The definition limits technique for the efficient regulation of the «diesel engine – pressurized turbocompressor» system for mobile compressor units

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Abstract

The mathematical model of a combined operation of a diesel engine (power plant) – pressurized turbocompressor system with the results of pilot studies of a high-pressure centrifugal stage with adjustable inlet guide vanes is developed. The functional relationship of effective regulation is obtained.

Keywords: turbocompressor; power plant; inlet guide vanes

1. Introduction

The regulation of operation modes of a turbocompressor (TC) used for pressurization of a diesel engine (DE) consists in bringing in compliance of its gas-dynamic characteristics (GDC) with external characteristics of the engine. The possibility of using of turbocharged diesel engines with aviation series-produced centrifugal stages with spatial axial-radial impellers (I) in mobile compressor units (MCU) for the oil and gas industry permits to increase substantially the power of such units. This is because the aviation impellers pressure is approximately twice as higher, compared with the existing stationary and transport power plants used for turbocharging. Aviation centrifugal impellers in diesel engines, including those, used as a drive in MCU were not applied for turbocharging before.

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The purpose of the paper was to establish the efficient regulation law of DE-TC system, i.e. obtaining a functional dependence between a TC rotor speed and turning angle of blades of TC inlet guide vanes to obtain the maximum power and torque on DE shaft.

2. Object of research

K39-01UTs-17 centrifugal stage is selected as an object to assess the possibility of its utilizing as a turbocharging unit of internal combustion engine for MCU. The stage includes axial inlet guide vanes (IGV), semi-open axial-radial impellers (I), vaned diffuser (VD) and axial annular diffusor. Key geometrical parameters of the stage are the diameter of I, \( D_2 = 0.268 \) m; the width ratio of I, \( b_2/D_2 = 0.0285 \); the axial part outside diameter of I, \( D_0 = 0.1522 \) m; the bushing diameter of I, \( D_{bush} = 0.10372 \) m; the blades number of I, \( z_2 = 32 \); the blades setting angle of I inlet/outlet, \( \beta_{b2}/\beta_{b1} = 71.33/35 \) degrees.

The research task was to establish the relationship between turbocharging regulation system elements, namely, between the turning angle of blades of TC IGV and rotor speed to obtain preset operational characteristics, for example, maximum power and shaft torque.

6DM-21A [3] diesel engine is chosen as an object of a power plant. In a diesel pressurization regular system turbocompressor TK-23B-31 is applied, according to head and rate characteristics it is similar to a turbocompressor of gas-turbine engine TVO-100, however, it has lower pressure [2,3].

The major characteristic for DE of mobile compressor units is the external high-speed characteristic representing the diagram of dependence on rotor speed: engine power, average effective pressure or engine torque proportional to it (pressure), economic parameters (specific fuel consumption and efficiency). Thus, this characteristic restricts an engine limit modes field.

The calculation of diesel and TC combined characteristics is based on the scheme where a turbocharging unit is rigidly connected with an engine shaft through a multiplier, with 28 gear ratio, i.e. a mechanical connection of a TC rotor with an engine crankshaft is chosen.

The following basic factors governing the working process in a combined engine with the mechanical connection are:

1. the «rigid connection» between shafts rotor speed: \( n_{rev}^c = i_c \cdot n_{rev}^{en} \), where \( i_c \) is the gear ratio between the compressor shaft and the piston engine crankshaft;
2. the identical air (mixture) consumption via the compressor and engine (working medium leakages can be neglected);
3. the compressor outlet total pressure is more than piston part inlet total pressure for the losses rate in coolers and pipelines.

3. Methods

On the basis of preliminary obtained experimental TC GDC of the gas-turbine engine TVO-100 with various blades setting angles of IGV towards starting position on different rotor speed [4] the technique of theoretical obtaining of centrifugal stage GDC based on underlying GDC (for the initial IGV blades position, \( \Delta \theta_b = 0^\circ \)) is developed. GDC recalculation's technique principles are stated in details in [5, 6].

Thus, using obtained TC GDC for various IGV blades turning angles and known high-speed characteristics of the 6DM-21A engine, the working process calculation of the diesel with pressurization by the aviation unit instead of a standard one was performed.

A mathematical model of DE-TC system is built on the relationships and ratios borrowed from Grinevetsky-Mazing and B. P. Baykov techniques [1,2] to calculate processes of filling, compression, combustion and expansion. Turbocharging compressor parameters obtained as a result of GDC calculation, DE known parameters (the diameter and piston stroke, the number of cylinders, the shaft speed, known external characteristics [3]), and also design coefficients recommended by professional and reference literature were set at the input. The calculation was carried out by the method of successive approximations to the convergence of key parameters. The uniform conventional average fuel composition in volume ratios (\( C=0.87, H=0.126 \) and \( O=0.04 \)) with the heat of combustion about \( Q_l=42287 \) kJ/kg is accepted in calculations.
A brief algorithm of the mathematical model. The engine rotor speed \( n_{en} \) and the fuel pump characteristic presented as the relationship \( B_e=f(n_{en}) \) for a rack position on full delivery stop determines the fuel consumption \( B_e \). Having estimated air consumption \( G_{en} \) according to \( n_{en} \), the excess air coefficient \( \alpha \) and engine indicator efficiency \( \eta_i \) are defined. Then the temperature and gas pressure in front of the turbine \( T_T \), pressure \( P_T \) and adiabatic work \( H_T \) are determined.

The relation between the engine power \( N_e \) and TC parameters determined by its GDC (relationship of pressure \( \pi^* \), mass consumption \( G_c \), adiabatic efficiency \( \eta_{ad,c} \)) could be presented as:

\[
N_e = \frac{250 \cdot G_c \cdot L \cdot T_c \cdot P_T \cdot \eta_m}{(318,4/13596)\alpha_x \cdot \eta_f \cdot P_e}
\]

(1)

where \( \alpha_x \) is a total excess air coefficient; \( \eta_m \) is a mechanical efficiency; \( \eta_i \) is an indicator efficiency; \( \eta_f \) is a filling coefficient, \( L \) is an available air amount for combustion;

Engine shaft torque

\[
M_{sw} = 974 - 9,806 \cdot N_e/n_{rev}^{en}
\]

(2)

If the values of air consumption, temperature, polytrope indicators or other parameters obtained at the end of calculations differ from accepted before, the calculation is made again up to the required convergence.

The TC and DE combined operation line was selected at each rotor speed in terms of the reference point compliance to maximum efficiency and stability margin no less, than the initial turbocharging unit, and also from the conditions of mass consumptions equality through TC (\( G_c \)) and DE (\( G_{en} \)). The air consumption through DE equals the sum of the air intaken \( G_{in} \) and the air coming in a scavenge period \( G_{sc} \) and can be determined by the following relationship:

\[
G_e = G_{en} + G_{sc} = V_i \cdot i \cdot n_{rev0} \cdot \eta_v \cdot \varphi \cdot \rho_c / (z \cdot 60)
\]

(3)

where \( V_i \) is a cylinder volume; \( i \) is a number of engine cylinders; \( z \) is a cycle coefficient; \( n_{rev0} \) is a nominal rotor speed of the engine, \( \text{rpm} \); \( \eta_v \), \( \varphi \) are respectively intaking and scavenging coefficients; \( \rho_c \) is an air density, \( \text{kg/m}^3 \).

4. Results and discussion

According to the obtained values graphic power relationships \( N_e=f(n_{rev}^{en}, \Delta \theta_b) \) (fig. 1) and torque \( M_{sw}=f(n_{rev}^{en}, \Delta \theta_b) \) (fig. 2) of the power plant at turbocharging compressor regulation modes due to IGV blades turning are built. These relationships allow determine preferable, in terms of maximum efficiency, variation range of turning angles of IGV blades at various engine shaft rotor speed. According to these recommendations for rational turning angles of IGV blades with regard to the adjustable turbocompressor of diesel unit 6DM-21A, complying with maximum power and torque values (for the considered range of rotor speed of drive shaft of power unit from 900 to 1500 rpm) are developed. The power and torque increase was from 2,9% (when \( n_{rev}=1260 \text{ rpm} \) and \( \Delta \theta_b=+5^\circ \)) to 11,2% (when \( n_{rev}=1020 \text{ rpm} \) and \( \Delta \theta_b=+20^\circ \)).

The relationship's diagram \( \Delta \theta_b = f(n_{rev}^{en}) \) for the diesel 6DM-21A is represented in fig. 3 and described analytically:

- for the range, when \( n_{rev}^{en} = 900 \ldots 1020 \text{ rpm} \): \( \Delta \theta_b = +20^\circ \);
- for the range, when \( n_{rev}^{en} = 1020 \ldots 1460 \text{ rpm} \): \( \Delta \theta_b = 39.5 + 0.0167 \cdot n_{rev}^{en} - 3.53 \cdot 10^{-5} \cdot (n_{rev}^{en})^2 \)
- for the range, when \( n_{rev}>1500 \text{ rpm} \): \( \Delta \theta_b = -15^\circ \).
Fig. 1. The relationship of power of the diesel power plant versus the shaft speed and IGV setting angle of the supercharging turbocompressor

Fig. 2. The relationship of the diesel power plant torque versus the shaft speed and IGV setting angle of the supercharging turbocompressor

Fig. 3. The diagram of TC GDC effective regulation (the relationship $\Delta \theta_{IGV}=f(n_{rev})$) for obtaining of maximum power and shaft torque on the diesel 6DM-21A
According to results of the given calculation example the regulation algorithm of the turbocharging unit (fig. 4) as a part of the diesel engine is developed. The control is carried out via the signals processing module in which the relationship of the maximum power and engine torque versus the shaft speed and IGV blades turning angle is included. The signal is transmitted via the unit control module to the IGV blades rotation control module.

Based on the conducted researches the determination methodology of design parameters of the turbocharged compressor with adjustable IGV (fig. 5) is developed. The methodology is applied at the design stage and development of the turbocharging system for the engine selected. It is that applying the developed technique of GDC recalculation for various IGV blades rotation angles to the chosen turbocompressor stage (from the available database), the range of possible operating modes of the turbocharging unit is calculated. Then the DE characteristics calculation (network characteristics for the turbocharging unit) is made. Further on, the construction of the shared modes line (operating modes line OML) of TC and DE with the set TC rotor speed (a multiplier gear ratio) is carried out. The calculation is made iteratively with a gradual revolutions increase (gear ratio) to the maximum permissible TC revolutions. Achieving the effective result (OML is close to TC efficiency maximum), means that the selected TC type can be applied to this DE. Otherwise, another type of TC is chosen from the database.
5. Conclusion

The results of the performed work shows the possibility of applying of aviation centrifugal compressor stages with adjustable inlet guide vanes as the turbocharging system unit of diesel power plants. The functional relationship establishing technique between the system elements of turbocompressor regulation of turbocharging system (blades turning of inlet guide vanes) for preset engine service conditions, e.g., to obtain maximum power is developed.

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