Development of radial turbines in low-power Gas-turbine engines

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Abstract. The article deals with the problem of providing required gasdynamic and strength parameters of turbines in low-power Gas-turbine engines. A method of turbine geometrical characteristics correction is offered. It includes blade profile evaluation and modification, turbine load complex modeling and CFD-modeling of gasdynamic processes. This method is applied to the turbine impeller prototype. The most loaded places of the impeller are revealed. Hazardous locations are places of blades joints with the disk in circumference where safety factor is close to the minimum acceptable value. One of the most effective methods of strength enhancement is use of parabolic joints between the wheel and blades. These joints not only can increment safety factor up to 1.3 but also preserves the gasdynamic efficiency of the turbine. This presented method which is based on CFD-modeling and strength analysis, allows to minimize fabrication time and costs of new micro-turbines’ developing process.

1. Introduction

Gas turbine engines are used widely in the electric power industry, heating supply systems and industrial facilities. The main component of such unit is a turbine: steam, gas or water powered [1]. One-stage centripetal axial-radial gas turbines are used in low power gas turbine engines (micro-turbines) because of their small dimensions (in comparison with axial turbines) and relatively high efficiency at the low gas flow rate [2]. A high-speed radial gas turbine engine, assembled on a single shaft with a high-frequency generator, composes the main part of a micro-turbine. In the market, fewer low power gas turbine engines are offered compared to high power engines [3].

Cost-effectiveness, reliability of development and enhancement of modern stationary gas-turbine engine construction depend on the capability of gas turbine engine flow range and flow process perfection. Micro-turbine designers achieve the task of using in designed micro-machines geometrical characteristics of existing prototype impellers with proven characteristics (e.g., aircraft impellers). It is necessary to demand higher standards of impeller durability while preserving required gasdynamic parameters of flow range, in order to provide the required frequency speed of micro-turbines. Thus, one of the most important tasks of micro-turbine design is radial turbine impeller gasdynamic and durability [4] optimization.
2. Method of optimization
In order to solve these tasks, a method of turbine geometrical characteristics correction is offered. The method makes possible to lower the stress that occurs in high loaded parts of impellers while preserving gasdynamic characteristics of prototype turbine flow range. Gas turbine engine turbine impeller optimization was conducted in several stages:

- cross-section estimation and prototype blade modification with profile input and output angle preservation;
- turbine impeller model creation according to the geometry of prototype flow range;
- turbine impeller strength calculation and comparison with prototype parameters;
- turbine impeller modification with consideration of greater demands in high-loaded parts;
- turbine gasdynamic calculation and comparison with prototype parameters.

3. Strength enhancement of turbine impeller
As prototype turbine impeller a typical turbine impeller with cooled blades was taken. Using Ansys CFX [5], the stress loads for 96000 rpm frequency speed and 700°C of the prototype turbine impeller were acquired. Prototype turbine impeller FEM strength calculation data is displayed in figure 1. Maximum loads appear at blade and disc connection areas (980 MPa) and on impeller hub (878 MPa). The ultimate stress of the material used ЧS88U-VI (ЧS88U-VI) at working temperature is equal to 960 MPa. Minimal safety factor is equal to 0.98 while allowed safety factor for continuous work is equal to 1.25 [6]. Prototype turbine impeller does not meet the strength requirement for micro-turbine rotor with nominal frequency speed.

![Figure 1. Stress distribution in the prototype of turbine wheel](image)

High-stress loads in impeller hub are caused by substantial peripheral blade thickness, therefore, higher values of rotating mass and corresponding stress loads are present. High values of stress loads in blade and disc connection areas are caused by blade flexural strain that comes out of centrifugal forces.

On impeller periphery, blades have a slope to the direction of rotation causing high-stress loads and deformation in disc and blade connection areas at high frequency speeds. On the first stage, to lower the centrifugal forces influence, the prototype turbine impeller blade profile is estimated with blade cross-section (perpendicular to rotation axis) analysis. The main task of the analysis is to define $\alpha_{av}$ — an average angle of blades cross-section center-line deviation from a radial direction. Next, basic
cylindrical cross-sections of prototype blade are separated layer by layer (figure 2(a)), rotated by rotation axis according to $\alpha_{av}$ in a way that the farthest from axis cross-section is rotated by $\alpha_{av}$ and all other are rotated proportionally to selected cross-section radius (figure 2(b)).

To reduce stresses in the impeller hub, decrease of a blade thickness is carried out by means of reducing thickness of profiles on the back part of a wheel (figure 2(c)). Model of a blade is designed considering turbine flow section and based on obtained basic profiles (figure 2(c)).

On the second stage, the wheel of initial impeller is combined with obtained blades. Coupling of elements is realized by means of a fillet with a radius of $1.4 \times 10^{-3}$ m. Then comparison of cross sections in initial blade and modified one is carried out. Profile analysis showed that they are partially deviated from radial direction in both sides. It can positively affect on centrifugal forces loads, because they will compensate each other.

Strength analysis in conditions of a nominal mode is conducted in designed turbine impeller to control the effectiveness of a suggested method. Minimal safety factor of a modified turbine impeller increased to 1.18. In this case the most loaded place is a blade joint with a wheel on a circumference. That’s why it’s offered to investigate and chose the most effective variant considering gasdynamic and strength parameters.

In this research practical measures are offered to reduce equivalent stresses in the most loaded areas: variant 1 with a radius of the fillet increased to $5 \times 10^{-3}$ m in joint of a blade and a wheel (figure 3(a)), variant 2 with a triangle support (figure 3(b)), variant 3 with a parabolic support (figure 3(d)). The foundation of suggested configuration variants is strengthening of a blade joint with a wheel in a circumference by means of supports of different configurations. The first variant has good strength parameters, but increasing of a fillet diameter leads to decrease of gasdynamic parameters of flow and efficiency more than 1.5% because of geometric impediment for gas fluid during its movement in a turbine flow section. The second variant allows to avoid changes in the flow section, but, due to fact that thickness of a wheel in circumference is limited, blade support in stress raiser has a small thickness and low value of safety factor.

To enhance blade support in the third variant, back blade surface was changed without any changes in configuration of a flow section in turbine. Radius of connection conic and flat back surfaces of blade are increased. In figure 3c a turbine blade is shown in a meridian section view of the impeller and also an area of strengthened material in the blade. Interface diameter from flat surface to conic one was increased without changes in an outlet blade edge.
Figure 3. Blade modifications and stress distribution in the modifications of turbine impeller: variant 1 with a radius of the fillet increased to $5 \times 10^{-3}$ m in joint of a blade and a wheel (a), variant 2 with a triangular support (b), enhancement of a blade back surface (c), variant 3 with a parabolic support (d).

Variant of a wheel joint modification, which is based on parabolic support, satisfies strength requirements and doesn’t affect on a turbine flow section.

4. Evaluation of gasdynamic efficiency in turbine impeller

On the last stage it’s necessary to evaluate an efficiency of a modified prototype impeller. To achieve that, using Ansys CFX (licence number 339001) research of gasdynamic parameters (gas flow rate, capability, efficiency) in a flow section considered in initial and modified impellers and comparison of obtained results are carried out.

Computational domain (figure 4) involves: inlet section 1, nozzle assembly 2, turbine computational domain 3, outlet section (to the recuperator) 4, recuperator section 5, the arrows show circular interfaces. Each domain consists of segment, which is closed in circumferential direction. Turbine computational domain doesn’t consider radial split between a blade and a stator, configuration of a blade joint with a wheel and a split from the side of the wheel.

The mathematical model for calculating the parameters of the stationary gas flow in the selected region is based on the following dependencies:

$$\nabla \cdot (\rho \mathbf{u} \times \mathbf{u}) = -\nabla p + \mu \nabla^2 \mathbf{u} + \frac{1}{3} \mu \nabla(\nabla \cdot \mathbf{u}) + \rho \mathbf{g}$$

(1)

$$\nabla \cdot (\rho \mathbf{u}) = 0$$

(2)

$$\text{div}(\rho c_p T \mathbf{u}) = \text{div}(\lambda \nabla T) + f$$

(3)

$$p = \rho RT$$

(4)

This is the Navier–Stokes equations (1), the continuity equation (2), the energy equation (3) and equation of state (4). They contain the value of $\mathbf{u}$ is the vector velocity of the working medium, $\mu$ is dynamic viscosity, $\rho$ – density, $p$ – pressure, $\mathbf{g}$ is vector of gravitational acceleration, $c_p$ is heat capacity at constant pressure, $T$ – temperature, $\lambda$ – thermal conductivity coefficient, $R$ is the gas constant, $f$ is a term of heat source. The mathematical model is supplemented by the equations of SST turbulence model.

Boundary conditions of this analysis are: inlet full pressure (inlet surface) is 350.8 kPa, full temperature is 1000K. On the outlet surface averaged static pressure is defined: 101.3 kPa. Computational turbine (domain 3) rotation speed is 96000 rpm.
High demands are imposed on the accuracy of the calculation, in particular, on the density of the finite element grid. A regular hexagonal finite element grid is used to solve this problem. The number of elements in the domains is chosen with a minimum loss of accuracy (less than 1% compared to the smaller grid). For the domain 3 no more than 1 million cells are enough.

The solution of the developed model was carried out using Ansys CFX. The convergence criterion is the minimum change of the total temperature on the output surface. As a result, distributions of pressure, velocities and temperature in the computational domain were achieved. Gasdynamic parameters in the flow section are presented in table 1 and also comparison of results of gasdynamic analysis of flow sections in the initial prototype and modified turbine impeller.

### Table 1. Comparison of gasdynamic parameters of flow sections in initial turbine and modified one.

| Parameter                           | Initial turbine | Modified turbine |
|-------------------------------------|-----------------|------------------|
|                                     | S2 inlet        | R3 outlet        | S2 inlet        | R3 outlet        |
| Gas flow rate (kg s⁻¹)              | 0.294           | 0.294            | 0.294           | 0.294            |
| Full temperature (K)                | 1100            | 839              | 1100            | 839              |
| Power (kW)                          | 91.8            | 91.7             |                 |                  |
| Ratio of expansion                  | 3.33            | 3.34             |                 |                  |
| Adiabatic efficiency in turbine impeller | 0.943          | 0.941            |                 |                  |
| Adiabatic efficiency in turbine stage | 0.923          | 0.922            |                 |                  |

Analysis of obtained parameters showed, that modified turbine has quite close parameters comparing with initial one’s. Because of a modification and a turn of blade inlet profile, there is an blow and jet separation on the inlet edge of blade. That’s why adiabatic efficiency slightly reduces (0,1-0,2%). But designed impeller provides initial parameters of gas flow rate, ratio of expansion and power and also satisfies strength requirements.

### 5. Validation of results

To verify and confirm the obtained calculated data on the basis of JIHT RAS, a stand of physical gasdynamic experiment was developed and installed [7]. The creation of such a gasdynamic stand allows for the implementation of a new technology for the study of impeller machines. This
technology includes tests of impeller machines in the form of models which can be manufactured quickly of cheap materials. This set of equipment will allow testing models of turbomachines of full-scale geometric dimensions and obtaining experimental data to verify all the calculated data and validation of the methods used.

6. Conclusion
Method of reducing stresses in a turbine impeller is developed. This method is tested by means of a prototype turbine impeller. Configuration of the prototype is modified in presented method: blade profile is estimated in case of deviation from radial profiles, a turn of blade profiles and development a new blade, design of a turbine impeller considering enhanced requirements related to high loaded places and strength analysis of prototype turbine impeller and modified one are carried out. In contrast to initial prototype, modified turbine impeller satisfies strength requirements in working mode of a micro-turbine. Results of gasdynamic analysis are: the main gasdynamic parameters and efficiency of modified turbine impeller have inconsiderable differences (less than 1%) from appropriate parameters of initial prototype. Due to modification, safety factor of turbine impeller increased in 30% without sacrificing of gasdynamic parameters. This method can be used not only in turbine impellers but in compressor axial-radial ones.

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