Proceedings of Global Power and Propulsion Society
ISSN-Nr: 2504-4400
GPPS Xi'an21
18th – 20th October, 2021
www.gpps.global

GPPS-TC-2021-0033

AERODYNAMIC IMPROVEMENT OF THE 16-STAGE AXIAL COMPRESSOR

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ABSTRACT

The paper describes the process of gas-dynamic modernization of a 16-stage axial compressor of an industrial gas turbine unit. During the modernization of the unit, it became necessary to significantly increase the working parameters of the engine, which resulted in a change in the operating conditions of the compressor. As a result, it was necessary to modify it while maintaining the design as much as possible.

Using the results of experimental studies of the initial compressor, a numerical model of the compressor was developed and validated. It was used to investigate the specifics of the compressor's working process and to propose solutions to the problem posed.

Modernization works were significantly hampered by the presence of many stages and many independent variables. For this reason, the problem was solved in several stages. A separate modernization of the first and rear groups of stages was performed. Then the working processes of the compressor parts were matched.

As a result of the research, a variant was found to modernize the existing 16-stage axial compressor, providing an increase in the air mass flow rate by 18%, adiabatic efficiency by 3.5%, and margins of gas-dynamic stability up to 16%.

INTRODUCTION

Every year, global energy consumption increases by an average of 2% per year [1]. This requires a steady increase in generating capacity. This requires a constant increase in generating capacity. One of the main sources of power generation in the Russian Federation and other large countries are gas turbine power plants that use natural gas as fuel. The share of electricity generated by such power plants in different countries is 21 ... 50% of all electricity consumed. Modern gas turbine power plants have low noise levels and have a less negative impact on the environment.

In the next 50-70 years natural gas will have a significant share in the power generation market, so research related to ensuring and improving the efficiency and reliability of ground-based GTUs will be relevant for a long time to come.

One of the companies successfully operating in the market of ground-based gas turbine units for power generation is Russian Power Machines [2]. In 1990s, the company developed a gas turbine unit to drive a 65 MW electric generator. It

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is a single-shaft GTE consisting of a 16-stage axial compressor, a combustion chamber, and a 4-stage axial turbine. Power for the drive of the electric generator is taken from the turbocompressor rotor (fig. 1).

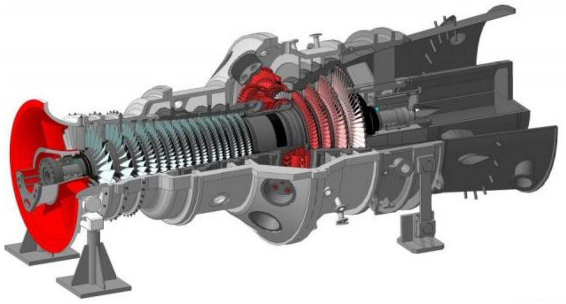


Figure 1 Gas Turbine Unit With a Power of 65 MW [2]

Power Machines is currently carrying out work on comprehensive modernization of the original GTU design, including, among other things, the modernization of the compressor. Cycle parameters were changed in order to bring the GTU's economic performance closer to the current level. For this reason, compressor operating conditions have changed and there was a need to modify it for the new operating conditions, as well as to eliminate the drawbacks found during the tests. The required changes to the compressor parameters are shown in Table 1. To solve this task, Power Machines approached the Samara University for assistance. At the same time, in order to reduce the cost of modernisation, the design of the initial engine had to be preserved as much as possible.

Table 1 - Requirements for modernization of a gas turbine compressor relative to the initial design

Parameter	Required value
Reduced air mass flow rate, Ga.red., kg/s	+ 5.6%
Pressure ratio, π^*c	+ 1.92%
Adiabatic efficiency, η^*ad	+ 1.2%
Gas dynamic stability margins, ΔKst, %	>15

The methods and tools for solving this problem and the results obtained are described in the article presented.

NUMERICAL MODEL AND THE RESULTS OF ITS VALIDATION

The research of the working process in the original compressor and then in its modified versions was carried out with the help of a numerical model created in the CFD software package NUMECA FINE/Turbo [3]. The geometry of the flow path of the baseline compressor was created in full accordance with its working drawings provided by JSC Power Machines. The deformations of the blades under the action of gas and centrifugal forces were not considered. The geometry of the computational domain considered the presence of cavities for air extraction from the flow path (a total of 5 bleeding points). The calculation model took into account the radial clearances above the rotor blades. Their values were provided by Power Machines and ranged from 1.2 to 0.45mm.

Computational area consists of rotating domains of the rotor wheels (RW), which were calculated in a relative coordinate system rotating at the speed of the rotor, and domains of stator vanes (SV), calculated in a stationary coordinate system (Fig. 2). Information between adjacent domains 1...15 stages of the computational model was transmitted using the Full Non-Matching Mixing Plane interface, averaging the parameter fields in the circumferential direction.

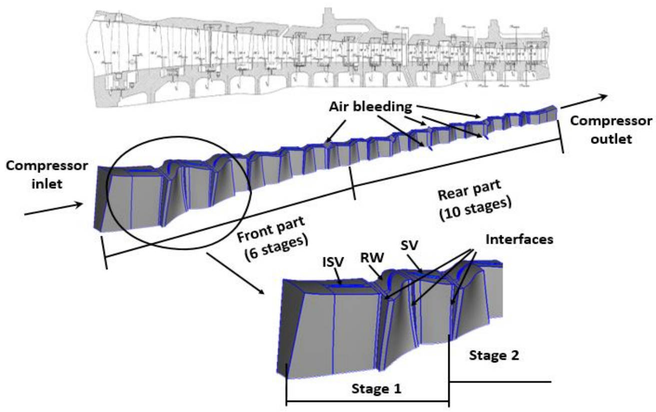


Figure 2 Appearance of the Compressor Computational Model

When creating computational models, the following assumptions were used:

- The gas flowing in all blade passages within one blade row was considered the same.
- The flow was considered in a steady statement.
- The properties of the working fluid were described by the ideal gas model considering the change in viscosity and heat capacity depending on temperature.
- Turbulence was simulated by isotropic one- and two-parameter models such as Spalart-Allmaras, k-ε, SST, and k-ω.
- Heat exchange between the walls of the flow path and the flow was not considered

The boundary conditions at the compressor inlet were the values of total pressures and temperatures in the form of radial diagrams obtained in the experiments. At the outlet of the compressor, the static pressure at the hub section was set. At the points of air bleeding from the compressor path, the values of the corresponding air mass flow rates were set.

All computational models used in research and optimization had similar settings. The differences were only in the shape of the blades.

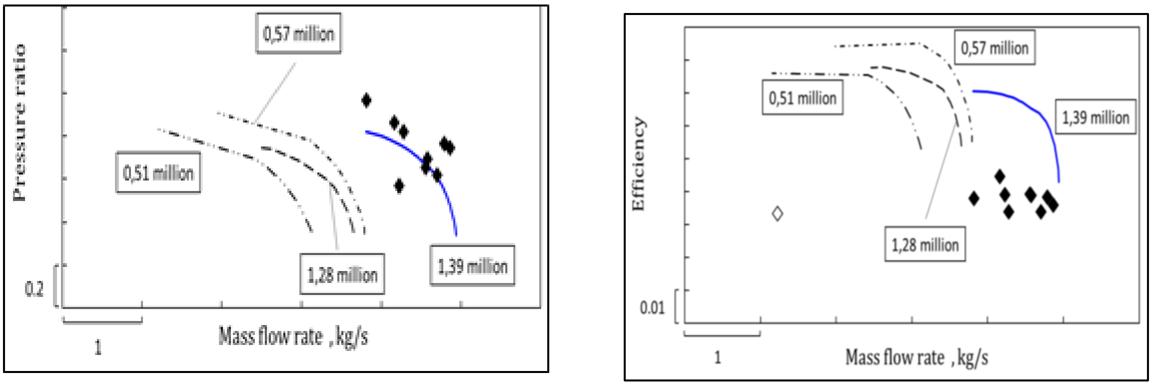
To verify the created computational model a mathematical model of the process in the front part of the compressor was created. This part was tested and Power Machines provided us with the results to validate the mathematical model. The information regarding the test results is provided by Power Machines exactly as given in the article. Unfortunately, we do not have detailed data on the measurement error and experimental conditions.

Four models with different finite volume meshes were created: from 0.51 to 1,39 million elements per blade row on average.

A model with 1.39 million cells has been built so that:

- the number of elements in single layer perpendicular to the blade axis (B2B layer) is approximately 21000;
- ER=1.2, where ER - cell growth factor, indicating how many times the height of one finite volume layer is greater than the next one;
- MR= 1000, where MR - channel cell height determined by the ratio of the maximum element height in the channel to the height of the closest cell to the wall.

The k-epsilon (extended wall function) turbulence model was used. The characteristics of the compressor front part obtained from the created models are shown in Figure 3. Calculation time per working point is 4 hours on a 12Tflops supercomputer.Calculation time per feature branch is approximately 50 hours. From the given results it can be seen that, the best match with experimental results is provided with the computational model having an average of 1.39 million mesh cells per blade row. This mesh has been accepted for the calculations. As a result, the mathematical model of the 16-stage compressor has 47.4 million elements. The relatively large difference between the calculation data and the experimental data is explained by some simplification of the calculation model (the influence of the near duct cavities and the deformation of the blades were not taken into account).



Pressure characteristics   Efficiency characteristics

◆ **Test data**

Figure 3 Comparison of the Calculated Characteristics of the Compressor Front Part Obtained Using Various Calculation Data with the Experimental Results

ANALYSIS OF THE WORKING PROCESS OF THE INITIAL COMPRESSOR CONFIGURATION

Using the accepted settings, a mathematical model of the entire 16-stage compressor was created. With its help, the characteristics of the compressor and detailed information about the working process of all its stages were obtained. There is a non-optimal distribution of the pressure head coefficient ($\bar{H}_T$) over the stages, a drop in the load in the first and middle stages (Stepanov load parameter ($\bar{H}_T / c_a/u$), which is less than the lower limit of the optimum area of 0.5), and a non-monotonic character of the full pressure ratio with a noticeable drop in stages 3, 5 and 6. The analysis of the rows and stages revealed the rows with a significant drop in aerodynamic efficiency - these are the rotor rows of the 1^{st}, 4^{th}, and 7^{th} stages. Also developed separation zones in the hub section of the compressor from stage 3 are detected.

The analysis of the characteristics of the compressor rear part revealed that it is oversized in mass flow rate by more than 9% for the original compressor version.

The real need to provide significantly greater compressor boost has been identified:

- by 22% in terms of full pressure ratio (an order of magnitude greater than previously shown in Table 1);
- by 3.6...4.2% in adiabatic efficiency (three times higher than indicated in Table 1).

Realization of the design value of the full pressure ratio increase in the inlet block will actually meet the requirement of the specification for gas-dynamic stability reserves. However, the improvement of the inlet stage group can only have a positive effect if there is a decrease in the reduced flow rate of the outlet unit.

MODERNIZATION OF THE COMPRESSOR

The analysis of working process of the initial compressor has shown, that by changing only the blade shape (keeping their number and shape of roots) it will not allow to solve the set task. Variation of incidence angles allowed to get compressor efficiency gain not more than 1%. For this reason it was decided to optimize all the blade rows, but retaining the shape of the discs.

The adopted ideology of modernization

Modernization of a 16-stage compressor entirely requires a lot of resources. In this regard, it was decided to first carry out the modernization of the compressor parameters independently by blocks:

- inlet block - first six stages;
- outlet block - stages from the seventh to the sixteenth and the stator vane of the sixth stage.

This approach made it possible to optimize the block workflows in parallel, as well as to reduce the requirements for computing resources. As a result, several dozen possible variants of both blocks were found. Then the found variants were combined into a single compressor and an analysis of its working process was carried out, the geometry of individual rows was adjusted. As a result, a compressor variant was found that satisfied all the aerodynamic requirements.

Inlet block modernization

During the modernization of the compressor inlet block researches had to increase the reduced air mass flow rate, provide the maximum possible polytropic efficiency and an increase in the total pressure ratio.

During the analysis of the inlet block parameters of the baseline compressor, a significant positive swirl at the inlet created by the ISV was noted. Such an inlet swirl, on the one hand, allows one to minimize the Mach number at the shroud of the RW in relative motion; on the other hand, the presence of an inlet positive swirl reduces the compressor head. This circumstance also led to a significant distortion of the radial field of total pressure right after the first stage RW. The presence of significant radial irregularity, increased residual swirl after the 1st stage SV increased in the next stages, which limited the head in the hub sections of the subsequent stages and did not allow increasing the head of these stages. To eliminate the noted defect in the new variant of the compressor, an inlet negative swirl was introduced on the hub sections (Fig. 4), which made it possible to reverse the total pressure in the group of inlet stages while minimizing the residual swirl when entering the next stage. To eliminate this defect in the new version of the compressor, it has been proposed to modify the design of the VNA to create an inlet negative swirl at the hub sections (Figure 4). This will produce a more favourable change in pressure fields along the radius in the first stages of the compressor. Due to the measures noted, the expended head coefficient of the first three stages was increased by 20%.

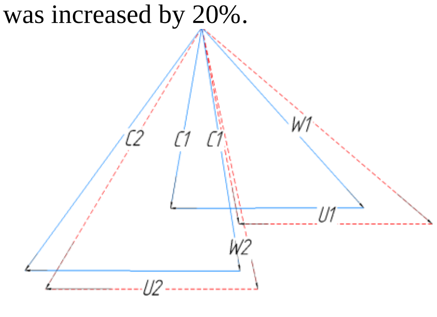


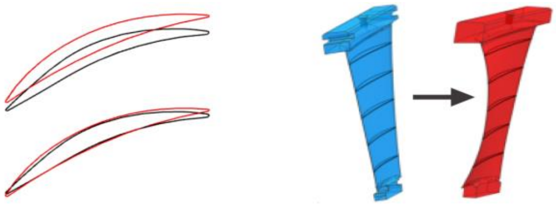
Figure 4 Comparison of Velocity diagrams of the First Stage of Baseline and Modernized Compressor (solid – base; dashed – modernized)

The choice of the geometry of the rows was carried out during a series of adjusting calculations. Initially, the first stage was modernized, to which the second stage was connected firstly, then the third. Further, stages four through six were added together. The use of this scheme is explained by the complex process of matching high-speed stages.

All compressor rows were created using double-circular-arc profiles. This type has advantages due to the possibility of selecting more rational aerodynamic shapes. While adjusting calculations, the geometry of the blade rows was changed. In individual rows, the maximum thickness and radius of the trailing edge were noticeably reduced.

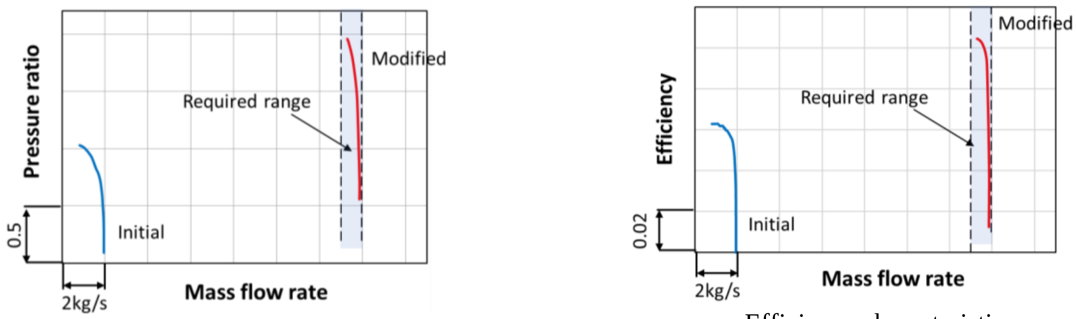
In total, more than 60 variants of the compressor inlet block were considered, based on the results of which the most acceptable geometry of the inlet block was determined. satisfying the set requirements. As a result, it was possible to achieve an increase in reduced air mass flow rate, adiabatic efficiency by 5,4%, the increase in the total pressure ratio was 24%.

Comparison of the shapes of the original and modernized inlet block blades is shown in figure 5. The characteristics of the modernised inlet block are shown in fig. 6. It should be noted that they were obtained without considering the influence of joint work with the outlet block.



black – base; red - modernized   blue – base; red - modernized

Figure 5 Example of Changing the Geometry of Inlet Block Blades



Pressure characteristics   Efficiency characteristics

Figure 6 Comparison of the Initial and Modified Characteristics of the Compressor Inlet Block

Outlet block modernization

The analysis of the baseline compressor showed that taking into account the increase in the head of the front part, it is necessary to shift the characteristics of the outlet block to the region of lower reduced mass flow rates at the inlet to the seventh stage. Therefore, in order to modify the compressor outlet block, it was necessary to decrease the reduced air mass flow rate at the seventh stage inlet, provide the maximum possible polytropic efficiency, provide an increase in the total pressure ratio.

When modernizing the outlet block a stepwise expert assessment of options and optimization were used.

Non-automatic modernization was carried out in a series of 3D CFD simulations with calculating characteristics. At first, the 7th and 8th stages of the baseline compressor were modernized, then the next two stages were added to them, then the next two, etc. At the inlet to the seventh stage, a radial diagram of the change in parameters was set, determined from the results of calculating the final variant of the inlet block.

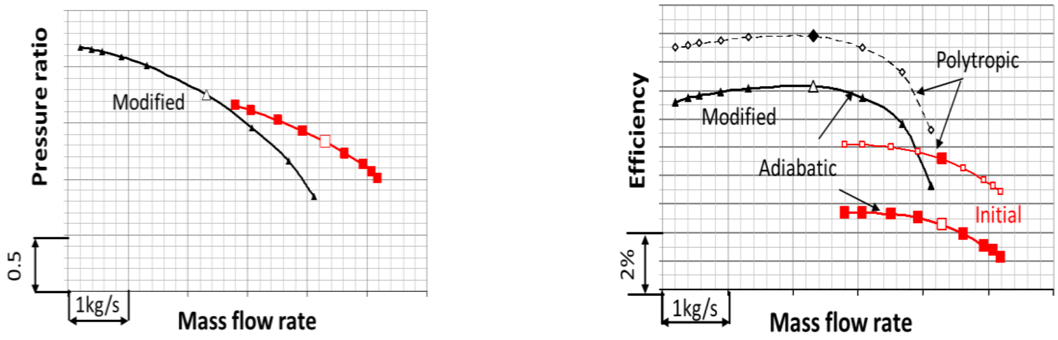
A decrease of the air mass flow rate through the rear part of the compressor required a decrease in stagger angles of almost all stages. Also during non-automatic modernization, the blade shapes were corrected. The greatest attention was paid to the shroud sections of the RWs and hub sections of SVs. These sections have been noticeably thinned. Also, the spatial profiling of the SV blades was carried out and their number in the rows was increased.

In the process of non-automatic modernization of the outlet block, more than 35 variants were considered. Figure 7 shows a comparison of baseline and best-of-breed performance. In the figure, the compressor operating points are highlighted. They are determined based on the results of calculating a full-size compressor based on the parameters at the inlet to the 7th stage. In this case, the front part variant obtained earlier was used. It can be seen from the Figure that although a decrease in the mass flow rate through the block was achieved, it was only 3.4% instead of the required 4.3%.

Since it was not possible to decrease the reduced air flow rate through the group of last stages in non-automatic mode, the decision was made to use mathematical optimisation methods with the IOSO program [9]. It is based on an optimisation method involving the construction of a response surface, which is refined and evolves each time the computational model is accessed. Each iteration of IOSO contains two steps. In the first step, a response function in the form of a multi-level graph is constructed based on early model accesses with different combinations of varying variables. The next step is to search for an extremum of the found function. This approach makes it possible to constantly adjust the response surface in the process of optimization. As a result, an unusually small number of initial points are required to start the optimization process in order to construct it and get the first results [9].

At each step optimizer IOSO forms vector of input data x1, x2, x3, …, xn. With its help in parametrizator program the geometry of blades of RW and SV is formed. On their basis a geometric model of the flow path is generated in *.GeomTurbo format. At the next step, the CFD calculations are performed in Numeca FineTurbo with the new mesh model. The processing of CFD results is carried out by a special script and small applications developed using NET Framework library. As a result, several output files are created containing operation parameters the outlet block of the compressor in the text format. These parameters are then passed to the optimizer IOSO.

In the course of the task IOSO was used as a ready-made product. No corrections were made to the optimization algorithm. IOSO was chosen due to a large number of positive examples of its application in turbomachinery optimization [10, 11].



Pressure characteristics Efficiency characteristics

Figure 7 Comparison of Initial and Modified During Non Automatic Modernization Characteristics of the Compressor Outlet Block

As a starting point for automatic optimisation, one of the best options found in non-automatic optimisation was used.

Automatic optimization was carried out in a two-criterion statement. A solution was sought that provides a decrease in the reduced air mass flow rate and a simultaneous increase in the pressure ratio at the stall point. The head characteristic will not become more flat during the optimisation process.

In the optimization, the blade width and section displacement for SV and stagger angles, blade angles at the inlet and outlet for RW and SV in 3 sections of each blade were varied.

The blades were parameterized in Numeca AutoBlade. The total number of independent variables was 224. The blade parameterization scheme is shown in Figure 8.

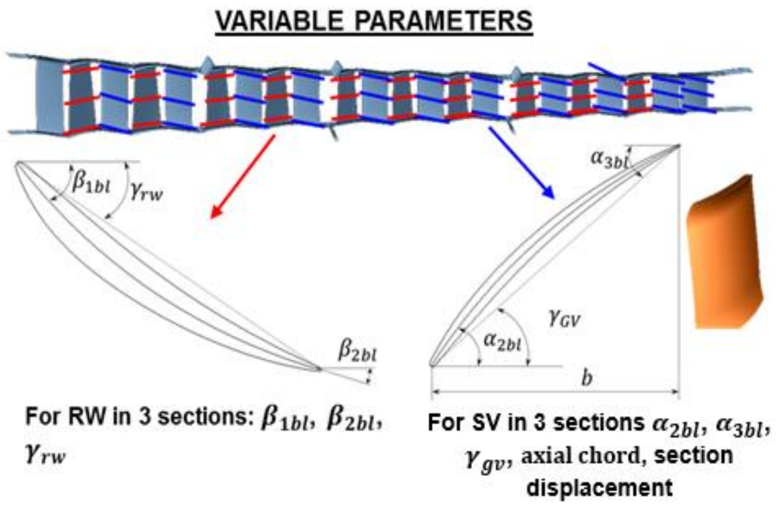


Figure 8 Scheme of Parameterization of RW and SV Blades

In each optimisation cycle, the compressor is first calculated at the design point. In this case, the air mass flow rate and the efficiency in the design point are monitored. This is necessary to obtain the initial data for CFD calculation of the characteristic point near the stall. Then, the calculation was carried out at the stall point (Fig. 9).

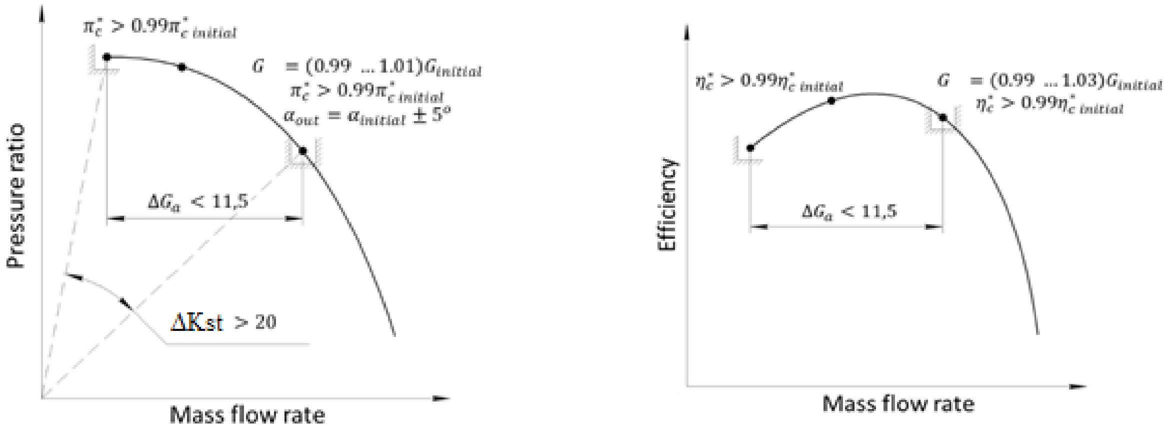


Figure 9 Scheme for the Task of Automatic Optimization and Accepted Constrains

After 2300 iterations with the computational model, the Pareto front was obtained by the parameters "pressure ratio at the stall point – compressor mass flow rate" and "efficiency at the operating point - compressor mass flow rate" (Fig. 10). It is seen that it was not possible to achieve a significant increase in the pressure ratio and efficiency with the mass flow rates achieved with non-automatic optimization. A further reduction in mass flow rate can only be achieved by reducing the pressure ratio. From the points obtained on the Pareto front, several promising options for the modernization of the output part were selected. It was decided to continue further modernization as part of a full-size compressor in non-automatic mode.

Matching of compressor parts

The modernization of the inlet and rear parts was carried out on separate models, which did not consider the mutual influence of their workflow. This especially affected the outlet block of the compressor, for the simulation of which constant fields were used at the inlet to the sixth stage stator vanes. In this regard, additional calculations were required after combining the input and output parts to coordinate the geometry of the blade rows as part of a full-size compressor. Based on the results of calculating the full-size compressor, an analysis of the characteristics was carried out, including for each separate compressor block. Mach number fields in the flow path were also analyzed. Based on the analysis carried out, subsequent decisions were made on the need to correct the blade rows. Basically, the geometry of individual rows of 5th… 16th stages was corrected.

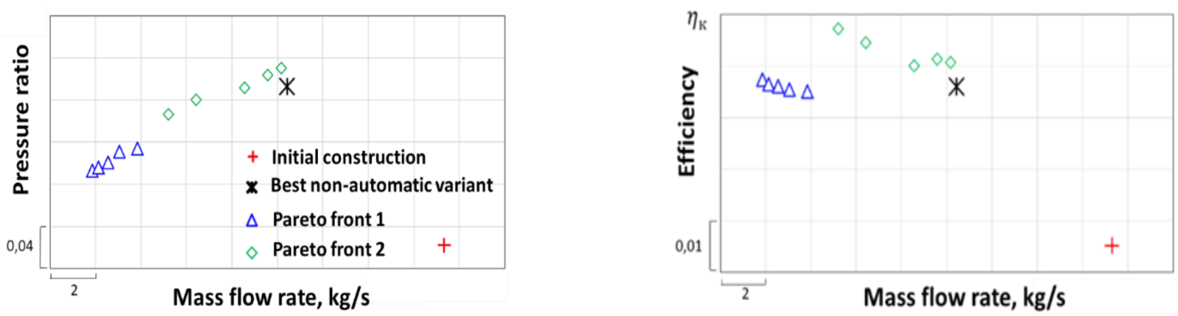
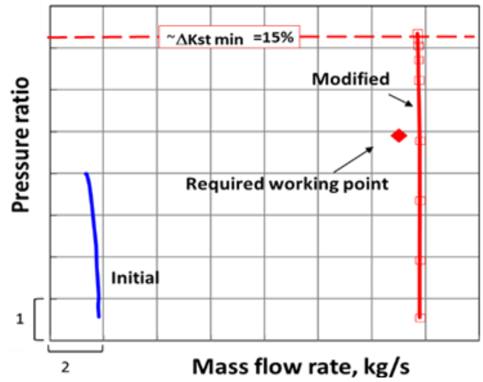


Figure 10 Pareto front «G_c and $\pi_{c.stall}$» and «G_c и η_c»

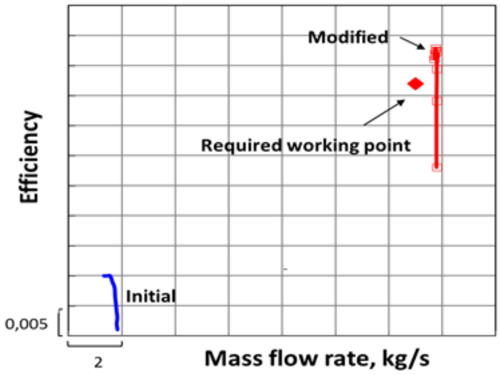
In total, more than 10 variants for modernizing a full-size compressor were considered. As a result, a compressor variant was obtained that met all aerodynamic requirements. Its characteristics are shown in the Figure 11.

The margins of aerodynamic stability were more than 15.5%, the adiabatic efficiency in the design point is higher than the required one by $\Delta\eta ad = 0.016$. The distribution of total pressure ratio across the stages has become smoother in the modernized variant. The adiabatic efficiency of all stages, except for the first one, became higher (Fig. 12).

Comparing characteristics of the inlet and rear parts (Fig. 13 and 14 with those of the full compressor before and after modification (Fig. 13 and 14 marked "initial" and «full modification») it is worth noting that it was the compressor inlet block that limited the performance of the full-size compressor in the initial design. After the modernization, the inlet block got a reserve in terms of the total pressure ratio of 8.1% relative to the maximum value as part of the full-size modernized compressor. At the same time, in the modernized compressor, the reserves of the total pressure ratio are now limited by the compressor outlet block. It follows from this that even though the modified compressor meets the requirements, it has a reserve of modernization by changing individual stages of the compressor outlet block.

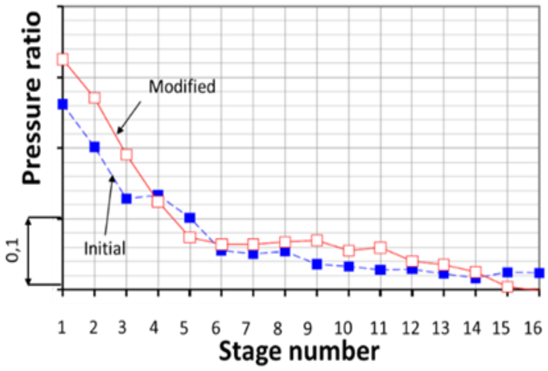


Pressure characteristics



Efficiency characteristics

Figure 11 Comparison of Initial and Final Characteristics of the Full-Size Compressor



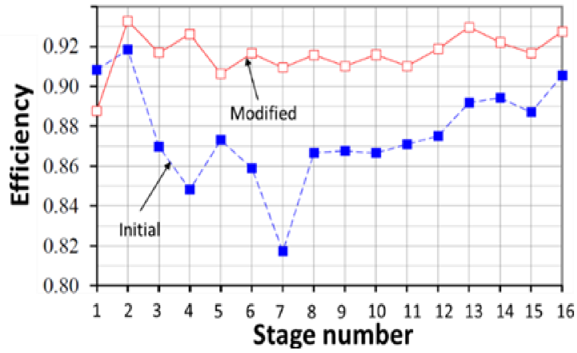


Figure 12 Comparison of Change in Polytropic Efficiency by Stages of Initial and Final Compressor Variants

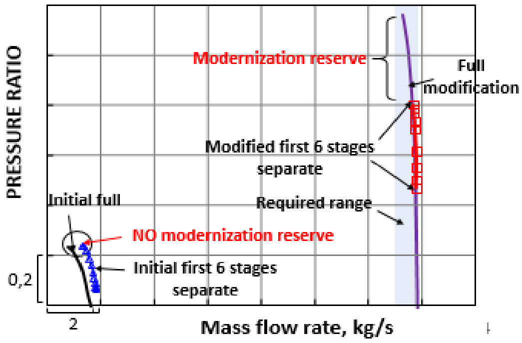


Figure 13 Comparison of Pressure Characteristics of the Front Part of Baseline and Modernized Compressors

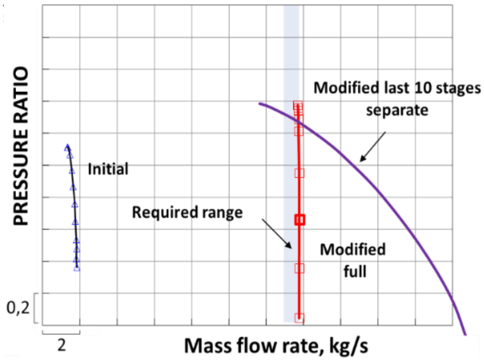


Figure 14 Comparison of Pressure Characteristics of the Rear Part of Baseline and Modernized Compressors

CONCLUSIONS

Our paper describes in detail the solution of a complex technical problem of aerodynamic adjustment of an axial 16-stage compressor for an industrial gas turbine unit. The complexity of the task lies in the fact that the task was set to not only achieve the design parameters of the compressor working process. Another serious problem was that the computational model of the working process of a 16-stage compressor requires large computing resources and is described by hundreds of variables, which makes it difficult to automatically optimize it.

At the beginning of the work, the CFD modelling was used to calculate the parameters that characterise the working process of the stages and separate rows at the various design points. This made it possible to carry out a detailed qualitative analysis and identify the most problematic stages. It was found that the front group of stages has low efficiency, and the rear group of steps is significantly oversized in terms of mass flow rates. In the future, such an analysis was used at further stages of work and made it possible to find compressor options close to the best one without using optimization methods.

The task of modernization was solved in several stages because of its complexity First, a separate adjustment of the first and rear groups of stages was given. For both parts, promising variants were found, from which the variant for the full compressor was formed. In total, 110 compressor variants were considered. 2300 variants of the rear part design were considered during its automatic optimization.

As a result of the study carried out, a variant was found to modernize the flow path of the existing 16-stage axial compressor, which provides an increase in the air mass flow rate by 18%, adiabatic efficiency by 3.5%, and a margin of aerodynamic stability up to 16%, which fully meets the requirements of the technical specification.

At the next stage, a 3D model of the compressor will be created. The possibility of its manufacture using the existing equipment will be studied. And the calculation for static and dynamic strength will be carried out. This will require another iteration of the compressor development. It is expected that meeting the stress requirements will lead to a significant decrease in aerodynamic efficiency.

Power Machines is also planning to manufacture and test the final version of the compressor, and we hope to get experimental confirmation of these results in the future.

NOMENCLATURE

c_a/u	flow coefficient;
$\bar{H}_T$	head coefficient;
$\bar{H}_т / c_a/u$	Stepanov coefficient;
π	pressure ratio;
η	efficiency;
G	mass flow rate;
Cp	static pressure rise coefficient of the compressor row;
$C_{p.eff}$	static pressure rise coefficient
γ	stagger angle.
β	blade angle;
b	axial chord;
SV	straightener vane;
SV	stator vane;
RW	rotor wheel;
GTU	gas turbine unit;
Bl	related to the blade;

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