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RAISING GAS-DYNAMIC STABILITY MARGIN OF AXIAL AND CENTRIFUGAL COMPRESSOR STAGES BY MEANS OF VANED DIFFUSER BOUNDARY LAYER CONTROL

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Abstract. Generalised research results that consider the upgradability of axial and centrifugal gas turbine engine compressors by means of gas-dynamic boundary layer control on bladed disks are demonstrated. Active and passive methods are used. Comparative analysis of the results has been carried out. The analysis is purposed to determine the influence of the flow circulation around the aerofoils on the performance of compressor single-row bladed disks with smooth blades and rough blades and under the condition that vortex generators are installed.

An increase in the efficiency of aviation gas-turbine engines and in their gas-dynamic stability margin support leads to the enhancement of the parameters and performance of compressors: increase in loading of aerodynamic bladed disks, improvement of their economical efficiency, improvement of margin of the continuous flow around the compressor grids, etc.

Airflow in the compressor grid is characterised by the flow region in the flow core and also by the flow regions in the wall boundary layers on the grid blades where shock waves, vortices, air swirls, and flow separation phenomena take place.

The principle objective of the work is to research the possibilities of influence on the parameters of the elements of compressors and overall performance of gas-turbine engines via the methods of active and passive flow regulation.

Active flow regulation is realised either by rendering the auxiliary gas mass to the blades boundary layer, or by suction (withdrawal) of the boundary layer (its part) from the surfaces of blades. Passive flow around regulation is characterised by influence on the boundary layer by means of energy redistribution in the flow itself.

Keywords: compressor gas-dynamic stability, bladed disks flow control, vortices generator, vaned diffuser.

1. Introduction

Investigation of the aerodynamic performance of compressor grids along with boundary layer control has shown that by affecting the boundary layer the performance of grids can significantly be improved (Kamisti 2000; Schmidt, Mueller 1989; Tereshchenko, Mitrakhovich 1996; Tereshchenko 1979, 1987; Shlikhting 1969; Chzhen 1979) thus raising the efficiency of the gas-dynamic engine and the aircraft itself (Tishkunov *et al.* 2008). This allows considering methods of active and passive control of the boundary layer on the surfaces of blades as a means of increasing aerodynamic loads on bladed disks in design modes and extending non-separated flow range in abnormal modes.

This work reveals the research results of axial and centrifugal compressor stages, where the extension of non-separated flow range about the bladed disks is provided by the application of passive and active control of the flow about the blades.

2. Analysis of the gas-dynamic stability margin

The analysis of the dependence of gas-dynamic stability margin, $\Delta K = f(\overline{n}_{\mathrm{re}})$, at reduced rotor rotation frequency, $\overline{n}_{re} = n/n_{de}$, for some types of compressors with single-row vaned diffusers allowed revealing the most significant feature of double-row bladed disks–their non-separated flow in deep deceleration modes (n, n_{de} – the actual and design rotation frequency values respectively).

The diagram in Fig. 1 shows the centrifugal compressor ΔK change in case of replacement of the single-row vaned diffuser with the optimal double-row one. The change in gas-dynamic stability margin is demonstrated by the formula:

$$\Delta\overline{K}_d = \frac{\Delta K_d}{\Delta K} - 1 = f(\overline{n}_{\mathrm{re}}),$$

where ΔK_d – gas-dynamic stability margin of the centrifugal compressor with double-row vaned diffuser; ΔK – gas-dynamic stability margin of the compressor with equivalent single-row vaned diffuser.

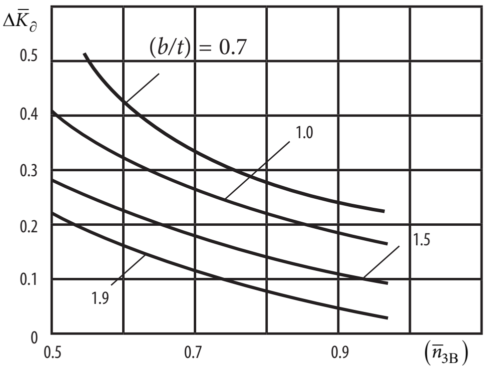

Fig. 1. The centrifugal compressor ΔK change in case of replacement of the single-row vaned diffuser

The analysis of the diagram in Fig. 1 reveals an increase in the gas-dynamic stability margin of 8–30% with the replacement of the single-row vaned diffuser by the double-row one at solidity $(b/t) = 0.66 - 1.92$ (where b – vane chord; t – distance between trailing edges of two adjacent vanes).

The flow stability in the compressor single-row vaned diffuser is disturbed when rotational stall arises in the diffuser. This phenomenon was registered during investigations of the flow in vaned diffusers in abnormal modes (Tereshchenko 1979, 1987).

With a decrease in the reduced rotation frequency to values $\overline{n}_{\mathrm{re}} = 0.6 - 0.7$, the effect from two-line grid application in the vaned diffuser increases and the gas-dynamic stability margin increases 20–48% in comparison with the compressor single-row vaned diffuser's ΔK.

The gas-dynamic stability margin is mostly affected by the application of low-solidity vaned diffusers. While at $(b/t) = 1.92$ $\Delta K = 1.2$ (which means it is growing 20%), at $(b/t) = 0.66$ $\Delta K = 1.48$ it grows 48% at $\overline{n}_{\mathrm{re}} = 0.64$.

3. Centrifugal compressor

Investigation of the gas-dynamic stability of compressors with vortex generators in the vaned diffusers revealed the most significant feature of vortex generators, that is their ability to shift the stall initiation into deeper deceleration modes (Fig. 2).

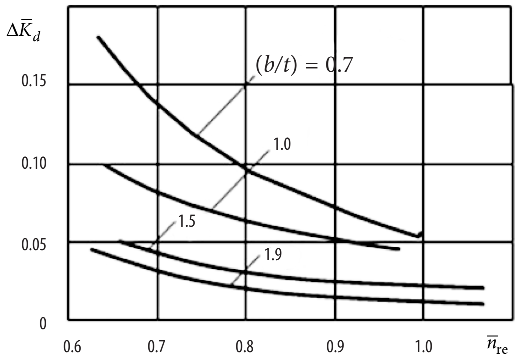

Fig. 2. The centrifugal compressor ΔK change with the vortex generators

The diagram in Fig. 2 shows the ΔK change in the centrifugal compressor with the vortex generators on the surface of the vaned diffusers. Gas-dynamic stability margins were determined for the steady modes, which correspond to the maximum values of the coefficient of performance at all compressor rotor rotation frequencies. At $\overline{n}_{\mathrm{re}} = 1$, the compressor gas-dynamic stability margin increases by 1–8% due to forced vortex generation. This effect occurs in the flow about bladed disks at small angles of attack, and the lower the solidity is, the more vortex generators affect the compressor performances.

At decreased modes of compressor operation, the forced vortex generation effect becomes more apparent, resulting in an increase in ΔK of 2–17% in comparison with ΔK of the compressor outlet. Vortex generators have the most significant influence on the flow about the grids, with solidity of $(b/t) < 1.5$.

Analysis of the gas-dynamic stability change $\Delta K' = f(\overline{n}_{re}, c_\mu)$ in the compressors with active control of the vaned diffuser flow (where c_μ –intensity coefficient of blow into the boundary layer) and $\Delta K = f(\overline{n}_{\mathrm{re}})$ of the conventional compressor enabled us to reveal one of the most important features of the vaned diffusers with blow into the boundary layer: their high resistance to initiation of stall modes at deceleration (Tereshchenko, Mitrakhovich 1996; Tereshchenko 1979).

Fig. 3 shows the change of ΔK of the centrifugal compressor when air is blown into the boundary layer on the surface of the vaned diffuser:

$$\Delta\overline{K}' = \frac{\Delta K'}{\Delta K} - 1 = f(\overline{n}_{\mathrm{re}}, b/t).$$

The increase in the gas-dynamic stability margin of compressors in the process of the vaned diffusers flow

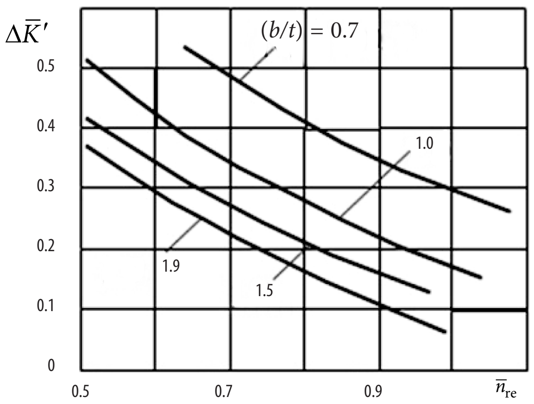

Fig. 3. The centrifugal compressor ΔK change when air is blown into the boundary layer

active control is the result of higher stability of the flow at the boundary layer. As in the flat grids, the application of blowing into the boundary layer on the vanes of low solidity diffusers is more effective.

4. Boundary layer in the vaned diffuser

Analysis of the dependence $\Delta\overline{K} = f\left(\overline{n}_{re}, m_{\text{suc}}\right)$ for compressors with suction of the boundary layer in the vaned diffuser (where m_{suc} –boundary layer suction coefficient in the vaned diffuser) showed that flow control results in an increase in the stability margins within a wide range of the compressors operation modes (Tereshchenko 1987) (Fig. 4).

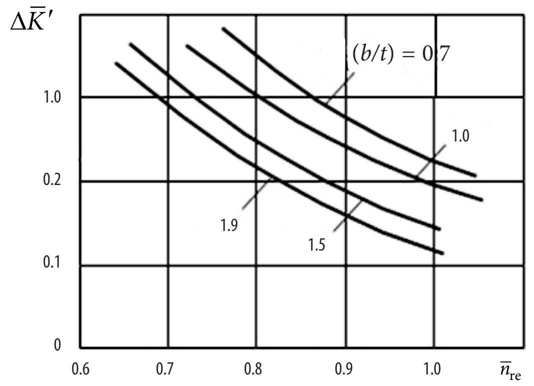

Fig. 4. The results of the analysis of diffuser compressors

Fig. 4 shows the results of the analysis of different diffuser compressors that use the effect of boundary layer suction.

Having considered the dependencies of $\Delta\overline{K}' = f\left(\overline{n}_{re}, b/t\right)$, it follows that in case of vaned diffuser boundary layer suction the gas-dynamic stability margin of ΔK changes over practically the entire operating range of the rotation frequencies. For compressors with vaned diffuser solidity $\left(b/t\right) = 0.66$, the change in ΔK varies between 27% (for $\overline{n}_{red} = 1$) to 43% (for $\overline{n}_{\text{red}} = 0.65$). The change in diffuser solidity to $\left(b/t\right) = 0.96$ results in an increase in ΔK of 22% for $\overline{n}_{red} = 1$ and 38% for $\overline{n}_{\text{red}} = 0.65$. The stability margins of compressors with diffusers that have solidity of $\left(b/t\right) = 1.46$ increase 15–35% at boundary layer suction. In compressors with diffuser solidity $\left(b/t\right) = 1.92$ the ΔK change is 10–33% at boundary layer suction.

Use of multiple-lane bladed disks in the axial compressor stages, especially with high rotation frequency, becomes significantly complicated due to the insufficient amount of research on the air elastic characteristics of such disks. Nowadays there are no sufficiently justified recommendations concerning which components of the axial stage (in the multistage axial compressor) should be designed as multiple-lane.

It is known that the use of double-row bladed disks in the guide vanes of some axial stages enables large-angle flow turns to be obtained in these components and to improve their stall characteristics under low degrees of reaction (Fig. 5). At the same time, the compression operation on degrees does not exceed 25–30 kJ/kg during rotation of up to 300 m/s. Basic double-row bladed disk characteristics of one of the researched running disks and the dependence of relative loss factor $\overline{\xi}$ on grid characteristics are shown in Fig. 5. The total angle of the flow turn in the bladed disks of the running disk, which is provided by the double-row grids (in design mode), varied from 19.6° (in peripheral intersection) to 48.3° (in the core section). With circumferential speed u_{com}=160 m/s, the compression ratio was H_{com} = 20 – 22 kJ/kg on some compressor stages. Bladed disk solidity of running disks varied from $\left(b/t\right) = 0.978$ in the core section to $\left(b/t\right) = 1.201 - 1.263$ in the peripheral intersection. With circumferential speed $u_{com} = 160$ m/s and airflow rate under the design mode $m_r = 6.6$ kg/s, the compressor provides the pressure increase ratio $\pi = 2.5$. Fig. 5 presents the diagram for defining the optimal relative position of aerofoils in the double-row bladed disks of the running disks. The optimisation condition is to obtain the lowest rate of total pressure loss $\overline{\xi}$.

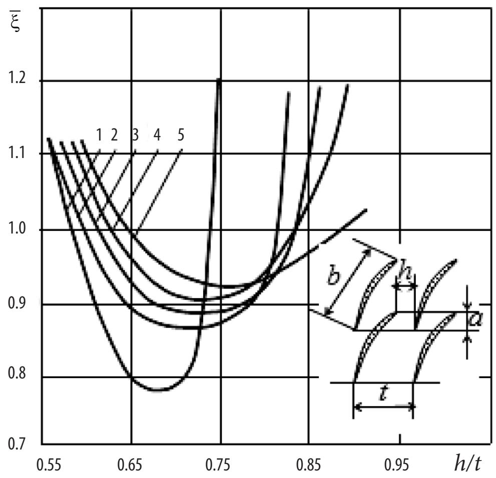

Fig. 5. The characteristics of basic double-row bladed disk

The research results of the axial stage of the compressor with double-row bladed disks of running disks showed that within the optimisation of characteristics of double-row bladed disks of running blades with minimum loss (under the relative row offset of $(h/t)=0.68-0.77$ and relative gap depth of $(a/b)=0.07-0.025$ on average bladed disk diameter), the loss ratio in the running disks of the compressor stage in abnormal modes is 10–20% lower than in the stage with the single-row bladed disks that have the same pressure in the design mode. The stall limit of such a stage (determined by calculations at the maximum load of the grids) is significantly shifted toward lower air consumption. In the assessment of the gas-dynamic stability margin under the value of ΔK determined for the design mode $\overline{\pi}^*_{com}=\frac{\pi^*_{com}}{\pi^*_{gr}}=1$, $\overline{q}(\lambda)=\frac{q(\lambda)}{q_c(\lambda)}=1$, an increase in ΔK from 0.3–0.35 to 0.6–0.65 is observed. $\overline{\pi}^*_{com}$, π^*_{com}, π^*_{gr} i $\overline{q}(\lambda)$, $q(\lambda)$, $q_c(\lambda)$ are relative, actual, and calculated value of the pressure growth ratio and the gas-dynamic function, and λ is the velocity coefficient. The research results of the axial stage of the compressor with double-row bladed disks of running disks showed that within the optimisation of the characteristics of double-row bladed disks of running blades with minimum loss (under the relative row offset of $(h/t)=0.68-0.77$ and the relative gap depth of $(a/b)=0.07-0.025$ on average bladed disk diameter), the loss ratio in the running disks of the compressor stage in abnormal modes is 10–20% lower than in the stage with the single-row bladed disks that have the same pressure in the design mode. The stall limit of such a stage (determined by calculations at the maximum load of the grids) is significantly shifted toward lower air consumption. In the assessment of the gas-dynamic stability margin under the value of ΔK determined for the design mode $\overline{\pi}^*_{com}=\frac{\pi^*_{com}}{\pi^*_{gr}}=1$, $\overline{q}(\lambda)=\frac{q(\lambda)}{q_c(\lambda)}=1$, an increase in ΔK from 0.3–0.35 to 0.6–0.65 is observed. $\overline{\pi}^*_{com}$, π^*_{com}, π^*_{gr} i $\overline{q}(\lambda)$, $q(\lambda)$, $q_c(\lambda)$ are relative, actual, and calculated value of the pressure growth ratio and the gas-dynamic function, and λ is the velocity coefficient.

Fig. 6 presents (in relative characteristics) the characteristics of the axial compressor stages with single-row and double-row bladed disks of the bladed disks under $\pi^*_{gr}=1.17$ and the circumferential speed under the design mode of $u_c=280\,\text{m/s}$, determined by the generalised characteristics method of the flat compressor grids. Analysis of these characteristics shows that in abnormal modes the pressure growth ratio on the double-row bladed disk stage is 10–15% higher than on the stage of the single-row bladed disk.

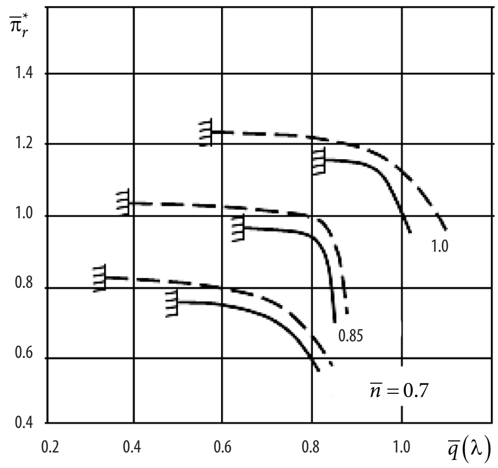

Fig. 6. The characteristics of the axial compressor

5. The characteristics of the axial compressor

Comparison of the characteristics of compressor stages with the double-row bladed disks (Fig. 5), the parameters of which were set up under the condition of obtaining the minimum value of the related total pressure discharge coefficient $\xi_{\min}$, shows significant differences (in Fig. 5: line 1– $(a/b)=0.071$; 2 – $(a/b)=0.047$; 3 – $(a/b)=0.024$; 4 – $(a/b)=0.008$; 5 – $(a/b)=0$).

While profiling of the double-row bladed disks, if $\xi_{\min}$ is in the design mode, the level of loss is lower than with the equivalent pressure on the single-row bladed disk axial stages. The stages with double-row vaned diffusers, profiled under the condition that $k=\left(c_y/c_x\right)_{\max}$, under the rated conditions have higher pressure, higher loss ratio, and lower efficiency factor relatively (c_y and c_x–lift coefficient and resistance force coefficient). In the abnormal modes (mainly in the left branches of the pressure lines), efficiency of the double-row bladed disk stages is slightly higher than the efficiency of the stages with the equivalent single-row bladed disks (Tereshchenko, Mitrakhovich 1996).

One of the less researched problems in compressor construction is the impact of the blade surface state on the performance and characteristics of the multi-stage compressors levels. Asperities and irregularities on the blade surface that do not exceed the dimensions of the boundary layer affect only the boundary layer, and by way of it they influence the flow in the inter-blade channel. Asperities and irregularities, the dimensions of which exceed the parameters of the boundary layer, directly influence the current in the flow core, leading to generation of active vortices. Some studies emphasise that the characteristics of axial compressors are significantly dependent on the surface condition of blades in bladed disks (Shlikhting 1969; Chzhen 1979).

6. Influence of the roughness of blade surfaces

Research regarding the influence of the roughness of blade surfaces on axial compressor characteristics show that the total roughness of the blade surface, which corresponds to the relative roughness height of $(K/b)=0.45\cdot 10^{-2}$, leads to a decrease in the ratio of pressure growth of almost 30% while the rate of airflow drops 15–20% simultaneously (in comparison with the values of the flow rate through a compressor with smooth blades) (Schmidt, Mueller 1989; Tereshchenko, Mitrakhovich 1996; Tereshchenko 1979). These results can be found by studying the characteristics of the triple-stage axial compressor with adiabatic compression operation of $H_{com}=67.3$ kJ/kg and air flow rate under the design mode of $m_{com}=10$ kg/s. The outer diameter of the compressor was $D_{com}=340$ mm, and circumferential speed in the design mode was $u_{com}=302.5$ m/s. For the simulation of roughness, emery granules were glued on the blade surface of the bladed disc. Moreover, the distribution of granules on the blade surface was strictly uniform.

Characteristics of the compressors with rough blades were researched at relative rotation frequency flow values, that is $\bar{n}=\frac{n}{n_{com}}=1$. The Reynolds number was changing in the range of $(3–8.5)\cdot 10^5$. Mach number on the inlet to the operating blades was equal to $M=0.75$.

The influence of blade roughness on the pressure line percolation under the rated rotation frequency is illustrated in Fig. 7. The related value of compression work:

$$\bar{H}=\frac{H_{com}}{H_{gr}}.$$

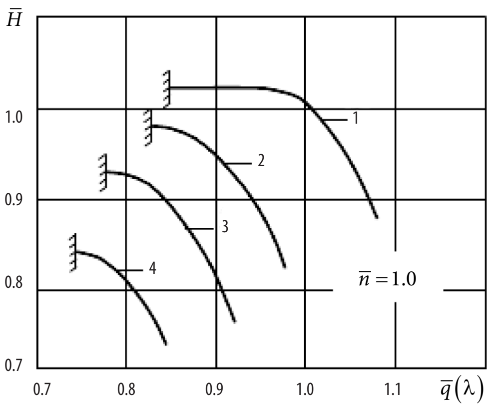

Fig. 7. The influence of blade roughness

Three variations of blade surface roughness (small, medium and large) were considered, which corresponds to certain values of the relative roughness parameter K/b, namely: 1 – compressor with smooth blades; 2 – relative roughness of $(K/b)=1.5\cdot 10^{-}$; 3 – relative roughness of $(K/b)=2.51\cdot 10^{-3}$; 4 – relative roughness of $(K/b)=4.51\cdot 10^{-3}$.

In comparison with the compressor with smooth blades, the compressor with rough blade surface under the rated rotation frequency had pressure lines shifted towards the lower values of airflow consumption.

Analysis of the diagram in Fig. 7 shows, that the increase in roughness uniformly distributed on the blades surface leads to a reduction in the compressor efficiency factor in the compressor nominal mode of 6% (with relative roughness of $(K/b)=4.51\cdot 10^{-3}$) (Kamisti 2000). The main reasons for these changes are the increase in loss and reduction in the deflection angles in the rough surface blade assembly.

Analysis of these characteristics showed that the shift in pressure lines towards lower losses is accompanied by an increase in the characteristics slope and reduction in the values of the pressure increase ratio, which is achieved at each rotational frequency.

These results correspond well enough to the data about the impact of the state of the streamlined surface, that is the level of its roughness on the resistance of the bodies that are flown round by cohesive liquid (Shlikhting 1969; Chzhen 1979) (Fig. 8).

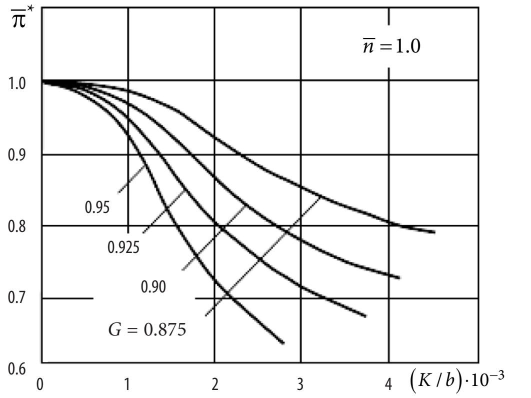

Fig. 8. The formation of artificial roughness on the blade surface

At the same time, according to Fig. 8, by formation of artificial roughness on the blade surface in the bladed disks, it is partially possible to affect the characteristics of the axial compressor, especially the pressure increase ratio: $\bar{\pi}^*_{com}=\frac{\pi^*_{\text{com.rough}}}{\pi^*_{\text{com.smooth}}}$; the inlet area blade back roughness of the bladed disk is distributed along the entire blade height, which leads to a shift in pressure lines downwards at all frequencies of rotor rotation, and the range of the non-separated flow about the stage expands: the limit of rotational stall shifts towards lower air flow rate.

Fig. 9 presents the characteristics of the axial compressor ratio with smooth blades (dashed lines) and with blades that have laminar boundary layer vortex generators on the inlet back area (solid line).

From analysis of the diagrams, it is concluded that the vortex generators in the inlet area of the blades flatten the characteristics and shift the rotational stall limit to lower air consumption.

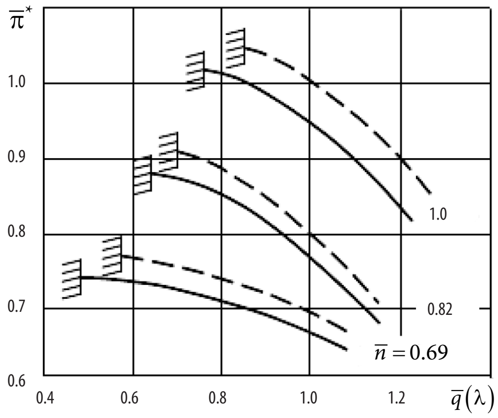

Fig. 9. The characteristics of the axial compressor ratio with smooth blades

7. Conclusions

The aforementioned research results suggest that the use of active and passive bladed disks flow control is a very effective way of improving the characteristics of axial and centrifugal compressors, as well as gas-dynamic stability. Analysis of results shows that when a single-row vaned diffuser is replaced by double-row at optimal grid solidity and the frequent rotor rotation gas-dynamic stability ΔK increases up to 48%. The pressure growth ratio increases up to 10–15%, which improves the efficiency factor of the double-row bladed disk stages. By means of generation of forced vortices, ΔK may be increased (up to 17%) when using certain modes of compressor operation. It is possible to affect the characteristics of the axial compressor by formation of artificial roughness on the blade surface in the bladed disks. An increase in roughness to $4.51 \cdot 10^{-3}$ uniformly distributed on the blade surface leads to a 6% reduction in the compressor efficiency factor in nominal mode.

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DUJŲ DINAMINIO STABILUMO RIBOS AŠINIO IR IŠCENTRINIO KOMPRESORIAUS PAKOPOSE DIDINIMAS ATLIEKANT STABILIZUOTO DIFUZORIAUS PARIBIO SLUOKSNIO KONTROLĘ

I. Lastivka

Santrauka. Šiuo tyrimu siekiama nustatyti sparno profilio aptekėjimo įtaką vienos eilės menčių kompresoriaus su lygiomis ir šiurkščiomis mentėmis darbui, esant įdiegtiems sūkurio generatoriams. Pagrindinis darbo tikslas – ištirti kompresoriaus elementų ir bendro dujų turbininių variklių darbo įtaką parametrams, taikant pasyvų ir aktyvų srauto reguliavimo metodus. Padidinus dujų turbininių variklių našumą ir jų dujų dinamikos stabilumo ribas, pagerėja kompresorių darbas ir parametrai: padidėja aerodinaminių diskų su mentėmis apkrova, jie tampa ekonomiškai našesni, padidėja nepertraukiamo srauto riba aplink kompresoriaus plokšteles.

Reikšminiai žodžiai: dujų dinaminis kompresoriaus stabilumas, diskų srauto valdymas, sūkurio generatorius, stabilizatoriaus sklaidytuvas.