with point where shaft is attached to the rotor. Weight of the rotor is not evaluated.

2) In the coordinate system \(O_2 X_2 Y_2 Z_2\) point \(O_2\) is tightly connected with the rotor. When the rotor rotates, center \(O_2\) moves in plane \(O_1, Y_1, Z_1\). Then displacement of \(O_2\) in the system of axes \(O_1 X_1 Y_1 Z_1\) is described by generalized coordinates \(\xi\) and \(\eta\).

3) System \(O_3 XYZ\) tightly connected to the rotor, moving with it about point \(O_2\). The rotor in movement can incline by small angles \(\phi_x, \phi_y\) about axes \(Y_2, Z_2\) and large angle \(\phi_z\) about axis \(X_2\). Coordinates of the rotor mass center is \(x_0, y_0, z_0\).

The kinetic energy when rotor shaft is rotating with constant angular velocity is described by Eq. (3) [7]. The influence of gyroscopic effect isn’t estimated here.

Fig. 6 Dynamic model of the air blower rotor: \(a_1 = 0.844\) m, \(a_2 = 0.5\) m

Generalized coordinates are taken:

\[q_i = \left\{\eta, \xi, \phi_x, \phi_y, \phi_z\right\}^T\]

(2)
\[ T = \frac{1}{2} \left[ \frac{1}{2} \left( m \left( \eta^2 + \zeta^2 \right) + I_x \omega_x^2 + I_y \omega_y^2 + I_z \omega_z^2 \right) \right] + \left[ \left( I_x \cos \theta + I_y \sin \theta \right) \omega_x \omega_y + \left( I_x \cos \theta - I_y \sin \theta \right) \omega_x \omega_y + m \eta \left( x_0 \omega_y \cos \theta - y_0 \omega_x \sin \theta \right) \right] + m \zeta \left( x_0 \omega_y - y_0 \omega_x \cos \theta - z_0 \omega_z \sin \theta \right) \] (3)

here: \( \omega_x \) - shaft (not rotor) angular velocity; \( \omega = \) constant; \( I_x, I_y, I_z, I_{xy}, I_{xz}, I_{yz} \) are moments of inertia of the rotor; \( m \) - weight of the rotor.

To estimate the influence of gyroscopic effect some new values must be included to Eq. (3). They were described in Eq. (4).

\[
\begin{align*}
\omega_x &= \omega + \phi_y \sin \phi, \\
\omega_y &= \phi_y, \\
\omega_z &= \phi_z \cos \phi, \\
\end{align*}
\] (4)

Kinetic energy equation is differentiate by generalized coordinates velocities \( \eta, \zeta, \phi_y, \phi_z \) and by displacements \( \eta, \zeta, \phi_y, \phi_z \).

The received eight differential equations of motion of the rotating system are described in compact matrix form.

\[
[A] \{q\} + [B] \{q\} + [C] \{q\} = \{F(t)\}
\] (5)

here \( \{q\} = \{\zeta, \eta, \phi_y, \phi_z\} \) is vector of generalized coordinates, \([A]\) is matrix of inertia, \([C]\) is matrix of stiffness, \([B]\) is matrix of damping and gyroscopic forces; \(\{F(t)\}\) is generalized vector of external excitation, components of which are the functions of time \( t \), angular speed \( \omega \), rotor moment of inertia.

4. Simulation results

The simulation results are shown in Fig. 8 (the plots scale is the same). They approved that vibration displacement of the rotor are caused not only by unbalance, but by gyroscopic effect too. In case of rotor center of mass shift from geometrical axis in 0.02 mm, when mass of the rotor is 2900 kg, rotational speed 3120 rpm, inertia force is 6190 N. Gyroscopic moment of the rotor have the same influence. Nominal rotational speed of the rotor is more than twice higher than its resonance frequency as experimentally measured ~23 Hz. The transient rotation running processes of BR acceleration and deceleration should be performed as fast as possible to pass resonance frequency.

Graph results corresponds the values on rotor and shat connecting point. To get the values on 3st bearing displacements \( Y_3 \) and \( Z_3 \) it is needed to multiply values of \( \zeta \) and \( \eta \) by the coefficient, equal to 0.524, to get 4nd bearing displacement \( Y_4 \) and \( Z_4 \) it is needed multiply the mentioned values by the coefficient equal to -0.279.
The simulation results acquired when: mass moment of inertia of the rotor $I_x = 436 \text{ kg m}^2$, $I_y = I_z = 287 \text{ kg m}^2$ and coefficients of stiffness of the support $k_{xy} = k_{xz} = 2 \times 10^8 \text{ N/m}$, $k_{yz} = k_{zy} = 1.5 \times 10^8 \text{ N/m}$, coefficients of damping of the support $c_{xy} = c_{xz} = 8000 \text{ Ns/m}$, $c_{yz} = c_{zy} = 4000 \text{ Ns/m}$.

5. Conclusions

1. The slow down mode is most dangerous running mode for the machines bearings. The shaft rubbing process starts at resonance and takes 14-15 minutes time interval till stoppage of the rotor.

2. The gyroscopic effect of the blower wheel changes shafts positions in the 7th and 8th bearings and increase rubbing process: the experiment measurements indicated that 7th baring shaft goes up (in vertical direction) ~200 µm.

3. Experimental testing results approved theoretical modeling and simulation data.

References


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THE ROTATING SYSTEM VIBRATION AND DIAGNOSTICS

Summary

The condition monitoring, protection and fault diagnostics system is inherent module in the failure prevention technology of modern machines. The paper is dedicated to research of dynamics, condition monitoring, fault diagnosis and modeling of high speed blower rotor failure prognosis. The specificity of the dynamic model comprises dynamics of the tilting-pad journal bearings. The blower rotor vibration displacements were monitored with contact less sensors during nominal load, run up and shout down modes. Hypotheses about physical nature of bearings failure effect are verified by experimental and theoretical research, where dynamical model was designed and simulated.

M. Vasylius, R. Didžiokas, P. Mažeika, V. Barzdaitis

ВИБРАЦИИ И ДИАГНОСТИКА РОТОРНОЙ СИСТЕМЫ

Резюме

Система мониторинга технического состояния и диагностики отказов является неотъемлемым модулем в технологии предотвращения отказов в современных машинах. Статья посвящена исследованию технического состояния, моделированию динамики и прогнозированию случайных отказов воздуходувки, рабочей в химической промышленности в продолжительном непрерывном режиме. Специфика динамической модели состоит в оценке динамики сегментных подшипников скольжения. Вибрации вала ротора в подшипнике измерены сенсорами индукционного типа при номинальной нагрузке и в режимах ускорения и торможения машины. Гипотеза о причинах повреждения подшипника скольжения проверена теоретическими и экспериментальными исследованиями.

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