Influence of temperature and geometric parameters of elements in a turbocompressor seal assembly on its operability

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Abstract. Currently, most power units using internal combustion engines are equipped with turbocharging and that is explained by the requirements for efficiency, environmental safety and specific power characteristics. At the same time, the turbocompressor (TCR) is one of the resource-determining elements of power units, which accounts for a significant number of failures. Studies conducted towards the analysis of turbocompressor failures have shown that the main reason for it lies in a malfunction of the seal assembly, which is a consequence of extremely unfavorable conditions for its operation, consisting in a high (up to 800 °С) exhaust temperature, aggressive environment, a high (up to 120,000 min\textsuperscript{-1}) rotor speed, the sliding speeds of the mating parts, up to 150 m/s. Oil for lubrication and cooling of the TCR is supplied through floating plain bearings, while the cavities of the turbine and compressor are isolated by means of the seal assemblies. Violation of their operability leads to the ingress of oil in a cavity of the impellers, to the imbalance of a rotor and, ultimately, to a turbocompressor failure. The determining factor affecting the operability of the seal assembly is a heat density of its parts. This factor is especially manifested in operating under forced conditions and engine shutdown, when cooling of a turbine volute is stopped by blown air, and a bearing assembly is stopped by circulating oil. In this case, a large part of the thermal energy is transferred from a heated turbine volute to the seal assembly and to the bearing assembly. The focus is on the temperature influence on performance of the turbocompressors’ sealing rings.

1. General information about turbocompressors usage

At present, the production of atmospheric diesels comprises not more than 40 % from the total volume of all production and the steady trend is being observed to its reduction at the world market. For trucks, building and agricultural machinery, the most widely used diesel engines are Kamins, Caterpillar (USA); Volvo, Scania (Sweden); Deutz, MAN (Germany). In Russia, the main diesel producers of this group are KAMAZ and YaMZ. At the same time, the turbocompressor (TCR), being the main unit of the boost system, significantly affects the equipment downtime, a cost of spare parts, repair and reliability of a power unit as a whole. In this regard, ensuring the reliability of a turbocompressor and developing the cost-effective repair technologies are the important tasks.
2. Problem and hypothesis statement

TCR of diesel engines [1, 2] (Fig. 1) operate under extremely adverse conditions, which include high (up to 800 °C) exhaust gas temperatures, aggressive environment, and a high (up to 120,000 min⁻¹) rotor speed, sliding speeds of the mating parts, up to 150 m/s.

![Figure 1](image1.png)

**Figure 1.** General view of a turbocompressor: 1 – a seal cover; 2 – a sleeve of a rotor shaft; 3 – a rotor shaft; 4 – a compressor wheel; 5 – sealing rings; 6 – a turbine wheel; 7 – an oil deflector; 8 – compressor housing; 9 – the medium body; 10 – turbine housing; 11 – an insert compressor; 12 – a bearing

One of the critical components of the TCR, affecting its operating cycle, is the lubrication system seal assembly. Severe temperature operating conditions of the seal assembly (especially from the turbine side) significantly affect the reliability of the turbocompressor and the power unit as a whole, where failures due to the TCR reach to 25.6 % (Fig. 2). At the same time, subject to operation requirements and timely performance of scheduled maintenance of the power unit, new TCRs, as a rule, satisfactorily fulfill the envisaged resource. The TCRs that underwent repair impacts, often even in the initial period of operation, demonstrate increased oil consumption, and the mean time between failures, as a rule, does not exceed 50 % of the operating time of a new turbocompressor.

A detailed analysis of failure causes of both new and restored TCRs is presented in [1]. The studies results reflecting the TCR failures are presented in Figure 3. As can be seen from the diagram, the main cause of failures is increased oil consumption, which is a consequence of the seal assembly poor operation. The reason for the wear of the turbine wheels and the compressor, as a rule, is a violation of the balancing of these parts due to carbon formation (in case of loss of tightness of the seal) and the presence of some solid particles from outside. The same reason leads to a break in the weld of the rotor shaft. Thus, up to 88 %, the reason for all turbocompressor failures is poor operation of the seal assembly.

To understand the reasons for this, we examine the seal assembly in detail. The oil seal of most modern turbocompressor of automotive engines is combined, which unites in itself the contact and labyrinth seals, formed respectively by the outer surface of the sealing ring and the housing, the ends of the seal and the walls of the groove (Figure 4). At the same time, the key role in this seal is played by elastic split metal rings of rectangular cross-section, installed in the grooves of the rotor bushings on the compressor and turbine sides. The effectiveness of this seal is most affected by the amount of clearance in the ring-groove mate, the elasticity of the seal ring, and the amount of thermal clearance in the seal ring.
Based on the analysis of TCR failures, the design of the seal assembly and its elements, and also taking into account that the majority of TCR failures occur suddenly, bypassing the stage of increasing manifestations of the malfunction, the authors made the assumption that the malfunction occurs due to the loss of elasticity of the sealing rings [3] and contact with a female part. Accordingly, a working hypothesis has been put forward and it states the following: the performance of the seal assembly is violated when extreme temperature conditions occur as a result of violations of the operating requirements, and can be caused by deformations of the sealing rings that extend beyond the elasticity range.

3. Mathematical apparatus for modeling temperature effect on details of seal assembly
In the framework of the hypothesis that has been put forward, the temperature fields of the extreme ring on the turbine side of the sealing ring were first theoretically determined as being most susceptible to temperature effects from the exhaust gases. To determine the temperature fields in this ring, we propose the following technique.
The exhaust gases in the turbine cavity have the property of internal friction, the tangential stress of which obeys the following law:

\[ \tau = \mu \left( \frac{du}{dn} \right), \]  

(1)

where \( \mu \) is the dynamic coefficient of viscosity; \( \frac{du}{dn} \) – a normal flow velocity gradient.

As you know, any gas flow (both laminar and turbulent) along the channel walls has a boundary layer, the flow rate in which tends to 0 as it approaches the walls. A similar situation arises in the formation of boundary layers. We should note that turbulent gas motion in heat conduction problems can be considered as internal friction, the tangential stress of which obeys the above law.

The temperature in this layer varies from the temperature of the boundary (wall) to the temperature of the main gas stream. The thickness of this layer is determined by the thermophysical properties of gas \([4]\). The laminar motion in the general case is described by the Navier-Stokes equation:

\[
\frac{D}{Dt} u_x = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \nu \left( \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2} \right),
\]

\[
\frac{D}{Dt} u_y = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \nu \left( \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial z^2} \right),
\]

\[
\frac{D}{Dt} u_z = -\frac{1}{\rho} \frac{\partial P}{\partial z} + \nu \left( \frac{\partial^2 u_z}{\partial x^2} + \frac{\partial^2 u_z}{\partial y^2} + \frac{\partial^2 u_z}{\partial z^2} \right),
\]

(2)

where: \( \frac{D u_x}{Dt}, \frac{D u_y}{Dt}, \frac{D u_z}{Dt} \) are the substantial derivatives of the velocity with respect to the coordinates of the space \( x, y, z \); \( \rho \) is the density of the liquid or gas in the stream; \( P \) is the pressure; \( \nu = \frac{\mu}{\rho} \) – the kinematic coefficient of viscosity of liquid (gas).

The system of equations 2, which describes the distribution of the flow velocities of a working fluid, is reduced (i.e., without viscous terms) to the system of the Euler equations of potential (in this case, friction creates vortex formation).

The system of equations 2, which describes the distribution of the flow velocities of a working fluid (gas or liquid), is reduced (i.e., without viscous terms) to the system of the Euler equations of potential motion.

It should be noted that turbulent gas motion in heat conduction problems can be considered as potential precisely because of its turbulence.

The temperature distribution is described by the Fourier-Kirchhoff equation:

\[
\frac{\partial T}{\partial t} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + \frac{\mu}{\rho \varepsilon_p} \cdot \Phi,
\]

(4)

where \( DT/Dt \) – substantial derivative temperature; \( \Phi \) is the Rayleigh dissipative function.

Since for most gases and liquids at moderate speeds the dispersive Rayleigh function is small, the energy equation 4 reduces to the following form:

\[
\frac{\partial T}{\partial t} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right),
\]

(5)

In case of stability loss, the laminar boundary layer can become turbulent. The condition for the existence of laminar motion on the walls is
\[
\frac{d}{dz} \cdot \left( \mu \cdot \frac{du_x}{dz} \right)_{z=0} < 0
\]  
(6)

where \(\frac{du_x}{dz}_{z=0}\) is the velocity gradient of the layer on the wall.

In this case, the stability of the layer is determined from the following expression:

It is obvious that the layer is stable at

\[
\frac{\partial^2 u_x}{\partial z^2} \bigg|_{z=0} < 0, \quad \text{i.e. } \frac{dT}{dz} > 0 \text{ at } \frac{\partial^2 u_x}{\partial z^2} \bigg|_{z=0} > 0.
\]  
(7)

Thus, if the gas in the turbine is cooled (it gives energy to the walls of the casing, to the parts of the rotor, including the rings and the turbine), then a laminar boundary layer is stored on the walls of these parts.

Using equations 6 and 7 and formulating the boundary conditions, we can find the distribution of gas flow velocities in the laminar (boundary) layer of the working fluid flow and calculate the temperature distribution in it, which, in turn, will allow, based on Fourier hypothesis, to determine the heat flux density given (perceived) by the walls of the parts from the working fluid (in our case, exhaust gas):

\[
q = -\lambda_f \cdot \left( \frac{dT}{dn} \right)_{n=0},
\]  
(8)

where \(\lambda_f\) is the coefficient of thermal conductivity of the working fluid (gas); \(\left( \frac{dT}{dn} \right)_{n=0}\) – temperature gradient in the boundary layer at the boundary with the wall.

Knowing the heat flux \(q\) and the temperatures \(T_\omega\) and the working fluid (exhaust gases) outside the boundary layer \(T_f\) allows to find the heat transfer coefficient introduced by I. Newton:

\[
\alpha = \frac{q}{T_f - T_\omega}.
\]  
(9)

The heat transfer coefficient, being a scalar value, is not a physical parameter; it is very convenient for generalizing empirical and calculated dependencies.

Given that the problem of determining, ultimately, the temperatures of specific parts of a turbocompressor is quasistatic, which allows to neglect the local derivatives of speed and temperature, and the boundary conditions (temperatures of the working fluid and body parts and gas pressure in the separated cavities are known from experiments), the solution for the task to be stated seems quite possible.

As a result of these considerations, we have obtained a mathematical model of the temperature distribution over the volume of parts, which allows to carry out some computational experiments with the operation of turbocompressor parts in various modes.

4. Computational experiment results
Calculations of the outermost ring temperature from the turbine side have shown (Fig. 5) that a significant temperature of the material of the body of the ring during operation is in the range 120...620°C. The highest temperatures are observed from the turbine side and along the surface of the ring lock.

Such an uneven temperature distribution can not but cause the ring deformation, the calculation results are presented in Fig. 5.

The analysis on the calculation results addressed the temperature deformations, which are presented in Figure 5, shows that the ring deformations reach 0.017 mm, which must be taken into account when designing the TCR geometric elements.

Taking into account the temperature deformation of the ring, in order to prevent “setting” and rotation of the rings with the sleeve, it is necessary to provide a gap of at least 0.1 mm across the width of the ring in the groove. In addition, for some manufacturers, the minimum value of the thermal gap in the lock of the ring is 0 with a positive upper deviation of, (Fig. 5.d) leads to a joint, which causes the additional force in this zone. The results relating to the stress calculations arising from this interaction
are presented in Fig. 6, and the results of calculating the safety factor for yield strength are shown in Fig. 7.

**Figure 5.** Temperature distribution in the sealing ring, the extreme one from the turbine side: (a) – view from the turbine side; (b) – view from the side of the central bearing; (c) – cross section of the ring; (d) – the section along the ring

**Figure 6.** Temperature distribution in the sealing ring, the extreme one from the turbine side: (a) – view from the turbine side; (b) – view from the side of the central bearing; (c) – cross section of the ring; (d) – the section along the ring
The stress data in the lock's zone, as suggested in the hypothesis, go beyond the yield strength, the safety factor for this indicator is significantly less than 1 (Fig. 7), which leads to the loss of operability of at least the extreme sealing ring on the turbine side.

5. Analysis of calculation results, recommendations problem solution and conclusion

As can be seen from Figures 5–7, the main reason for the malfunction of the seal assembly is the absence of a thermal gap in the ring lock under the influence of temperature, which leads to unnormalized excess stress and deformations beyond the yield strength of the material. As one of the ways to solve this problem, the authors propose to significantly increase the thermal gap in the lock to a value exceeding the value of thermal deformation, i.e., according to the above calculations, to a value of 0.15 mm for the TCR under consideration, at least for a ring extreme side of the turbine.

The calculation results towards deformations, stress, and safety factor for yield strength are presented in Fig. 8.
Figure 8. The calculation results (a) – deformations; (b) – voltages; (c) – tensile strength by yield strength

The calculation results on deformations, voltages and the safety factor for the yield strength of the TCR-7 turbocompressor, O-ring of the extreme side of the turbine, having a thermal gap of 0.15 mm, showed that this thermal gap allows avoiding a “joint” even under the most adverse conditions of the TCR operation in the lock, which, in turn, eliminates the deformation of the ring beyond the yield strength, and, therefore, allows to maintain the health of the sealing ring and the seal assembly as a whole.

Given that an increase in the lock in the free state entails an increase in oil consumption, it is advisable to use rings with the size of the lock in the free state proposed by the authors only for the extreme rings from the turbine side. At the same time, it is possible not to manufacture special rings, but to use selective assembly when picking the TCP.

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