MODERN AVIATION AND SPACE TECHNOLOGIES

UDC 629.735.03 (045)
DOI: 10.18372/2306-1472.74.12295

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INVESTIGATION OF INTERFERENCE INFLUENCE OF BLADE ROWS ON LOSSES IN AXIAL COMPRESSOR STAGE

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Abstract

Purpose: *the purpose of this work is to investigate the total pressure losses in the input guide vanes, in the rotor wheel and the guide vanes of the axial compressor stage taking into account their gas-dynamic interference.* ***Methods:*** *the study was carried out by numerical simulation of three-dimensional flow in the stage of an axial compressor. An unstructured adaptive computational grid is constructed. The gas dynamic calculation of the flow in the stage of the axial compressor is performed using the Navier-Stokes system of equations, which was closed by the SST turbulent model.* ***Results:*** *a series of gas-dynamic flow calculations was performed at different values of the axial velocity at the inlet to the compressor stage. The circuit velocity at the peripheral radius in the calculated mode was u= 238.64 m/s. The coefficient of velocity at the entrance to the stage varied in the range λ_c =0.35…0.7. Based on the calculation results, the dependences of the total pressure loss coefficients from the input velocity coefficient for various elements of the stage of the axial compressor were constructed. Analysis of the results of the study shows that the greatest contribution to the overall balance of total pressure losses is made by losses in the guide vanes.* ***Discussion:*** *losses in the guide vanes increase due to the gas-dynamic interference of the rotor wheel and the guide vanes. The mutual influence of the blade rows of the rotor wheel and the guide vanes leads to a significant transformation of the velocities and pressures in the interblade channels. As a consequence, there is a redistribution of losses caused by circumferential and radial flow irregularity, end-flow and centrifugal forces. It can be expected that a reduction in the level of losses in the step of the axial compressor can be achieved by influencing the boundary layer in the end clearance of the rotor wheel.*

Keywords: blade row; boundary layer; flow modeling; guide vanes; secondary losses; total pressure losses.

1. Introduction

The study of total pressure losses in the blade rows is one of the most important phases in the study of the working process of an axial turbomachine. The total pressure losses in the blade rows can be divided into the following components:

– profile losses associated with the formation of a boundary layer on the blade surface;

– end losses caused by friction of gas on the end surfaces of the casing and hub at the section corresponding to the axial length of the blade row;

– secondary losses associated with the formation of secondary flow (areas of secondary vortex motion) in angles formed by the surfaces of the blades and the end surfaces of the casing and hub. The formation of secondary flow is caused by the interaction of the boundary layers of the blades' end cross-sections and the limiting end surfaces;

– gap losses caused by gas flowing through radial gaps between the rotating rotor wheel and the fixed surface of the casing.

This division of losses in the blade row is conditional. Under real gas flow conditions, individual components interact with each other, which affects the structure of the flow in the blade row, and on the overall balance of losses [1].

2. Analysis of the researches and publications

A large number of experimental and theoretical studies have been devoted to the investigation of the flow in a compressor. Today, special interest is evoked by the study of flow in turbomachines by numerical simulation methods. These methods make it possible to investigate flows with a high degree of accuracy and in a short period of time. In work [2], the methods of numerical experiment are critically analyzed. The review of models of turbulent viscosity, CFD codes is presented. Advantages, disadvantages of various CFD codes for turbomachines are described.

The characteristics of compressors were calculated in [3, 4] using numerical flow simulations. In work [3] the four-stage compressor is investigated. Calculated losses of the total pressure in the compressor and the degree of pressure increase at different operating modes. In [4], the characteristic of an axial transonic compressor is obtained by the streamline curvature method. The total losses in the compressor are calculated. Total losses in the axial compressor at different operating modes were investigated in [5]. In more detail, the authors consider secondary losses. In work [6] simulates the flow in a single-stage compressor, the stator-rotor and the rotor-stator interaction are considered. The losses caused by this interaction are calculated. In work [7], the influence of the stator hub configuration on the total pressure losses in an axial compressor was considered. The change in losses is investigated depending on the geometric parameters of the stage. The losses in the stator elements of the compressor are considered in [8, 9]. The results of [8] show that an increase in losses in the guide vanes entails an increase in losses in the rotor wheel. In [9] a universal algebraic model was proposed for calculating losses in the inlet guide vanes. The model was tested on several inlet guide vanes with different geometric parameters and different profiles. The influence of radial clearance parameters on the level of losses in the stage of the axial compressor was considered in works [10, 11]. In [10], losses in the end clearance of a single-stage compressor are analyzed for different parameters at the inlet duct. The authors of [11] proposed a technique for calculating the total pressure loss in the end clearance of the rotor wheel of an axial compressor.

3. Aim of the paper

An analysis of the result of works [3–11] has shown that studies of the mutual influence of the rotor and stator elements in the compressor is an actual problem. At the same time, the tasks associated with detailed investigation of the losses in individual blade rows of the compressor stage and their interaction remains unresolved. The purpose of this work is to investigate the total pressure losses in the inlet guide vanes, in the rotor wheel and the guide vanes of the axial compressor stage taking into account their gas-dynamic interference.

4. Method

The study was carried out by numerical simulation of three-dimensional flow in the stage of an axial compressor. An unstructured adaptive computational grid is constructed. The gas dynamic calculation of the flow in the stage of the axial compressor is performed using the Navier-Stokes system of equations, which was closed by the SST turbulent model.

5. Solution of the assigned task

The object of the study was the stage of the axial compressor, which consists of an inlet guide vanes (IGV), a rotor wheel (RW), and a guide vanes (GV). The blade row of the IGV consists of 30 blades, the RW and GV each have 24 blades. The geometric parameters of the step are given in [12]. Fig. 1 shows a 3D model of the stage of the axial compressor under study.

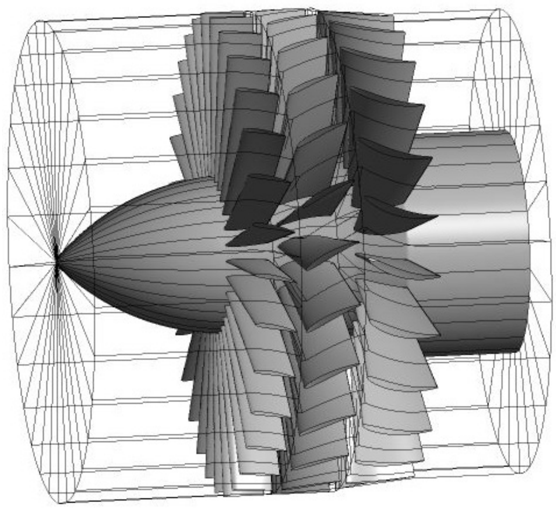

Fig. 1. Solid model of axial compressor stage

A series of gas-dynamic flow calculations was performed at different values of the axial velocity at the inlet to the compressor stage. The circuit velocity at the peripheral radius in the calculated mode was u= 238.64 m/s. The coefficient of velocity at the entrance to the stage varied in the range λ_c=0.35…0.7.

The coefficient of velocity was calculated by the formula

$$\lambda_c = \frac{c_0}{a}, \quad (1)$$

where c_0 – the axial flow velocity at the entrance to the stage, a – velocity of sound, k – the adiabatic index.

Based on the results of numerical simulation of the flow in the compressor stage, the values of the total pressure loss coefficient in individual blade rows were calculated.

The total pressure loss coefficient in the inlet guide vanes was calculated by the formula:

$$\xi_{igv} = \frac{p_1^* - p_2^*}{\frac{\rho_o c_{1m}^2}{2}}, \quad (2)$$

where p_1^* and p_2^* – average values of the total pressure at the inlet and outlet of the inlet guide vanes, c_{1m} – average absolute flow velocity in the inlet guide vanes.

The total pressure loss in the rotor wheel was calculated as:

$$\xi_r = \frac{p_3'^* - p_4'^*}{\frac{\rho_o w_m^2}{2}}, \quad (3)$$

where $p_3'^*$ and $p_4'^*$ – mean values of the total pressure in the relative movement at the inlet and outlet of the rotor wheel, w_m – average relative flow velocity in the rotor wheel.

The coefficient of total pressure loss in the guide vanes was calculated by formula:

$$\xi_{gv} = \frac{p_5^* - p_6^*}{\frac{\rho_o c_{3m}^2}{2}}, \quad (4)$$

where p_5^* and p_6^* – mean values of the total pressure at the inlet and outlet of the guide vanes, c_{3m} – average absolute flow velocity in the guide vanes.

Based on the calculation results, the dependences of the total pressure loss coefficients from the input velocity coefficient for various elements of the stage of the axial compressor are constructed (Fig. 2).

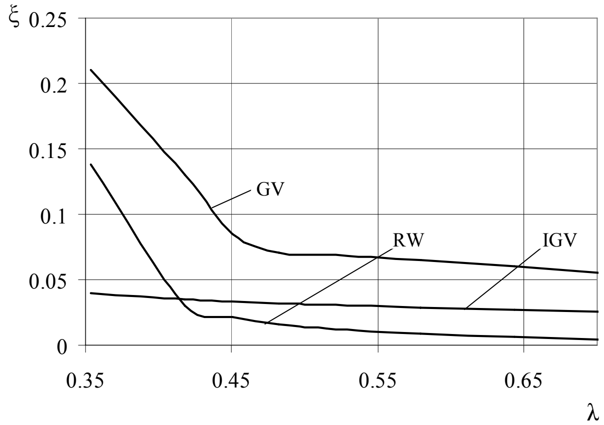

Fig. 2. Dependence of the total pressure loss coefficients in various elements of the axial compressor stage from the input velocity coefficient

Analysis of the results of the study (Fig. 2) shows that the greatest contribution to the overall balance of total pressure losses is made by losses in the guide vanes.

The relative level of total pressure loss in GV varies in the range ξ_{gv}=0.21…0.061 when the values of the velocity coefficient changes from λ_c=0.35 to λ_c=0.7.

The relative level of total pressure loss in the RW varies from ξ_r=0.14 (when λ_c=0.35) to ξ_r=0.005 (when λ_c=0.7).

For IGV, the relative level of total pressure loss in the range ξ_{igv}=0.035…0.05 when the velocity coefficient changes in the range λ_c=0.35…0.7.

In the range of flow rates at the inlet of the corresponding λ_c=0.35…0.4, the greatest contribution to the total level of relative losses is made by RW and GV (to 87%).

At values of velocity coefficient λ_c=0.45…0.7 the main level of relative losses of the total pressure is the losses in the guide vanes (≈65%).

The velocity fields in IGV, RW and GV of the investigated stage at λ_c=0.48 are shown in Fig. 3. Their analysis allows a qualitative assessment of the effect of various factors on the level of losses in the elements of the stage of the axial compressor.

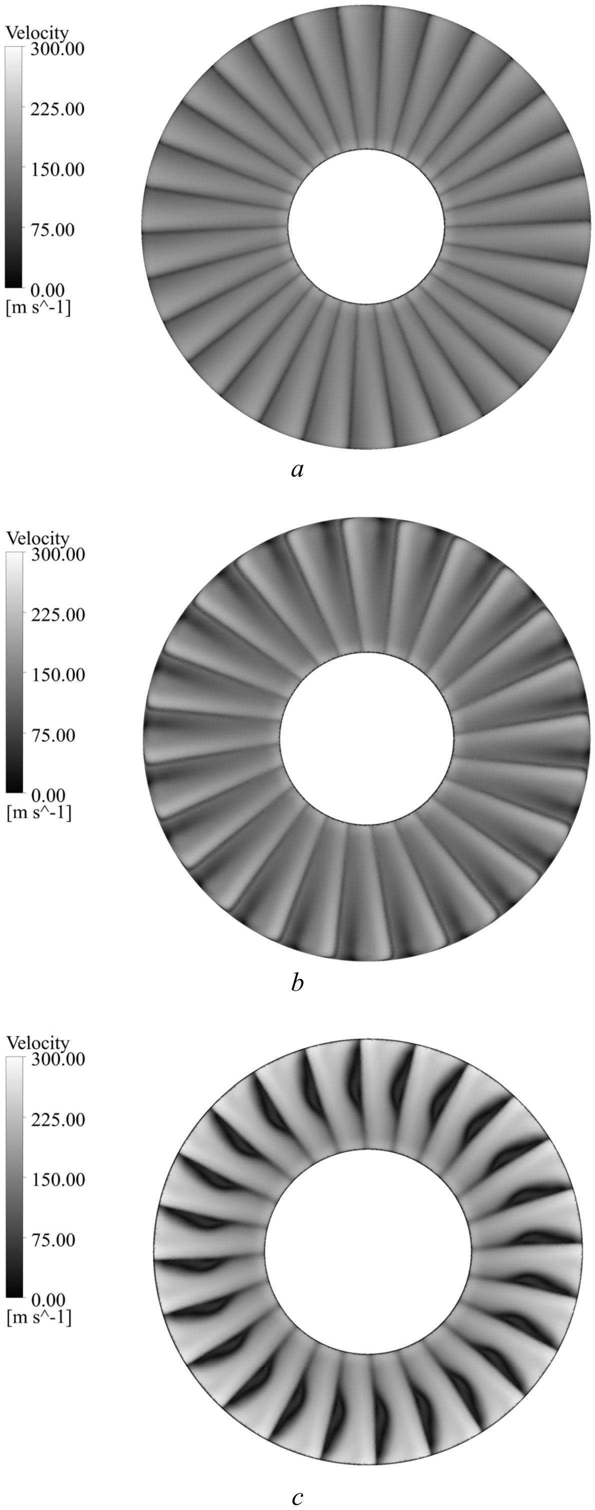

Fig. 3 Velocity field in the compressor blade rows
a – inlet guide vanes, *b* – rotor wheel, *c* – guide vanes

The presence of secondary flows in the boundary layer leads to a significant displacement and accumulation of the boundary layer in the flowing part. The presence of end clearances, circumferential and radial velocity gradients, the action of centrifugal forces and the relative motion between the ends of the blades and the casing lead to an increase in losses associated with the increase in the boundary layer.

In IGV (Fig. 3, a), the velocity gradient across the channel in the flow core causes secondary flows in the circumferential direction. In addition, the radial velocity gradient leads to radial overflow of the boundary layer. The boundary layer occupies an area encompassing the guide vanes from the casing to the sleeve. In this case, there is a radial overflow of the boundary layer. The outer part of the boundary layer twists, which leads to a vortex wrapping.

In the rotor wheel (Fig. 3, b), the presence of an end clearance contributes to a transverse velocity gradient. With the interaction of a thinner boundary layer on the casing and thicker on the blade surface, vortices are formed. Vortex wrapping in the peripheral zone helps to reduce radial flow. I.e. in the rotor wheel, the effect of centrifugal forces affects the decrease in the radial velocity gradient. However, the accumulation of a boundary layer with a low energy level near the surface of the blade causes perturbation of the flow and loss, leading to a deflection of the flow angles and a change in the angles of attack in the subsequent blade row.

The increase in the boundary layer on the blades of the rotor wheel leads to an inevitable increase in losses in the guide vanes (Fig. 3, c). In the guide vanes, the effect of a radial velocity gradient is added, which leads to an increase in the zone of reduced velocities behind the blades.

To summarize, it can be expected that a reduction in the level of losses in the step of the axial compressor can be achieved by influencing the boundary layer in the end clearance of the rotor wheel.

6. Conclusions

The work represents the results of numerical simulation of the flow in the stage of an axial-flow compressor at a velocity coefficient λ_c =0.35…0.7. It is shown that the losses in the guide vanes increase due to the gas-dynamic interference of the rotor wheel and the guide vanes.

The mutual influence of the blade rows of the rotor wheel and the guide vanes leads to a significant transformation of the velocities and pressures in the interblade channels. As a consequence, there is a redistribution of losses caused by circumferential and radial flow irregularity, end-flow and centrifugal forces.

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Received 15 November 2017

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Дослідження впливу інтерференції лопаткових вінців на втрати в ступені осьового компресора

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Мета: дослідження втрат повного тиску у вхідному напрямному апараті, в робочому колесі й напрямному апараті ступеня осьового компресора з урахуванням їх газодинамічної інтерференції. **Метод дослідження:** дослідження виконано методом чисельного моделювання тривимірної течії в ступені осьового компресора. Побудовано неструктуровану адаптивну розрахункову сітку. Газодинамічний розрахунок течії в ступені осьового компресора виконано з використанням системи рівнянь Нав'є – Стокса, яка замикалася моделлю турбулентної в'язкості SST. **Результати:** було проведено серію газодинамічних розрахунків течії при різних значеннях осьової швидкості на вході в ступінь компресора. Колова швидкість на периферійному радіусі на розрахунковому режимі складала u=238.64 м/с. Коефіцієнт швидкості на вході в ступінь змінювався в діапазоні λ_c=0.35…0.7. За результатами розрахунку побудовано залежності коефіцієнтів втрат повного тиску від коефіцієнта швидкості на вході для вхідного напрямного апарату, робочого колеса і напрямного апарату. Аналіз результатів дослідження показує, що найбільший внесок у загальний баланс втрат повного тиску вносять втрати в напрямному апараті. **Обговорення:** унаслідок газодинамічної інтерференції робочого колеса і напрямного апарату відбувається збільшення втрат в напрямному апараті. Взаємний вплив лопаткових вінців робочого колеса і напрямного апарату призводить до суттєвої трансформації швидкостей і тисків в міжлопаткових каналах. Як наслідок, має місце перерозподілення втрат, обумовлених коловою і радіальною нерівномірністю потоку, кінцевими перетіканнями і дією відцентрових сил. Можна очікувати, що зменшення рівня втрат в ступені осьового компресора можна забезпечити шляхом впливу на пограничний шар в кінцевому зазорі робочого колеса.

Ключові слова: лопатковий вінець; пограничний шар; моделювання течії; напрямний апарат; вторинні втрати; втрати повного тиску.

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Исследование влияния интерференции лопаточных венцов на потери в ступени осевого компрессора

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Цель: исследование потерь полного давления во входном направляющем аппарате, в рабочем колесе и направляющем аппарате с учетом их газодинамической интерференции. **Метод исследования:** исследование выполнено методом численного моделирования трехмерного течения в ступени осевого компрессора. Построена неструктурированная адаптивная расчетная сетка. Газодинамический расчет течения в ступени осевого компресора выполнен с использованием системы уравнений Навье – Стокса, которая замыкалась моделью турбулентности SST. **Результаты:** была проведена серия газодинамических расчетов течения при различных значениях осевой скорости на входе в ступень компрессора. Окружная скорость на переферийном радиусе на расчетном режиме составила u= 238.64 м/с. Коэффициент скорости на входе в ступень изменялся в диапазоне λ_c=0.35…0.7. По результатам рассчета построены зависимости коэффициентов потерь полного давления от коэффициента скорости на входе для входного направляющего аппарата, рабочего колеса и направляющего аппарата. Анализ результатов исследования показывает, что наибольший вклад в общий баланс потерь полного давления вносят потери в направляющем аппарате. **Обсуждение:** вследствие газодинамической интерференции рабочего колеса и направляющего апарата происходит увеличение потерь в направляющем аппарате. Взаимное влияние лопаточных венцов рабочего колеса и направляющего апарата приводит к существенной трансформации скоростей и давлений в межлопаточных каналах. Как следствие, имеет место перераспределение потерь, обусловленных окружной и радиальной неравномерностью потока, концевыми перетеканиями и действием центробежных сил. Можно ожидать, что уменьшение уровня потерь в ступени осевого компрессора можно обеспечить путем влияния на пограничный слой в концевом зазоре рабочего колеса.

Ключевые слова: вторичные потери; лопаточный венец; моделирование течения; направляющий аппарат; пограничный слой; потери полного давления.

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