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EXPERIMENTAL STUDY OF NONLINEAR VIBRATION CHARACTERISTICS OF A TWO-ROTOR TURBOFAN ENGINE SYSTEM WITH AN INTERSHAFT SUPPORT

The layout of the turbofan engine with coaxial rotors and intershaft rear support of the high-pressure rotor is the most compact one since the rear supports of both rotors are located in one oil cavity, provided with a joint system of communications for oil supply and suction, and the force diagram of the aircraft engine stator is simplified. The disadvantage of this layout is the mutual influence of high-pressure (HP) and low-pressure (LP) rotors through the intershaft support, which leads to significant vibration activity due to the operating conditions of the intershaft bearing. The purpose of this work was to identify the characteristic features of vibration instability during the testing of turbofan engines with intershaft support, the factors influencing it, and the methods of elimination within the framework of the general reduction of vibration levels and the improvement of the efficiency of engine vibration diagnostics. The research was performed on serial engines manufactured by MOTOR SICH JSC. Based on the vibration recording data, approximate curves of the vibration response in the near-resonance area are plotted. The directions of the rotation frequency change and hysteresis zones with stable and unstable oscillations are marked. Bench tests with registration of broadband vibration sensor signals were performed. The moment spectra of the vibration signals for two stable oscillation modes with the same rotor speed in the resonance region were calculated. The phenomenon of vibration instability during the operation of a turbofan engine with intershaft support in the steady-state mode of operation is analyzed. The main features of the vibration state of such an engine are identified as follows: 1) the presence of the vibration jump phenomenon when passing the resonance frequency of HP rotor; 2) the presence of combination frequencies of the rotors and characteristic frequencies of the intershaft bearing in the vibration signal spectrum; 3) the presence of a constant phase shift of the signal from the front and rear suspension plane sensors with the frequency of HP rotor in modes with vibration instability; 4) the occurrence of a chaotic phase shift in the vibration signal of the rear suspension plane when vibrating with the frequency of LP rotor. An assumption was made about the presence of rolling element skidding in the intershaft bearing as the main cause of the described vibration instability. The influence of the inlet oil pressure on the instability occurrence was detected.

Keywords: *turbofan engine; coaxial rotors; intershaft support; bench test; vibration; instability.*

Introduction

When designing modern aviation gas turbine engines, the key criteria that are taken into account, in addition to compliance with specific requirements and tasks of a particular aircraft, are mass-dimensional characteristics and efficiency parameters. Both criteria stipulate the necessity of increasing the parameters of the gas dynamic cycle, application of circuit solutions with minimum material intensity, which in turn contributes to an additional increase in the engine dynamic load, reducing reliability and lifetime parameters. One of the widespread layouts discussed in this paper is the dual-rotor turbofan engine with coaxial rotors and the rear support of HP rotor, which is an intershaft support. Such a layout with a general trend to reduce the number of rotor supports is the most compact, since the rear supports of both rotors are located in one oil cavity, are provided with a joint system of communications for oil supply and suction, and the stator force diagram is simplified [1].

The disadvantage of this layout is the mutual influence of HP and LP rotors through the intershaft support, which leads to significant vibration activity due to the intershaft bearing (ISB) operating conditions [2] and the generation of additional oscillation forms of the rotor system taking into account the variable speed ratio of rotors at different operating modes caused by gas dynamics [3, 4]. Such phenomena, which are not related to defects identified by vibrodiagnostic methods, complicate the evaluation of the technical state, and require more detailed design development and maturation of the twin-rotor system.

The purpose of this work is to identify the characteristic features of vibration instability during testing of turbofan engines with intershaft support, the factors influencing it and ways of elimination within the framework of general vibration level reduction and improvement of engine vibration diagnostics efficiency. The study was performed on serial engines manufactured by MOTOR SICH JSC (engine layout - see Fig. 1), having the mentioned features in the design and vibration state.

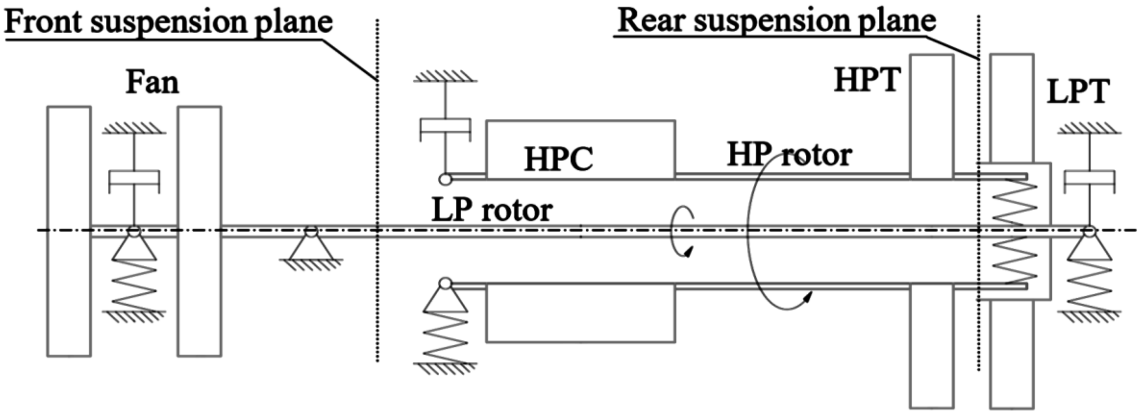

Fig. 1. Layout of turbofan engine with intershaft support

One of such peculiarities of operation, met in the practice of gas turbine vibration metering, and usually defined as signs of defects by vibrodiagnostic statements, can be vibration level instability during engine operation at steady-state mode, occurring as a periodic inverse change of vibration velocity of pulse "sawtooth" character. The specified phenomenon is observed in the range of the vibration spectrum containing the main "first" rotor harmonic.

The description of such phenomena is found in the actual literature [5, 6], dedicated to the solution of applied problems related to the design improvement and frequency tuning of specific engine models. A number of works have considered and statistically substantiated the relationship of vibration instability with changes in the direction of the low-pressure rotor axial force, clearances in labyrinth seals, gas-dynamic parameters of the engine, eigenfrequencies of the exhaust device, bearing clearances and, as a consequence, their stiffness.

It should also be noted that the close in occurrence phenomenon of sudden oscillations amplitude change with frequency of the first rotor harmonic varying the rotor rotation speed in the range of resonance frequencies («jump phenomenon») is well enough studied in rotor dynamics [7]. This phenomenon is related to the nonlinear stiffness of the rotor-stator dynamic system elements at unbalanced rotor vibrations, it is mentioned in some literature as bistable vibrations. However, unlike the considered change in vibration velocity at steady-state mode of operation, the vibration jump occurs when the excitation force frequency changes with the transition from one stable branch of oscillations of a nonlinear system to another. In this case, the amplitude of vibrations depends on the previous system state, i.e., at increasing and decreasing rotational speeds, the vibration jump is observed at different rotational speeds.

The objectives of the study are to analyze bench test statistics, determine if the vibration instability can be affected during testing, and analyze the results of broadband vibration recording.

1. Vibration analysis during acceptance bench tests

According to the results of statistical analysis of serial engines bench tests in addition to the phenomenon of unstable vibrations in steady-state mode under investigation, the following general features were determined:

- HP rotor resonance peak in the rotation speed about n_{HP} = 290 Hz;

- hysteresis of vibration peak when passing resonance frequencies with increasing and decreasing operation mode.

Within the framework of statistical analysis, the mentioned hysteresis during the "passage" operation (smooth increase and decrease of rotors rotation speed in the range from idle to maximum mode) is studied. A sample of 50 serial engines, which had vibration level at the rear suspension with HP rotor frequency V_{RS} from 60% to 140% of the limit when passing the acceptance test, was studied.

Based on the vibration recording data, approximate curves of vibration response in the near-resonance region are plotted, directions of rotation frequency change and hysteresis zones with stable and unstable oscillations are marked (Fig. 2).

The results of preliminary analysis of statistical data demonstrate the correspondence of the rotors vibration response character to the rigid amplitude-frequency characteristic of the nonlinear system at slow resonance passage [8]. The frequency response of such a system shows two stable oscillation forms with different amplitudes near resonance frequency. When the frequency increases, the jump from the form with larger amplitude to the form with smaller amplitude occurs at a higher frequency than the reverse transition

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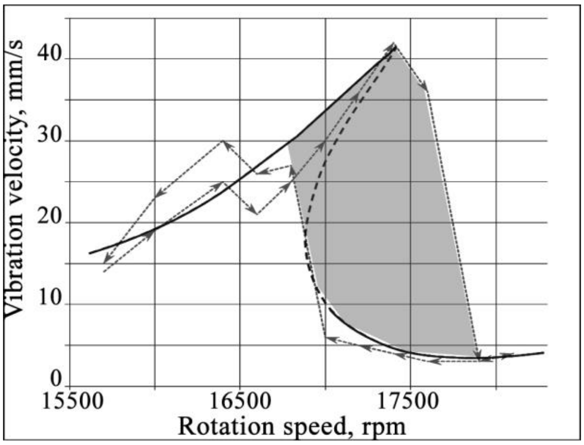
from smaller amplitude to a larger one when the frequency decreases. This is explained by the nonlinear properties of supports, in particular, for the hydrodynamic damper with an elastic element, applied in the design of the considered engine in the low-pressure turbine (LPT) support.



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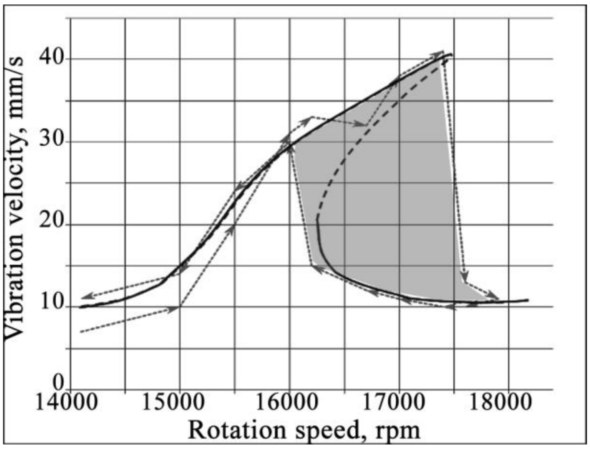

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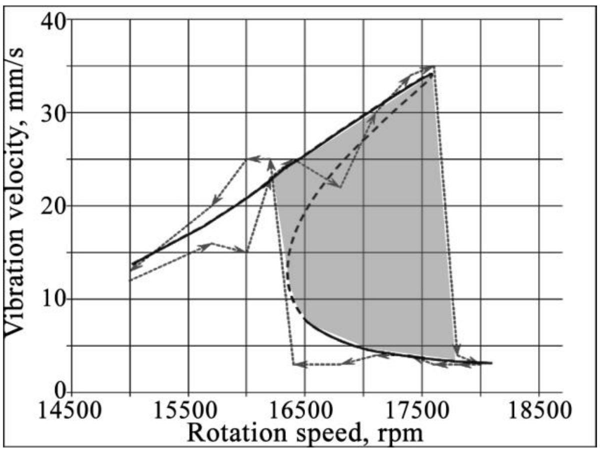

c

Fig. 2. Vibration response curves
of the near-resonance region:
a) – engine No. 1; b) – engine No. 2;
c) – engine No. 3

2. Experimental studies of nonlinear vibrations

In order to detailed study of the non-stationary vibration processes at engine operation, bench tests with registration of vibration sensors broadband signal were carried out.

The engine is equipped with four piezoelectric accelerometers type CA-135 (“MEGGITT Vibro-Meter”) in the front and rear suspension plane measuring vibrations in vertical, horizontal and axial directions with the following symbols: VFS (vertical, front suspension), VRS (vertical, rear suspension), HFS (horizontal, front suspension), HAFS (horizontal-axial, front suspension). For the needs of this work mainly used data from the VFS and VRS sensors, which have the most typical behavior of vibration velocity at steady-state mode. Signal processing is performed using software package WinПОС and MATLAB.

For a present engine, the jumps in vibration velocity are observed with different amplitude A at two steady-state operating modes (Fig. 3):

- Cruise (Cr), n_{HP} = 275 Hz, A = 20 mm/s;
- 0,6 Max, n_{HP} = 282 Hz, A = 27 mm/s.

Using FFT, the moment spectra of the vibration signal for two states in terms of vibration amplitude were calculated: $V_{VRS\ HP}$ = 30 mm/s (hereafter “state 1”) and $V_{VRS\ HP}$ = 4 mm/s (hereafter “state 2”) at constant rotor rotation speed at steady-state mode. Spectrum components and vibration sources were determined for all significant peaks (Fig. 4, Fig. 5). Comparison of spectra shows in general identical composition by frequency components, the differences are due to the various intensity of harmonics associated with HP rotor and combination frequencies.

Observed characteristic features of the vibration frequency spectrum:

- presence of a peak at the frequency proportional to rotation speed n_{HP} with a factor of 1,91, the amplitude of which varies insignificantly at operation of this engine at different modes;
- growth of LP rotor harmonics in “state 2”, while their level generally remains quite low ($V_{ЗПНД} \leq 7$ mm/s at tolerance level of 40 mm/s);
- prevailing of $2n_{LP}$ harmonic compared to n_{LP} in “state 1”, which may indicate the operation of the rotor with axis misalignment [7];
- significant growth of n_{HP} harmonic in “state 1”, accompanied by occurrence of rotors combination frequencies set, present in “state 2” at the level of noise amplitude;
- the presence of distinctive side bands around the n_{HP} harmonic in “state 1”, which are offset by the characteristic frequency of the intershaft bearing ($n_{HP} \pm f_{ISB}$). Since both rings of the intershaft bearing

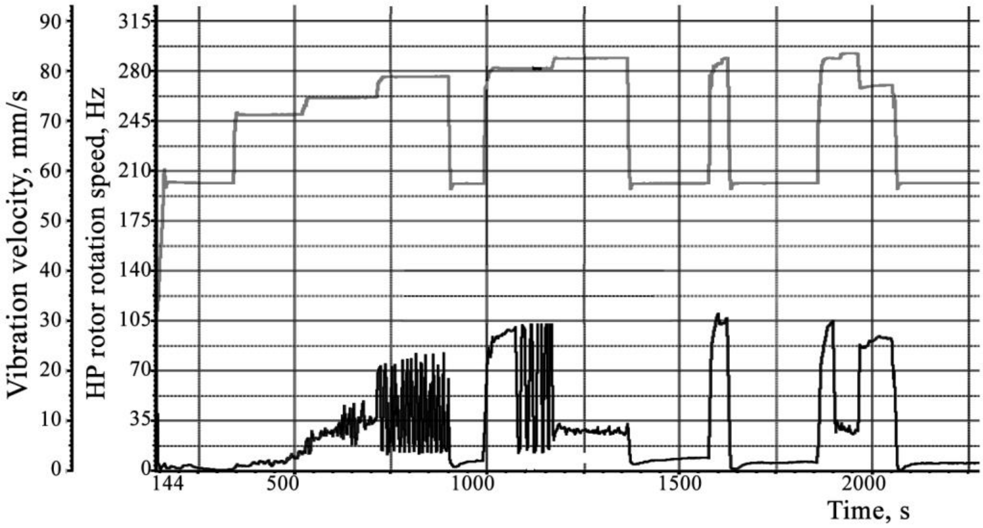

Fig. 3. Vibration recordings of the VRS sensor with frequency n_{HP}

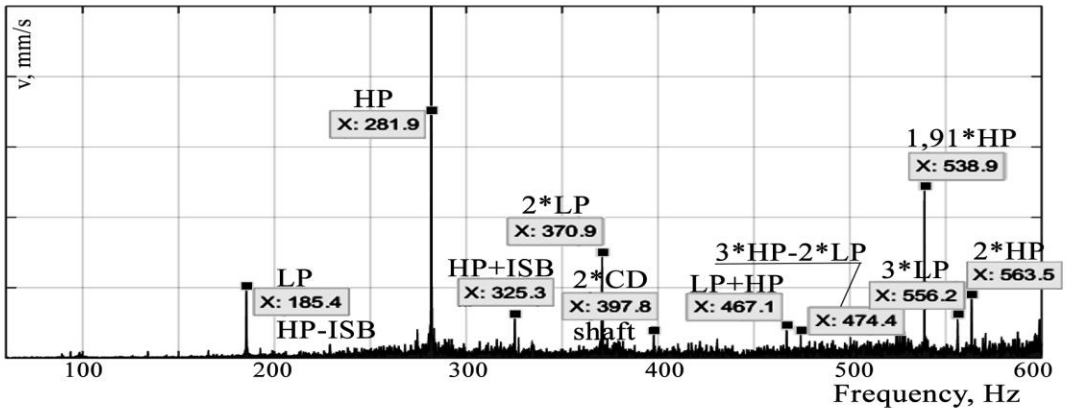

Fig. 4. Frequency spectrum of the vibration signal at the peak point of the VRS vibration ("state 1")

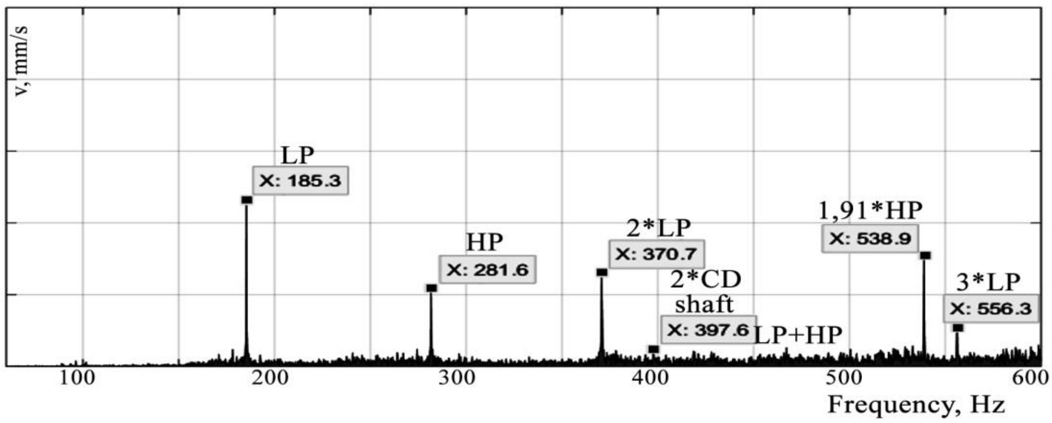

Fig. 5. Frequency spectrum of the vibration signal at the roll-off point of the VRS vibration ("state 2")

rotate, its characteristic frequencies appear on the measurements of the VRS sensor in relation to the rotors' rotation speeds and amplitude-modulate them.

The phase trajectories for "state 1" and "state 2" are shown in Fig. 6.

Taking into account the prevailing frequency of n_{HP} harmonic in the spectrum of "state 1", the contribution of other components on the phase plane is not so noticeable. At the same time, "state 2" is characterized by comparable amplitudes of frequencies n_{LP}, n_{HP}, $2n_{LP}$, $1{,}91n_{HP}$, due to which the trajectory becomes more complex, loops and chaotic behavior appear.

Comparison of vibration signal phases from the VFS and VRS sensors at steady-state mode with vibration instability was performed to analyze the change of rotor precession shape at vibration amplitude drop. For this purpose, the signals are filtered by rotor

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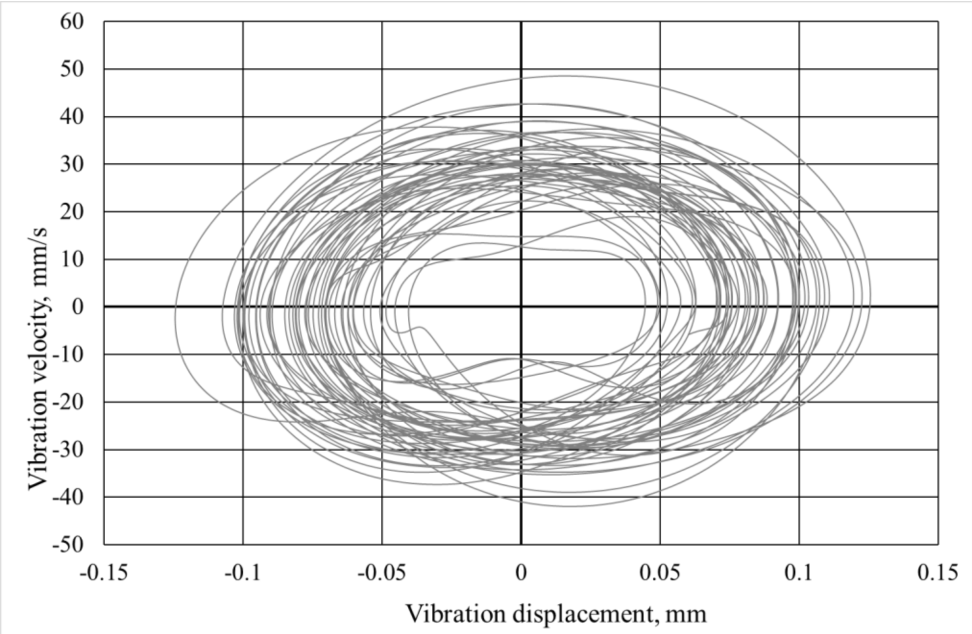

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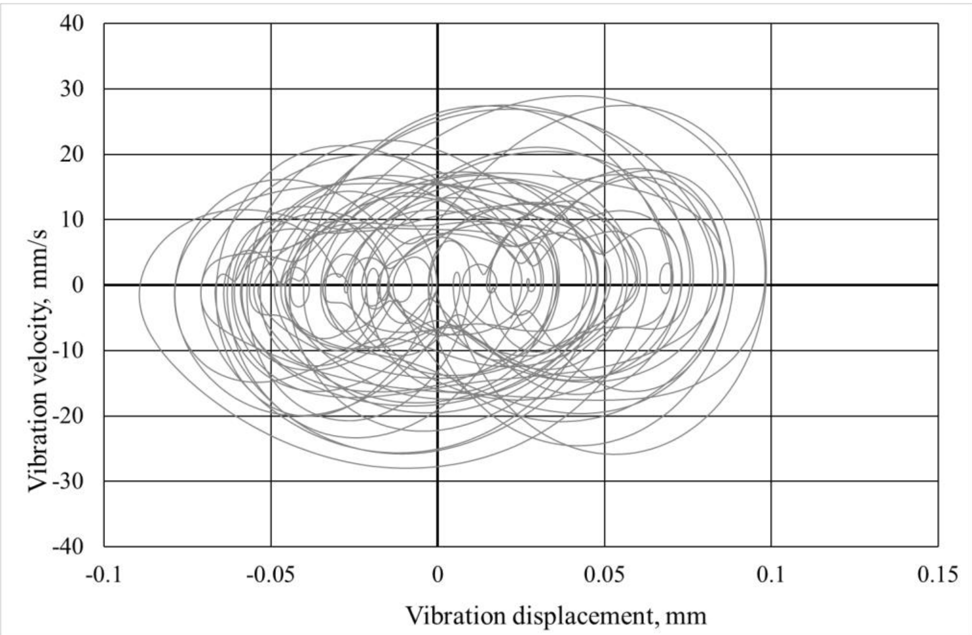

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Fig. 6. Phase trajectory: a) “state 1”; b) “state 2”

rotation speeds. According to the results of the analysis for the time mapping of the signal, there is a constant phase shift of vibrations with n_{HP} frequency between VFS and VRS sensors in “state 1” and “state 2”, which is $\varphi_{HP} \approx 2{,}3$ rad (Fig. 7). This is due to the different location of the suspension planes relative to the rotors and the different paths by which the vibrations pass through the casing parts to the vibration sensors, taking into consideration their placement. Since the constant phase shift of the same magnitude is also inherent in all lower engine operation modes, it is assumed that this value of the shift corresponds to a forward synchronous precession, excited by the unbalance of HP rotor [9].

At the same time, the indicated phase shift φ_{HP} has some ripple, which is clearly seen in the Poincare map on the phase plane plotted for the oscillations recorded by VRS sensor (the VFS sensor data serves as a “counter” for HP frequency oscillations period – see Fig. 8). The Poincare map shows a dispersion of the point cloud for both states within the second quadrant, but given the significant amplitude modulation observed in “state 2” and the lower amplitudes at that, some of the values fall into consecutive areas of the other quadrants.

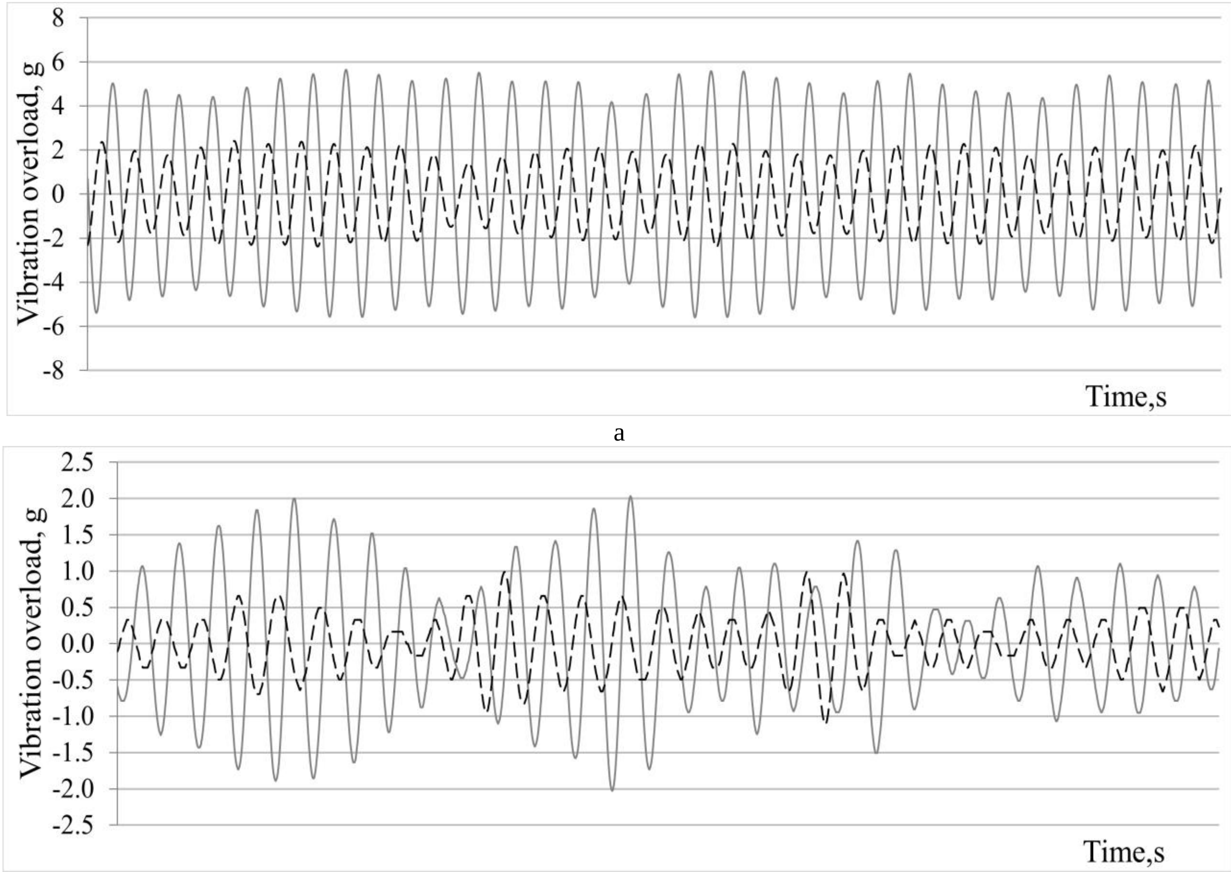

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Fig. 7. Oscillogram of VFS (dashed line) and VRS (solid line) signals with the frequency of HP rotor:
a) “state 1”; b) “state 2”

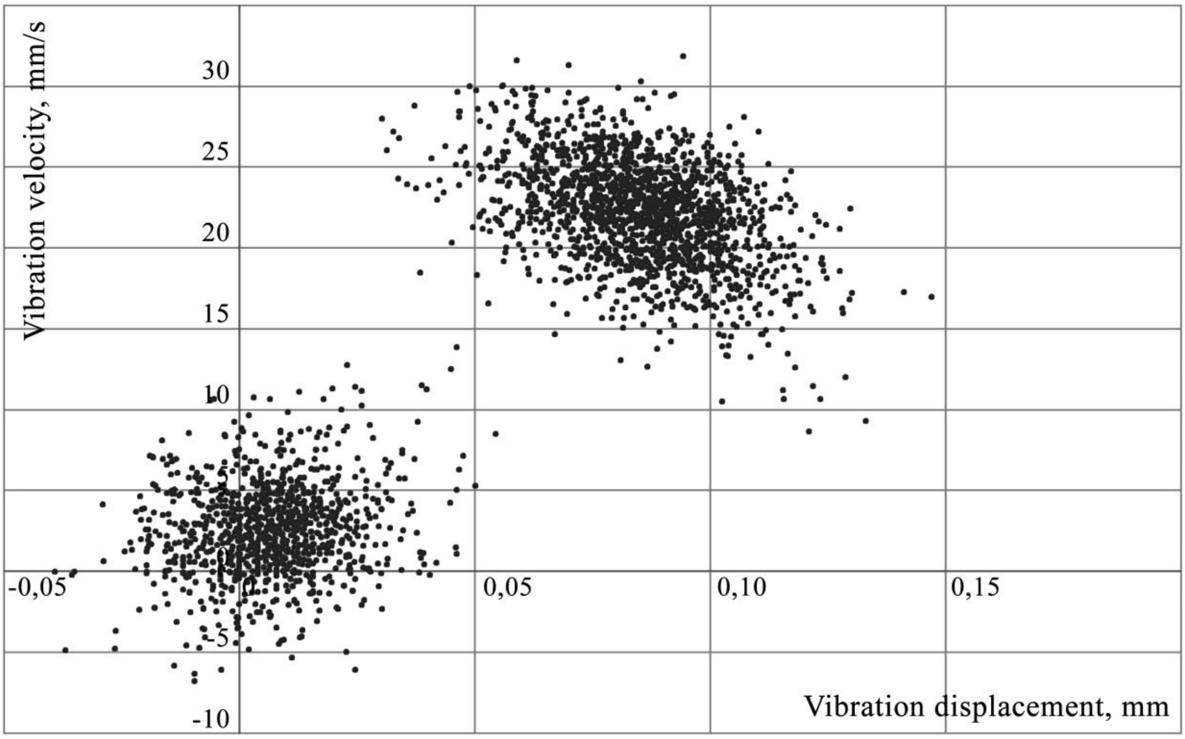

Fig. 8. Poincare map of HP rotor vibrations. The "State 1" points are grouped
at the top center and the "State 2" points are grouped at the bottom left

ISSN 1814-4225 (print)
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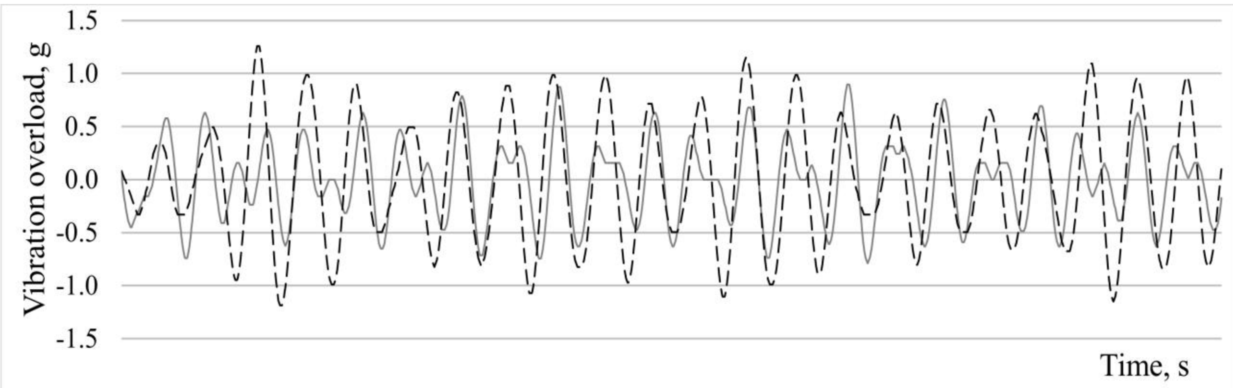
Thus, HP rotor oscillations amplitude drop does not lead to a change in the precession form of this rotor, which coincides with theoretical studies of the hard bifurcation at the "classical" jump [10].

Same analysis of the signal phase relation from VFS and VRS sensors with LP frequency shows the presence of a constant shift $\varphi_{LP} \approx 0.5$ rad in "state 2" and chaotic changes in vibrations on the rear suspension (VRS), causing a variable phase shift in "state 1" (Fig. 9). Such a vibration response is specific to damped rotor system models when occurring fast-flowing random transients, commensurate with the rotor rotation period, e.g., rotating with nonconstant rubbing or scuffing [11, 12].

Taking into account the presence of intershaft bearing characteristic frequencies in the spectrum, as well as the typicality of the rolling element skidding for intershaft bearings in general due to the small radial load, this phenomenon is the most probable cause of vibration instability. At skidding there is a loss of contact of rollers with raceways, re-entry of the roller into the load zone with significant acceleration and large heat generation in the process of friction [13], at the same time the load zone of the bearing due to the precession of rotors rotates, which creates the effect of random perturbations. Also, heating of rolling surfaces during skidding affects the reduction of the bearing radial clearance [14] and, under certain conditions, can cause a transition between stable oscillation branches of a nonlinear system.

To check the effect on vibration instability, additional runs of the engine under study with the adjustment of oil inlet pressure within 0.5 kg/cm^2 were performed. It was defined that with increasing oil inlet pressure, vibration instability at steady-state mode is not observed, the vibration level corresponds to the "state 1" described above. This can be explained by both the change in stiffness and damping of the turbine elastic-damping support [13] and by the change in heat transfer to the oil, preventing the realization of the described transition of states.

It is also observed that vibration instability is not affected by a further reduction of oil pressure compared to the pressure value at which instability occurs. Unstable vibration modes persist when engine thrust is adjusted by fuel consumption, when both the frequency of both rotors and the ratio of the rotation frequencies of the HP and LP rotors change.



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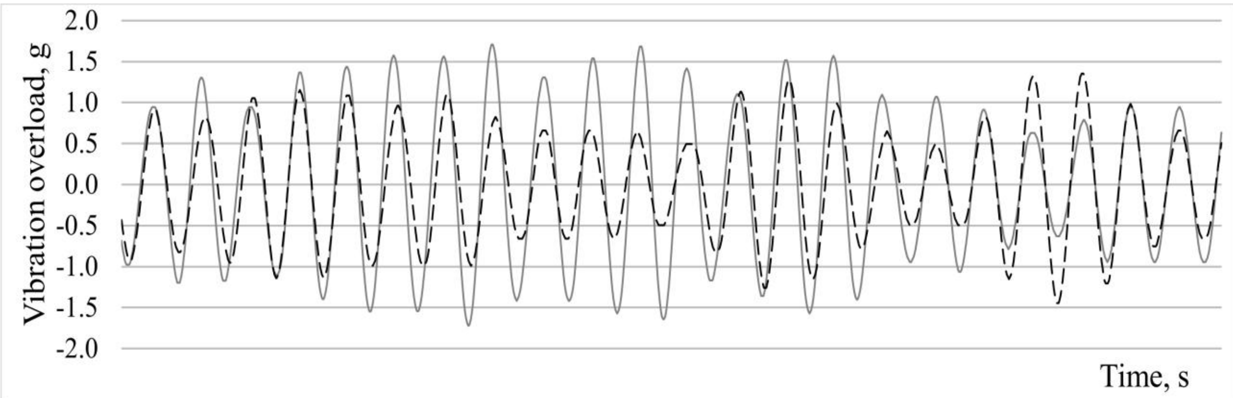

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Fig. 9. Oscillogram of VFS (dashed line) and VRS (solid line) signals with the frequency of LP rotor: a) – "state 1"; b) – "state 2"

Conclusions

In this work, the phenomenon of vibration instability during operation of a dual-rotor turbofan engine with an intershaft support in the steady-state mode has been studied. Statistical analysis of acceptance tests is performed and a vibration recording of such an engine is carried out. As a result, the main features of its vibration state have been determined:

1) the vibration jump phenomenon presence when passing the resonance frequency of HP rotor, caused by the nonlinear stiffness of the elements of the rotor-stator dynamic system;

2) presence of combinational frequencies of rotors and characteristic frequencies of intershaft bearing in the vibration signal spectrum;

3) the constant phase shift of the front and rear suspension vibration signals with the frequency of HP rotor in modes with vibration instability;

4) occurrence of chaotic phase shift in the rear suspension vibration signal with the frequency of LP rotor.

The influence of oil pressure adjustment, fuel consumption adjustment to ensure the specified thrust parameters on the occurrence of vibration instability has been studied. It is defined that increase of oil pressure at the engine inlet can lead to disappearance of vibration instability phenomenon. In this case, the level of vibrations when passing rotor speeds, at which instability was observed earlier, corresponds to the state with a higher amplitude.

The assumption is made about the presence of rolling element skidding in the intershaft bearing as the main cause of a transition between stable oscillation branches of a nonlinear system which is the vibration instability under consideration.

In the course of further research, to confirm the assumptions made in this work, it is necessary to perform modeling of the two-rotor dynamic system with intershaft bearing, taking into account the nonlinearity and the operation of the intershaft bearing with variable radial clearance caused by heating at rolling element skid.

Contributions of authors: methodology – **Sergey Filipkovsky, Sergey Chaikin**; formulation of tasks – **Sergey Filipkovsky**; analysis – **Sergey Filipkovsky, Sergey Chaikin**; development of model – **Sergey Filipkovsky, Sergey Chaikin**; experimental studies – **Sergey Chaikin**; verification – **Sergey Filipkovsky**, **Sergey Chaikin**; analysis of results – **Sergey Filipkovsky, Sergey Chaikin**; visualization – **Sergey Filipkovsky, Sergey Chaikin**; writing – review and editing – **Sergey Filipkovsky**.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

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Data availability

The Manuscript has no associated data.

Use of Artificial Intelligence

The authors confirm that they did not use artificial intelligence methods while creating the presented work.

All the authors have read and agreed to the published version of this manuscript.

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ЕКСПЕРИМЕНТАЛЬНЕ ДОСЛІДЖЕННЯ ХАРАКТЕРИСТИК НЕЛІНІЙНИХ КОЛИВАНЬ ДВОРОТОРНОЇ СИСТЕМИ ТРДД З МІЖВАЛЬНОЮ ОПОРОЮ

С. В. Філіпковський, С. В. Чайкін

Схема ТРДД із співвісними роторами та міжвальною задньою опорою ротора високого тиску (ВТ) є найбільш компактною, оскільки задні опори обох роторів розміщуються в одній масляній порожнині, забезпечуються загальною системою комунікацій для підведення та відкачування оливи, спрощується силова схема статора авіаційного двигуна. Недоліком даної схеми є взаємний вплив роторів ВТ та низького тиску (НТ) через міжвальну опору, що призводить до значної віброактивності через умови роботи міжвального підшипника. Метою роботи є встановлення характерних особливостей нестійкості вібрацій при випробуваннях ТРДД з міжвальною опорою, факторів, що впливають на неї, та способів усунення в рамках загального зниження рівня вібрацій та підвищення ефективності вібраційної діагностики двигуна. Дослідження виконано на серійних двигунах, виготовлених АТ «МОТОР СІЧ». Ґрунтуючись на даних вібрографування, побудовано приблизні криві вібраційного відгуку в навколорезонансній ділянці, позначені напрями зміни частоти обертання та зони гістерезису зі стійкими та нестійкими коливаннями. Виконано стендові випробування з реєстрацією широкосмугового сигналу вібродатчиків. Розраховані миттєві спектри вібросигналу для двох стійких режимів коливань з однаковою частотою обертання ротора в області резонансу. Розглянуто явище нестійкості вібрації під час роботи ТРДД з міжвальної опорою на стаціонарному режимі роботи. Встановлено основні особливості вібростану такого двигуна: наявність явища стрибка вібрацій при проході резонансної частоти ротора ВТ; 2) наявність у спектрі вібросигналу комбінаційних частот роторів та характерних частот міжвального підшипника; 3) наявність постійного зсуву фаз сигналу з датчиків передньої та задньої площини підвіски з частотою ротора ВТ на режимах з нестійкістю вібрацій; 4) прояв хаотичного зсуву фаз у вібросигналі задньої площини підвіски при вібрації з частотою ротора НТ. Зроблено припущення про наявність прослизання тіл кочення в міжвальному підшипнику, як основної причини нестійкості вібрацій, що описується, виявлено вплив тиску оливи на вході в двигун на прояв нестійкості.

Ключові слова: турбовентиляторний двигун; співвісні ротори; міжвальна опора; стендові випробування; вібрація; нестійкість.

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