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RELIABILITY IMPROVEMENT OF ACCESSORY GEARBOX BEVEL DRIVES

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Abstract

Bevel gear drives of modern aviation engines operates with high rotational speeds and transmitted torques. Dynamic loads in gear mesh actuates gear rim oscillations.

Coincidence of dynamic load frequency and bevel gear natural frequency can cause oscillation amplitude grow and gear rim breakdown. In this paper method of bevel drive vibration condition modeling is submitted. Efficiency of Coulomb friction damper application for resonance oscillation amplitude decrement is shown.

1 Problem definition

During exploitation of aviation motors some defects of central drive and angle drive bevel gear take place. They are caused not by tooth bending stress or contact stress because of the force in the gear mesh, but by alternating stress in gear rim because of gear's resonance oscillations [1]. In most cases fatigue crack has been developed from minor module tooth slot, where bending stresses because of the force in mesh are low.

2 Harmonic response analysis of bevel gear

Natural forms of bevel gears oscillations, which are situated near its operating range, are forms of oscillations by one, two or three nodal diameters, or by "umbrella" form. For the estimation of the most dangerous natural form in terms of first principal stresses in bevel gear rim harmonic response analysis by finite elements (FE) is made. Solid model of studied bevel gear is shown in Fig.1.

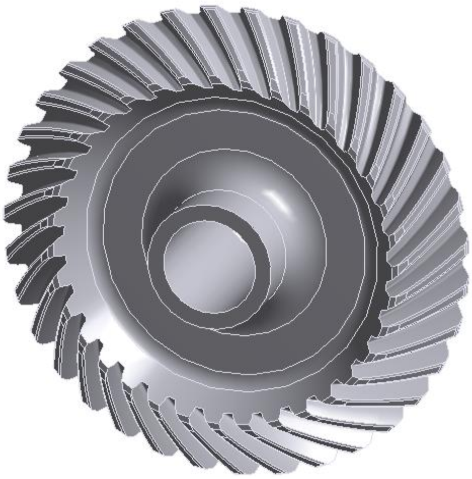

Fig. 1. Solid model of bevel gear.

FE model of bevel gear is supported by bearings. A force in axial direction with harmonic law of variation is applied to the gear tooth.

Structural damping was taken in account by damping ratio coefficient, which is estimated by this equation:

$$\xi = \frac{\delta}{2\pi} \qquad (1)$$

where δ - damping decrement in gear material.

Before harmonic response analysis a modal analysis was performed. System response during harmonic analysis is calculated as a superposition of calculated natural forms oscillations.

Amplitude-frequency characteristic and phase-frequency characteristic by first principal stresses in tooth root is calculated (Fig. 2)

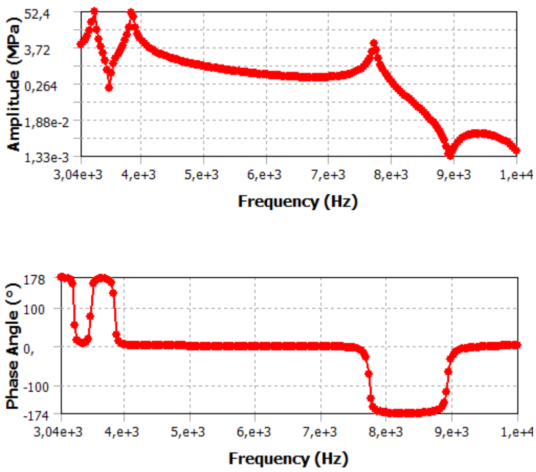

Fig. 2. Amplitude-frequency characteristic and phase-frequency characteristic by first principal stresses.

As it is shown in Amplitude-frequency characteristic, most dangerous are oscillations by “umbrella” and two nodal diameters natural forms. During such oscillations tooth slot is a stress raiser (Fig. 3).

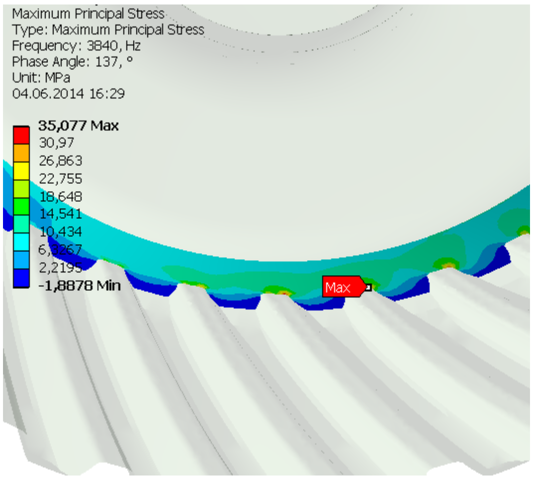

Fig. 3. Deflected mode of gear during oscillations by two nodal diameter natural form.

3 Bevel gears design specificity

An existing bevel gears strength calculations take into account only such geometry parameters, as normal or transverse module, shaft angle, helix angle, pressure angle at normal section, etc.

Diaphragm configuration has to be checked by natural frequencies estimation and Campbell diagram definition [1]. If natural frequency of bevel gear occurs in operating range, configuration of diaphragm should be changed. By the way, big amount of bevel gears in exploitation has a natural frequencies in operation range. The reason is that some machines has a big operation range. Moving out of this range one natural frequency can cause another frequency appearing in operation range.

Mentioned above requirement to bevel gear rim has a shortcoming. From the one side, in case of resonance oscillations a fatigue safety factor can be sufficient. From the other side, in case of near resonance oscillations in bevel gear diaphragm a high stresses in gear rim could occur.

In case of defect detection in product reliability of bevel drive should be approved by tests. During this tests strain gauging should be performed. The criteria of gear strength is a level of stress, detected by strain gauges and calculated using a stress concentration coefficient to a realm of the highest stress concentration. Such kind of tests that suppose current collector mounting in gearbox and rotating gear preparing is very labor-consuming.

4 Analysis of disturbing force, generated by gear mesh

Development of bevel gears vibration condition modeling methods is one of the ways of gear failure prevention and increasing of their reliability.

During harmonic response analysis it is assumed that disturbing force has a harmonic law of variation. In real gear mesh it is not so. By the influence of static component of transmitted torque gear rim is being deformed. It causes to a shift of the teeth profile curvature surfaces [2]. Teeth in mesh engage in contact before theoretical line of contact, what causes to an impact interaction between teeth.

Amplitude of disturbing force in gear mesh depends on following factors:

1) Design
2) Technology
3) Exploitation

Design factors are diaphragm configuration (stiffness and mass distribution) and stiffness of gear bearings. Technological factors are machine tool adjustment which provide a gear modification [3]. Gear modification is a variation of a tooth theoretical working profile to balance its deformation under the operating load.

To calculate a gear mode of deformation, a model, which can handle all mentioned above factors should be developed.

5 Gear vibration state calculation in time domain

Finite Element Analysis (FEA) is used for bevel gear vibration state modeling. A solid model of gear and pinion is developed. A contact interaction between teeth curvature surfaces is modeled (Fig. 4).

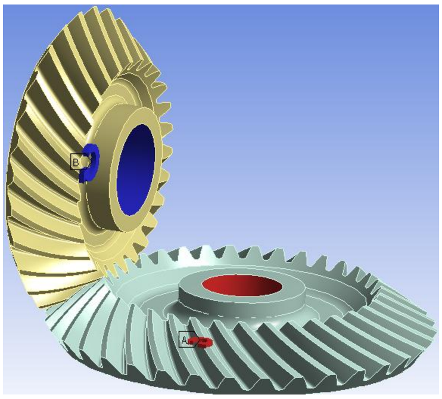

Fig. 4. Dynamic model of gear mesh

As a boundary conditions model is supported in bearings. A rotation angle is applied to driving pinion. A transmitted torque is applied to driven gear. An integration step is $5 \cdot 10^{-6}$ s.

Dynamic behavior of gear mesh is described by the following equation:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K(u)]\{u\}=\{F(t)\} \qquad (2)$$

where [M] – mass matrix,

[C] – damping matrix

[K(u)] – nonlinear stiffness matrix,

u – node displacement,

F(t) – disturbing force.

Solution of this equation is found by Newmark method. Displacement of nodes are developed in Taylor row by the following equation:

$$\{\dot{u}_{n+1}\}=\{\dot{u}_n\}+[(1-\delta)\{\ddot{u}_n\}+\delta\{\ddot{u}_{n+1}\}]\Delta t \qquad (3)$$

$$\{u_{n+1}\}=\{u_n\}+\{\dot{u}_n\}\Delta t+[(0.5-\alpha)\{\ddot{u}_n\}+\alpha\{\ddot{u}_{n+1}\}]\Delta t^2 \qquad (4)$$

Parameters α and δ are calculated by following equations:

$$\alpha=\frac{1}{4}(1+\gamma)^2 \qquad (5)$$

$$\delta=\frac{1}{2}+\gamma \qquad (6)$$

where γ - numeric damping value of Newmark method.

A s a result of calculation full information of bevel gear dynamic state in operating modes is obtained, including disturbing force in gear mesh, contact pressure distribution (Fig. 5) and first principal stresses in gear rim in time domain (Fig. 6).

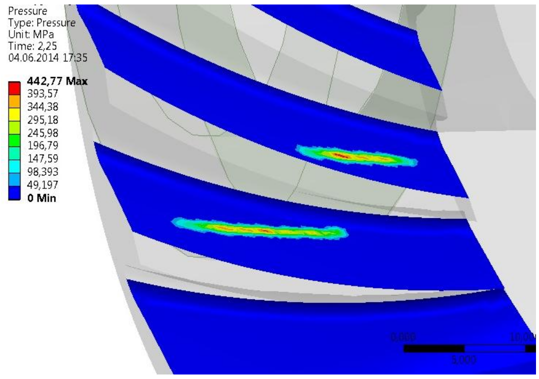

Fig. 5. Contact pressure in working surface of driving

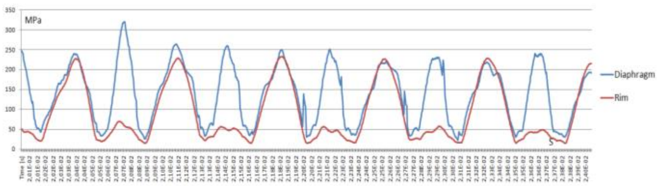

Fig. 6. Graph of first principal stresses in gear rim and diaphragm

Obtained graphic of maximum principal stresses can be used to determine a fatigue safety factor.

$$n_{-1}=\frac{\sigma_{-1}}{\sigma_{eqv}} \qquad (6)$$

where σ_{-1}- endurance strength of gear material,

σ_{eqv} - equivalent alternating stresses [4].

Calculated specter of rim vibrodisplacement is shown in Fig. 7

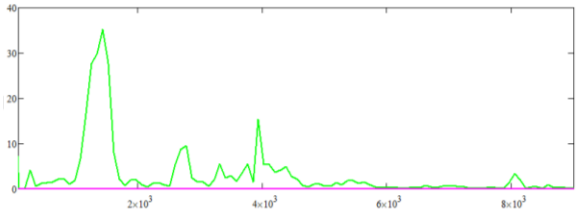

Рис. 7. Calculated gear rim vibrodisplacement specter

In concerned case disturbing force frequency is 2780 Hz. In rim vibrodisplacement specter spike near 3920 Hz is caused by “umbrella” natural form, near 8180 Hz – by three diameter natural form.

In case of crack appearing in gear rim stiffness matrix [K(u)] and mass matrix [M] will be changed. Consequently, natural frequency of gear will be changed also. Thereby, in time of permanent spectral analysis with high resolution shift of the natural frequency in oscillation specter can be used as a diagnostic sign of fatigue crack appearing and development in gear rim.

6 Using of coulomb friction damper to decrease variable stress in gear rim

Selection of an optimal damper configuration is a very complicated task. It can be solved after development of a mathematical model and performing of a set of dynamic calculations.

There are two main types of a bevel gear dampers: ring damper and plate damper.

Ring damper is a flexible ring inserted in a slot with pretension, which is situated in an inner surface of gear rim. Plate damper is a plate spring, pressed to a face of a gear. In this paper plate damper is considered. Engineering design of a plate damper is shown in Fig. 8.

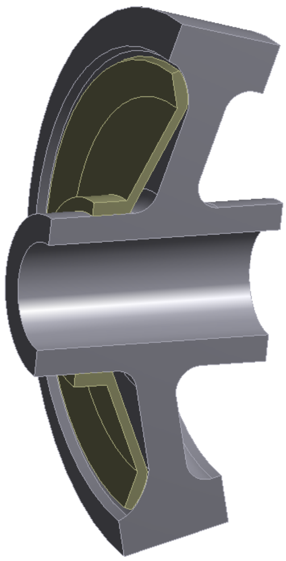

Fig. 8. Bevel gear with frictional damper (teeth are not shown)

To estimate efficiency of plate damper it is necessary to compare amplitude of resonance oscillation oa gear with damper and without it. It is impossible to use to use harmonic response analysis to research behavior of construction with coulomb damper because frictional contact is not linear.

A forced oscillation of gear without damper with variable force, applied to a gear tooth with frequency equal to a natural frequency of gear oscillations by three diameter (7821 Hz) and amplitude 200 N is modeled. As a result of calculation an amplitude equal to 98 microns is obtained.

To estimate an efficiency of a plate gear damper it is necessary to estimate a natural frequency of a gear – damper system. As a first iteration a middle meaning of three diameter natural frequency of gear without damper (7821 Hz) and of three diameter natural frequency of gear with damper with bonded contact between damper and gear (9019 Hz) and

As a result of resonance oscillation analysis of bevel gear with frictional damper it was estimated, that amplitude of force application point is 2 microns. Thereby, applying of frictional damper can decrease amplitude of resonance oscillations in 49 times.

7. Conclusion

1. Stress concentration areas in case of most dangerous modes of oscillations has been estimated.

2. A dynamic model with teeth contact by curvature surfaces, which can handle it's modifications has been developed.

3. A calculated specter of forced oscillations under transmitted torque has been obtained.

4. Shift of the natural frequency in oscillation specter can be used as a diagnostic sign of fatigue crack appearing and development in gear rim.

5. Applying of frictional damper can decrease amplitude of resonance oscillations.

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