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Investigation of the Effect of the Gas Turbine Compressor Supports on Gas Flow Circumferential Nonuniformity

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Abstract: This study discusses the possibility to reduce the non-uniformity of the circumferential gas flow forcing on the blades of the rotor wheel of a gas turbine engine stage. Circumferential non-uniformity is caused by the availability of the middle support behind the compressor stage. This results in higher resonant stresses in the rotor blades which are made with anti-vibration blade shrouds. In order to exclude the antivibration blade shrouds it is necessary to reduce circumferential non-uniformity of gas flow that would lead to falling of the amplitude of the exciting harmonics. In the Software Package Numeca Fine Turbo there was developed a full-circle model of the compressor flow passage with middle support which allows to determine the distribution of the gas load on the working blade row and to investigate the effect of the supports on the amplitudes of exciting harmonics. Variation in the width and number of supports has led to a 2 fold reduction in the most dangerous harmonics.

Key words: Gas turbine engine, compressor, blade, support, bearing, ANSYS, numeca, forced oscillations

INTRODUCTION

Rotor blades of the fifth stage of the intermediate pressure compressor of engine NK36-ST (Fig. 1) in a regular configuration are made with anti-vibration blade shrouds. It is related to the increased level of fluctuating stresses in blades because of their excitation by circumferential nonuniformity of the airflow originating because of a flow-over of middle support bearing racks (Kuz'michev and Morozov, 1991). The middle support is arranged between intermediate and high pressure compressors. Bearing racks have different width and are arranged nonuniformly in a circumferential direction. A regular support has seven bearing racks. Application of blade shrouds on blades of the fifth stage of the compressor degrades efficiency of the engine, its adaptability to manufacture and is an obstacle to application of the gas generator on the long life products because of wear of the blade shroud contact surfaces (Shklovets *et al*., 2012). Objective of the study is elimination of a airfoil shroud platform from a blade of the fifth stage rotor wheel. One of provisions on lowering excitation from the support may be change in the number and width of its bearing racks (Bochkarev *et al*., 1993). The CAD Model of the middle support is presented in Fig. 2.

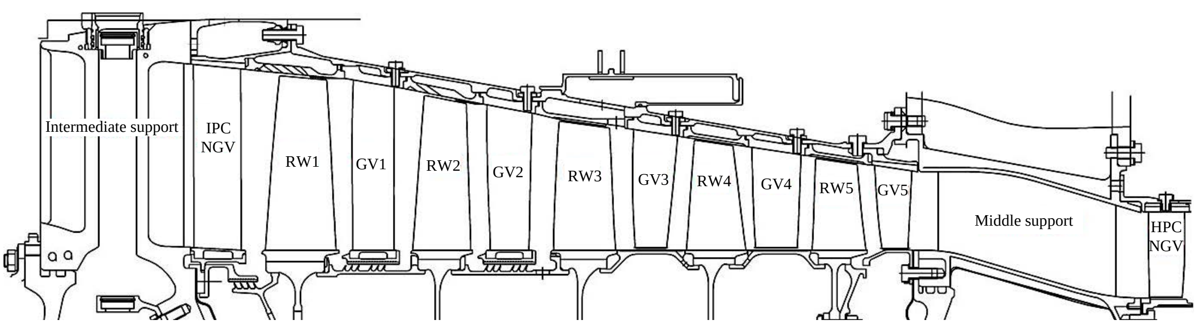

Fig. 1: The scheme of investigated compressor blading

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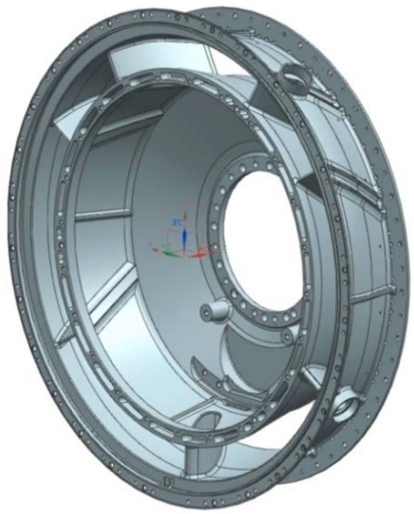

Fig. 2: Middle support casing

CFD-COMPUTATIONS OF THE COMPRESSOR BLADING SECTION WITH A REGULAR SUPPORT

In order to detect a level of excitation from a gas flow a gas-dynamic computation of the Intermediate Pressure Compressor (IPC) has been performed in Software Package Numeca Fine Turbo for the basic operating mode gas-compressor plant. A full-circle model of IPC for definition of boundary conditions at an inlet into the 4 Nozzle Guide Vanes (NGV) has been created. The geometry of the model in a greater part matches to drawings of the compressor, however, the following assumptions have been used:

- The middle support was simulated by a one bearing rack of a medium width. Upon that the number of bearing racks was equal to 7
- As the given study is directed on investigation of a blade of the fifth IPC Rotor Wheel (5RW) without the anti-vibration blade shroud, the specified blade was simulated without it

When building the computational domain of the full-circle model radial clearances, numbers of blades and fillets of blade rows were considered. The average number of elements per a blade row in the full-circle model has made 500000 elements. The total size of the grid is equal to 7600000 elements (Fig. 3).

The total pressure, the full temperature and the flow direction (30 degrees in relation to the axis) at the inlet were set in the capacity of boundary conditions. A static pressure on the hub was set at the outlet from the Nozzle Guide Vane of the High-Pressure Compressor (HPC NGV)

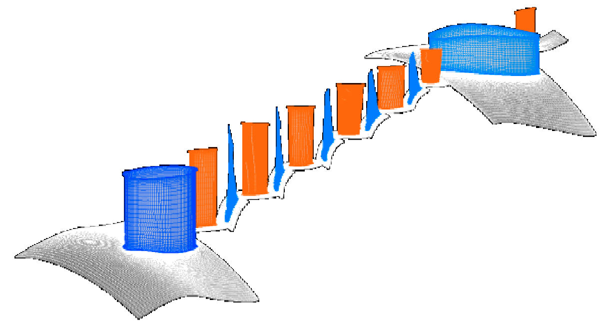

Fig. 3: IPC full-circle model with the support

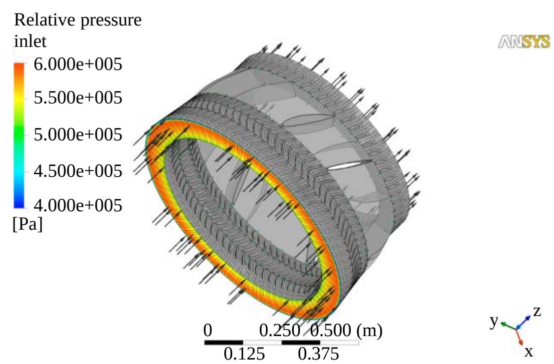

Fig. 4: Full-circle model

(Kuz'michev, 2012). As a result of computation of the full-circle model radial profiles of total pressure, full temperature and a flow angle in the section upstream 4 IPC NGV have been defined.

The created model for computation of circumferential nonuniformity comprised the blade rows 4 GV, 5 RW, 5 GV IPC, bearing racks of a middle support and HPC NGV (Fig. 4). In the computational model blade rows were featured not by one blade passage but by the full round.

The grid for the computational model was formed in the software complex Numeca AutoGrid5. The number of elements for one blade passage (for 4 GV, 5 RW, 6 GV and HPC NGV) made the order of 75000. The number of grid elements in a middle support has made the order of 4019393. The total size of a grid of a computational model is equal approximately to 40,000,000 elements.

As a result of computation essential effect of supports on circumferential nonuniformity of a gas flow is determined. Figure 5 shows decrease in flow velocity near to the thickest bearing rack.

Figure 6 shows distribution of static pressure on the medium diameter of RW5 at a cruise rating mode. It is possible to identify seven peaks of pressure corresponding to bearing racks.

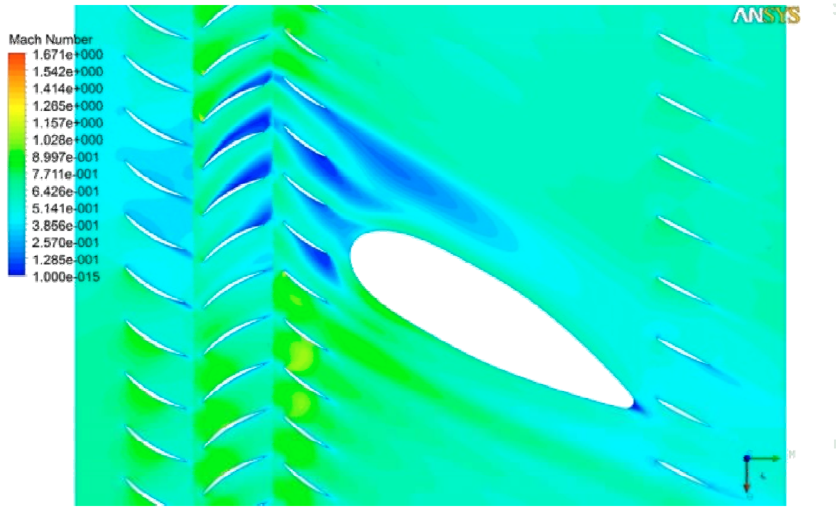

Fig. 5: Decrease in a Mach number near to the bearing rack

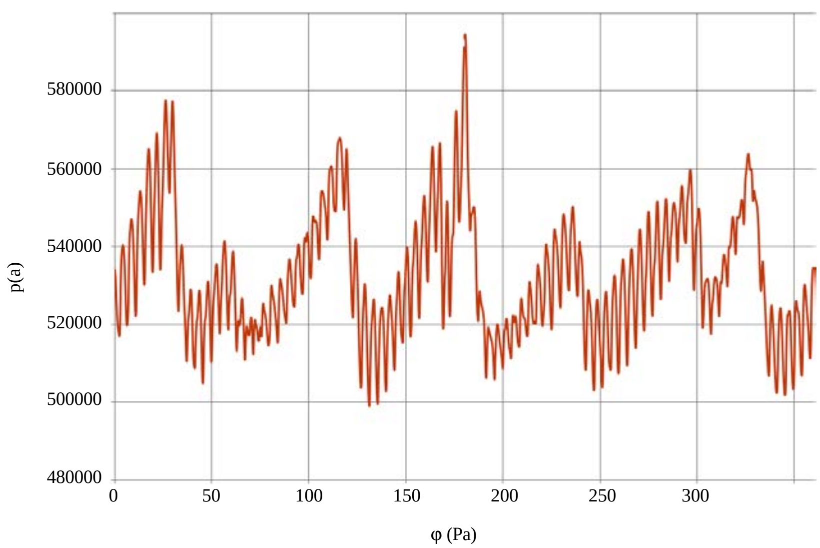

Fig. 6: Diagram of static pressure variation in the section downstream RW5 at the medium diameter

CFD-COMPUTATION OF THE COMPRESSOR BLADING SECTION WITH THE MODERNIZED SUPPORT

When selecting a new construction of a middle support following requirements were considered:

- The excitation level from bearing racks decreases with decreasing of bearing rack width
- Increasing of bearing racks number leads to lowering of resonance rotation frequencies (tune-out downwards)
- Width and number of bearing racks should ensure sufficient strength of the middle support (Bochkarev *et al*., 1993)
- Number of bearing racks is limited by the requirement on provision of a necessary flow section and admissible losses on a flow-over of the support
- Dimensions and positions of separate bearing racks should ensure requirements on arrangement of drive units of aggregates, oil drain, etc.

Figure 7 shows the finite-element model of the designed 13-bearing racks support. After completion of

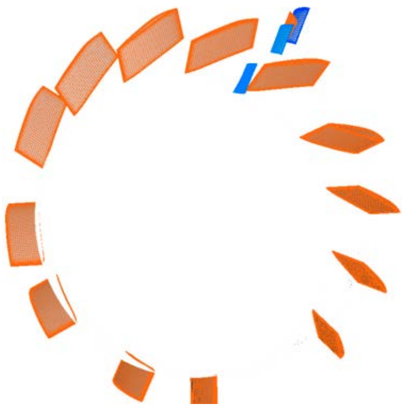

Fig. 7: Modernized bearing racks of the middle support

gas-dynamic computations diagrams of variation of the relative static pressure in the section downstream RW5 at the medium diameter in the operating mode for a version of the support with 13-bearing racks have been built. On the diagrams the base version is displayed by dark blue color and the modernized by red color. The separate peaks of pressure matching bearing racks and sections near to them are marked.

Figure 9 and 10 show comparison of static pressure and Mach numbers fields near to bearing racks of the first and third section for versions with seven-bearing rack and thirteen-bearing rack supports. The significant lowering of Mach number decreasing level and increasing of the static pressure near to bearing racks are observed.

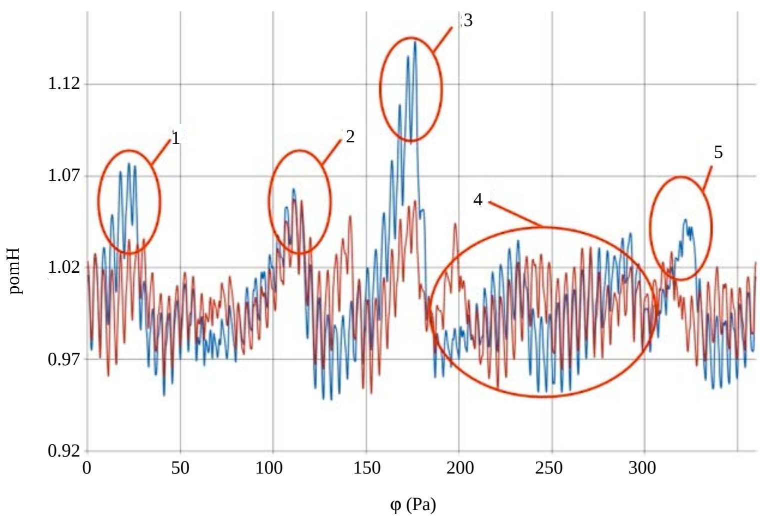

Fig. 8: Relative static pressure variation diagrams for two versions of support execution

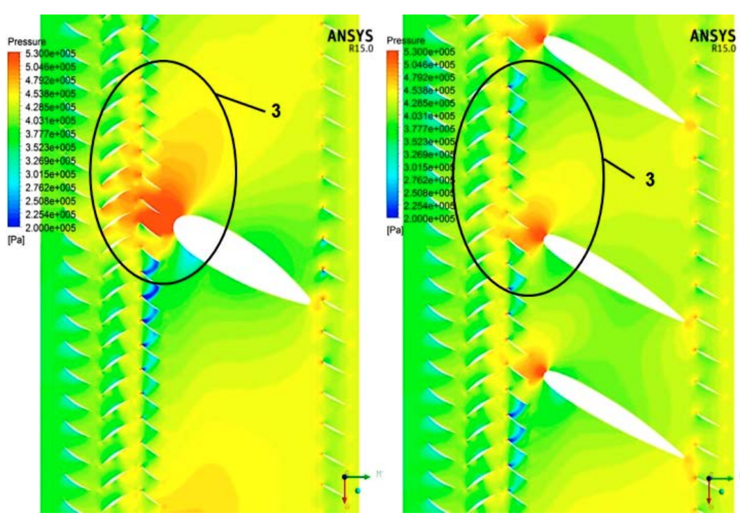

Fig. 9: Comparison of static pressure fields at a medium diameter of the stage for versions with 7 and 13-bearing rack support

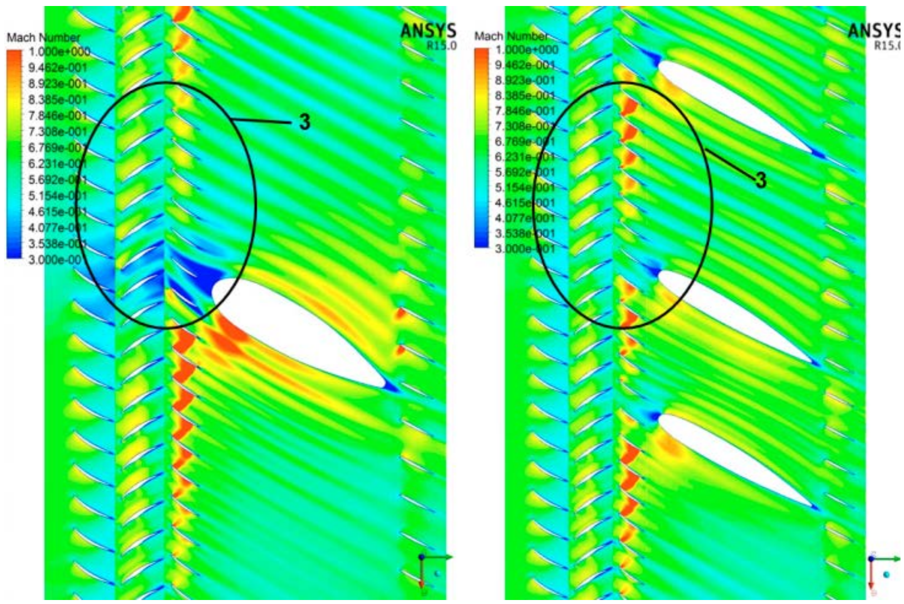

Fig. 10: Comparison of static pressure fields at a medium diameter of the stage for versions with 7 and 13-bearing rack support

ANALYSIS OF A DEGREE OF EXCITING LOADING REDUCTION

Gas-dynamic force Q_g is a periodic value, i.e., $Q_g(\alpha) = Q_g(\alpha+2\pi)$, therefore it can be expanded into a Fourier series (Maslov *et al*., 1976):

$$Q_g = \sum_{m_б}^{\infty} Q_m \cdot \cos(m_б\alpha - \gamma_m) = \sum_{m_б}^{\infty} Q_{gm} \qquad (1)$$

Where:

Q_m = Amplitude of a harmonics component
$m_б$ = The number of harmonics
α = The central angle
γ_m = Phase angle difference in a circumferential direction

The expansion Eq. 1 allows the gas loading having the complex circumferential distribution character to present in the form of the sum of harmonics components each of which represents a loading wave train which are being packed in full circle of the blading section (Kuz'michev and Morozov, 1991).

For a rotating rotor wheel any of components in expansion, Eq. 1 is the exciting harmonics representing a train of back progressive waves. The loading rotates with an angular velocity ω equal to angular velocity of rotor wheel rotation, thus, circumferential nonuniformity of a gas flow for the wheel is equivalent to effect of an infinite aggregate of exciting harmonics each of which represents a train $m_б$ of back progressive waves of a loading performing simple harmonic motions in a time with frequency $f_б = m_б n_c$ where, $n_c = \omega/2\pi$.

To define dangerous operating modes of a gas-compressor plant the Campbell diagram for an investigated blade 5RW (Fig. 11) has been built. As is clear from the diagram, minimum harmonics from which the resonance is possible is 9th harmonics.

The dynamic test result for rotor wheels of the compressor has shown that fracture of shroudless blades of the fifth stage happens at a resonance to 12th harmonics (Zhukov *et al*., 1985; Korzh *et al*., 1973a). To review qualitative convergence of CFD-computation with the experiment the level of amplitude of some exciting harmonics with which the resonance is possible was evaluated. The procedure of estimation of an exciting harmonics level in a gas flow has been developed. The gas loading (pressure) was imported from a blade surface in CFD-computation on the finite-element model of a blade row created in ANSYS Mechanical. Per se, the blade is similar to a probe intended to read a loading in a gas flow. Considering a significant number of blades of the fifth rotor wheel (84 pieces) it is possible to neglect a discrete disposition of blades in a circumferential direction.

Figure 12 presents CAE Model of the 5RW blade. Using possibilities of programming language APDL,

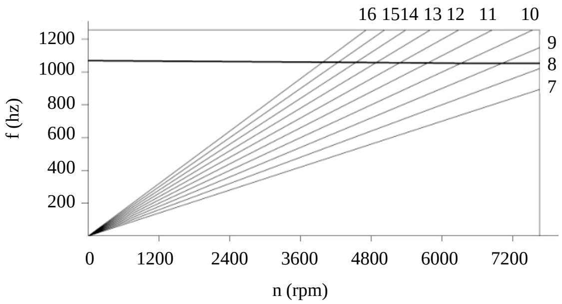

Fig. 11: Campbell diagram for a fifth stage blade

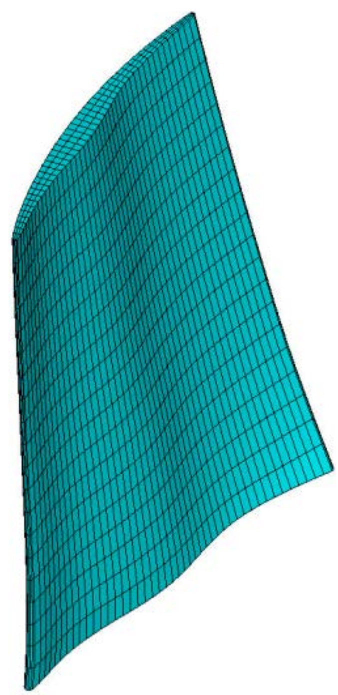

Fig. 12: FE Model of the 5RW blade

the gas loading was interpolated by nodes of the finite-element grid of a blade in ANSYS Mechanical.

Further, a gas loading was expanded in a Fourier series in similar nodes (the nodes laying on one circle). To define the summarized amplitude of exciting harmonics Q_m and integral criterion of an excitation level from a particular harmonic, amplitudes in all nodes of the blade were summarized (Korzh *et al*., 1973b; Shklovets *et al*., 2013). Figure 13 shows the summarized amplitudes of the first 14 harmonics in all nodes of the blade for versions of the compressor with 7 and 13-bearing rack supports. As it follows from the diagram, the amplitude of dangerous 12th harmonic is halved.

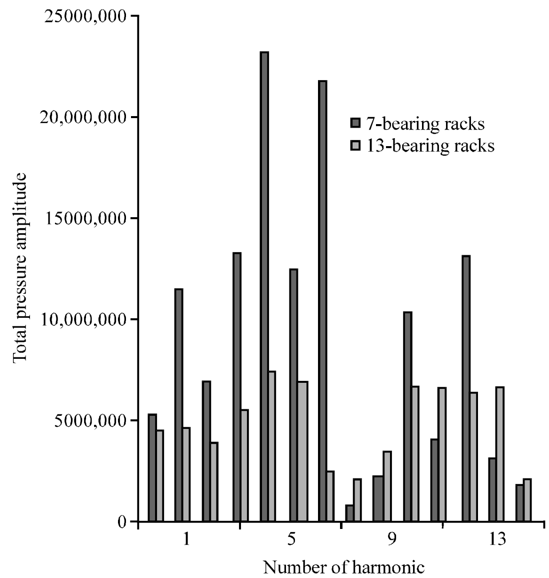

Fig. 13: Summarized amplitudes of exciting harmonics

CONCLUSION

Thus, the following results were achieved in the course of this research:

- Application of 13-bearing rack support allows to achieve more periodic circumferential distribution of a gas loading over the rotor wheel
- Halving of the summarized amplitude of the most dangerous exciting harmonic is gained the resonance with which led to fracture of blades of the examined rotor wheel
- From the technological point of view, the modification of the middle support is the most expedient way to reduce a level of an exciting loading, unlike a modification of the angle and the pitch of installation of guide vanes of the examined compressor stage
- Development of optimization CFD Model of the compressor support makes it possible to change a width, number and circumferential distribution of bearing racks as a result, it is possible to achieve more significant reduction of exciting harmonic amplitudes

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