The Results of Gas Dynamic and Strength Improvement of Turbocharger TK-32 Axial Turbine

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Abstract: The results of strength and gas dynamic improvement of turbocharger TK-32 axial turbine are presented. Turbocharger was manufactured by LLC "Penzadieselmash" (Penza, Russian) and is used as unit boost for diesel locomotive. The goal of this work was to ensure turbine work capacity when rotor speed is increased by 10% without efficiency reduction. The strain-stress state analysis indicated the region of high stresses on rotor blade body at the level of 2/3 of root. These stresses exceed allowable values when rotor speed is increased. The variant of peripheral rotor blade section tangential displacement, allowing to reduce the level of stresses by 20%, was found. Gas dynamic calculation showed that 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 on a mass turbocharger TK-32.

1 INTRODUCTION

Turbocharger TK-32 (Figure 1) was developed at LLC "Penzadieselmash" (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 revolutions per minute (rpm). In this regard LLC "Penzadieselmash" applied for SSAU to assess the forcing effect on the stress strain state of the turbine TK-32 and its gas-dynamic efficiency, and make recommendations for their improvement (Tikhonov, Matveev, 1982).

2 GAS DYNAMIC CALCULATION OF TURBINE BASIC DESIGN

Three-dimensional computational model of the flow in the turbine stage, which includes zone of flow around the nozzle guide vane (NGV), zone of flow around the rotor wheel (RW) and free flow area at the outlet of the turbine, was developed in *Ansys CFX* program (ANSYS - Simulation Driven Product Development, 2014). This model was used for investigation of gas dynamic performances of the existing turbocharger's axial turbine. The flow models of NGV and RW contain only one blade passage for reducing required computer resources and calculation time (Bonh, Heuer, Kusterer, 2005). Therefore, the periodic boundary conditions were implemented on lateral boundaries of the computational domain (Figure 2).

Finite element mesh was created such as to provide a value of y + no more than three. The total number of elements was 250000 in the NGV mesh, and 500000 elements in RW mesh. Tip clearance was simulated when the RW mesh were 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:

- mass flow rate (G = 5,34 kg/s), total temperature (T* = 773 K) and flow direction (perpendicular to the face) were set at the computational domain's inlet (NGV inlet);

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- outflow boundary was set constant adjustment of the flow static pressure (p = 105000 Pa) constant at all channel height was set at the computational domain's inlet (RW outlet);
- to account for the RW rotation, this area was calculated in the rotating reference frame with rotor speed n = 25500 rpm (nominal conditions), n = 28000 rpm (forced mode);

The model of turbulence was *SST k- ω*. The calculation was performed in a stationary formulation (Bochkarev, Dmitriev, Kulagin, Makeenko, Mosoulin, Mossoulin, 1993). Flow parameters at RW inlet and outlet were averaged in the circumferential direction (*Mixing Plane approach*).

The flow pattern, as well as flow parameters in all points of considered flow region at nominal mode *(n=25500 rpm)* and forced conditions (*n=28000 rpm*) were obtained. Analysis of the flow structure in the turbine blade passage found no areas with unfavorable flow pattern. Some flow parameters distribution fields at turbine nominal mode *(n=25500 rpm)* are given at Figure 3 and 4. Predicted value of turbine efficiency at this mode was η_m*=83,6%*.

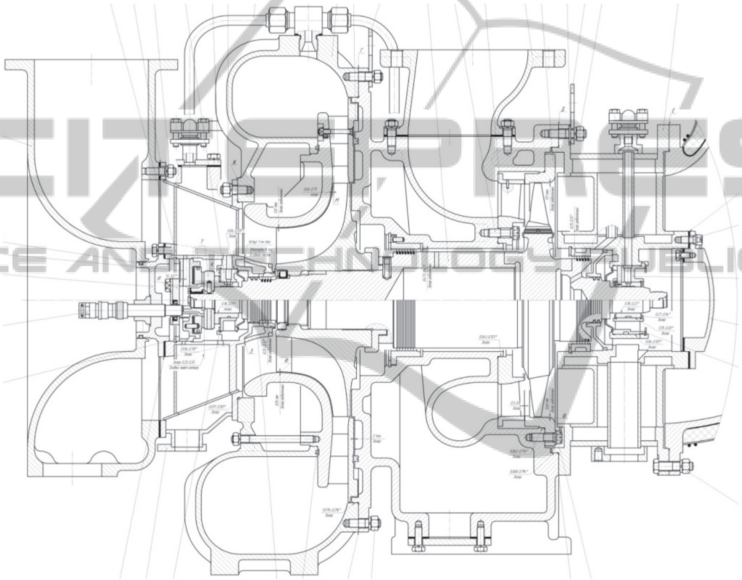

Figure 1: Turbocharger TK-32.

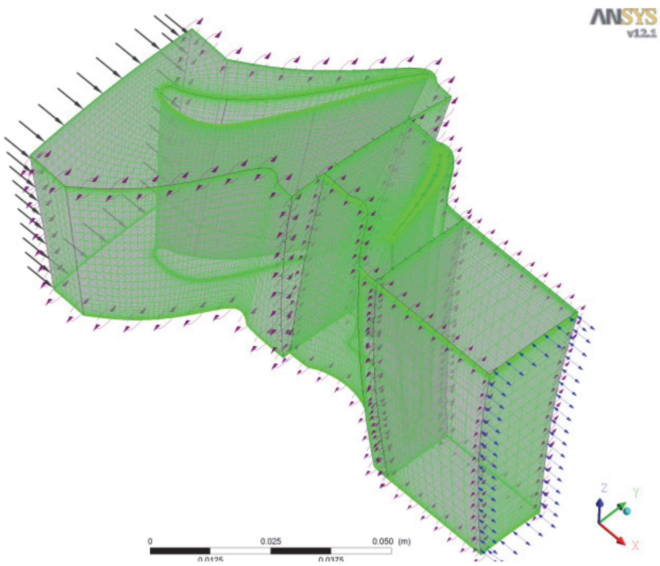

Figure 2: Computational model of the flow in turbocharger TK-32 turbine.

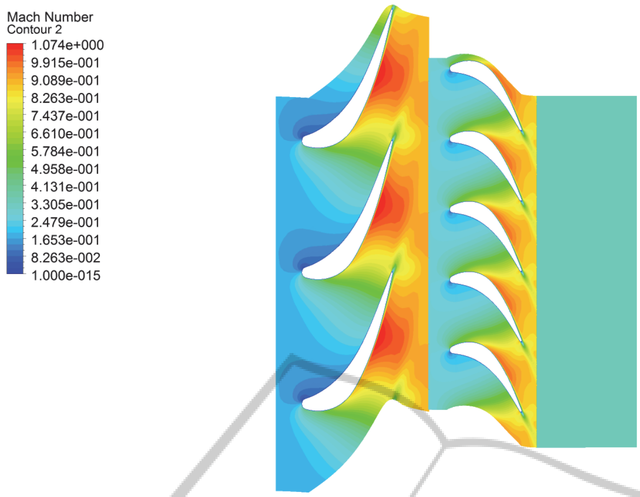

Figure 3: The field of Mach number's value distribution in absolute reference frame at turbine middle diameter.

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Figure 4: The field of static pressure distribution at turbine middle diameter.

3 THE STRENGTH ANALYSIS OF TURBINE

The pressure and temperature fields at blades surfaces obtained from gas dynamic calculation were used as boundary conditions in turbine rotor wheel's static strength calculation by means of *Ansys Mechanical* program. The modal for strength calculation contained a whole rotor wheel, consisting of a disc, blade attachment and blade aerofoil. Since the computational model had cyclic symmetry, then only the sector containing one blade was modelled during the research. The periodic boundary condition was implemented on its lateral surfaces (Figure 5).

The computational model was loaded with gas (obtained earlier in the Ansys CFX program) and centrifugal forces. 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 mesh of Solid 185 and Solid 186 finite elements. Special contact elements, limiting the movement of parts, were used in areas of fir-tree root teeth contact with disk slot.

Stress-strain state was evaluated at two modes: nominal mode (n = 25500 rpm) and forced conditions (n = 28000 rpm).

Figure 5: Computational model for turbocharger TK-32 turbine rotor wheel's strength analysis.

The results obtained in the computation indicated that turbocharger's basic turbine satisfies the strength conditions at the nominal mode (n = 25500 rpm) as a whole. However it should be noted that the derived load factors are dangerously close to the minimum value. Equivalent stress maximum value was 600 MPa at forced mode (n=28000 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 disc and blade.

Noteworthy is the fact that the maximum value of stresses can be evidenced in the upper part of the blade body at the level of two thirds from 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 cases of the upper third turbine blades shedding of the turbocharger available in use.

4 THE MODERNIZATION OF TURBINE

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 the sections centring on height. As a result, centrifugal forces acting on the periphery part, cause the increased torque, that bends the blade body.

To reduce bending stresses peripheral sections of the blade have to be "shifted". Hereinafter the term "shift" means the displacement of blade body sections of the pen in the circumferential direction.

To reduce the bending stresses in turbocharger TK-32 turbine rotor blade body peripheral sections have to be shifted in the circumferential direction toward the suction side.

The effect of shift of three peripheral sections in circumferential direction on the rotor wheel blades' stress strain state was investigated. The variant allowing to reduce the maximum value of stress up to 506.8 MPa (18%) (peripheral section shifted by the value of 0.05h towards the suction side) (Figure 7) at forced mode, which corresponds to the factor load 1.49 (Figure 8), was found. It should be noted that derived value of the load factor at the forced mode (n = 28000 rpm) does not exceed the value of the basic turbine version load factor at nominal conditions (n = 25500 rpm). The flow in the modernized turbine was investigated using *Ansys CFX*.

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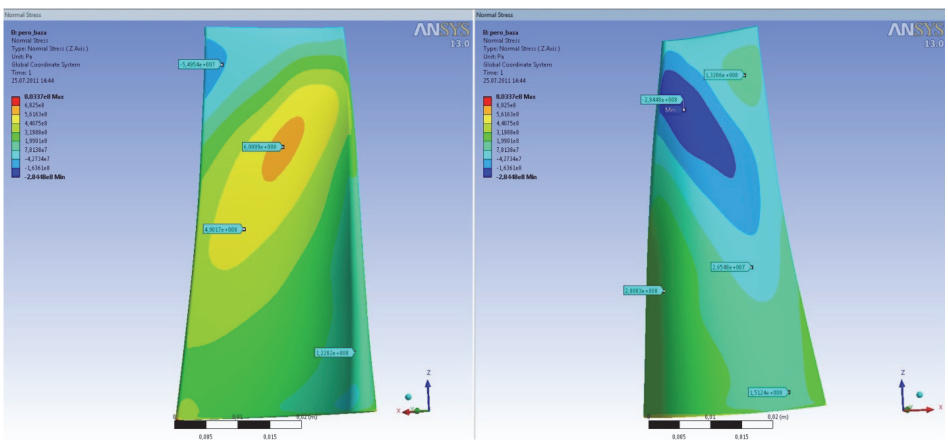

Figure 6: Normal stress distribution on the basic blade body at n=28000 rpm (pressure side – on the left; suction side – on the right).

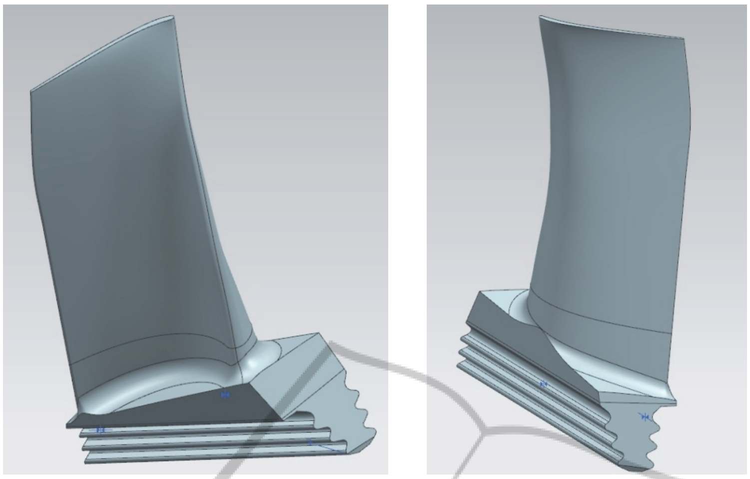

Figure 7: The appearance of modernized blade version.

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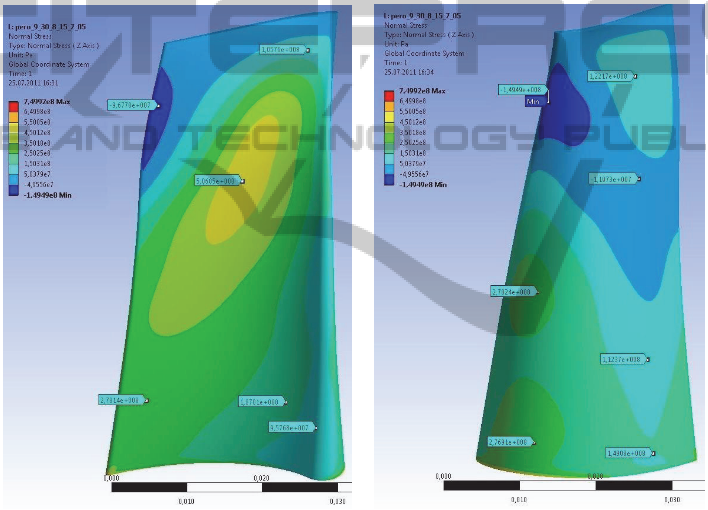

Figure 8: Normal stress distribution on the modernized blade body at n=28000 rpm (pressure side – on the left; suction side – on the right).

It was found that recommended variant of the peripheral sections slope at the mode with $n = 25500$ *rpm* increases the turbine efficiency by *0.4%* (absolute).Modernization of Blade Attachment and Blade Number Selection.

Alternative larger typical size of fir-tree root for the elimination of plastic deformation in blade attachment’s was selected. This in turn required reduction the number of blades from *49 to 43* by allocation on disk conditions.

The effect of rotor blades number 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 guide vanes was not changed.

From the Figure it can be seen that the reduction of RW blade number increases the turbine efficiency by more than 1% for all versions of blade body. It is related to the reduction of skin friction, number of edged wakes decreasing and relative size of the secondary vortices reduction. The value of efficiency begins to drop again if the number of blades more than 40 due to the reduction of torque on RW blades.

Noteworthy the fact that blade version with peripheral sections shift exceeds the base variant in the gas-dynamic efficiency.

Analysing the diagram in Figure 9 it can be concluded that the decrease of RW blades number from initial *49* to *43...41* does not worsens the gas dynamic turbine efficiency, but also improves it to *0.8...1.0%.*

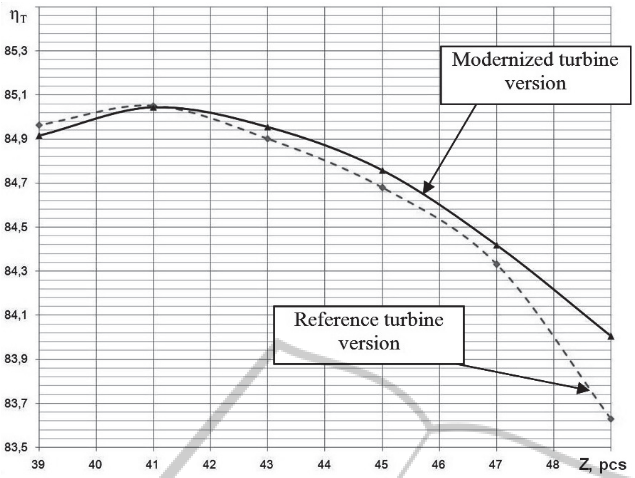

Figure 9: The turbine efficiency from number of RW blades dependence with constant number of NGV blades (dashed line – basic blade version; solid line – blade with sections removal).

5 CONCLUSIONS

As the result of calculation research it was found that turbine rotor blade of turbocharger TK-32, manufactured by LLC "Penzadieselmash", will not meet the strength conditions in case of engine forcing up to *n = 28000 rpm*. The trouble spots of the design are the blade bode and blade attachment.

In the course of the research it was found that stress in the blade can be significantly reduced through shifting of the three upper sections on *0,05h* in the circumferential direction towards the suction side, replacing the blade attachment to another from industry-specific standard (OST 1.10975-81) with an opening angle φ = 30 °, tooth pitch S = 3.2 mm, as well as reducing the number of RW blades to 43 units, while maintaining the number of NGV.

The recommended variant of the basic blade body modernization allows to satisfy the strength conditions at all modes and to increase turbine efficiency by *1%.*

Turbocharger with number of rotor blades *43* was made and tested. The blade attachment and blade body form was basic. The experiment showed an increase in turbine efficiency by *1%*, which fully confirms the conclusions drawn by the authors.

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