Improving of the working process of axial compressors of gas turbine engines by using an optimization method

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Abstract. The article presents one optimization method for improving of the working process of an axial compressor of gas turbine engine. Developed method allows to perform search for the best geometry of compressor blades automatically by using optimization software IOSO and CFD software NUMECA Fine/Turbo. Optimization was performed by changing the form of the middle line in the three sections of each blade and shifts of three sections of the guide vanes in the circumferential and axial directions. The calculation of the compressor parameters was performed for work and stall point of its performance map on each optimization step. Study was carried out for seven-stage high-pressure compressor and three-stage low-pressure compressors. As a result of optimization, improvement of efficiency was achieved for all investigated compressors.

1. Introduction

Compressor design process involves several stages from preliminary 1D calculation to the final 3D design with stacking blade sections along height. At the step of 3D design, software of numerical simulation of the workflow (CFD software systems) is usually used. They allow to predict reliably the characteristics of the compressor taking into account even the most minor features of the three-dimensional geometry of the flow part and to assess the impact of variation of any geometric or operational parameter on them.

Application of CFD software for the calculation of compressor operation does not cause any trouble when the designer has the geometry of the blades and the flow part. In case if the designer must find the geometry of the blades and flow part for specific operating parameters of the compressor, it is difficult to use CFD software. The main reason is that the flow part shape of axial compressor stages is described by large number of variables. At least two sections, each of which is determined by dozens of independent variables, must be determined to describe the shape of one blade. Furthermore, it is necessary to describe the relative position of the sections relative to each other, shape of meridional section of the flow part, etc. As a result, the number of independent variables for a complete description of one stage reaches several tens which affect contradictory compressor characteristics. Man is not capable physiologically to analyze the problem of such dimension. In addition, the search time for optimal combination of variables will be prohibitively high. The problem becomes proportionally larger if it is necessary to investigate the operation of multistage compressor.
In this case, the optimization program can help the engineer. Their use allows the automatization of the search for the optimal combination of independent variables by their automatic variation and analysis of the results based on optimization algorithms. This article shows the technique for improving the working process of axial compressor using optimization software IOSO and CFD software NUMECA Fine/Turbo and shows examples of the application of this technique on real seven-stage high-pressure compressor and three-stage low-pressure compressors of gas turbine engine (figure 1).

Figure 1. Investigated three-spool compressor

The solution of this problem was carried out in the study of upgrade options of gas turbine engine with power of 25 MW held in Samara University [5]. Compressor of the engine (figure 1) consists of three spools: a low-pressure compressor (LPC), an intermediate pressure compressor (IPC) and a high pressure compressor (HPC). Optimization of LPC and HPC are presented in the article. The aim of optimization was to investigate which maximum efficiency is possible to obtain in each compressor [3]. The criteria and constraints for the optimization problems of each compressor have been selected on the basis of thermodynamic calculations of the engine [1].

2. Developed method of optimization of axial compressors
Indirect Optimization on the base of Self-Organization (IOSO) [9, 10] is used in this research. IOSO Technology is based on the response surface methodology approach. At each IOSO iteration the internally constructed response surface model for the objective is being optimized within the current search region. This step is followed by a direct call to the actual mathematical model of the system for the candidate optimal point obtained from optimizing internal response surface model. During IOSO operation, the information about the system behavior is stored for the points in the neighborhood of the extremum, so that the response surface model becomes more accurate for this search area. The following steps are internally taken while proceeding from one IOSO iteration to another:
- the modification of the experiment plan;
- the adaptive adjustment of the current search area;
- the function type choice (global or middle-range) for the response surface model;
- the adjustment of the response surface model;
- the modification of both parameters and structure of the optimization algorithms;
- if necessary, the selection of the new promising points within the search area.

Specified actions are executed automatically by the program IOSO, and data exchange with a mathematical model of the process under investigation (in this case the numerical model of the working process in the compressor) is performed by text files.

The developed algorithm of the compressor optimization using IOSO is shown in figure 2.
At each step of the optimization, the optimizer IOSO creates a design of experiment and generates a Vector of variable parameters $x_1, x_2, x_3, \ldots, x_n$. This vector is a set of variables that describe the geometry of the compressor blade in a parametric form. The Vector of variable parameters is transferred to the block of reprofiling. In this block, specialized «in-house» programs perform a conversion of compressor blades based on the Vector of variable parameters and save them in the form of geometry files (GF) of blades in a format *.geomturbo [4] which is suitable for importing blade geometry in software systems of numerical simulation Numeca FineTurbo [6]. Used parametric models of the blades and reprofiling method will be described hereinafter.

At the next step, construction of computational mesh in the software Numeca AutoGrid 5 is initially performed in the block of gas-dynamic parameters’ calculation (figure 3) using the files with geometry of reprofiled compressor blades GF1, GF2 ... GFn. Then, creation of computational model is carried out in the software Numeca FineTurbo and its calculation is performed. Processing of CFD-calculation results is performed in the software package Numeca CFView. As a result, several output files are created containing operation parameters of compressor (constraint vector and optimization criteria) in the text format. These parameters are passed to the optimizer IOSO. At the final stage, the vector of output parameters (optimization criteria and constraints) $y_1, y_2, \ldots, y_n$, formed by the results of the running of the blocks of gas-dynamic and strength parameters’ calculation is transferred to the optimizer IOSO. Analysis of obtained data is performed in the optimizer as it was indicated above in the description of one IOSO iteration. Then, a new Vector of variable parameters is generated. And the cycle (iteration) of optimization is repeated.

![Figure 2. The algorithm for solving optimization problem of compressors.](image)

3. Numerical model of compressors used for optimization
The numerical models of the compressors created by means of the mesh generator NUMECA AutoGrid 5 were used to calculate operational parameters of HPC and LPC. Geometrical model of computational domain of the flow in the investigated compressors was created based on design documentation of engine. Figure 3 shows created geometrical models of LPC and HPC.

![Figure 3. Geometrical models of investigated compressors](image)
The created model was divided into finite volumes to block-structured grid by internal tools of the program NUMECA. Two computational models (“light” model and “heavy” model) were created for each compressor [11]. Light model contained 300 thousand elements per blade row on an average; the value of $y^+$ was 3-5. Heavy model contained 500 thousand elements per blade row on an average; the value of $y^+$ was 1-2.

For data transmission between the area of the GV and BW the built-in software package NUMECA interface Full Non Matching Mixing Plane is used. It averages the flow parameters in the circumferential direction in the upstream area and transmits as the boundary condition in the area located downstream.

As the boundary conditions at the inlet of the HPC the value of the total pressure was set equal to $p^* = 101.325 \text{kPa}$ and total temperature was equal to $T^* = 288.15 \text{K}$. Parameters of turbulence at the inlet boundary are $k=5 \text{m}^2/\text{s}^2$, $\varepsilon=30000 \text{m}^2/\text{s}^3$.

In the calculation of characteristics the stall borderline was defined as the point with the minimum flow rate of the working fluid, in which it managed to get the coincided decision.

To prove the validity of the numerical models LPC (figure 5) and HPC (figure 6) characteristics were calculated at several operational modes. Then they were compared with the available experimental data for the compressors.

The calculated parameters of the compressor are presented in relative form:

$$\bar{n}_{\text{cor}} = \frac{n_{\text{cor}}}{n_{\text{cor BASE}}} \times 100\%,$$

where $n_{\text{cor BASE}}$ – the rotor speed at the primary operational mode of the base LPC; $n_{\text{cor}}$- corrected rotor speed,

$$n_{\text{cor}} = n \sqrt{\frac{288.15}{T_H^*}},$$

where $n$ - specific rotor speed; $n$ – physical rotor speed; $T_H^*$ - air temperature at engine inlet.

**Normalized Efficiency** – efficiency referred to efficiency at the primary operational mode of the base LPC.

**Normalized Pressure Ratio** – pressure ratio referred to efficiency at the primary operational mode of the base LPC.

**Normalized Mass Flow** – air flow rate referred to efficiency at the primary operational mode of the base LPC.

![Figure 5. Comparison of calculated and experimental parameters of HPC.](image-url)
Figure 6. Comparison of calculated and experimental parameters of LPC.

As can be seen from figure 5, both created numerical models show good qualitative coincidence with the experimental results. However, the model №2 shows significantly better quantitative coincidence with experimental results. The difference of values for efficiency and the compression ratio is not more than 1%.

Apprently from figure 6, calculated pressure characteristics of the LPC qualitatively repeat the experimental ones. The maximum deviation of calculated values of the pressure ratio from the experimental value is 1% (abs.) at the mode $\bar{n}_{cor} = 84\%$. Also, figure 6 shows that calculated curves of efficiency characteristic of the base LPC qualitatively repeat the experimental ones at all modes. The maximum difference between calculated and measured efficiency is 2.9% (abs.) at the mode $\bar{n}_{cor} = 84\%$. At the mode $\bar{n}_{cor} = 100\%$, the maximum deviation is 0.7% (abs.).

Based on the above, it was concluded that, despite some quantitative discrepancies with the available experimental data, created numerical model can adequately describe the operation of the LPC and can be used for search of configuration with maximum efficiency. For this reason, the model №2 was used for further studies.

4. Optimization of the HPC of engine

The optimization problem of the HPC was testing and was solved in order to prove the possibility of increasing the efficiency of the multistage compressor with the help of the developed optimization method. Objective functions were assigned to the values of efficiency at the modes corresponding to the rotational speed of 95% and 100% from the primary mode; and all stagger angles of the HPC blade rows were selected as varying variables.

The criteria for this optimization problem:
- increase in HPC efficiency at the mode corresponding to the rotational speed of 95%;
- increase in HPC efficiency at the mode corresponding to the rotational speed of 100%.

In order to prevent the shift of characteristics of the compressor, the following restrictions were set in the optimization:

- flow rate of the working fluid through the HPC at relative frequency of rotation of 95% was not supposed to be different from the respective flow rate of the base compressor more than $\pm1.3\%$;
- flow rate of the working fluid through the HPC at relative frequency of rotation of 100% was not supposed to be different from the respective flow rate of the base compressor more than $\pm0.6\%$;
- value change of HPC pressure ratio compared with the basic compressor at points of the maximum efficiency at relative rotational speeds of 95% and 100% allowed within $\pm1.5\%$.

In formulating the optimization problem and the appointment of restrictions the changes in stall margin of HPC did not take into account, in order to reduce the time of receipt of the decision.
Evaluation of changes in stall margin of HPC is conducted at the analysis phase of optimization results.
Schematically, the optimization criteria and restrictions used in the formulation of the optimization problem, shown in figure 7.

![Figure 7. Formulation of the optimization problem by varying the stagger angles of compressor blades.](image)

As varied variables the stagger angles of all rotor blades, guide vanes and IGV of the HPC were selected (figure 8). The range of change of stagger angles of the vanes of each blade row has been selected so that during the blades rotation their profiles fits into existing blade locks. The number of blades in the row has not changed. This solution allows to find the variant to increase the efficiency of the HPC, which would not require modification of the disk and the body parts of the compressor. The total number of changed variables was 15.

![Figure 8. Blade rows changed during optimization](image)

Optimization was performed according to the algorithm given in section 2. The whole optimization process was completely automated. To solve the formulated problem of optimization the software package IOSO had 446 calls to the numerical model of the HPC. Each call to the numerical model is calculation of two points on the characteristic of the HPC (points of maximum efficiency on the branches corresponding to the relative frequency of rotation of 95% and 100%) in the software package NUMECA FineTurbo.

As a result, a lot of unimprovable solutions (Pareto set) were obtained, which is a compromise between increase of efficiency at the relative rotational speed of 95% and increase of efficiency on the relative rotational speed of 100% (figure 9). Each point of Pareto set corresponds to the unique geometry of the HPC represented as an array of stagger angles of all blade rows of HPC.

![Figure 9. Pareto set of HPC optimization.](image)
maximum efficiency is 1.8% (abs.) at a substantially constant maximum efficiency at relative rotational speed of 100% (point 1 of Pareto set in figure 9). When the relative rotational speed of 100% the highest increase of maximum efficiency is 0.6% (abs.) at the increasing of maximum efficiency at relative rotational speed of 80% to 1% (point 2 Pareto set in figure 9). However, for further research one of midpoints of the set Pareto has been selected (point 3 in figure 9), providing increasing of the efficiency as at relative rotation frequency of 100% (0.5% (abs.)) and at the relative frequency rotation of 95 % (1.2% (abs.)).

To analyze the results of the optimization the numerical model of HPC variant was created, which corresponding to the selected point 3 of the Pareto set. By this numerical model the characteristics of optimized version of HPC at the relative rotational speeds of 95% and 100% were obtained as well as their comparison with the characteristics of HPC base version (figure 10) and with the searching results of the optimal combination of stagger angles at the first three stages, described above, performed.

The result of comparison revealed the following characteristics:
- stall margin of operation of optimized HPC compared to the base case at the investigated frequencies of rotation have changed slightly;
- changing the values of the air flow rate and compressor pressure ratio of the optimized HPC at points of maximum efficiency at the investigated frequencies of rotation is within the accepted limits;
- HPC efficiency at the relative rotation frequency of 95% has increased by 1.2% (abs.) and at relative rotation frequency of 100% increase of efficiency was 0.5% (abs.).

5. Optimization of the LPC of engine

The aim of LPC optimization was to reprofile the compressor for new operating conditions. Required parameters of the compressor operation have been obtained as a result of thermodynamic analysis of the engine. On conditions of the problem, it was necessary to ensure the growth of the LPC efficiency by 1.5% (absolute value) with 4% increase of pressure ratio, 2% of the rotor speed and decrease of the working fluid flow rate by 8% with respect to the parameters of the original engine. The goal is to be achieved with unchanged diametrical and axial dimensions of the compressor flow, while maintaining acceptable safety margins of all the rotor elements.

Optimization problem was solved in a two-criteria formulation. As optimization criteria were selected (figure 11):
- increase of LPC efficiency at rotor speed $n = 102\%$ (relative to rotor speed of the base compressor);
- decrease of the relative flow rate of air the through the compressor.
The constraints that determine the position of the working point on the LPC characteristic specified for the optimization problem:
- minimum value of the air flow rate in the design point is limited by the range: \(0.91 \ldots 0.96 \cdot \text{base}\);
- the total pressure ratio was maintained at the design point a predetermined range: \(1.009 \ldots 1.046 \cdot \text{base}\);
- the flow angle changing at the LPC outlet was limited by the range: \(\pm 5\) degrees.

Varying variables were selected as follow. Slightly modified approach from [7] was used to describe the shape of the camber line. Changing the camber line shape of the rotor blades and the relative position of the cross sections relative to each other was carried out by moving the spline middle control points in the circumferential direction in the global coordinate system, as well by varying the stagger angle (figure 12, a). This solution helped to keep the value of the blade chord in control sections, which is important in terms of maintaining the stress strain state. The guide vane profile was changed by moving the middle point of the spline in the circumferential direction and moving the point of the trailing edge along both coordinates (figure 12, b).

Reforming algorithm of the blade profile along the height with the approaches described above has been implemented in the «in-house» program, developed at the Department of Aircraft Engines Theory of SSAU [18, 19]. This program allows to convert a table of coordinates, which describes the blade shape in design drawing into text files of source data to build turbomachinery computational models in programs like Auto Grid and TurboGrid.

To solve the formulated problem of optimization the software package IOSO had 446 calls to the numerical model of the HPC. Each call to the numerical model is calculation of two points on the characteristic of the HPC (points of maximum efficiency on the branches corresponding to the relative frequency of rotation of 95% and 100%) in the software package NUMECA FineTurbo. As a result, the Pareto set of best possible criteria according to two criteria - the relative efficiency and the relative flow rate of the working fluid through the compressor were obtained (figure 13).

Three points were selected for the optimization results validation and analysis (figure 13):
1. The pressure characteristics were calculated for each point at the rotor speed of 102% of the base LPC speed and compared with the initial characteristic of the compressor (figure 14).
Comparing the characteristics of the different variants, it can be seen that all the variants have approximately the same efficiency at the desired air flow rate through the compressor. Moreover, this value is greater than efficiency of the base compressor by 1.3% (absolute). Figure 17. Comparison of relative characteristics of the selected LPC variants with characteristic of original compressor Variant №2 was adopted as a final embodiment of the compressor, because it allows to obtain the desired pressure ratio (variant №3 has less value), has a higher efficiency than the variant №1 and stall margin for this variant does not differ from the original compressor stall margin.

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Figure 13. Pareto frontier of the problem.

Figure 14. Comparison of characteristics of the selected LPC variants with characteristic of original compressor

5. Conclusion
Summing up the work done it can be concluded that the goal has been reached. The developed algorithm showed its effectiveness for the repilling multistage turbomachinery.

HPC variant has been found that provides an increase in the efficiency at the operation mode corresponding to 95% of the rotational speed by 1.2% (absolute) and at the mode corresponding to 100% of the rotational speed by 0.5% (absolute).

Also a variant of the shape of low pressure compressor blades has been found that provides an increase in efficiency by 1.3% (abs.), while increasing the pressure ratio by 4%, the rotational speed by 2% and decrease in the mass flow rate of working fluid flow by 8% relative to compressor of base engine.
In this case compressor parameters have been changed only by changing the shape of the blades, while maintaining unchanged the remaining design elements, including the blade number and shape of the grooves for blade attachment. During the search for improved shapes of blades, measures for the conservation of their stress strain state in the former borders were taken. It would be difficult to obtain a similar result with traditional methods of designing compressor as the base variants of the compressor already had a high level of efficiency, and the requirements were contradictory.

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