Abstract. The article discusses the design of a magnetic damper to fix the problem of resonance occurrence, leading to vibration of the shaft at high speeds. When creating engines of large bypass ratio and large rotor speeds are crucial questions speakers. The most effective means of reducing the level of vibrations and dynamic loads at the nodes of the motors is the use of damper supports. For the final determination of the effectiveness of the type of damper selected and designed for this engine, it is necessary to perform experimental work. When fine-tuning the engine D-436T1 (special examination for the fan wheel) at 5900 rpm fan…6100 rpm had increased engine vibration at frequencies of the rotor due to resonance vibrations of the shaft of the fan. These oscillations were caused by the bending of the shaft of the fan and tapping it into stator and rotor parts of the compressor low pressure. A change in the design of the fan shaft to eliminate resonance was impossible. Domain service fan wheel and fan design coil spring / oil damper was not provided to prevent resonance. To solve the problem of design of a magnetic damper that allows you to change the technology of Assembly of the engine and provide its operation on all modes. The analysis of magnetic materials that can be used in the damper based on the design features and maximize the stiffness of the damper. Analyzed three systems: Sm-Co, Fe-Nd-B and Fe-Nd-C, from which we can conclude that magnetic characteristics of the most printlevel may be the system Fe-Nd-B. At the moment developed the design of the fan shaft with magnetic damper is at the stage of experimental work.

Introduction. In the aircraft industry the main goal of the developers of turbojet engines is the creation of highly economical and reliable, with low specific gravity propulsion. The life and reliability of modern machines with elements rotating in a large extent depends on the level of vibration, durability of bearings, the magnitude of the dynamic loads caused by rotor imbalance. One of the reasons that causes more vibrations in an aircraft engine, is the increased imbalance of the rotor (or rotors), which can lead to the excitation of one or more critical speeds, vibration of the rotor, and cause the resonance phenomena in the reference design [1].

When you create engines heavily decontrol with large rotor speeds are crucial questions speakers. The most effective means of reducing the level of vibrations and dynamic loads at the nodes of the motors is the use of dampfern supports. Supports of rotor are separate modules, which are included in the constructive force of an engine system. Design dampfer supports very diverse, but regardless of this they have the following main functions:

– to reduce the support stiffness of the rotor, which leads to a change of dynamic elastic system of the engine (this reduces the natural frequency of the system, eliminates resonances in the operating modes);
– the absorption of the vibration energy of the engine dynamic system, turning it into heat, which does not allow to develop large amplitude vibrations, dynamic loads and stresses in engine parts.

Nowadays in aviation engines use three basic types dampfern supports:
– hydrodynamic (oil) damper;
– hydrodynamic (oil) damper and elastic element type "squirrel wheel" (elastic oil damper);
– with elastic ring.

For the final determination of the effectiveness of the type of damper selected and designed for this engine, it is necessary to perform experimental zavodnye work. But since experimental evidence is of high value, it is advisable to use numerical simulation to evaluate variants of the system rotor-bearing:

1. Determination of critical speeds redemptive system.
2. The calculation of the system response, steady-on imbalance, i.e., forced oscillations.
4. Investigation of the stability of the system rotor-bearing on the basis of the calculation of the critical speed taking into account damping or the application of sustainability criteria.
5. Analysis of the transient unsteady processes or reactions of the rotor to imbalance and impact exciting force and also the study of transients caused by the phenomenon of self-excitation.

Abroad these issues were addressed by many authors in a different setting. In the CIS such researches were considered in the MAI and CIAM. In Ukraine the company "Ivchenko-Progress" developed a methodology and software package for the determination of natural frequencies and forms of oscillations of a system of rotors, and also for definition of amplitude-frequency characteristics, which are caused by the response of the system is established on imbalances, on structural friction and viscous friction. To determine the natural frequencies used in the method of initial parameters for systems which have many relations [2,3].

Design features magnetic mounts engine D-436T1. On GP "Ivchenko-Progress" in fine-tuning the engine D-436T1 (special examination for the fan wheel) at 5900 rpm fan…6100 rpm had increased engine vibration at frequencies of the rotor due to resonance vibrations of the shaft of the fan. These oscillations were caused by the bending of the shaft of the fan and tapping it into stator and rotor parts of the compressor low pressure. A change in the design of the fan shaft to eliminate resonance was impossible. Domainservice fan wheel and are available in fan design elastic oil damper was not provided to prevent resonance. To solve the above mentioned problems together with the SE "Ivchenko-Progress" according to the agreement on cooperation was tasked to design such a damper that would allow the available geometrical dimensions do not change the technology of Assembly of the engine and provided the performance of the engine during all modes of operation. A separate task was to ensure maximum rigidity of the damper due to the selection of magnetic material with optimal properties [4-7].

On the basis of the foregoing, of the design of the fan
shaft with a magnetic damper [8-10], which would allow to eliminate the resonance frequency region of the operating range of the engine and to prevent the increase of vibration level. The scheme also had the task of testing the possibility of applying this type of damper as extra support for long shaft engines. Developed the design of the support does not require the traditional oil supply for cooling of the bearing and is devoid of typical defects mevalonic supports (proconsulare, the complexity of the supply and discharge of oil, the complexity of the collection and ensure that special requirements for the bearing).

In Fig. 1 shows the fan shaft of aircraft engine D-436 T1 with magnetic support [5,11].

![Fig. 1. The fan shaft of aircraft engine D-436 T1 with magnetic bearing: 1 – bearing of the fan, 2 – fan shaft, 3 – the oil feed pipe, 4 – magnetic damper, 5 – bearing the low-pressure compressor, 6 – shaft front compressor low pressure.](image)

In Fig. 2 shows the structure of the rings of the rotor and the stator.

![Fig. 2. The Design of the rings of rotor and stator: a – the inner ring with magnets, b – outer ring with magnets.](image)

Developed magnetic damper consists of a portion that rotates the rotary and stationary part (stator). Rotary part of the magnetic damper consists of a set of radially magnetized sectors gathered in a ring of non-magnetic steel (EP 33-VD) and installed in the cage with magnetom eco steel (EP 609-W) of alternating magnetic poles, each pair of segmented rings. The fixed part of the damper is assembled from segments radially magnetized permanent magnets assembled in a ring of non-magnetic steel (EP 33-VD) and installed in the cage with magnetom eco steel (EP 609- W) of alternating magnetic poles, each pair of segmented rings. Thus, the poles of the magnets of the rotor and stator create a whole range repulsive force. In Fig. 3 shows a bushing with magnets.

Design must provide:

1. The mutual displacement in axial direction by the amount L of the sleeve with the magnets 1 and sleeve with magnets 2 (Fig. 4).
2. Measurement of axial force Grew and the value of L when mutual displacement of the sleeve with the magnets 1 and sleeve 2 with magnets (see Fig. 4).
3. The mutual displacement in the radial direction by the amount d of the sleeve with the magnets 1 and sleeve 2 with magnets.
4. Measuring radial forces of Rad and the value of d when mutual displacement of the sleeve with the magnets 1 and sleeve 2 with magnets (see Fig. 4).
5. Rigid fixation of one of the plugs with magnets in the measurement of axial and radial ROS Red forces.

6. The step width in the measurement of axial and radial forces – 0.1 mm.
7. Needs to be secured and the installation of the hub axle with the magnets 1 and sleeve 2 with magnets in adjacent axes (see Fig. 4).
8. Weight of the subjects and items of equipment should not affect the accuracy of measurement of forces.

![Fig. 3. Bushing with magnets: 1 – sleeve, 2 – ring, 3 – sleeve with magnets, 4 – adjusting ring.](image)

According to the scheme of the test magnetic damper (see Fig. 4), the value razanajato (axial) forces $P_{ax} \geq 1500$ N, the value of radial force $Rad \geq 500$ N.

**Features choice of permanent magnet material**

The requirements that apply to magnetic materials:

- high coercive force;
- low specific weight;
- low cost;
- stable characteristics over a wide temperature range and time;
- sufficient physical and chemical properties;
- high electrical resistance.

The use of permanent magnets with a high value of coercive force, which characterizes the resistance Romagnan arising from the characteristics of the suspension: of the magnet in the field of another magnet. The required values razanajato forces can be achieved provided that there is no significant mutual demagnetization of support elements. This condition is responsible cobalt alloys and composite materials based on alloy Nd-Fe-(b,C). When looking at the second type alloys are also the third condition is low cost compared to cobaltocene alloys. The specific weight of the obtained sintered magnets – 9375 kg/m$^3$.

High resistance of magnetic materials is necessary to minimize losses from eddy currents generated by the rotation of the shaft.

Also, the design of a magnetic damper was used, known at that time, magnetic material based on the alloy Sm-Co with known characteristics. However, the high cost of the Sm, high energy costs for obtaining cobalt and the need to preserve the basic characteristics of quite a long time does not give them sufficient widespread use. To reduce the cost of such magnets you can use magnets based on alloys of the system Fe-Nd-B and Fe-Nd-C. Table 1 presents comparative characteristics of the magnets of the systems Sm-Co, Fe-Nd-
B and Fe-Nd-C (TC – Curie temperature, IS the spontaneous on-magnet, HA – anisotropy field, (BH)max – the maximum magnetic energy). The working temperature of about 470 K.

Table 1. Comparative characteristics of the magnets of the systems Sm-Co, Fe-Nd-B and Fe-Nd-C.

<table>
<thead>
<tr>
<th>Connection</th>
<th>Tc, K</th>
<th>Is, A.m</th>
<th>Br, T</th>
<th>Hc, kA/m</th>
<th>(BH)max, kJ/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>SmCo</td>
<td>1020</td>
<td>1,14</td>
<td>0.77</td>
<td>1.3</td>
<td>140</td>
</tr>
<tr>
<td>Nd₂Fe₁₃C</td>
<td>535</td>
<td>1.5</td>
<td>0.8</td>
<td>7.6</td>
<td>450</td>
</tr>
<tr>
<td>Nd₂Fe₁₄B</td>
<td>585</td>
<td>1.6</td>
<td>1.2</td>
<td>5.4</td>
<td>512</td>
</tr>
</tbody>
</table>

Technical characteristics of the magnets:
1. Magnetic induction Br ≥ 0.9 T
2. The coercive force HC ≥ 600 kA/m.
3. Operating temperature T≥ 493 K.

The calculation of magnetic forces in the magnetic damper. Informed by the results of the research supports damper with a permanent magnet the Bekkers were made the following conclusions: by increasing the radius of the support its capacity increases linearly, and the weight of the shaft increases with the square. That is, under certain conditions, the bearing will compensate only the weight of the shaft. Therefore, the shaft must be run empty and the inner ring of the bearing – minimal [12-20].

To calculate the discharge power was used the following formula:

$$ F = \mu_0 \cdot J^2 \cdot l \cdot (r + x/2) \cdot e^{2x/\mu_0 - \sin \varphi} $$

Magnetic saturation can be defined as:

$$ J = \frac{B}{\mu_0} $$

In total, this design uses magnets 390. The size of the magnets that are used in design, 6x12x16 mm, height 6 mm, the working part of the magnet 12x16 mm, thus this plane can be divided into 12 elementary magnets of square section. Thus, for a single magnet, the value of the discharge power obtained by the formula 1, should be increased in 12 times.

Table 2. Magnetic properties [21-24].

<table>
<thead>
<tr>
<th>Magnetic system</th>
<th>The coercive force, Hc, kA/m</th>
<th>Residual induction, Br, T</th>
</tr>
</thead>
<tbody>
<tr>
<td>Magnetoplasma composition Nd₁₅²Fe₁₄₁₈Cu₁₃₅Co₁₀₃₅Ti₁₃₈</td>
<td>1320</td>
<td>0,79</td>
</tr>
<tr>
<td>Anisotropic magnet composition Nd₁₅²Fe₁₄₁₈Cu₁₃₅Co₁₀₃₅Ti₁₃₈ after TH</td>
<td>1260</td>
<td>1,125</td>
</tr>
<tr>
<td>Plate magnets KS-37</td>
<td>1300</td>
<td>0,77</td>
</tr>
<tr>
<td>Rosentiech force for the minimum values of Hc and Br</td>
<td>600</td>
<td>0,9</td>
</tr>
</tbody>
</table>

Using the data in table 2, determined by the discharge power according to the formula (1). To compute the dependence of the force from the magnitude of the gap in the magnetic damper, it was believed that the sine of the maximum angle of rotation [25].

It is known that the smaller the gap in the damper, the greater the discharge power can be achieved [5,26-32]. However, the features of production do not allow to reduce the gap to a minimum. In the proposed design, the clearance is 1 mm. figure 7 shows the dependence razanajato forces from the gap in the magnetic damper.

In Fig. 5-7 shows the dependence razanajato power from step razanajato (axial) forces from the gap and radial forces from clearance for sintered under the initial pressure of 12 MPa and a temperature of 1023 K magnets of the composition Nd-Fe-B doped with Cu, C, Ti in the magnetic damper.

Radial support stiffness is calculated according to the following formula:

$$ F = \frac{C}{x} $$

The specific lift force per unit surface is calculated by the following formula:

$$ f = \frac{F}{x} $$

Fig. 5. Dependence razanajato (axis) from the force of the step sintered under the initial pressure of 12MPa and a temperature of 1023 K magnets of the composition Nd₁₅²Fe₁₄₁₈Cu₁₃₅Co₁₀₃₅Ti₁₃₈.

Fig. 6. Dependence razanajato (axial) forces from the gap in the magnetic damper:
1 – Baked under an initial pressure of 12 MPa and a temperature of 1023 K magnets of the composition Nd₁₅²Fe₁₄₁₈Cu₁₃₅Co₁₀₃₅Ti₁₃₈; 2 – Plate magnets KS-37; 3 – Anisotropic magnets of the composition Nd₁₅²Fe₁₄₁₈Cu₁₃₅Co₁₀₃₅Ti₁₃₈ after HT; 4 – Rosentiech power for the minimum values of Hc and Br.
In Fig. 8 shows the dependence of the radial stiffness of the support from the gap in the magnetic damper:

C1 – Baked under an initial pressure of 12 MPa and a temperature of 1023 K magnets of the composition Nd$_{15}$Fe$_{24}$Co$_{15}$B$_{5}$C$_{0.5}$Ti$_{1.18}$; C2 – Anisotropic magnets of the composition Nd$_{15}$Fe$_{24}$Co$_{15}$B$_{5}$C$_{0.5}$Ti$_{1.18}$ after HT; C3 – Rosentieh force for the minimum values of HC and Br; C4 – Plate magnets KS-37.

Thus, the proposed design fan shaft with magnetic damper and the analysis of magnetic materials that can be used in the damper based on structural features. It is shown that the maximum stiffness of the damper and taking into account the cost of finished products the most successful are the magnets made from the alloy Nd$_{15}$Fe$_{24}$Co$_{15}$B$_{5}$C$_{0.5}$Ti$_{1.18}$. It is also shown that such a design of the support does not require the traditional oil supply for cooling of the bearing and is devoid of typical defects between the bulk of the supports.

**Conclusions**

1. The proposed design fan shaft with magnetic damper that had the task of testing the possibility of applying this type of damper extra support for long shaft engines. Such a design of the support does not require the traditional oil supply for cooling of the bearing and is devoid of typical defects mevalonic supports.

2. The scheme tests the magnetic damper, the calculated discharge power for different types of magnetic systems.

3. The analysis of magnetic materials that can be used in the damper based on structural features. Analyzed three systems: Sm-Co, Fe-Nd-B and Fe-Nd-C, from which we can conclude that magnetic characteristics of the most printlevel may be the system Fe-Nd-B.

4. Developed the design of the fan shaft with magnetic damper is at the stage of experimental work to determine the effect of magnetic field on the details that surround the damper in the first place, on the parts of the bearing and the efficiency of damping.

**REFERENCES**


ja technika i pneumatika – Compressors and pneumatical, 3(4)-12-13, 107-109. [in Rus]


reshenie. [in Rus]

Магнитный демфер для авиадвигателя Д-436Г

Т. В. Гуляева, О. В. Таратухин, Л. В. Гуляев

Аннотация. В статье рассматривается конструкция магнитного демпфера, который предназначен для устранения проблемы возникновения резонанса, приводящего к вибрации вала на большинстве образцов. При создании двигателей большей степени двухконтурности с высокими частотами вращения роторов чрезвычайно большое значение имеют вопросы динамики. Наиболее эффективным средством снижения уровня вибраций и динамических нагрузок в узлах двигателей является применение демпферных опор. Для окончательного определения эффективности типа демпфера, который может быть выбран и рассчитан для данного двигателя, необходимо провести специальные исследования. С этой целью предлагается демпфер, основанный на магнитном поле.

Ключевые слова: магнитный демфер, осевая сила, радиальная сила, жесткость демпфера, остаточная индукция, демпфирование.

Magnetic Damper for Aircraft Engines

T. V. Gulyaeva, O. V. Taratukhin, L. V. Gulyaev

Abstract. The article discusses the construction of a magnetic damper designed for removing the problem of resonance, which causes vibration of the shaft on most of the samples. When building engines of higher degree of two-cascade with high frequencies of rotation of the rotors, it is very important to study the issues of dynamics. The most effective means of reducing the level of vibrations and dynamic loads in the engine units is the use of damper supports. For the final determination of the efficiency of the type of damper, which can be selected and calculated for the given engine, it is necessary to perform special investigations. For this purpose, a damper is proposed, which is based on the magnetic field.

Keywords: magnetic damper, axial force, radial force, stiffness, residual induction, damping.