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# EXPERIMENTAL-AND-COMPUTATIONAL PREDICTION OF AEROELASTIC STABILITY IN BLADE ASSEMBLIES AGAINST SUBSONIC FLUTTER

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**Abstract:** *The basic principles of the experimental-and-computational complex for predicting the subsonic flutter stability of aircraft gas turbine compressor blading are stated. Based on the approaches of physical modeling of phenomena observed under full-scale conditions, the similarity criteria are defined, which allow modeling the behavior of blade assemblies at their interaction with flow. The methodology is described for the experimental determination of non-stationary aerodynamic forces and moments acting on blades during their in-flow vibrations; the calculation of the dynamic stability of a blade assembly against flutter; the aerodynamic rig design and peculiar features of its components to perform testing of airfoil cascades. The results of testing of the developed experimental-and-computational complex are presented.*

**Keywords:** ***blade assembly, modeling, airfoil cascade, subsonic flutter, aeroelastic stability prediction***

## 1. Introduction

The most dangerous vibrations of axial compressor blades are those that occur most frequently during operation of the modern aircraft gas-turbine engines (AGTE) and can cause fatigue failures, namely: resonant vibrations, which are caused by circumferential nonuniformity of the flow due to gas dynamic wakes from the stator blades, struts and etc., nonresonant vibrations induced by rotating stall, nonresonant vibrations induced by turbulent flow fluctuations and flutter (self-excited vibrations in fluid flow).

Each of the mentioned type of vibrations has its own physical vibration generation mechanisms and methods of their control. Special attention is given to flutter wherein an abrupt increase in the vibration amplitude can lead to a loss of the lifetime or engine failure over a short time period. The most susceptible to flutter are compressor blade assemblies, especially under conditions close to separation of the flow from the blades. That is why expenses related to ensuring their vibraiton reliability take a significant amount of general time and money, which are spent on development and manufacturing application of AGTE.

It is known that about 90% of causes of possible failure are detected and eliminated during engine development and the rest are detected and eliminated during operation, 20% of which cause accidents.

There are many methods for the prediction of flutter of blade assemblies, which are based on finding unsteady aerodynamic loads (forces and moments) acting on blades during their vibration in a flow. Different methods for determining such loads are used in practice. Thus, modern numerical methods make it possible to solve the problem of blade vibration in a flow by using their different models, including three-dimensional ones.

However, to consider the viscousity of the fluid that flows around the assembly, it is nessecary to solve the Navier–Stokes unsteady equations, whence practical use of direct solution requires considerable computational resources as far as it involves very fine computational grids. The use of different turbulence models significantly reduces the requirements for computer facilities, however, correct selection of these models relative to the specific problem requires a comparison with the experimental data.

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Full-scale engine testing for blade stability against flutter is very expensive. Laboratory testing of aerodynamic loads and determination of dynamic stability of blade assemblies of rotor wheels make it possible to reduce their scope of testing. This exactly defines the purpose of the paper, which lies in scientific and technical justification and development of the experimental and computational complex for predicting the stability of AGTE blade assemblies against subsonic flutter, which involves the experimental determination of aerodynamic loads acting on blades during their vibration in a flow and computation determination of the flutter stability limit by calculating the natural frequencies of coupled vibrations of blades in a flow using the obtained experimental data.

## 2. Simulation of Unsteady Force Interaction of Blades Vibrating in a Flow

From an analytical point of view, vibrations of the blade assembly in a flow can be represented in the matrix form as:

$$[M_b]\{\ddot{q}\}+[C_b]\{\dot{q}\}+[K_b]\{q\}=\{Q(t)\}, \tag{1}$$

where $[M_b]$, $[C_b]$, $[K_b]$ are the matrices of inertial, dissipative, and elastic characteristics of the assembly, respectively; $\{Q(t)\}$ , $\{q\}$ are the column vectors of aerodynamic forces acting on the blades and of their displacements.

Aerodynamic loads $\{Q(t)\}$ can be presented as a sum of loads $\{Q_D(t)\}$, which are caused by nonuniformity of the flow and do not depend on the blade motion, and $\{Q_A(t)\}$ that are induced by blade vibrations. For flutter analysis of blades it is enough to consider only the second load component.

Let us consider the problem of simulation of unsteady aerodynamic loads QA that act on the blade during its vibrations in a gas flow. They are linearly dependent on small vibrations of blades and can be presented through complex dimensionless aerodynamic influence coefficients (AIC) (Gorelov et al., 1971, Tsymbalyuk et al., 2001):

$$\{Q_A(t)\}=q_V h[A]\{q\}, \tag{2}$$

where $q_V = 0{,}5\rho V^2$ is the velocity pressure, $\rho$ and $V$ are the air density and relative flow velocity ahead of the blade assembly, $h$ is the blade length, $[A]$ is the matrix of generalized AICs.

It is known that the main principle of simulation is that using the results of experiments with models one should be able to describe the phenomena observed in full-scale tests. In this case, an obligatory condition is that the similarity criteria (the dimensionless complex of determining physical quantities) of the phenomenon under study and its model should be equal, and thus other dimensionless characteristics will be equal as well.

According to the principles of simulating the processes in turbomachines (Kholschevnikov et al., 1986), which are based on the $\pi$-theorem of dimensional theory, the load $Q_A$ generally depends on the following parameters:

$$Q_A = Q_A\left(b/D, u/c, K, M, \mathrm{Re}, k_a, Fr, \mathrm{Pr}\right), \tag{3}$$

where $b$ is the characteristic blade size (chord); $D$ is the characteristic diameter; u is the peripheral velocity; c is the absolute velocity and similarity criteria: $K=\dfrac{\omega\cdot b}{V}$ is the reduced vibration frequency (the Strouhal number); $M=\dfrac{V}{a}$ is the Mach number; $\mathrm{Re}=\dfrac{bV\rho}{\mu}$ is the Reynolds number; $Fr=\dfrac{Dg}{c^2}$ is the Froude number; $\mathrm{Pr}=\dfrac{\mu\cdot C_p}{\lambda_g}$ is the Prandtl number, $k_a=\dfrac{C_p}{C_v}$ is the adiabatic exponent. The following symbols are introduced here: $V$, $a$, $\rho$ are the relative velocity, sound speed and density of the mainstream flow, respectively; $\lambda_g$ is the heat conductivity coefficient of gas; $g$ is the free fall acceleration; $C_p$ is the heat capacity of gas at constant pressure; $C_v$ is the heat capacity of gas at constant volume.

It was justly mentioned by Kholschevnikov that the above-mentioned similarity criteria are too cumbersome to be used in practice. Futhermore, the experimental investigations are usually performed on models whose scale is not equal to unity, and therefore it is practically impossible to achieve complete similarity. That is why based on the analyzed processes typical of turbomachines it was shown that some similarity criteria can be ignored in simulation, namely:

- The Froude number *Fr* can be neglected, since the effect of gravitational forces in turbomachines is usually insignificant.
- The Prandtl number *Pr* and the adiabatic exponent $k_a$ can be considered as constant and be excluded from the determining parameters, if the model is tested in the same gas and at approximately the same temperature as the object of simulation.
- The Peynolds number *Re,* which represents the correlation between inertia forces and viscous forces. It has an impact on both the earodynamic resistance forces, which play a secondary role at flutter, and the formation of unstalled and stalling flow. The forms of these zones and pressure distribution in them are constant for compressor blade assemblies at $\mathrm{Re} \geq \mathrm{Re}_{\min} = 1{,}5 \cdot 10^5 \div 2 \cdot 10^5$ (Tereshchenko, 1979).

Based on the above assumptions, both unsteady aerodynamic loads $Q_A$ acting on the blade assembly and AIC are the function of three similarity criteria, namely: reduced frequency, the Mach number *M* and the peripheral-to-axial velocity ratio, which for airfoil cascades is replaced with the angle of attack *i*. This means that the matrix of generalized AICs [*A*] can be presented as a function:

$$[A] = [A(i, K, M)]. \tag{4}$$

For the Mach number M< 0.5, the effect of gas compressibility is insignificant and similarity in terms of the Mach number can be disregarded (Lamper, 1990). Moroever, in the case of low values of the reduced frequency, where the stationary hypothesis is valid, the gas compressibility can be considered by using the Pradtl-Glauert correction until the shock-install instant (Abramovich, 1976).

## 3. Straight Cascade of Airfoils

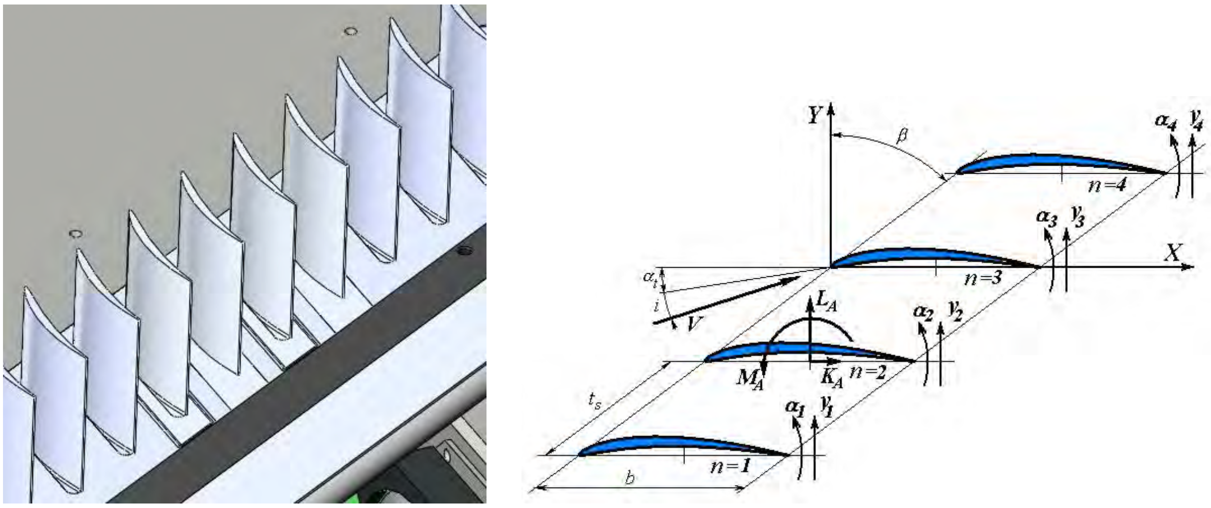
The most susceptible to subsonic flutter are the first flexural and torsional modes of blade vibrations. As far as such modes of vibrations are concerned, virtually all exchange of energy with flow is implemented in the peripheral section of the blades where the vibration amplitudes are the largest. That is why during analyzing their dynamic stability against flutter it can be sufficient to determine AICs for peripheral section, and these values can be regarded as constant along the blade length provided that the flow surfaces are close to cylindrical.

Considering the above-mentioned, the experimental investigations on determining aerodynamic loads have been performed using the straight cascade of airfoil models shown in Fig.1, *a*. These models are located parallel in a cascade and their cross sections are equal in height, which simulate the development of the chosen cylindrical section of the blade assembly. Then, the blade airfoil model is considered to be known as airfoil, whereas the straight cascade of airfoil models is referred to as airfoil cascade.

To various vibration modes of the blade assembly there correspond different combinations of translational (*x, y*) and angular ($\alpha$) displacements of airfoils, while aerodynamic loads acting on them can be represented in terms of forces $L_A$, $K_A$ and moment $M_A$, as is seen in Fig. 1, *b,* where $\beta$ is the stagger angle; $t_s$ is the cascade spacing.

Based on the test results presented by Gorelov et al. (1971), the following assumptions can be introduced:

- the influence of x on aerodynamic loads is slight;
- the force $K_A$ can be neglected, since it is small;
- the effect of other cascade airfoils on a given airfoil decreases very quickly as the distance to them increases.




(а) (б)

*Fig. 1 Straight cascade of airfoils (a) and scheme of acting loads (b)*

From this it can be concluded that:

1) an airfoil can be simulated by the system with two degrees of freedom;

2) the total unsteady aerodynamic load on a chosen cascade airfoil depends on its natural vibrations and vibrations of some nearest adjacent airfoils, which are called to be determining and denoted as «n». According to the results, obtained by Tsymbalyuk (1994), it is sufficient to consider the influence of no more than five determining airfoils on a given airfoil ($-2 \le n \le 2$) with stalled flow regime and of no more than three determining airfoils with unstalling flow regime ($-1 \le n \le 1$);

3) if each airfoil, which makes it possible to perform specified vibrations, is provided with the device for measuring unsteady aerodynamic loads (let us call them active airfoils), which are induced by vibrations of five determining airfoils ($n = 0$), then for conducting assessment of aerodynamic loads on the chosen airfoil ($-2 \le n \le 2$) three active airfoils are found to be sufficient, whereas two active airfoils are sufficient for assessment of the influence of three determining airfoils ($-1 \le n \le 1$). This follows from the fact that, owing to periodicity, the impact of $n = -1$ on $n = 1$ is similar to the impact of $n = -2$ on $n = 0$, while the impact of $n = 1$ on $n = -1$ and that of $n = 2$ on $n = 0$ are the same.

## 4. Unsteady Aerodynamic Loads Measurement Procedure

To determine the AIC matrix, it is required to measure the aerodynamic force $L_A$ and moment $M_A$ for only several linearly-independent combinations of vibrations $y$ and $\alpha$.

Within the context of the present paper, the procedure for experimental determination of aerodynamic loads, which basic statements are presented by Tsymbalyuk (1996), has been improved. The procedure described makes it possible to measure the force $L_A$ and moment $M_A$ simultaneously at arbitrary combinations of translational displacements $y$ and $\alpha$ of the objects under study. With this aim in view, excitation of vibrations of the elastic suspension is performed and its scheme is shown in Fig. 2. In accordance with the Ampere law, to determine aerodynamic forces and moments, it is required to measure the current in moving coils of the vibrator both with flow and without it, moreover, the specified airfoil vibrations are maintained unchanged.

Reduction of the error in determining aerodynamic loads from the difference between two measurements and enhancement of the sensitivity of the measuring device can be achieved, if the following conditions are fulfilled:

1) The center of mass of the elastic suspension should coincide with the rotation axis, i.e. $x_m = 0$. In this case the flexural and torsional vibrations of the elastic suspension will be mechanically uncoupled, which makes independent control of vibrations easier;

2) Natural frequencies of flexural $\Omega_y$ and torsional $\Omega_\alpha$ vibrations of the elastic suspension should be equal to the operating frequency of the excitation $\omega$;

3) Mechanical damping of vibrations should be insignificant. To do this would require a small mass and the minimum number of joints for vibrating parts of the suspension. The realization of the latter condition will contribute to enhancing the stability of the vibrating system properties. At the same time, the elastic suspension of the given airfoil should be vibroinsulated from the bench test and elastic suspensions of other airfoils.

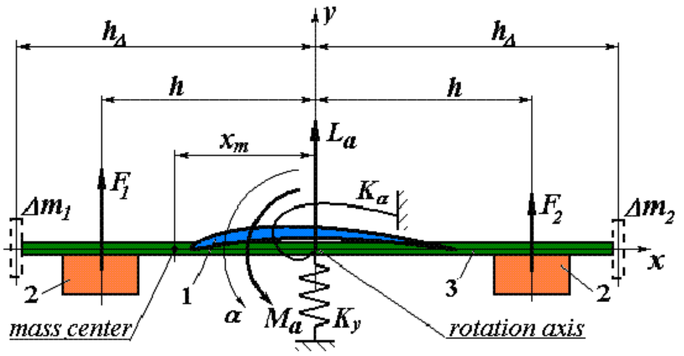



*Fig. 2 Scheme of the elastic suspension of the airfoil: 1 – cross beam; 2 – moving coil of the vibrator; 3 – airfoil under study*

Apart from the above-mentioned conditions, the relation $k_\omega = \Omega_1 / \omega$, where $\Omega_1$ is the first natural frequency of the airfoil vibrations, which is, as a rule, flexural, should be sufficiently high to achieve similarity between vibration amplitudes along the airfoil. For instance, the relation between the maximum and minimum amplitudes of the airfoil displacement should not exceed 1.065 at $k_\omega \geq 4$.

## 5. Test Bench for Determination of Unsteady Aerodynamic Loads

Measurement of aerodynamic loads, which act on the airfoils vibrating in a flow, is performed using the developed automated test bench, which scheme is shown in Fig.3 and its special features are presented in Table.

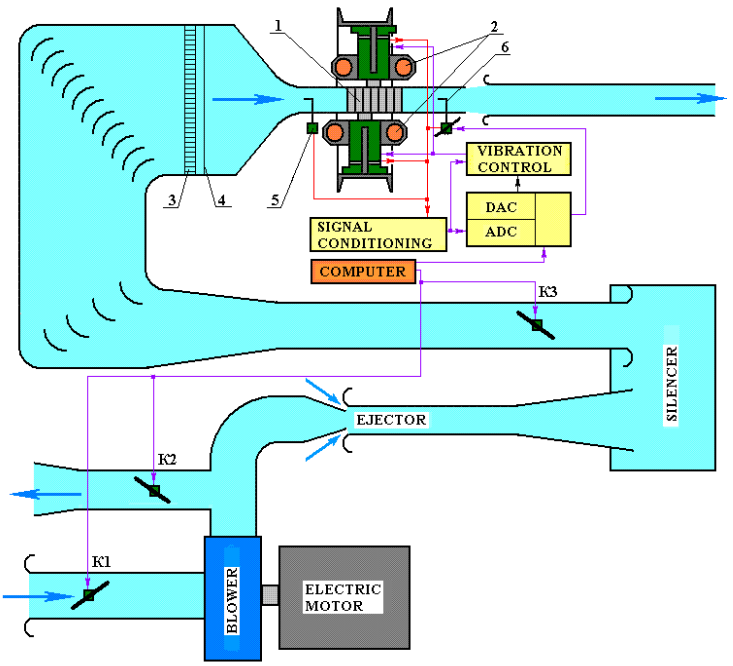



*Fig. 3 Scheme of the test bench for determining unsteady aerodynamic loads acting on blades vibrating in a flow*

*Table –Special Features of the Test Bench*

| | Objects under Study | Airfoil Cascades |
|---|---|---|
| 1. | Mach number ahead of the cascade | no more than 0.75 |
| 2. | Angles of attack | -15°...+20° |
| 3. | Airfoil length | 70.0 mm |
| 4. | Number of airfoils in a cascade (depending on the stagger angles and spacing) | 7...13 |
| 5. | Number of active airfoils | Max. 4 |
| 6. | Number of degrees of freedom of active airfoils | 2 |
| 7. | Vibration modes of active airfoils | Angular, translational vibrations and their combinations |
| 8. | Aerodynamic loads to be measured | Forces and moments |
| 9. | Operating frequency | Up to 250 Hz |

The test bench is the open circuit subsonic wind tunnel with which the air is drawn from the atmosphere and rejected back into the surroundings. Since the centrifugal blower generates higher pressure than is required for the test bench, an injector is located at the inlet of the test bench, where due to excess pressure additional airflow is induced through the working section. A settling chamber contains honeycomb 3 and mesh screens 4, which are used for creating a uniform air flow. Behind the settling chamber there is an accelerating nozzle with a large contraction ratio and walls that are curved along the so-called double-sine cross section.

To ensure the preset rates and angles of attack of the flow around airfoils, the cascade of airfoils is put into the working section shown in Fig.4. The maximum four central airfoils 7 can be active. Each of them is fixed on individual vibration unit 8, which structure and function will be provided in detail below. The other (background) airfoils are mounted rigidly to the rotating base plates 1. Setting of the airfoil cascade to the required angles of attack is performed by rotating the base plate 1 and frame with vibration units.

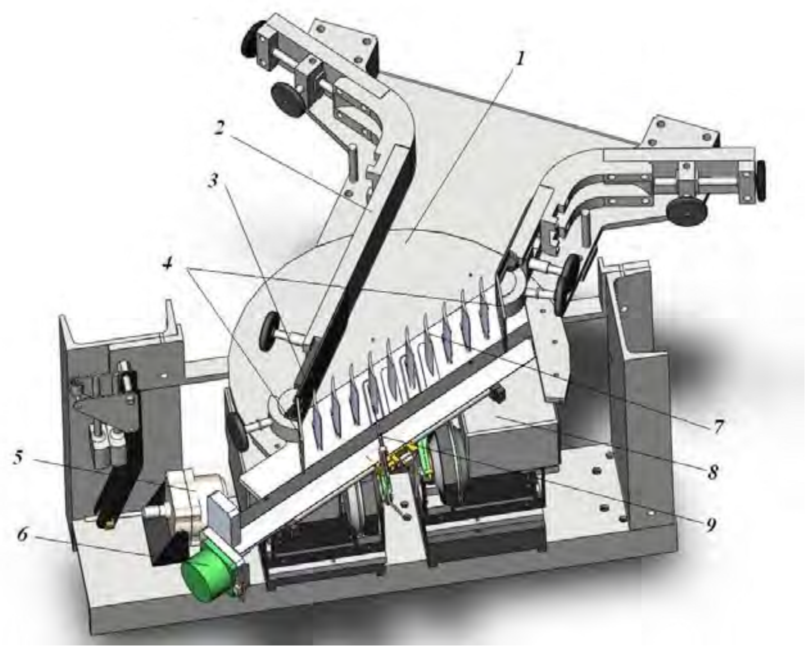
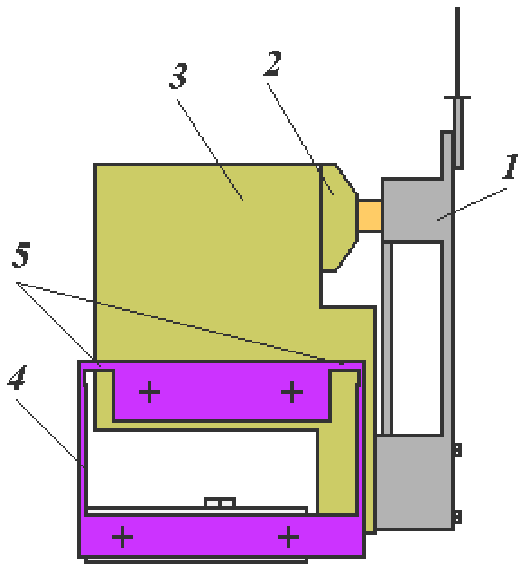



*Fig. 4 Scheme of the working section of the aerodynamic test bench: 1 – rotating base plate; 2 – nozzle walls; 3 – blades; 4 –rotating vanes; 5 – frame displacement drive unit; 6 – step-servo motor; 7 – airfoil cascade; 8 – vibration unit; 9 – traversing gear*

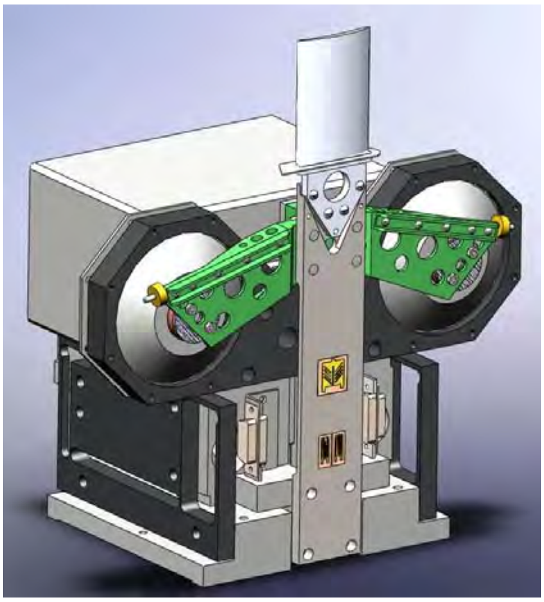
The working section has a regulated nozzle with walls 2 that can be adjusted to the side airfoils. The wall length can be adjusted by the moving blades 3. The rotating vanes are fixed at the end of the blades, they control the degree of diffuser (confuser) of the channels at the end of the cascade. The all mentioned adjustments are applied to provide the same conditions of flow in active airfoil cascades, which is controlled through the measurement of steady and unsteady components of aerodynamic forces and moments as well as through total pressure traversing and downwash behind the cascade.

To weaken the relationship between the airfoils within the test bench as well as to prevent undesirable flow fluctuations due to vibrations transferred to the wind tunnel walls that influence the accuracy of measurement, vibration isolation for elastic suspension of the airfoils is required. In the modernized test bench the elastic suspension of the airfoil is assembled in the so-called vibration unit shown in Fig.5.

The results from the performed experimental and computational investigations showed that the reason of the enhanced mechanical coupling between vibration units lies in a high rigidity of the elastic parallelogram in the vertical direction and a long distance between the centers of mass of the elastic suspension and foundation in the vertical direction. Introducing of the horizontal elastic elements 5 and lifting of the center of mass of the foundation made it possible to improve vibration isolation significantly and thus to increase the accuracy of measurement of unsteady aerodynamic loads acting on the vibrating airfoils. Use of the proposed structure of vibration unit ensured a fivefold reduction in metal consumption for the frame with vibration units.




(a)



(b)

*Fig. 5 Scheme (a) and general view (b) of the vibration unit: 1 – elastic suspension; 2- magnetic circuit of the vibrator; 3- foundation; 4- elastic parallelogram; 5- horizontal elastic elements*

The structure of elastic suspension in Fig.6 has been designed based on the requirements to procedure of measurement of unsteady aerodynamic loads. It comprises two elastic elements 4 of different width. The auxiliary (narrow) elastic element does not impede the twisting of the main (wide) element about its own longitudinal axis, and during flexural vibrations these elements form an elastic parallelogram, which ensures the equal vibration amplitudes along the airfoil length 1 upon the fulfillment of the condition $k_{\omega} = \Omega_1 / \omega$.

This unit also makes it possible to change the natural frequencies of the elastic suspension by using replaceable main elastic elements differing only in thickness. Here, the equality $\Omega_y = \Omega_\alpha = \omega$ is essentially not violated. Final adjustment of natural frequencies is made by small adjusting masses 8 placed either in the center ($\Omega_{0y} > \Omega_{0\alpha}$) or at the ends ($\Omega_{0y} < \Omega_{0\alpha}$) of the cross-beam.

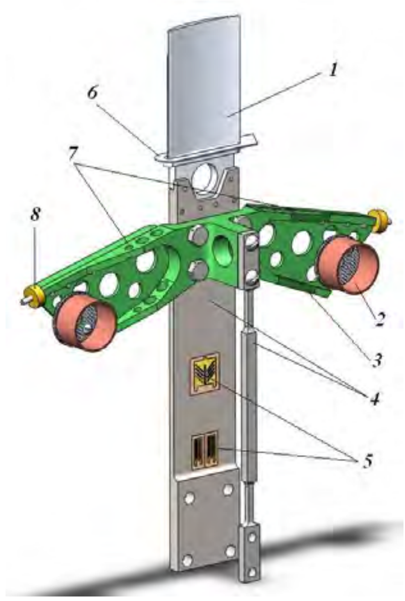



*Fig. 6 Elastic suspension: 1 - airfoil, 2 – moving coils of the vibrator, 3 – cross-beam, 4 – elastic elements, 5 – displacement strain gages; 6 – aerodynamic washer; 7 – attachment points for calibration masses; 8 – small adjusting masses*

The elastic suspension of the airfoil is secured to the foundation 3 that has magnetic circuits of the electrodynamic vibrators 2 (Fig.5). The foundation, in turn, is attached to the elastic elements 4, which form an elastic parallelogram. Their rigidity is such one that the natural frequency of vibrations of the foundation, which mass is approximately 100 times larger than the reduced mass of the elastic suspension with airfoil, is by 2 orders of magnitude lower than the natural frequency of the elastic suspension with airfoil.

## 6. Procedure for Determining the Flutter Stability Limit Using the Experiment Data

It is known that flexural mode of vibrations of twisted blade with asymmetric airfoil is accompanied by rotation of its cross sections. In a rectangular coordinate system *XY*, which is associated with the middle of the chord, the displacement of a peripheral section of the blade during its vibration can be regarded as rotation relative to the point *O* with coordinates ($X_t$ , $Y_t$) as it is shown in Fig. 7.

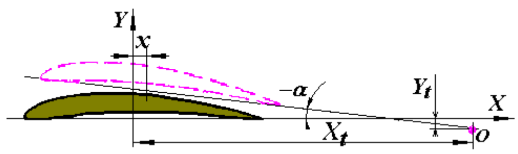



*Fig. 7 Scheme of the blade cross section displacement*

By taking into account the insignificant effect of the displacement component on unsteady aerodynamic loads in the direction *X*, it will be considered that point *O* lies on the extension of the chord, i.e. $Y_t = 0$.

If the experimental AIC values $l_{ny} \ldots m_{n\alpha}$ for the chosen cylindrical sections of the blade assembly are known, they can be used for determination of the intensity distribution of unsteady aerodynamic loads on the blades at these sections.

As a rule, the AIC values are reduced to the axis of angular displacements, which is at the middle of the chord *b*. However, recalculation of the AIC values can be performed for vibrations and can be reduced relative to the point *O*, as it is shown by Gorelov et al. (1971).

To determine the stability limit of the blade assembly against flutter, the following assumptions are made:

- there is no mechanical coupling between the blades;

- the inertia and elastic forces of a blade are much higher than the aerodynamic forces, i.e. vibration modes in a flow and in vacuum differ slightly;

- each blade has one degree of freedom and the blade assembly consisting of $N$ blades has $N$ degrees of freedom.

The equation of motion of the chosen discrete model of the blade assembly can be written as:

$$I_k \rho_k \ddot{\alpha}_k + g_{k\alpha} \dot{\alpha}_k + k_{k\alpha} \alpha_k = \tilde{M}_k \text{ ; ( } k = 1, 2, ..., N). \qquad (5)$$

Here, $I_k$, $g_{k\alpha}$, $k_{k\alpha}$, are the generalized volume moment of inertia relative to the point $O$ and coefficients of mechanical resistance and stiffness of the blades, respectively; $\rho_k$ is the density of the blade material; $\alpha_k$ are the generalized displacements of blades that correspond to the displacements of their finite cross sections; $k$ is the blade number in the assembly; $\tilde{M}_k$ is the generalized aerodynamic moment acting on each blade in the assembly during their vibration:

$$\tilde{M}_k = q_h \cdot b_h^2 \sum_{n=k-2}^{k+2} \tilde{m}_{(n-k)\alpha} \alpha_n , \qquad (6)$$

where the generalized AIC values accounting for the vibration mode $\varphi$ of the blade in vacuum are as follows:

$$\tilde{m}_{(n-k)\alpha} = \frac{1}{q_h b_h^2} \int_0^h q(z) b(z)^2 \overline{m}_{(n-k)\alpha}(z) \cdot \varphi^2(z) dz . \qquad (7)$$

The index «$h$» denotes that the specified parameters refer to the section $z = h$.

Generally for integration of distributed aerodynamic loads along the blade length it is required to determine the AIC values for several airfoil cascades that simulate different annular cross-sections of the blade assembly.

The solution to such equation (5) will be sought for in the form:

$$\alpha_k = A_k e^{pt} , \qquad (8)$$

where $A_k$ is the complex vibration amplitude, which accounts for the phase shift between vibrations of neighboring blades, $p = \varepsilon + j\omega$, $\varepsilon$ is the damping coefficient, $j$ is the imaginary unit. Since the AIC values are determined in harmonic vibrations, the solution for the stability limit should be substituted into Eq. (5), when $p = j\omega$.

Let us introduce $\lambda = \frac{1}{\omega^2} = -\frac{1}{p^2}$. Then, by substituting Eq. (8) into Eq. (5) and by classifying the members of the equation for the same amplitudes of vibration, the standard form of the problem for eigenvalues is reached. By solving the obtained system of equations, the characteristic roots N or eigenvalues $\lambda$ are derived, which are generally presented as complex values, which means that both stable ($\varepsilon < 0$) and unstable ($\varepsilon > 0$) motion of the blades is found to be possible. They are associated with the damping coefficient$\varepsilon$, angular frequency of vibrations $\omega$ and logarithmic decrement of vibrations $\delta_A$ via the following relations:

$$\lambda = -\frac{1}{(\varepsilon + j\omega)^2} ; \quad \delta_A = 2\pi \frac{\varepsilon}{\omega} . \qquad (9)$$

Moreover, the damping coefficient sign and sign of logarithmic decrement of vibrations coincide with the sign for imaginary part of the eigenvalue$\lambda$.

The eigenvalues $\lambda$ make it possible to determine the stability coefficient $\delta$ for the blade assemblies against flutter:

$$\delta = \max \operatorname{Im}(\lambda). \tag{10}$$

If the inequality $\delta < 0$ is true, the blade assembly is stable against flutter and its stability limit meets the condition $\delta = 0$.

## 7. Implementation of the Experimental and Computational Complex for Predicting the Stability of Blade Assemblies against Subsonic Flutter

As an example, Fig. 8 provides the results of determining the stability limits of blades for GTE compressor wheel against subsonic flutter. From the given data it can be concluded that with flexural vibrations operating conditions of the compressor under study are in the region of stability far from the flutter limit. At large angles of attack (both positive and negative) stability limit shifts in the direction of higher frequencies of vibrations. The operation frequency decreases with an increase in the angle of attack and approaches the stability limit, i.e. stability margin of the flexural mode of vibration of the blade assembly decreases.

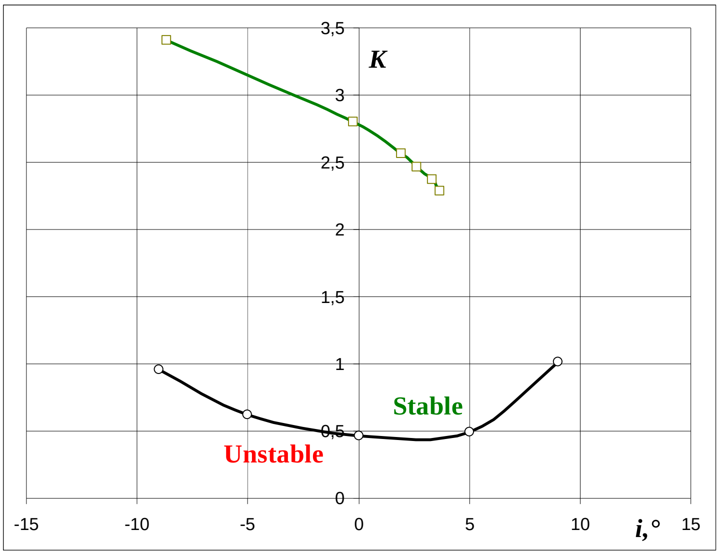



*Fig. 8 Dependence of critical reduced frequency of vibrations K on the angle of attack and for positions of axis of rotation of the blade sections $\overline{X}_t$ equal to 0.5 (o). □ - dependence of the reduced frequency for flexural mode of vibrations of full-scale blade under operating conditions of GTE*

## 8. Conclusions

Based on the principles of simulation of aeroelastic processes the complex of determining similarity criteria for simulating the behavior of compressor blade assemblies during their interaction with flow was developed. It was shown that in the experimental determination of unsteady aerodynamic loads acting on blade assemblies of GTE compressor wheels it is sufficient to consider only three similarity criteria, namely, the reduced frequency, the Mach number and angle of attack.

A methodology has been developed for experimental determination of unsteady aerodynamic loads acting on GTE compressor blade assemblies under laboratory conditions, which ensures the enhancement of accuracy of test results and substantiated selection of the parameters for the model and test bench.

The computational model of the blade assembly with flexural-torsional vibrations of its blades and procedure for determination of the flutter stability limit based on it have been developed.

The results of approbation of the proposed methods using the analysis of dynamic stability of AGTE compressor blade assemblies as an example were provided.

The test bench, which has been developed at the G.S.Pisarenko Institute for Problems of Strength, makes it possible to determine aerodynamic loads (forces and moments) acting on blades in a flow by implementing unique experimental and computational complex and based on their use to predict stability of AGTE compressor blade assemblies against subsonic flutter over a wide range of variation of their mechanical parameters and flow characteristics. Moreover, it should be noted that the developed complex is a fully automated system with modern measurement and computer facilities.

The developed experimental and computational complex aimed at predicting the stability of blade assemblies against subsonic flutter can be deployed for the design and manufacture of gas turbine engines, which are used in aircraft manufacturing, ship building, steam turbines and other fields, also it can be adjusted to predicting the dynamic stability of tube bundles of steam condensers and heat exchangers of NPP.

## 9. Acknowledgement

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