Gas Turbine Driven GPU Diagnostics by the Gas Path Parameters

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Abstract. The issue of diagnostics of a gas turbine engine path is mainly related to the complexity of direct measurements inside it under operating conditions. The spaces between the axial compressor stages and the turbine expansion stages are small, characterized by an intense gas flow (for the turbine, also high temperatures), and vane wheels are rapidly rotating in close proximity to them. Therefore, installation, routine operational testing, and replacing sensors are difficult, moreover, any violation of body part integrity entails a change in stiffness and, as a result, the risk of destruction from vibrations.

In this case, the mathematical simulation techniques gain special relevance, which provide the missing parameters required to estimate the gas path conditions as a result of calculations and allow the diagnostics–forecasting the development of destructions or the consequences of malfunctions. The paper represents one of such techniques and provides data on its application for diagnostics of a gas turbine engine–a GPU (gas pumping unit) drive with a free turbine.

1. Introduction

Gas turbine drives, as a rule, derived from aeroengines and adapted to the ground operating conditions, are currently used everywhere to drive gas pumping units. This drive type is especially relevant for areas where connecting power lines to install electric drives is difficult.

Fig. 1 shows a typical diagram of a gas turbine engine (GTE) operating on natural gas and driving a main centrifugal compressor to transport gas. Such an engine has 14 compression stages in an axial compressor, two expansion stages in a power turbine rigidly connected by a shaft with an axial compressor, an annular combustion chamber, and a three-stage free turbine transmitting torque to the compressor through a coupling or multiplier [1, 2, 3, 5].
2. The study relevance
The principle of diagnostics of such a gas turbine unit (GTU) is based on the gas path or
thermogasdynamic parameters and in the best case, should be based on estimating dimensionless
parameters (which depend less on the types of fuel and its injection into the combustion chamber,
operating mode, and ambient humidity-temperature conditions) [1, 2, 4, 5].

3. Research objective
Since diagnosing the current state of a GTU as part of a GPU by dismantling control (shutting down the unit) is impossible and requires a vibration diagnostics expert to be invited for a detailed unit examination, a technique should be developed that will allow determining the GTU gas/air path conditions with sufficient accuracy using the readings of standard instruments and some mathematical apparatus [1, 6, 7].

4. Theoretical
Research technique. The engine ACS (automated control system) controls its main gas-dynamic parameters such as shaft speed, air temperature at the engine and the combustion chamber inlets and the exit of the turbine rotating the compressor, and fuel consumption. balance

As a basis, the power balance (compressor and turbine) in steady-state operating modes is used. However, the listed parameters depend on the GPU operating mode and therefore, are not informative enough to diagnose the technical conditions of this unit. The technique for studying the possibility of more accurate diagnostics comprises calculating (on their basis) dimensionless parameters or coefficients depending less on the GPU operating mode. Such coefficients can be got only by mathematical simulation of the processes occurring in the flow path of the converted aeroengine.

The main mathematical simulation stages include determining the below parameters, not directly measured but necessary ones [3, 8, 9].
5. GTE compressor shaft power

Let us denote the axial components of the flow velocity in the air path (averaging the velocity over the flow area—hydraulic approximation is applied) between the axial compressor variable guide vane (VGV) and its impeller as \(v_1\), and the axial component of the air flow velocity at the impeller exit as \(v_2\), then the power consumed at one compression stage of an axial compressor:

\[
N_i^K = \omega R_i^K Q(\omega R_i^K - v_1 t g \alpha_i^K - v_2 t g \beta_i^K),
\]

Where \(Q\) is the mass air flow rate at the compressor stage; \(i\) is the stage sequential number; \(v_1\) is the air velocity at the impeller inlet; \(v_2\) is the air velocity at the impeller outlet; \(R_i^K\) is the average impeller radius along the vane height; \(\omega\) is the angular velocity; \(t g \alpha_i^K\) is the tangent of the angle of the VGV trailing edge at the i-th stage inlet; \(t g \beta_i^K\) is the tangent of the angle of the impeller vane trailing edge (angles to the rotation axis) [10, 12].

Since \(Q = \rho_i v_1 S_i^K = \rho_{i+1} v_2 S_{i+1}^K\), where \(\rho_i\) is the air density at the i-th compression stage exit; \(S_i^K\) is the flow area at the i-th compression stage exit, then transforming the power equation, we get:

\[
N_i^K = \omega R_i^K Q \left( \omega R_i^K - \frac{q \gamma g \alpha_i^K}{s_i^K \rho_i} - \frac{q \gamma g \beta_i^K}{s_{i+1}^K \rho_{i+1}} \right);
\]

According to the condition of almost adiabatic air compression in an axial compressor: \(\frac{T_i^K}{\rho_i^{-1}} = \frac{T_{i+1}^K}{\rho_{i+1}^{-1}}\), where \(\gamma=1.4\) is the adiabatic index for air; \(T_i^K\) is the air temperature at the i-th compression stage exit. In this case, the air pressure at the compressor inlet is adopted as \(P_0 = \frac{R}{\mu_a} \rho_0 T_0\), where \(R\) is the universal gas constant; \(\mu_a\) is the molar mass of air; \(\rho_0\) is the air density at the compressor inlet; \(T_0\) is the air temperature at the compressor inlet.

Let us denote the temperature growth at the compression stage as \(\Delta T^K\) (it is conditionally considered the same for all stages) [4, 10, 11, 13].

Based on the adiabatic compression condition, let us find the air density along the engine air path (L is the number of compression stages) using the chain of equations:

\[
T_i^K - T_{i+1}^K = \Delta T^K, \quad \frac{T_i^K}{\rho_i^{-1}} = \frac{T_{i+1}^K}{\rho_{i+1}^{-1}} \Rightarrow \rho_{i+1} = \rho_i \sqrt{\frac{T_i^K + \Delta T^K}{T_i^K}}; \quad i = 0,1,\ldots,L - 1
\]

Then, the equation for the heat balance inside the combustion chamber is used: \(Q T_i^K + q \lambda = (Q + q) T_i^{KC}\), where \(T_i^K\) is the air temperature at the compressor exit (at the entrance to the combustion chamber); \(q\) is the fuel consumption; \(T_i^{KC}\) is the gas temperature in the combustion chamber (average); \(\lambda\) is the fuel calorific value [17, 19].

6. GTE turbine shaft power

Let us denote the axial components of the gas velocity in the path after the combustion chamber between the axial turbine nozzle guide vane and impeller as \(v_1\), and the axial component of the gas flow velocity at the impeller exit as \(v_2\), then the power at one turbine impeller:

\[
N_i^T = \omega R_i^T Q(v_1 t g \alpha_i^T + v_2 t g \beta_i^T - \omega R_i^T)
\]

Where \(Q\) is the mass gas flow rate in the turbine; \(i\) is the turbine stage sequential number; \(v_1\) is the gas velocity at the impeller inlet; \(v_2\) is the gas velocity at the impeller outlet; \(R_i^T\) is the average turbine impeller radius along the vane height; \(\omega\) is the angular velocity; \(t g \alpha_i^T\) is the tangent of the angle of the
i-th stage nozzle guide vane trailing edge; \( t g \beta_i^T \) is the tangent of the angle of the i-th stage impeller vane trailing edge (angles to the rotation axis) [14, 15].

Since \( Q = \rho_i v_i S_i^T = \rho_{i+1} v_{i+1} S_{i+1}^T \), where \( \rho_i \) is the gas density at the i-th stage; \( S_i^T \) is the flow area at the i-th stage, then the turbine stage power will be:

\[
N_i^T = \omega R_i^T Q \left( \frac{Qtg \alpha_i^T}{S_i^T \rho_i} + \frac{Qtg \beta_i^T}{\rho_{i+1} S_{i+1}^T} - \omega R_i^T \right)
\]

According to the condition of adiabatic gas expansion in the turbine: \( \frac{T_i^T}{\rho_i^T} = \frac{T_{i+1}^T}{\rho_{i+1}^T} \), where \( \delta = 1.3 \) is the adiabatic index for the gas obtained by mixing air, the oxygen of which is used for combustion, with the products of this combustion; \( T_i^T \) is the gas temperature at the i-th stage turbine exit. In this case, the gas pressure at the turbine inlet is adopted as \( P_0^i = \frac{R}{\mu_g} \rho_0 T_0^i \), where \( \mu_g \) is the molar mass of gas; \( \rho_0 \) is the gas density at the turbine inlet (despite the coincidence of density denotations, it is clear that this is not the same as for the compressor); \( T_0^i = T_0^{i+1} \) is the gas temperature at the turbine inlet. Then, using the chain of equations:

\[
\frac{T_i^T}{\rho_i^T} = \frac{T_{i+1}^T - \Delta T^T}{\rho_{i+1}^T} \Rightarrow \rho_{i+1} = \rho_i \sqrt{\frac{T_i^T}{T_{i+1}^T}} \delta^{-1} ; \quad i = 0,1,\ldots,M - 1,
\]

where: \( \Delta T^T \) is the temperature drop at the turbine impeller (it is considered the same for all stages), \( M \) is the number of expansion stages [14, 20].

Thus, considering the equality of the shaft powers (compressor and turbine),

\[
N = \sum_{i=1}^{L} N_i^k = \sum_{j=1}^{M} N_j^T,
\]

conditional (diagnostic) efficiency of the working compressor and turbine are determined, respectively:

\[
\eta^k = \frac{\frac{R}{\mu_a} N \gamma}{N \gamma - 1} \quad \text{and} \quad \eta^T = \frac{\mu_g N}{R M \Delta T^T Q} \delta^{-1}
\]

‘Conditional’ means that they are physically defined similarly to polytropic efficiency values but may largely differ from them due to simplifications (mathematical model assumptions). These parameters are the basic ones for diagnostics (their systematic changes allow concluding about the development of malfunctions in the gas turbine engine path).

It is now required to clarify which parameters are initially known (constitutive for the engine and physically constant), which ones can be measured during operation by sensors of the engine monitoring and control system (operating - current), and which parameters can be found using a mathematical model (identification - diagnostic). Thus, pressure \( P_0 \) and temperature \( T_0 \) at the compressor inlet; temperature \( T_k^E \) at the compressor outlet (at the combustion chamber inlet, while the pressure drop in it is neglected); temperature \( T_M^T \) at the turbine outlet (i.e. at the GPU drive free turbine inlet), and angular velocity \( \omega \) (shaft rotation speed) are measured as current – operating parameters. All other equation parameters are initially known, except for the flow rate \( Q \) and temperature drops at the compressor \( \Delta T^k \) and turbine \( \Delta T^T \) impellers [4, 16, 18, 20].

The temperature drops \( \Delta T^k \) and \( \Delta T^T \) vary when calculating by the mathematical model (i.e. the equation chains (1) and (2)), while the temperature values \( T_k^E \) and \( T_M^T \) are corrected (deviation of calculated values from the measured ones is reduced to minimum). Due to the monotonicity of the corresponding dependencies, these computational procedures allow determining the drops with an accuracy of their variation step magnitudes.
Thereat, the flow rate $Q$ is still unknown and found (at each variation step) by the equation (3), i.e., if we denote

$$A = -\sum_{i=1}^{L} R_k^i \left( \frac{tga_k^i}{\rho_k^i + S_k^i} + \frac{tga_q^i}{\rho_q^i + S_q^i} \right), \quad B = \sum_{i=1}^{L} (R_k^i)^2 \omega$$

for the compressor and

$$C = \sum_{j=0}^{M} R_j^f \left( \frac{tga_j^1}{\rho_j^1 + S_j^1} + \frac{tga_j^2}{\rho_j^2 + S_j^2} \right), \quad D = -\sum_{j=0}^{M} (R_j^f)^2 \omega$$

for the turbine, then the flow rate $Q$ is determined by the equation (3) rewritten as $AQ^2 + BQ = CQ^2 + DQ$.

Here, the difference in the air and gas mass flow rates through the compressor $Q$ and the turbine $Q + q$, respectively, is neglected since the fuel mass flow rate $q$ is substantially less than the air mass flow rate (hundredfold or more).

After determining these values, the diagnostic efficiencies of the compressor $\eta_k$ and the turbine $\eta_t$ are found, which can be used to estimate the conditions of the corresponding units (they may be $>1$-conditional values).

This technique can also be implemented for two-shaft and three-shaft engines with the difference that the equation chains (1) and (2) and the power equations (3) can be written separately for each shaft. The number of varying temperature drops increases accordingly to the number of shafts, and the gas flow rate through the engine path $Q$ remains the same. This allows building the many-fold number of equations for powers and temperatures after passing the compressors and turbines (current-operating parameters).

The informativeness of the conditional (diagnostic) efficiencies of compressors and turbines is that their systematic and simultaneous decrease may testify to the development of malfunctions such as nicks and the vane fastening plays, gas erosion, and an increase in sealing gaps. This is described below [3, 4, 21].

7. Free GPU turbine
To diagnose a GPU with a free turbine by thermogasdynamic parameters, the basic parameters characterizing its operation should be determined.

One of these parameters is the free turbine shaft power $N_0$. The Figure shows the dependencies between the rated (reduced) power $N$ and the speeds of the free turbine $n_f$ and GTE (also called the gas generator) $n_{gg}$.

$$N_0 = N(n_{ft}; n_{gg}) \frac{288 \cdot P_{air}}{10^5 \cdot \text{air}}$$

The power characteristic of the free turbine (as part of the GPU) has been built based on the data of testing on a hydraulic brake test bench under stationary conditions. The power changes when adjusting the VGV angle $\beta_{VGV} = 16^\circ$ and $28^\circ$ and changing the GTE speed.
To determine the useful power $N$ ($n_{ft}$, $n_{gg}$), the plots should be approximated. Upon building the equations for each of the lines in the form $N(n_{ft}, n_{gg})=ay^2+by+c$ (shown in the diagram), the following coefficients have been got:

|   | $a$     | $b$     | $c$     |
|---|---------|---------|---------|
| $N_1$ | -0.0154 | 23.916  | -5013   |
| $N_2$ | -0.0086 | 15.961  | -2234.2 |
| $N_3$ | -0.009  | 17.112  | -2561.3 |
| $N_4$ | -0.0073 | 15.366  | -1681.5 |
| $N_5$ | -0.0085 | 18.008  | -2353.6 |

A certain GTE (gas generator shaft) speed $n_{gg}$ (designated above as $\omega$) corresponds to each equation:

| Equation          | Speed ($n_{gg}$) |
|-------------------|------------------|
| $N_1 = -0.0154y^2 + 23.916y - 5013$ | $n_{gg}=889.66$ rad/s |
| $N_2 = -0.0086y^2 + 15.961y - 2234.2$ | $n_{gg}=858.26$ rad/s |
| $N_3 = -0.009y^2 + 17.112y - 2561.3$ | $n_{gg}=837.33$ rad/s |
| $N_4 = -0.0073y^2 + 15.366y - 1681.5$ | $n_{gg}=826.86$ rad/s |
| $N_5 = -0.0085y^2 + 18.008y - 2353.6$ | $n_{gg}=805.93$ rad/s |

Based on the data obtained, let us plot the dependence of the coefficients $a$, $b$, and $c$ on the gas generator speed $\omega = n_{gg}$:
Figure 3. The Dependence of the Coefficients a, b, and c on the Gas Generator Speed.

The a, b, and c values expressed through \( x = n_{ft} = \omega \) (Fig. 3) can be substituted in the equation \( N(n_{ft}, n_{gg}) = ay^2 + by + c \), where \( y = n_{ft} \), and find the reduced power in any given point of the area of the free
turbine operating parameters. This is a way of approximating or smoothing experimental data, which can further be used in diagnostics of gas turbine units by thermogasdynamic parameters.

It is now possible to determine the supercharger efficiency (as another diagnostic parameter):

\[
\eta_{sc} = \frac{\gamma R Q_m (T_{out} - T_{in})}{\mu Q_m N_0}
\]

where \( \gamma = 1.31 \) is the adiabatic index of the pumped (natural with the main methane fraction) gas,
\( \mu = 0.016 \text{ kg mol}^{-1} \) is the gas molar mass,
\( \rho_0 = 0.68 \text{ kg m}^{-3} \) is the \( \text{CH}_4 \) density under normal conditions,
\( R = 8.31 \text{ J} \text{ grad}^{-1} \text{ mol} \) is the universal gas constant,
\( Q_m \) is the gas mass flow rate \( \left( \text{kg s}^{-1} \right) \) related to the volume flow rate \( Q_V \left( \text{m}^3 \text{ h}^{-1} \right) \) by the known dependence:

\[
Q_m = Q_V \rho_0 \frac{3600}{\pi}
\]

8. Diagnostic technique
Along with the diagnostic parameters having a physical meaning of the compressor \( \eta^K \), GTE turbine \( \eta^T \) (4), and supercharger \( \eta_{sc} \) (5) efficiencies, this technique allows outlining another one—the ratio \( \chi = \frac{q}{Q} \) called ‘combustion quality’, the excess of which can testify to non-optimal fuel (as a rule - fuel gas) consumption.

9. Conclusions
Using this mathematical model, the GTU and GTE failure can be determined with an accuracy of 82-86 %, which is confirmed by studies in real industrial conditions of the Gazprom Transgaz Krasnodar compressor station GPUs.

As compared to the conventional methods, the diagnostic technique proposed is less expensive and labor-intensive, does not require the unit shutdown, and has a sufficient accuracy.

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