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Strength and gas dynamic methods of development of the axial turbine turbocharger

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Abstract. The results of strength and gas dynamic improvement of the turbocharger TK-32 axial turbine are presented. The turbocharger is manufactured by LLC “Penzadieselmash” (Penza, Russian) and is used as unit boost for a diesel locomotive. The aim of this work was to ensure the turbine work capacity when the rotor speed is increased by 10% without efficiency reduction. The strain-stress state analysis indicated the region of high stresses on the rotor blade body at the level of 2/3 of root. These stresses exceed allowable values when the rotor speed increased. The variant of tangential displacement of the peripheral rotor blade section, allowing reducing the level of stress by 20%, was found. Gas dynamic calculation showed that the variant of rotor blade modernization results in an increase of efficiency by 0.4%. Also it was shown that the increase in turbine efficiency by 1% can be reached if the number of rotor blades is reduced by 13%. This recommendation was implemented and confirmed experimentally by the example of mass turbocharger TK-32 [1].

1. Introduction

Turbocharger TK-32 (Figure 1) was developed at LLC “Penzadieselmash” [2] (Penza, Russian Federation) for use on diesel generator 1A-9DG manufactured by LLC “Kolomensky Zavod”. During turbocharger’s operation, there was a necessity of engine forcing. As a result, the turbocharger operating condition was modified. In particular, the rotor speed of the turbocharger increased from 25500 to 28000 rpm. In this regard, LLC “Penzadieselmash” applied for SSAU to assess the effect of the force on the stress strain state of turbine TK-32 and its gas-dynamic efficiency, and make recommendations for their improvement.

2. Gas dynamic calculation of the reference turbine

A three-dimensional computational model of the flow at the turbine stage, which includes a zone of the flow around the nozzle guide vane (NGV), a zone of the flow around the rotor wheel (RW) and a free flow area at the outlet of the turbine, was developed in Ansys CFX program [3]. This model was used for investigation of gas dynamic performances of the existing turbocharger’s axial turbine [4]. The flow models of NGV and RW contain only one blade passage for reducing required computer resources and calculation time. Therefore, the periodic boundary conditions were implemented on lateral boundaries of the computational domain (Figure 2).

A finite element mesh was created so that to provide a value of $y + n$ no more than three [5,6]. The total number of elements was 250000 in the NGV mesh, and 500000 elements - in the RW mesh. The
tip clearance was simulated when the RW mesh was created. The value of tip clearance was taken as 1 mm in accordance with the engineering drawing.

The following boundary conditions were set during calculation [7-9]:

- the mass flow rate \( G = 5.34 \text{ kg/s} \), the total temperature \( T^* = 773 \text{ K} \) and the flow direction (perpendicular to the face) were set at the computational domain’s inlet (NGV inlet);
- the outflow boundary was set as a constant adjustment of the flow static pressure \( p = 105000 \text{ Pa} \), the constant at all channel heights was set at the computational domain’s inlet (RW outlet);
- taking into account the RW rotation, this area was calculated in the rotating reference frame with rotor speed \( n = 25500 \text{ rpm} \) (nominal conditions), \( n = 28000 \text{ rpm} \) (forced mode);

The model of turbulence was \( \text{SST } k - \omega \). The calculation was performed in a stationary formulation. Flow parameters at the RW inlet and outlet were averaged in the circumferential direction (Mixing Plane approach).

\[ \text{Figure 1. Turbocharger TK-32.} \]

\[ \text{Figure 2. The computational model of the flow in the turbocharger TK-32 turbine.} \]
The flow pattern, as well as flow parameters at all points of the considered flow region in the nominal mode \( (n=25500 \text{ rpm}) \) and forced conditions \( (n=28000 \text{ rpm}) \) were obtained. Analysis of the flow structure in the turbine blade passage found no areas with an unfavourable flow pattern. Some flow parameters distribution fields in the turbine nominal mode \( (n=25500 \text{ rpm}) \) are given in Figures 3 and 4. The predicted value of turbine efficiency in this mode was \( \eta = 83.6\% \).

3. Turbine strength analysis

The pressure and temperature fields on blades surfaces from gas dynamic calculation were used as boundary conditions in turbine rotor wheel’s static strength calculation by means of the Ansys program [10]. The strength calculation model contains a complete rotor wheel, consisting of a disc, a blade footing and a blade body. Since the computational model had cyclic symmetry, the research was modelled only within the sector containing one blade. The periodic boundary condition was implemented on its lateral surfaces (Figure 5).

![Figure 3. The field of Mach number M of the value distribution in the absolute frame of reference throughout the turbine mid-diameter.](image)

![Figure 4. The field of static pressure distribution throughout the turbine mid-diameter.](image)

![Figure 5. The computational model for strength analysis of the turbocharger TK-32 turbine rotor wheel.](image)
The computational model was loaded with gas load (obtained earlier in the Ansys CFX program) and centrifugal forces. The disk temperature was adopted by the thermometry data provided by LLC "Penzadieselmash". Since the turbine disk is welded to the shaft, the RW calculation model was fixed by the front and rear flanges.

The computational model was divided by the Solid 185 and Solid 186 finite elements mesh. Special contact elements, limiting the movement of parts, were used in areas of fir-tree root teeth contact with the disk slot.

The stress-strain state was evaluated in two modes: nominal mode \((n=25500 \text{ rpm})\) and forced conditions \((n=28000 \text{ rpm})\).

The results obtained in the computation indicated that the reference turbine of turbocharger TK-32 satisfies the strength conditions in the nominal mode \((n = 25500 \text{ rpm})\) as a whole, but it should be noted that the derived load factors are dangerously close to the minimum value. The equivalent stress maximum value was 600 MPa in the forced mode \((n=28000 \text{ rpm})\), which corresponds to a load factor of 1.25. This value is below the allowable value (permissible value of 1.3). It was also revealed that there is plastic deformation in footing parts of the disc and the blade.

It is noteworthy that the maximum value of stresses is evidenced in the upper part of the blade body at the level of two thirds of the root (Figure 6), which indicates that stresses are caused by the blade bending. This conclusion is supported by the fact that compression stresses acts on the suction side area, located beyond the region of maximum stress. This conclusion is indirectly confirmed by the available in use instances of the upper third turbine blades shedding of turbocharger TK-32.

![Figure 6](image)

**Figure 6.** Normal stress distribution throughout the reference blade body at \(n=28000 \text{ rpm}\) (pressure side – on the left; suction side – on the right).

4. **Turbine modernization**

Elevated bending stresses are the result of a specific form of the rotor blade body. Its top sections are greatly expanded relatively bottom ones, violating sections centring by height. As a result, centrifugal forces acting on the periphery part cause an increased torque, bending the blade body. To reduce bending stresses, the peripheral sections of the blade can be “removed”. Hereinafter, the term "removal" means the displacement of blade body sections of the pen in the circumferential direction.

To reduce the bending stresses in turbocharger TK-32, the peripheral sections of the turbine rotor blade body have to be shifted in the circumferential direction toward the suction side.
The effect of three peripheral sections removal in the circumferential direction on the stress strain state of the rotor wheel blades was investigated. The variant allowing reducing the maximum value of stress to 506.8 MPa (18%) (the peripheral section shifted by the value of 0.05h towards the suction side) (Figure 7) in the forced mode, which corresponds to the factor load of 1.49 (Figure 8), was found. It should be noted that the derived value of the load factor in the forced mode (n = 28000 rpm) does not exceed the value of the load factor of the reference turbine version under nominal conditions (n = 25500 rpm).

The flow in the modernized turbine was investigated using Ansys CFX. It was found that the recommended variant of the peripheral sections slope in the mode with n = 25500 rpm increases the turbine efficiency by 0.4% (absolute).

Figure 7. The appearance of the modernized blade version.

Figure 8. Normal stress distribution throughout the modernized blade body at n=28000 rpm (pressure side – on the left; suction side – on the right).

5. **Blade attachment modernization and the number of blades selection**

An alternative larger typical size of the fir-three root for the plastic deformation in blade attachment’s elimination was selected. This, in turn, required a reduction of the number of blades from 49 to 43 by allocation on disk conditions. The effect of the number of rotor blades on the efficiency value was carried out in Ansys CFX in order to evaluate the impact of this decision on the turbine efficiency. The resulting dependence is shown in Figure 9. The number of nozzle blades was not changed.
From the diagram above it can be seen that the reduction of RW blade number increases the turbine efficiency by more than 1% for all versions of the blade body. It is related to the skin friction reduction, the decreasing number of edged wakes and a relative size of the secondary vortices reduction. The value of efficiency begins to drop again if the number of blades is more than 40 due to the torque on RW blades reduction.

It is noteworthy that the blade version with peripheral sections removals exceeds the base variant in the gas-dynamic efficiency.

Analysing Figure 9, it can be concluded that the number of RW blades decreases from initial 49 to 43...41, which does not worsen the gas dynamic turbine efficiency, but improves it up to 0.8...1.0%.

6. Conclusion
Eventually, the calculation research has found that the turbine rotor blade of turbocharger TK-32, manufactured by LLC "Penzadieselmash", will not strengthen conditions in case of forcing up to \( n = 28000 \) rpm. Design trouble spots are the blade bode and the blade attachment.

In the course of the research, it was found that stress in the blade can be significantly reduced through implementation of the three upper sections removal \((0,05h)\) in the circumferential direction towards the suction side, replacing the blade attachment with the attachment according to industry-specific standard (OST) 1.10975-81 with opening angle \( \varphi = 30^\circ \) and tooth pitch \( S = 3.2 \) mm, as well as reducing the number of RW blades to 43 units, while maintaining the number of nozzle vanes.

The recommended variant of the reference blade body modernization allows one to satisfy the strength conditions in all modes and to increase the turbine efficiency by 1%.

Currently, LLC "Penzadieselmash" are preparing the blade modernized in this way for manufacture. Moreover, the turbocharger with number of rotor blades reduced to 43 was made and tested. The blade attachment and blade body form are former. Experimental results showed an increase in the turbine efficiency by 1%, which fully confirms the conclusions drawn by the authors.

![Figure 9. The turbine efficiency of turbocharger TK-32 dependence of the number of RW on the constant number of NGV blades.](image)

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