Methodology for assessing the aerodynamic imbalance of GTE impellers

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Abstract. Increased vibration during testing and operation of gas turbine engines may be caused by aerodynamic imbalance. As usual, aerodynamic imbalance is eliminated on stands simulating the operating conditions of the impellers. Existing stands have a high cost of testing or do not allow to make allowances for all the factors during the operation of the impellers. A review of models and techniques has shown the importance of considering operating parameters when calculating aerodynamic imbalance. The paper provides a methodology for calculating the aerodynamic imbalance of the impeller. The methodology is based on the use of finite element modeling (FEM). The proposed method for assessing the aerodynamic imbalance of the impellers makes it possible to take into account the rotational speed, the actual geometry of the blades based on the measurement results, as well as the assembly conditions and the operating conditions of the impeller. The technique was used through the example of a specially designed and manufactured impeller with geometric deflections. Theoretical and experimental studies have been carried out. The discrepancy between theoretical and experimental data did not exceed 15.5%. The developed model can be used at enterprises that manufacture gas turbine engines, gas compressor units, steam turbine plants and wind power plants.

1. Introduction
Dynamic balancing of the impellers is an integral part of the technological process of assembling the rotors of gas turbine aircraft engines. The rotation frequency at the stage of balancing of serial machines is 10...50% of the working one. The quality assessment of the balanced rotor of a gas turbine engine is carried out according to the interstate standard [1]. Dynamic balancing does not allow eliminating the aerodynamic imbalance of the rotor, since it does not take into account the real rotational speed, the environment where it operates and the factors. The aerodynamic imbalance of the impeller can be up to 20% of the total imbalance.

Aerodynamic imbalance arises due to the uneven gas flow through the passageway of the impeller. Mechanical and aerodynamic imbalances are similar in that during operation there is an unbalanced force vector pointed to the radial direction of the impeller. The above vectors can be positioned to each other quite randomly. Aerodynamic imbalance results from the following factors:

- the manufacturing error of blades and disks;
- the manufacturing tolerance of the installation of the blades in the disks and their pitches [5, 6];
- the spread in values of the blade airfoils deformations under the action of centrifugal and gas forces, resulting in a different change in the angles of profiles and pitches installation [6];
- the spread in values of the tensions over the contact surfaces of the blade shrouds [7] and other factors.

Mechanical imbalance of the impeller is eliminated using balancing stands by removing or adding corrective masses, and aerodynamic imbalance is eliminated using stands that simulate the operating conditions of the unit. At most enterprises, measures to eliminate the aerodynamic imbalance of the impellers are not carried out due to the high cost of the stand and the significant costs of testing. Another option is the use of acceleration benches, which allow the units to be accelerated to operating speeds. The use of acceleration benches does not allow to reduce the aerodynamic imbalance, since the factors acting during the operation of the impellers are not taken into account.

The invention [2] describes a method for balancing the aerodynamics of a vane wheel, based on an amplitude value estimation of the radial forces of the blades when they are selected to the scheme of their balance. The paper [3] presents a method of low-speed balancing of the mass of a high-speed blade rotor, which consists in balancing the rotor using a balancing machine. There is a way [4] of balancing the vane wheels of machines, which consists in the selection of blades with close chord angles. Their measurement is carried out in a certain section of the blade airfoil to the wheel plane, and they are installed in pairs in mutually opposite grooves of the disc. The described methods increase the labor intensity and cost of finishing work, and also do not take into account the temperature, pressure and composition of the gas air environment during operation. The paper [8] describes the importance of taking into account the total pressure and air inlet temperature \( (P_{in}, T_{in}) \) when simulating the flow of a gas medium through a gas turbine engine. The operation of an aircraft engine is associated with a change in \( P_{in} \) and \( T_{in} \), which change the forces acting on the blades and on the impeller as a whole.

Analytical and FEM do not have these disadvantages. As the study [9] shows, the aerodynamic imbalance of the impeller can be balanced by the method of mass correction for one of the operating modes. This is because the emerging force is directly proportional to the angular velocity of the rotor, and to compensate it, the centrifugal force of the correcting mass is used, which is proportional to the square of the angular velocity. The problem of aerodynamic imbalance also arises in wind turbines [10] as a result of deformation of the blades. The paper provides a forecast of the frequency and damping coefficient depending on the rotational speed of the blade.

Computer modeling makes it possible to improve the aerodynamic and aeroacoustic characteristics of the blade airfoils [11], as well as to predict the dynamic stability of the blade units of the GTE compressor [12]. This makes it possible to design the geometry of the blades for the required operating conditions at the design stage. The paper [13] presents an analytical model for calculating the dynamic and aerodynamic imbalance for the Vestas V80-2MW wind turbine. The article [14] investigates the influence of the main geometric parameters of the axial stage on the averaged and unsteady airloads acting on the rotor blades. Aerodynamic imbalance also affects the energy performance of engines [15].

A review of models and techniques has revealed the importance of considering operating factors when calculating aerodynamic imbalance. The paper proposes a method for assessing the aerodynamic imbalance of the impellers which takes into account the rotational speed, the actual geometry of the blades based on the measurement results, assembly conditions and operating conditions.

2. Methodology for assessing the aerodynamic imbalance of the impeller

2.1. Preparatory stage

The technique for assessing the aerodynamic imbalance of the impellers is presented in the form of a flowchart in figure 1.

Let us consider the application of the proposed technique on the example of a specially designed and manufactured impeller, taking into account its geometric deflections.
Figure 1. Block diagram of a technique for assessing the aerodynamic imbalance of the impeller.

At the preparatory stage, the geometry of the impeller blading was formed using the ANSYS software product with the aid of the Vista AFD and BladeGen modules. The initial parameters of the process were the characteristics of a mobile balancing machine used for small rotors: shaft speed $n = 5800$ rpm; shaft power $N = 100$ W; shaft torque $M = 0.17$ N·m; the outer diameter of the impeller $D = 84$ mm. As a result of the calculation in the Vista AFD module, the geometry of the blade airfoil was formed and the following parameters of the impeller were determined: air flow rate $G = 0.195$ kg/sec and its total pressure increase $P = 800$ Pa, the number of blades was 7 pcs. The resulting geometry was named Impeller No. 1.

The calculated nominal geometry of impeller No. 1 has been revised. The blade airfoils were changed and the impeller was constructed in the BladeGen module. The impeller has small dimensions, and the aerodynamic imbalance is mainly significant for oversized fan impellers, therefore, the location of the two blades had to be profoundly changed relative to the nominal position. The changed parameters of impeller No. 2 relative to the nominal (No. 1) are presented in table 1.

Table 1. Changed parameters of impeller No. 2.

| Blade, number | The angular position of the blade relative to the nominal | The angle of the blade relative to the nominal |
|---------------|----------------------------------------------------------|---------------------------------------------|
| 1             | 5°                                                       | -10°                                        |
| 2             | -51.43°                                                  | 0°                                          |
| 3             | -102.86°                                                 | 0°                                          |
| 4             | -159.29°                                                 | 10°                                         |
| 5             | -205.71°                                                 | 0°                                          |
| 6             | -257.14°                                                 | 0°                                          |
| 7             | -308.57°                                                 | 0°                                          |

Geometric deflections in 1 and 4 of the impeller blades will cause flow separation and the appearance of unbalanced forces on them. Due to the opposite rotation of two blades relative to each other, the total projection of the force vector will be headed in one direction.

2.2. Measuring the flow path of the impeller

The measurement was carried out with a RangeVision PRO optical scanner using a rotating table. Impeller No. 2 was manufactured using FDM additive technology. The measurement of impeller No. 2
was carried out in order to control the accuracy of its manufacture and to verify the adequacy of the developed technique for assessing the aerodynamic imbalance. Based on the measurements obtained, a 3D model will be built, taking into account the geometric deviations identified during its measurement. Scanning was carried out under the operating conditions of the first zone to ensure the highest accuracy. Impeller No. 2 was measured in three positions with a discreteness of 15 degrees in automatic mode. The measurement process is shown in figure 2.

Removal of outlying values, noises and stitching of the resulting 72 scanned copies was carried out in ScanCenter NG. As a result of the final alignment, the estimated stitching accuracy of the impeller scans was 0.025 mm. The claimed measurement accuracy of the RangeVision PRO scanner in the first zone is 0.018 mm. The resulting polygonal model, shown in figure 3, was exported to *.stl format for further processing. Impeller measurement can also be carried out by the contact method using a coordinate measuring machine. In this case, the predetermined sections of the blades will be controlled. For convenience of further use, a parametrized 3D model can be created that changes its geometry according to the measured coordinates of the surface points. In the case of creating a parameterized 3D model, the next step of the methodology is not required.

2.3. Building a 3D model that takes into account real deviations obtained from measurements

The current stage is being implemented using the Siemens NX CAD system based on a scanned polygonal model. Since a solid 3D model is required for further steps, it was first converted to a sheet body. Small unscanned facet elements were interpolated with cubic splines, and on their basis the “patch” surfaces were built. Surfaces not involved in further calculations have been removed or replaced with planes or cylinders. The next step was to transform the surfaces into a solid 3D model of impeller No. 2 and build its flow path using a Boolean operation. The CAD model of the part from the NX system is saved in *.x_t format. Then their geometry is imported into the CAE system.

2.4. FEM of the gas flow through the passage volume of the impeller

The calculations were performed in the CFX module of the ANSYS software product. The following parameters were set as the initial parameters: air flow rate \( G \), the magnitude of the total pressure increase behind the impeller \( \Delta p \), shaft speed \( n \), and the environmental parameters. The CFX module simulated the operation of impellers No. 1 and No. 2. The divergences of forces no more than 0.0001 N along the Ox, Oy, and Oy axes between iterations were used as repeatability criteria. Figure 4 shows a solid 3D model of the impeller flow volume, and figure 5 shows the lines of gas flow through it.
The result of the numerical calculation was: the projection of forces, co-directional to the coordinate axes of the rectangular coordinate system, the torque and power arising from the gas flow through the impeller.

2.5. Aerodynamic imbalance calculation

The calculation was carried out on the basis of the obtained projections of forces directed in the radial direction. To perform calculations of the obtained data, the MATLAB software package was used.

The aerodynamic imbalance $D_a$ was calculated using the expression:

$$D_a = \sqrt{F_x^a + F_y^a},$$

where $F_x^a$, $F_y^a$ – the resulting projections of gas forces on the axis $O_x$ and $O_y$, accordingly, resulting from the gas flow around the blades; $w$ – rotational speed of the impeller.

Elimination of aerodynamic imbalance is carried out by adding a balancing weight, or by removing the weight from the ends of the impeller. The mass of the correction weight can be found by the formula:

$$m = \frac{D_a}{e},$$

where $e$ – is the position radius of the correction weight.

The angle of the corrective weight is determined from the formula:

$$\beta = \arccos \left( \frac{\overrightarrow{F_x^a} + \overrightarrow{F_y^a} \cdot \overrightarrow{F_x^a}}{|\overrightarrow{F_x^a} + \overrightarrow{F_y^a}| \cdot |\overrightarrow{F_x^a}|} \right),$$

where $\overrightarrow{F_i}$ – is the vector of forces directed along the axes $O_x$ and $O_y$.

Next, the repeatability of the proposed model will be tested.

3. Model verification

The model was verified in the following sequence:

1) elimination of the mechanical component of the total imbalance of impeller No. 2;
2) determination and elimination of aerodynamic imbalance of impeller No. 2;
3) assessment of the discrepancy between the results obtained using the model and during the experiment.

The first step was to eliminate the mechanical imbalance. To eliminate the influence of the gas air environment, the impeller was balanced in a VV-1 high-vacuum chamber (figure 6). Balancing of impeller No. 2 was carried out using a developed mobile balancing stand, which includes a BalKom-4 device.

![Mobile balancing stand installed in a high-vacuum chamber.](image)

The balancing was carried out at a residual pressure of $7.1 \times 10^{-7} \text{ bar}$ maximum and included 3 steps. Each step included removal of air, measuring, and normalizing the pressure inside the chamber to atmospheric. Elimination of the mechanical component of the imbalance was carried out by corrective weights installed in specially provided holes with a diameter of $2.5 \text{ mm}$, located on the separating radius of $18 \text{ mm}$. A weight of $0.15 \text{ g}$ was used as a test one. The electronic balance used in the experiment had a division value of $0.005 \text{ g}$. The residual mechanical imbalance of impeller No. 2 as a result of balancing in vacuum was $0.1 \text{ g} \cdot \text{mm}$, with a vibration level of $0.640 \text{ mm/s}$.

At the second stage, the mobile balancing stand was placed in atmospheric conditions, where after the aerodynamic imbalance was determined, as well as the position and weight of the corrective mass. After the introduction of the corrective mass, the residual imbalance was $0.1 \text{ g} \cdot \text{mm}$ with a residual vibration level of $0.613 \text{ mm/s}$.

The repeatability of the simulation results with the experimental data for impeller No. 2 was assessed by calculating the absolute deviations:

$$\Delta_u = P_w - P_e,$$  \hspace{1cm} (4)

where $P_w$ – is the parameter obtained as a result of modeling; $P_e$ – is the parameter obtained during the experiment.

Relative deviations were determined from the formula:

$$\Delta_o = \frac{\Delta_u}{P_e},$$  \hspace{1cm} (5)

Table 2 presents the results obtained using the model and during the experiment for the calculated impellers.
Table 2. Theoretical and experimental data of the impeller.

| Results                        | The magnitude of aerodynamic imbalance, g·mm | Corrective weight mass, g | The position of the corrective weight, deg |
|-------------------------------|---------------------------------------------|---------------------------|--------------------------------------------|
| FEM of impeller No. 1         | 0.056                                       | 0.0031                    | 103.27                                     |
| FEM of impeller No. 2         | 0.929                                       | 0.0516                    | 290                                        |
| Experiment carried out on impeller No. 2 | 1.1                                          | 0.059                     | 326                                        |
| Δa                            | 0.155                                       | 0.0074                    | 36                                         |
| Δa %                          | 15.5                                        | 12.5                      | 11                                         |

The discrepancy between theoretical and experimental data when using the above test and measuring equipment was no more than 15.5%.

The discrepancy between the simulation and experimental results can be explained by the following reasons:

1) errors in measuring and creating models of impeller surfaces;
2) measurement errors of the BalKom-4 device, electronic scales;
3) the assumptions made when developing a model for predicting aerodynamic imbalance.

The developed model can be used at enterprises that manufacture gas turbine engines, gas compressor units, steam turbine plants and wind power plants.

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