Reduction of gas flow nonuniformity in gas turbine engines by means of gas-dynamic methods

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Abstract. Gas flow nonuniformity is one of the main sources of rotor blade vibrations in the gas turbine engines. Usually, the flow circumferential nonuniformity occurs near the annular frames, located in the flow channel of the engine. This leads to the increased dynamic stresses in blades and as a consequence to the blade damage. The goal of the research was to find an acceptable method of reducing the level of gas flow nonuniformity as the source of dynamic stresses in the rotor blades. Two different methods were investigated during this research. Thus, this study gives the ideas about methods of improving the flow structure in gas turbine engine. On the basis of existing conditions (under development or existing engine) it allows the selection of the most suitable method for reducing gas flow nonuniformity.

1. Introduction

Circumferential nonuniformity of gas flow is one of the main sources of turbomachinery rotor blades destruction [1-5]. There are several reasons for this phenomena: violation of axial symmetry at turbomachinery inlet; presence of guide and nozzle vanes in the flow passage of turbomachinery; impact of load-bearing elements (annular frame struts and bearings); operation of blow-off system and air bleeding system in compressor; operation of bringing fuel nozzles of combustion chambers; distortion of load-carrying casing; warping of flame tubes of combustion chambers [1].

Due to the circumferential nonuniformity of gas flow, the RW blades are under higher dynamic stresses. All this leads to the oscillations of the blades and its damage.

Several approaches are implemented in practice to reduce the level of gas flow nonuniformity. The main idea of these methods is rather common. It is to eliminate the negative effect of nonuniformity on the problem domain of the turbomachinery, which can be achieved in two ways. Firstly, it is possible to modify the design of turbomachinery itself by changing the axial clearance between the RW and GV or by changing the stagger angles and pitch of GV blades to non-uniform around the circumference [6-8]. Secondly, the modifications can be implemented for the annular frame, for example by changing the number of struts or using the inclined struts.

The main drawback of these both ways is that it is difficult to use them for existing engines. These approaches require a large number of significant changes in the engine construction.

This article is dedicated to reducing the circumferential nonuniformity of gas flow form the annular frame of axial compressor. In turn, the goal of this research was to find the ways of the flow non-
uniformity reduction, which would lead to a decrease in the level of dynamic stresses in RW blades located in front of the annular frame, but would not require a fundamental engine design changing.

Both ways of reduction of flow nonuniformity including variations in design of compressor and frame were considered in the paper. The main advantage of this research is in detailed consideration of the reasons of negative effects of flow nonuniformity in present case, as well as possible ways to reduce the nonuniformity and results that can be obtained through the use of these approaches.

2. Motivation of the research. Description of the compressor
The motivation for this study was the real-life problem of the rotor blades’ destruction of the fifth rotor wheel of intermediate pressure compressor (RW5 in the Figure 1). This compressor is part of a gas turbine engine NK-36ST (Figure 1), which has been designed and manufactured at Samara enterprise of aviation and space propulsion engineering [9].

The cause for the blade destruction is circumferential flow nonuniformity caused by struts of GTE middle annular frame. The struts of annular frame have different thickness and are distributed with different angular displacement relative to each other (Figure 2). The struts cause the local areas of high pressure, passing through which the impeller blades experience the dynamic impact. This impact leads to blades’ forced vibrations and high dynamic stresses (Figure 3).

As a consequence of this problem, the blades of RW5 had mid-span shroud. However, this solution has several drawbacks [10]: the shroud increases the weight of a compressor; shrouded blades have low efficiency and life cycle, and greater complexity in manufacturing.

3. Method of calculation of the level of gas flow nonuniformity
Calculation method of the gas flow nonuniformity level after the rotor blades of the fifth IPC stage was developed at the initial stage. This method consists of two steps (Figure 4).
Step 1. CFD simulation of the sector IPC model was performed at the main GTE modes. Sector IPC model consists of the following domains: intermediate annular frame, all blade rows of IPC, middle annular frame, and inlet guide vanes of high-pressure compressor. All blade domains were simulated as one blade passage with periodical boundary conditions; domains of annular frames were simulated as a sector of one strut.

To transfer the flow parameters between the domains, the Mixing Plane interface averaging the flow parameters in the circumferential direction was used [11].

The aim of this calculation was to determine the distribution of the flow parameters along the height of flow section (total pressure, total temperature and flow angle) in the section behind the RW4. These parameters will be used as boundary conditions for subsequent calculations.

Figure 3. Mach number (a) and pressure (b) fields between struts and close to struts

Figure 4. Scheme of method for calculation compressor rotor blades forced oscillations
**Step 2.** Calculation of the "full circle" model was performed.

The reason for using the full circle model for the further calculations is the desire to detect the flow circumferential unevenness, which arises from the presence of annular frame struts, unevenly located along its circumference in the compressor passage.

Full circle model consists of blade row domains of the fourth GV, the fifth RW and GV, the middle annular frame and HPC IGV. Gas dynamic load acting on the all blades as well as static pressure distribution along the circumference of the impeller are determined in this calculation.

Finite volume mesh of the full circle model was created in the module *AutoGrid 5* of software package *Numeca FineTurbo* [11]. Parameters of finite volume mesh and configuration of computational models have been chosen according to the recommendations given in [12-14]. Computational models have been verified in studies [6] in the evaluating of the vibration level of the rotor blades. This shows the reliability of calculations carried out in the described study.

Total pressure and total temperature, and flow direction at the inlet obtained from sector model calculations [14] and outlet static pressure were set as boundary conditions.

To transfer parameters between rotating and stationary domains of the full circle model, the standard interface *Frozen Rotor* was used. This interface transmits the flow parameters between domains without their averaging in the circumferential direction in contrast to the *Stage* interface [15]. Thus, it can be used to calculate the circumferential flow nonuniformity.

### 4. Gas-dynamic methods to reduce gas flow nonuniformity

#### 4.1 Modifications of Compressor Design

One of the methods of reducing the circumferential nonuniformity is to use guide vanes with a different circle pitch and stagger angle in front of annular frame struts [7-8]. It is possible to redistribute the flow between the different blade passages and to adjust the position of high pressure zones [16].

To implement the proposed method, a parametric model of the GU5 was created. All the guide vanes were divided into 7 groups according to the location of seven struts of the annular frame. Changes in the stagger angles and pitch were performed within each group. With the introduction of different stagger angles, the minimum number of variable vanes was a key factor, since the production of a large number of blades with different geometry greatly increases the manufacturing charges.

The stagger angle of GV of the fifth stage has not been changed for vanes that are located in the symmetry plane of the strut, the first and the last vane in the group. Changing in the stagger angles within the groups was carried out according to the linear law. The vanes located on the opposite sides of the symmetry plane of the strut were rotated in opposite directions relative to the initial position (Figure 5). When the vanes are located closer to the strut, they were rotated by a larger angle. If the vane was rotated for closing (an increase in the stagger angle), there is sign "+" before the angle if the vane was rotated for the opening (a decrease in the stagger angle), there is sign "-" before the angle. It should be noted, that stagger angels were measured from the leading edge plane.
In case of pitch change, the number of variable vanes was not restricted. Different pitch was set within -0.35...+0.35 of the initial pitch. The sign "-" indicates that the pitch between vanes is decreased, while the sign "+" indicates the pitch increase. The number indicates the maximum increase (decrease) in the pitch between the vanes in the group in relative values from the initial pitch with evenly spaced vanes. The position of extreme vanes in the groups has not been changed with the introduction of different pitch. The law of the pitch changing was also linear.

There were investigated 11 different configurations of the fifth GU with circumferentially different stagger angles and pitch (Table 1).

**Table 1. The results of the parametric IPC model calculation.**

| No. of variant | Parameter of different stagger angles, maximum stagger angles (No. of blades) for groups: | Parameter of alternating blade pitch for the groups: | Number of variable blades | Dynamic stresses MPa |
|---------------|---------------------------------------------------------------------------------|--------------------------------------------------|--------------------------|----------------------|
| 1             | 0, 0, 0, 0, 0, 0                                                                  | 0, 0, 0, 0, 0, 0                                  | 0                        | 86.7                 |
| 2             | 0, 0, 0, 0, 0, 0                                                                  | 0.3, 0.3, 0.3, 0.3, 0.3, 0.3                     | 0                        | 115.98               |
| 3             | 3 (6), 3 (6), 3 (8)                                                                | 0, 0, 0, 0, 0, 0                                  | 42                       | 43.791               |
| 4             | 3 (6), 3 (6), 3 (8)                                                                | 0.3, 0.3, 0.3, 0.3, 0.3, 0.3                      | 42                       | 31.877               |
| 5             | 3 (6), 6 (6), 9 (8)                                                                | 0, 0, 0, 0, 0, 0                                  | 42                       | 37.536               |
| 6             | 0, 0, 0, 0, 0                                                                      | 0.35, 0.35, 0.35, 0.35, 0.35, 0.35                | 0                        | 86.95                |
| 7             | 0, 0, 0, 0, 0                                                                      | -0.3, -0.3, -0.3, -0.3, -0.3, -0.3                | 0                        | 105.94               |
| 8             | 0, 0, 0, 0, 0                                                                      | -0.15, -0.15, -0.15, -0.15, -0.15, -0.15         | 0                        | 137.741              |
| 9             | 6 (2), 6 (2), 6 (2)                                                                | 0, 0, 0, 0, 0, 0                                  | 14                       | 46.737               |
| 10            | 0, 6 (2), 6 (2)                                                                   | 0, 0, 0, 0, 0, 0                                  | 8                        | 57.214               |
| 11            | 6 (2), 6 (2), 6 (4)                                                                | 0, 0, 0, 0, 0, 0                                  | 16                       | 44.426               |
Analysis of the results showed that the use of circumferentially different stagger angles and pitch makes it possible to achieve a significant reduction in the level of gas flow nonuniformity, and. The positive impact of circumferentially different stagger angles and pitch on the value of dynamic stresses can be explained from gas-dynamic point of view. Flow structure becomes more uniform, with less high pressure zones due to the redistribution of the flow (Figure 6). In addition, the values of pressure peaks behind the RW5 have been reduced (Figure 7). In turn, these improvements lead to the decrease in dynamic stresses in rotor blades (Table 1). That was confirmed in [17] with the help of method of calculation of dangerous harmonic amplitudes proposed by researches A.I. Ermakov and A.O. Shklovets.

Thus, the introduction of circumferentially different stagger angles and pitch of GU vanes located in front of the annular frame reduces the circumferential flow nonuniformity. However, the implementation of this method leads to a complication of manufacturing technology, since it will be necessary to produce the guide vanes using not versatile technology in this case.

In addition, due to the change in the vane pitch, a new technology for grooving will be required.

4.2 Modification of Annular Frame Design

There are experimental studies that prove that the airfoil cascade, which is located in front of the cylinder, streamlined by the flow, enhances the upstream transmission of the high pressure zone [16].

The proposed method for reducing the circumferential nonuniformity of the gas flow is based on the results of the study [16]. It is in distancing the leading edge of annular frame struts (in this case it is the cylinder) from the trailing edge of GV blades located upstream (in this case it is the cascade) in order to neutralize the negative effects. At the same time, an important requirement is the need to maintain the internal cavities of the annular frame struts for placement of the engine system.

Struts can be divided into three groups according to the maximum thickness of profile Cm: thick (strut No. 1), medium (struts No. 2,4,5) and thin (struts No. 3,6,7) (Figure 2).

![Figure 7](image)

**Figure 7.** Graphs of the relative static pressure behind the fifth impeller of the basic variant (a) and GU variant No. 9 (b)

To implement the chosen method of the nonuniformity reduction, the parametric models were developed for each type of strut. The models make it possible to adjust the position of the leading edge, pressing the profile of the strut, but maintaining the thickness and the rest of the profile by specifying the desired value of shifting ($\Delta b$) (Figure 8). Parametric models take into account the limitations related to the location of the engine systems in the struts.
To analyze the effect of leading edge shifting on the circumferential nonuniformity of the gas flow, the set of computational models of the compressor was created using the parametric models of struts. All of them were full circle models and were consisted of five domains: GV of the fourth stage, RW and GV of the fifth stage, middle annular frame, and IGV of HPC. At the same time, the annular frame contained only two struts disposed facing each other with the same thickness in each model (Figure 9). Such models were called elementary, and were created to assess the contribution of each type of strut in the total circumferential flow non-uniformity. In other words, they were used to calculate the pressure peaks occurring opposite each type of strut.

Moreover, leading edge shifting of struts with $\Delta b = 0$, $\Delta b = 0.2$, $\Delta b = 0.4$, $\Delta b = 0.6$, were set for each type of struts in different models. Thus, the total number of computational models was 12.

The calculation and analysis of the results were also carried out in Ansys CFX using the supercomputer of SSAU "Sergei Korolev". Computational models were verified in [6,17] while assessing the level of vibrations in rotor blades. This indicates the reliability of calculations described in this paper.

The circumferential distributions of relative static pressure in the section behind the RW for each type of strut with all values of leading edge shifting were obtained. It should be noted that obtaining of the pressure distribution only for one model took a long time, both in terms of creation model, and the calculation process. The total time for testing of just one variant is approximately 12 hours.

These distributions were averaged to obtain the so-called elementary peaks (Figure 10). In addition, the level of mean pressure after the RW (red line) was correlated with zero of y-axis for a visual representation of the elementary peaks.

Figure 11 shows a field of static pressure distribution for case of thick strut with of $\Delta b = 0$.

It is obvious from the graphs form Figure 10 that annular frame struts lead to the appearance of complex forms of pressure peaks after the RW. The peaks have not only the maximum jump $h$ (positive upper part of the graph), but also go to the negative part of the graph into the decline $h_1$. A similar pattern is seen in Figure 11 by following in the positive direction of the angle $\varphi$.

The complex peak shape can be caused by the rotation of the RW blades in the area of the circumferential nonuniformity of gas flow.
Maximal jump of pressure peaks $h$ decreases (upper part of graph) while the value of the leading edge shifting is increasing. It is worth noting that the maximum failure of pressure peaks $h_1$ (the bottom part of the graph) does not depend on the shift of the leading edge and it is constant in all charts. Similar results were obtained for other types of struts.

To verify the adequacy of the obtained dependences, the graph from the individual elementary peaks of each type of strut was assembled and compared with the graph for the reference design of seven-strut annular frame (Figure 12a).

Also, the distribution of relative static pressure after the RW of the similar compressor, but with experimental 13-strut annular frame was obtained in previous studies [17]. Thick and medium struts in this annular frame correspond to the same struts in seven-strut annular frame, and thin struts have the twice smaller thickness than the thin strut in seven-strut annular frame.

A similar investigation aimed on the identification of elementary peaks was conducted for that annular frame. Comparison of chart from the elementary peaks with the chart received in the calculation of the reference design of 13-strut annular frame is shown in Figure 12b.

In general, graphics derived from the elementary peak, describe the pressure distribution after the RW well. The results can be used for further research stages.
4.2.1 Searching for the equations of the peak height dependence on the leading edge shifting. In order to use the obtained results, it is necessary to describe the dependence of pressure peaks on the shift of the strut leading edge via equations. That was the next step of the study.

Quadratic equations were derived for each type of strut using the regression analysis techniques, describes the effect of the shift of the leading edge (Δb) on the pressure peak height (h). Equation 1 corresponds to the thick strut, Equation 2 to the medium strut and Equation 3 to the thin one.

\[
\begin{align*}
    h &= 0.001\Delta b^2 - 0.135 \Delta b + 7.5 \\
    h &= 0.001\Delta b^2 - 0.135 \Delta b + 7.5 \\
    h &= 2.64 \times 10^{-18} \Delta b^2 - 0.02 \Delta b + 2
\end{align*}
\]

Thus, the so-called surrogate model was created, by means of which the value of pressure peak behind the RW can be predicted at the preliminary stage of investigation under certain provisions of the leading edges of the annular frame struts.

Application of leading edge shift of annular frame struts allows for not only quantitative changes in the values of pressure peak behind the impeller (Figure 13). The proposed method also provides better quality structure of the flow in the compressor: field becomes more uniform, high-pressure zone reduces its impact on the blade rows, located upstream.

Figure 12. Comparison of the graph of relative static pressure after RW for reference design of seven-strut (a), 13-strut annular frame (b) and graphic obtained by elementary models

Figure 13. Circumferential distribution of the relative static pressure behind the RW for the reference design of the 7-strut annular frames and annular frame with shifted leading edges
The received regression equations were used for optimization of the location and shift of annular frame struts to reduce the dynamic stresses on the rotor wheel of the fifth stage of the IPC [18].

Conclusion

Several approaches to decrease the gas flow nonuniformity in the axial compressor arise because of the presence of annular frame have been developed as a result of conducted research. Each of the proposed approaches made it possible to achieve the goal that was to reduce the level of pressure peaks after the rotor blades of the last compressor stage.

Firstly, it was suggested to eliminate the negative influence of flow nonuniformity by modifying the guide vanes of the last compressor stage. It was found that the different blades stagger angles and alternating blade pitch implementation allows "flattening" the circumferential variation in the section following RW5. In addition, more periodic and uniform flow field is achieved for all specific areas.

Secondly, the circumferential unevenness of gas flow in the compressor can be reduced by a minimum interference in design. It is enough to alienate its source (the support rack) from the elements, experiencing the negative impact of circumferential unevenness of gas flow (rotor blades located upstream). This is accomplished by shifting the leading edges of the support racks.

The advantage of the second method is that it makes possible to predict the pressure peak level behind the RW to the particular design of the support racks without the long process of preparation of the computation model, calculation and post-processing of the results by using surrogate models. This can accelerate the design process of new compressors and design refinement of existing units.

The obtained results were used to optimize the design of the support to reduce the dynamic stresses in the RW5. The surrogate model was integrated into the strength module. Pressure peaks were decomposed to Fourier series, for further calculation of dangerous harmonics.

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References

[1] Ivanov V P 1983 Kolebaniya Rabochego Kolesa Turbomashin (Vibrations of Turbomachinery Impeller) (Mashinostroenie, Moscow) p 244
[2] Vorob'ev Yu S 1988 Kolebaniya Lopatochnogo Apparata Turbomashin (Vibrations Turbomachinery Blading) (Kiev, Nauk. Dumka) p 244
[3] Cohen H, Rogers G F C and Saravanamuttoo H I H 1996 Gas Turbine Theory. 4-th edition (Longman Group Limited)
[4] Logan E Jr 2003 Handbook of Turbomachinery (CRC Press; 2 edition)
[5] Carter T J 2005 Common failures in gas turbine blades Engineering Failure Analysis 12(2) 237–247
[6] Kolmakova D, Popov G, Shklovets A and Ermakov A 2014 Techniques and Methods to Improve the Dynamic Strength of Gas Turbine Engines Compressor Rotor Wheels Proc. of the ASME 2014 Gas Turbine India Conference GTINDIA2014-8203
[7] Sladojević I, Sayma A I and Imregun M 2007 Influence of stagger angle variation on aerodynamic damping and frequency shifts Proc. of the ASME Turbo Expo 5 683–700
[8] Yang Y, Yang A, Dong R, Chen E and Dai R 2012 Influence of stagger angle on aerodynamic sound performance of compressor cascade Hangkong Xuebao/Acta Aeronautica et Astronautica Sinica 33(4) 588-596
[9] JSC "Kuznetsov", Accessed July 20, (2016). http://www.kuznetsov-motors.ru/en
[10] Cumpsty N A 2004 Compressor Aerodynamics (Krieger Publishing Compan, Edition 2)
[11] NUMECA, User Manual AutoGrid5 Release 8.4, NUMECA.inc., Belgium, January 2008.
[12] Matveev V N, Popov G M, Goryachkin E S and Smirnova Y D 2014 Effect of Accounting of Air
Bleed from the Flow Passage of the Multi-Stage Axial Low Pressure Compressor on its Design Performances Research J. of Applied Sciences 9(11) 784-788

[13] Popov G, Goryachkin E, Baturin O and Kolmakova D 2014 Development of Recommendations on Building of the Lightweight Calculation Mathematical Models of the Axial Turbines of Gas Turbine Engines Int. J. of Engineering and Technology 6(5) 2236-2243

[14] Ermakov A, Shklovets A, Popov G and Kolmakova D 2014 Investigation of the effect of the gas turbine compressor supports on gas flow circumferential nonuniformity Research J. of Applied Sciences 9 684-690

[15] ANSYS® Release 12.0, Help System, Ansys CFX-Solver Modeling Guide, ANSYS, Inc.

[16] Saren V 1984 Flow around irregular lattice of plates placed in front of the cylinder Technical Report (Moscow: Central Institute of Aviation Motors) p 36

[17] Popov G, Kolmakova D, Shklovets A and Ermakov A 2015 Optimization of the axial compressor flow passage to reduce the circumferential distortion 9th International Conference on Compressors and Their Systems

[18] Kolmakova D, Popov G, Shklovets A and Ermakov A 2016 Rational Design of Gas Turbine Engine Compressor to Provide the Required Dynamic Strength Level of Rotor Blades Proc. of the ASME Turbo Expo GT2016 GT2016-56991