GEOTECHNICAL AND MINING MECHANICAL ENGINEERING, MACHINE BUILDING

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AEROACOUSTIC CHARACTERISTICS OF THE AXIAL COMPRESSOR STAGE WITH TANDEM IMPELLER

Purpose. Comparative evaluation of aeroacoustic characteristics of the axial compressor stages with a single and equivalent tandem impeller.

Methodology. The research was performed using the numerical experiment. The flow parameters in the stage of the axial compressor were calculated by solving the non-stationary system of Navier − Stokes equations. The equations were closed by the model of turbulent viscosity SST. The stage of the compressor consisted of inlet guide vanes, the impeller and the guide vanes. The impeller of the modified axial compressor stage is designed as an equivalent tandem row. Acoustic sources of the axial compressor stage were calculated using the Ffowcs Williams − Hawkings equation.

Findings. The results of a comparative evaluation of the aeroacoustic characteristics of the axial compressor stages with a single and equivalent double-row impeller are obtained. The use of a tandem row instead of an equivalent single row in the impeller of the axial compressor stage makes it possible to increase the pressure ratio by 1−15 %. In the design mode, the pressure ratio is increased by 8.5 %. A stage with a tandem impeller has greater acoustic efficiency than a compressor stage with a single impeller. The obtained results demonstrate a significant decrease in the acoustic pressure of the dipole source. In the design mode, the acoustic pressure of the dipole source decreases by more than 70 %, the acoustic pressure of the quadrupole source decreases by more than 10 %.

Originality. For the first time the results of a comprehensive study of the aerodynamic and acoustic characteristics of the axial compressor stage were obtained. The results allow estimating the efficiency of application of tandem row in the impeller of the low loaded subsonic stage of the axial compressor.

Practical value. The received recommendations can be used while designing impeller machines with a low level of acoustic emission.

Keywords: *axial compressor stage, tandem impeller, dipole acoustic source, quadrupole acoustic source, pressure ratio*

Introduction. One of the important problems of the aircraft engine building is to increase the efficiency of gas turbine engines and to ensure gas-dynamic stability in a wide range of operating conditions. The parameters of the working process and the loss level in the elements of the engine determine the perfection of the gas turbine engine. First of all, improving the fuel efficiency and engine efficiency is ensured by increasing the gas temperature before the turbine. The gas temperature growth before the turbine requires a simultaneous increase in pressure ratio of the compressor. Pressure ratio growth is provided by raising the rotor speed, the aerodynamic loading growth of the stages, and also by increasing the number of stages. However, increasing the number of stages entails the overall mass characteristics growth of the engine [1]. A promising way to increase the aerodynamic loading of stages is the use of tandem blades.

In addition to high engine efficiency and fuel efficiency, modern gas-turbine engine should have a high acoustic perfection. One of the main noise sources of gas-turbine engine is the compressor. The sound-absorbing materials are widely used for noise reduction. However, the noise reduction achieved with the help of the sound-absorbing materials is not enough. Therefore, research studies aimed at improving the aerody-

namic and acoustic characteristics of tandem axial compressor stages are relevant.

Analysis of the recent research and problem statement. Today quite a lot of experimental and theoretical works are devoted to the study of aerodynamic characteristics of tandem blade rows. The results of 2D flow simulation in a single and tandem impeller of a low-speed axial compressor are presented in [2]. The authors analyzed the distribution of the pressure field in the interblade channel. The flow visualizations in single and tandem cascade of the impeller are shown. Paper [3] presents the results of numerical simulation of the flow in a blade row with tandem blades. The influence of geometric parameters of a tandem blade row on its aerodynamic characteristics is analyzed. A method for designing of the compressor stage with tandem blade rows is proposed in [4]. The authors obtained and analyzed the characteristics of stage with single impeller and guide vanes, tandem impeller and guide vanes. It is shown that loading of the stage with tandem blades increases. Work [5] is devoted to the investigation of the effectiveness of tandem variable guide vanes. The characteristics of a stage with tandem variable guide vanes are analyzed in article [5]. It is shown that tandem blade row allows improving pressure ratio of the stage. The analysis of [2−5] shows that the use of tandem blade rows makes it possible to improve the aerodynamic characteristics and increase the compressor loading.

Many theoretical and experimental works are devoted to the understanding of noise formation mechanism in the elements of turbomachines. The noise sources in the near field of modern aircrafts are analyzed in [6]. The authors present critical review of different methods for calculating the acoustic characteristics in the near field. It is shown that compressor and fan are one of the main noise sources in engine. In [7], different factors that affect the noise level in the axial fan are investigated. On the basis of experimental data, the generation of tonal components of the acoustic radiation of an axial fan at low subsonic speed is analyzed. The noise generation of a four-stage axial compressor at high subsonic speed is considered in [8]. The interaction effect of rotor and stator in the calculation of the acoustic characteristics of the compressor is investigated in [9]. Also, the authors identified the dependence of tonal noise from the shape of the leading edge of the rotor blades.

As shown in [6−9], the components of the acoustic radiation vary according to different operation modes, as a result of the gas-dynamic interaction of elements of the rotor-stator system, the presence of turbulence and many other factors.

Analysis of the research presented in [2−9] has shown that one of the methods to increase aerodynamic loading of the axial compressor stages is the use of tandem blade rows. However, the acoustic characteristics of tandem blade rows have not been investigated to the present day.

The objectives of the article. The aim of present work is to improve the acoustic characteristics of axial compressor stage. To accomplish the aim of the research, the following tasks were set.

1. To perform a calculation analysis of aerodynamic loading of the axial compressor stage with single and equivalent tandem impellers in different operational modes.

2. To estimate acoustic characteristics of the axial compressor stage with a single row and equivalent tandem row of the impeller in different operational modes.

Research technique. The research was performed using the numerical experiment. The flow parameters in the stage of the axial compressor were calculated by solving the non-stationary system of Navier − Stokes equations. The equations were closed-formed by the turbulent viscosity model SST. A 3D model of the axial compressor stage was built. The stage of the compressor consisted of inlet guide vanes, the impeller and the guide vanes. An unstructured adaptive grid was used for calculation. Computational grid consisted of approximately 2 million cells. According to condition of periodicity, the design area of each blade was composed of a single blade and an interblade channel.

Acoustic sources of the axial compressor stage were calculated using the Ffowcs Williams − Hawkings equation.

Reliability of the obtained results was ensured by performing the test [10].

Main results. The object of research was axial compressor stage. The stage of the compressor consisted of the inlet guide vanes, the impeller and the guide vanes. The blade row of the inlet guide vanes had 30 blades; the impeller and guide vanes had 24 blades.

Table gives the geometric characteristics of the base axial compressor stage with single blade rows [12]: r − the cross-section radius; t/b − relative pitch of the airfoil cascade; β_1 − the angle of flow inlet; β_2 −angle of flow outlet; γ − the angle between profile chord b and the rotor rotation plane.

The modified stage of the axial compressor consisted of three blade rows: inlet guide vanes (30 blades), impeller (24 blades), and guide vanes (24 blades). The blade row of

Table

Geometrical characteristics of the blade rows

Inlet guide vanes					
r, mm	80	104	136	168	200
t/b	0.48	0.624	0.816	1.008	1.2
γ	89°18'	87°37'	85°10'	83°	81°46'
Impeller					
r, mm	80	104	136	168	200
t/b	0.468	0.589	0.742	0.878	1.005
β_1	62°37'	57°55'	51°47'	46°42'	42°09'
β_2	101°43'	96°27'	88°30'	80°03'	71°13'
γ	85°01'	80°23'	73°40'	66°57'	60°02'
Guide vanes					
r, mm	80	104	136	168	200
t/b	0.475	0.55	0.633	0.699	0.747
γ	76°30'	74°40'	72°17'	69°59'	67°36'

the impeller is made with an equivalent tandem blade row. The blade row of the tandem impeller is designed according to the recommendations obtained in [11].

A series of calculations of flow parameters in the stage of an axial compressor with a single and equivalent tandem impeller were carried out. The calculations were performed for modes with a circumferential velocity at the peripheral radius $u_1 = 272.7$ m/s, $u_2 = 238.7$ m/s, $u_3 = 204.7$ m/s, $u_4 = 170.7$ m/s.

To evaluate the aerodynamic loading of an axial compressor stage, the pressure ratio for four rotation speeds was calculated. Based on the calculation results the dependence of pressure ratio in stage from the gas-dynamic functions $q(\lambda_c)$ was built.

The pressure ratio was calculated by the formula

$$\pi = \frac{p_2^*}{p_1^*},$$

where p_1^* is the average value of total pressure at the stage inlet; p_2^* is the average value of total pressure at the outlet of the stage. The averaging of the flow parameters was carried out according to the principle of average mass averaging over the radius.

Gas-dynamic function $q(\lambda_c)$ was determined from the relation

$$q(\lambda_c) = \lambda_c \left(1 - \frac{k-1}{k+1}\lambda_c^2\right)^{\frac{1}{k-1}} \left(\frac{k+1}{2}\right)^{\frac{1}{k-1}},$$

where $\lambda_c = \dfrac{c}{a}$ is the velocity coefficient; c is axial flow velocity at the inlet; a is sound speed; k is the adiabatic index.

Fig. 1 shows the compressor stage characteristics in the form of dependence of pressure ratio on the gas-dynamic function $q(\lambda_c)$. The solid line shows the results for the base stage of the axial compressor with a single impeller, the line with points shows the results for the modified stage of axial compressor with an equivalent tandem impeller.

Analysis of the graphs in Fig. 1 shows that the stage with a tandem impeller in all operational modes has greater pressure ratio than the compressor stage with a single impeller.

At rotational speed n_4, the minimum value of pressure ratio on the left branch increases from 1.08 to 1.105, maximum − from 1.095 to 1.106. At rotational speed n_3, the minimum value of pressure ratio on the left branch increases from 1.1 to 1.133, maximum − from 1.135 to 1.16. The largest growth of pressure ratio is observed at a rotational speed of n_2 and n_1. At rotational speed n_2, the minimum value of pressure ratio on the left branch increases from 1.123 to 1.183, maximum − from 1.175 to 1.235. At rotational speed n_1, the minimum value of pressure ratio on the left branch increases from 1.18 to 1.26, maximum − from 1.21 to 1.35. At the design operation mode, the pressure ratio increases from 1.21 to 1.314. The use of an equivalent tandem impeller gave the opportunity to improve the
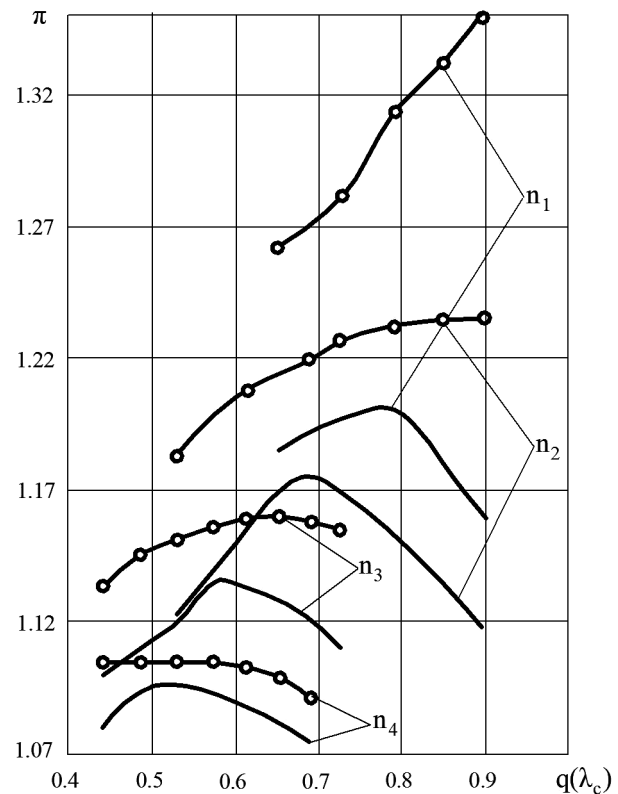

Fig. 1. Characteristics of axial compressor stage

pressure ratio value on the left branches of pressure line to maximum values of axial compressor stage with a single impeller.

This phenomenon can be explained as follows. Passive control of the boundary layer takes place in a tandem impeller. That is why the deflection angle of a tandem impeller is more than the deflection angle of a single impeller. Thus, increasing the aerodynamic loading of the axial compressor stage can be achieved by use of a tandem impeller. Obtained result agrees well with the results of other authors [4, 5].

The next objective of the research was comparative estimation of acoustic characteristics of axial compressor stages.

The aerodynamic loading acting on blades of impeller and guide vanes is continuous in time. In the general case they are non-stationary random functions and for constant operation modes of the impeller can be represented as

$$F = \Phi(t)e^{i\omega t},$$

where $\Phi(t)$ is a stationary random function; $e^{i\omega t}$ is deterministic multiplier, which takes into account the periodic nature of function change. A stationary random function $\Phi(t)$ may be considered as the sum of the static, i.e. time-independent, and dynamic or pulsating components.

The acoustic radiation is formed during the action on non-stationary force environment. Non-stationary force is analogous to the periodic action on environment by impeller blades. Acoustic radiation spectrum is superposition of spectra of static and pulsating loading

components. Static component of aerodynamic loading corresponds to discrete emission spectrum, and pulsating component is continuous in frequency spectrum radiation. Acoustic emission of axial compressor stage according to effect on the non-stationary loading environment may be divided, depending on nature of loading change in time, into continuous and pulse, and depending on kind of radiation spectrum on the harmonic and broadband.

The impulse type radiation occurs in interaction of vanes with uniformly distributed over the wheel disk unevenness in the form of turbulent tracks behind the impeller blades. The impulse type radiation can also be generated by the impeller blades when inlet guide vanes are installed in front of the impeller. Pulse radiation spectrum is a superposition of harmonic and broadband components.

There are several possible models for interaction of flow unevenness with the impeller and the guide vanes, which leads to sound formation. One of the models is dipole. The dipole model explains the sound appearance by the action of variable pulsating forces on the blade. Another model is a quadrupole model. It connects the sound generation with fluctuations of Reynolds stresses in rotational flow.

The discrete components are dominated at the frequency of impeller blades and its harmonics at subsonic speeds of impeller in spectrum on the background of broadband noise.

The appearance of broadband and discrete components of the noise spectrum is due to stationary and random flow unevenness at subsonic relative speeds on the impeller blades. Pressure fluctuations arise as a result of uneven flow interaction with the blades, and turn into sound waves. Stationary flow unevenness is associated with the general flow unevenness that enters the input device. The random unevenness is associated with turbulence of input flow, boundary layer on the walls and on blades, as well as with turbulence in the tracks behind the blades and turbulence in secondary flows on the impeller periphery.

Acoustic radiation of the compressor stage is caused by aerodynamic sources. Discrete and broadband components are present in noise spectrum of axial compressor stage. A discrete component is generated by the dipole and monopole acoustic sources. The dipole acoustic source is caused by aerodynamic loading of axial compressor stage. The monopole acoustic source is due to displacement noise, but it has quite a small value for compressors and fans.

The quadrupole acoustic source is formed by vortex noise. The quadrupole acoustic source is a broadband component of total acoustic radiation.

The total acoustic radiation of an axial compressor stage is characterized by the dipole, quadrupole and monopole acoustic sources. The dipole acoustic source dominates in general acoustic radiation for a subsonic impeller. Acoustic sources of the axial compressor stage were calculated using the Ffowcs Williams − Hawkings equation. It represents solution for inhomogeneous wave equation with solid boundaries.

$$\rho = \frac{1}{4\pi c_0^2} \frac{\partial^2}{\partial x_i \partial x_j} \int_V \frac{T_{ij}dV}{r} + \frac{1}{4\pi c_0^2} \frac{\partial}{\partial x_i} \int_S \frac{\rho u_i u_n + p_i}{r}dS + $$
$$+ \frac{1}{4\pi c_0^2} \int_S \frac{\partial}{\partial t}(\rho u_n)\frac{dS}{r}, \qquad (1)$$

where ρ is density; c_0 is sound speed in a stationary environment; x_i, x_j are coordinates of observation points; V is gas volume; T_{ij} is Lighthill turbulence stress tensor; r is radial coordinate of the point of observation in a cylindrical coordinate system; S is streamlined surface; u_i is flow velocity in the direction x_i; u_n is normal velocity near the solid surface; p_i expresses the flow force acts the boundary in the direction x_i on the unit of the surface S.

The components of expression (1) can be interpreted as follows. The first item represents the sound radiation by quadrupoles, which are distributed in a turbulent flow at some distance from the boundaries. The second item is radiation from dipoles distributed on the surface S.

These sources are determined by pressure pulsation and viscous stresses. The value $\rho u_i u_n$ characterizes the rate of momentum change, it is zero in case of a rigid plane or a plane oscillating in its own boundary, while for moving boundaries it characterizes the transfer of momentum plane, which is nearby. Parameter p_i represents the force flow impact on the boundaries. The third item is the monopole sound sources, which are located on the surface S. It characterizes the fact that due to boundary surface movement, the liquid is pushed out of the area which it has occupied. In the absence of solid boundaries, second and third integrals disappear and equation becomes the well-known Lighthill solution.

Based on the fact that the monopole acoustic source does not make a significant contribution to general acoustic radiation of the axial compressor, the dipole and quadrupole acoustic sources were calculated in the work.

Fig. 2 shows the acoustic characteristics of base and modified stages of the axial compressor in the form of dependence of the quadrupole source acoustic pressure on gas-dynamic functions $q(\lambda_c)$ for four rotation speeds.

The value of quadrupole source acoustic pressure of an axial compressor stage with a single row and equivalent tandem row of impeller increases with the increasing rotational speed and inlet flow velocity. Replacement of the single row with an equivalent tandem row of an impeller led to insignificant increasing quadrupole source acoustic pressure at small values: $q(\lambda_c)$ at rotational speed n_4 $q(\lambda_c) = 0.4...0.45$, at rotational speed n_3 $q(\lambda_c) = 0.4...0.52$, at rotational speed n_2 $q(\lambda_c) = 0.48...0.6$, at rotational speed n_1 $q(\lambda_c) = 0.6...0.67$. However, in modes which correspond to the right branches of pressure lines, the value of quadrupole source acoustic pressure decreases. For rotational speed n_4 at $q(\lambda_c) = 0.5...0.64$ acoustic pressure is reduced by 500...1450 Pa. For rotational speed n_3 at $q(\lambda_c) = 0.55...0.675$ acoustic pressure is reduced by 500...1900 Pa. For rotational speed n_2 at $q(\lambda_c) = 0.64...0.85$ acoustic pressure is reduced by 700...3300 Pa. For rotational speed n_1 at $q(\lambda_c) = 0.7...0.85$ acoustic pressure is reduced by
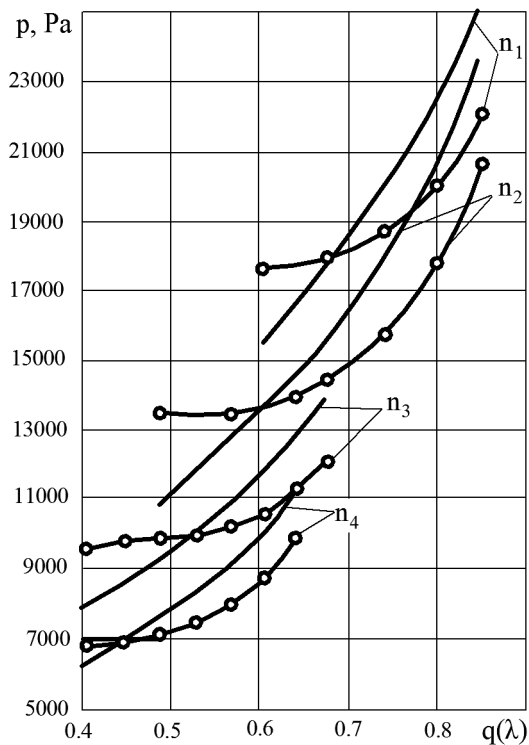
Fig. 2. Acoustic characteristics of an axial compressor stage with a quadrupole source

700...3400 Pa. The quadrupole source acoustic pressure decreases from 22500 to 20 000 Pa at calculated design operation mode.

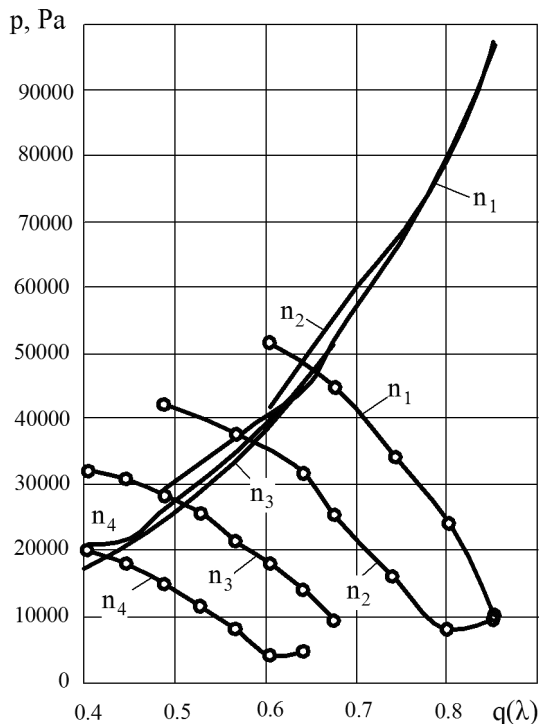Fig. 3 shows the acoustic characteristics of base and modified stages of an axial compressor in the form of dependence of the dipole source acoustic pressure from the gas-dynamic functions $q(\lambda_c)$ for four rotational speeds.

The value of dipole source acoustic pressure of an axial compressor stage with a single row of impeller increases with the increasing value of the gas-dynamic functions and rotational speed. Growth of acoustic pressure in the axial compressor stage with a tandem row of an impeller is observed at rotational speed n_4 at $q(\lambda_c) = 0.4...0.42$, at rotational speed n_3 at $q(\lambda_c) = 0.4...0.5$, at rotational speed n_2 at $q(\lambda_c) = 0.48...0.57$, at rotational speed n_1 at $q(\lambda_c) = 0.6...0.64$. However, in the operation modes that correspond to the right branch of pressure lines, dipole source acoustic pressure is significantly reduced. For frequency n_4 at $q(\lambda_c) = 0.6$ the acoustic pressure decreases from 40 000 to 4100 Pa. For frequency n_3 at $q(\lambda_c) = 0.675$ the acoustic pressure decreases from 51 000 to 9150 Pa. For frequency n_2 at $q(\lambda_c) = 0.8$ the acoustic pressure decreases from 80 000 to 8200 Pa. For frequency n_1 at $q(\lambda_c) = 0.85$ the acoustic pressure decreases from 97 200 to 10 300 Pa. In the calculated design operation mode the dipole source acoustic pressure decreases from 81 000 to 24 000 Pa. This is because the value of dipole source acoustic pressure depends on the aerodynamic loading of the stage. Aerodynamic loading in a stage with an equivalent tandem row increases and this contributes to a significant reduction of dipole source acoustic pressure.

So the use of a stage with a tandem row of an impeller makes it possible to improve the acoustic characteristics of an axial compressor stage in near field. For an axial compressor stage with a single row of an impeller dominant noise source is the dipole. The use of an equivalent tandem row of an impeller leads to significant reduction of dipole source acoustic pressure, and , in certain modes, the dominant source is quadrupole. Therefore, for further improving of acoustic efficiency of the axial compressor stage it is necessary to affect the quadrupole acoustic source.

The method of acoustic emission determination on the basis of application of the Ffowcs Williams — Hawkings equation does not take into account the interference of dipole and quadrupole sources. Therefore, the task for future research is to create calculation methods of monopole, dipole and quadrupole sound sources interference of the axial compressor stage.

Conclusions.

1. The results of the research showed that replacement of single row of impeller on the equivalent tandem row of an impeller in an axial compressor stage leads to the growth of pressure ratio from 1−15 %. In the design operation mode, the pressure ratio increases by 8.5 %.

2. For the first time, the results of comprehensive research on aerodynamic and acoustic characteristics of an axial compressor stage are obtained, which allow estimating the efficiency of application of a tandem row of an impeller of a subsonic low-pressure axial compressor stage.

Comparative estimation of acoustic characteristics of an axial compressor stage with a single row and equiv-

Fig. 3. Acoustic characteristics of an axial compressor stage with a dipole source

alent tandem row of an impeller in different operation modes showed that a stage with a tandem row of an impeller has greater acoustic efficiency than a compressor stage with a single row of an impeller. Obtained results indicate significant decreasing dipole source acoustic pressure. In the design operation mode, the dipole source acoustic pressure decreases by more than 70 %, while quadrupole source acoustic pressure decreases by more than 10 %.

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Аероакустичні характеристики ступеня осьового компресора із дворядним робочим колесом

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Мета. Порівняльна оцінка аероакустичних характеристик ступенів осьового компресора з однорядним і еквівалентним дворядним колесом.

Методика. Дослідження виконано за допомогою чисельного експерименту. Параметри течії у ступені осьового компресора розраховувались шляхом розв'язання нестаціонарної системи рівнянь Нав'є – Стокса. Рівняння замикались моделлю турбулентної в'язкості SST. Ступінь компресора складалася зі вхідного напрямного апарату, робочого колеса й направляючого апарату. Робоче колесо модифікованого ступеня осьового компресора виконано як еквівалентний дворядний лопатковий вінець. Акустичні джерела ступеня осьового компресора розраховувались за допомогою рівняння Фокс Вільямса – Хоукінгса.

Результати. Отримані результати порівняльної оцінки аероакустичних характеристик ступенів осьового компресора з однорядним і еквівалентним дворядним робочим колесом. Застосування дворядного лопаткового вінця замість еквівалентного однорядного в робочому колесі ступеня осьового компресора може забезпечити збільшення ступеня підвищення тиску на 1–15 %. На розрахунковому режимі ступінь підвищення тиску збільшується на 8.5 %. Ступінь із дворядним робочим колесом має більшу акустичну ефективність, ніж ступінь компресора з однорядним робочим колесом. Отримані результати свідчать про суттєве зменшення акустичного тиску дипольного джерела. На розрахунковому режимі роботи акустичний тиск дипольного джерела зменшується більше, ніж на 70 %, акустичний тиск квадрупольного джерела зменшується більше, ніж на 10 %.

Наукова новизна. Уперше отримані результати комплексного дослідження аеродинамічних і акустичних характеристик ступеня осьового компресора, що дозволяють оцінити ефективність застосування дворядних лопаткових вінців у робочому колесі дозвукового низьконапірного ступеня осьового компресора.

Практична значимість. Отримані рекомендації можуть бути використані при проектуванні лопаткових машин з низьким рівнем акустичної емісії.

Ключові слова: *ступінь осьового компресора, дворядне робоче колесо, дипольне акустичне джерело, квадрупольне акустичне джерело, ступінь підвищення тиску*

Аэроакустические характеристики ступени осевого компрессора с двухрядным рабочим колесом

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Цель. Сравнительная оценка аэроакустических характеристик ступеней осевого компрессора с однорядным и эквивалентным двухрядным колесом.

Методика. Исследование выполнено с помощью численного эксперимента. Параметры течения в ступени осевого компрессора рассчитывались путем решения нестационарной системы уравнений Навье – Стокса. Уравнения замыкались моделью турбулентной вязкости SST. Ступень компрессора состояла из входного направляющего аппарата, рабочего колеса и направляющего аппарата. Рабочее колесо модифицированной ступени осевого компрессора выполнено как эквивалентный двухрядный лопаточный венец. Акустические источники ступени осевого компрессора рассчитывались с помощью уравнения Фокс Вильямса – Хоукингса.

Результаты. Получены результаты сравнительной оценки аэроакустических характеристик ступеней осевого компрессора с однорядным и эквивалентным двухрядным рабочим колесом. Использование двухрядного лопаточного венца вместо эквивалентного однорядного в рабочем колесе ступени осевого компрессора может обеспечить увеличение ступени повышения давления на 1−15 %. На расчетном режиме степень повышения давления увеличивается на 8.5 %. Ступень с двухрядным рабочим колесом имеет большую акустическую эффективность, чем ступень компрессора с однорядным рабочим колесом. Полученные результаты свидетельствуют о существенном уменьшении акустического давления дипольного источника. На расчетном режиме работы акустическое давление дипольного источника уменьшается больше, чем на 70 %, акустическое давление квадрупольного источника уменьшается больше, чем на 10 %.

Научная новизна. Впервые получены результаты комплексного исследования аэродинамических и акустических характеристик ступени осевого компрессора, которые позволяют оценить эффективность применения двухрядных лопаточных венцов в рабочем колесе дозвуковой низконапорной ступени осевого компрессора.

Практичная значимость. Полученные рекомендации могут быть использованы при проектировании лопаточных машин с низким уровнем акустической эмиссии.

Ключевые слова: *ступень осевого компрессора, двухрядное рабочее колесо, дипольный акустический источник, квадрупольный акустический источник, степень повышения давления*