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DETERMINING THE WORKING CHARACTERISTICS OF MECHANICALLY DRIVEN CENTRIFUGAL COMPRESSORS N-46-6

ODREĐIVANJE RADNIH KARAKTERISTIKA MEHANIČKIH CENTRIFUGALNIH KOMPRESORA N-46-6

Originalni naučni rad / Original scientific paper
Rad primljen / Paper received: 15.03.2024

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Keywords
- diesel engine
- working characteristics
- testing
- test bench

Abstract

The paper presents a mathematical model and research methodology, i.e., determination of working characteristics of mechanically driven centrifugal compressors used in the process of supercharging special purpose high power diesel engines (780 HP) with intake air. A special research laboratory installation was created for research with the aim to check and adapt compressors to the needs of the operating cycle of new or generally overhauled special purpose diesel engines. The paper provides a complete testing methodology and presents a mathematical model that can be very interesting and useful to other researchers in similar applications of this compressor type. After determining the compressor operating characteristics on the research installation, the compressor is then installed on the engine and the declared engine output parameters are monitored on a special research installation for engines.

Ključne reči
- dizel motor
- radne karakteristike
- ispitivanje
- probni sto

Izvod

U radu je predstavljen matematički model i istraživačka metodologija, odnosno, određivanje radnih karakteristika mehanički pogonjenih centrifugalnih kompresora dizel motora velike snage (780 KS). Specijalna istraživačka laboratorijska instalacija je kreirana u svrhu istraživanja. Cilj istraživanja je provera i prilagođavanje pomenutih kompresora potrebama radnog ciklusa novih ili remontovanih specijalnih dizel motora. Rad pruža potpunu metodologiju testiranja kao i prikaz korišćenog matematičkog modela, što može biti veoma interesantno i korisno drugim istraživačima zainteresovanim za slična polja primene ovog tipa kompresora. Nakon određivanja radnih karakteristika kompresora na probnom stolu, kompresor se zatim montira na motor i onda se obavlja praćenje deklarisanih izlaznih parametara motora na probnom stolu za ispitivanje motora.

INTRODUCTION

Optimal coupling of devices for supercharging engine cylinders in internal combustion engines with air (mechanical compressors, turbochargers) and engines themselves involves intertwining of two separate mechanical engineering branches: one related to the study of turbomachinery /1, 2/, and the other related to the study of internal combustion engines /3, 4/. Only a detailed analysis of operational characteristics of these two units, even in the early stage of designing these assemblies, provides the opportunity to achieve their optimal coupling, /5/.

The testing procedure of the mechanically driven centrifugal compressor N-46-6, designed to supply high-power diesel engines (780 HP) with air, is conducted to verify compliance with specified requirements /6/, ensuring its optimal integration with the engine. The mentioned document prescribes operating regimes and references effective characteristics that the tested compressor must meet after installation on the engine, and also gives values for reference parameters. For rotational speed 1300 rpm: Δp_{ref} = 32.35 kPa, ΔT_{ref} = 38 K, m_{ref} = 0.7 kg/s, P_{ref} = 27.96 kW; for rotational speed 1800 rpm: Δp_{ref} = 69.63 kPa, ΔT_{ref} = 67 K, m_{ref} = 1.1 kg/s, P_{ref} = 80.93 kW; and for rotational speed 2000 rpm: Δp_{ref} = 88.24 kPa, ΔT_{ref} = 82 K, m_{ref} = 1.3 kg/s, P_{ref} = 117.72 kW. The reference effective characteristics of the compressor, Δp_{ref} [kPa] and ΔT_{ref} [K], are provided in a format not commonly found in technical literature on turbomachinery /1, 2/. The parameter of pressure rise Δp_{ref} [kPa] corresponds to the quantity known as the total pressure ratio in the compressor ($\pi_{ktot} = p_{2tot}/p_{1tot}$), while the temperature rise ΔT_{ref} [K] represents a measure of the quality of compression performed and corresponds to the isentropic efficiency η_{is}.

Since the initial document /6/ does not contain a procedure for calculating effective characteristics of the compressor from measured quantities, nor a method for reducing them to reference atmospheric conditions, it was necessary to supplement the measurement and data processing procedure. It was also necessary to develop a reliable, original method for assessing the quality of the compressor and to establish an appropriate test bench to avoid complex testing of the entire engine - mechanical compressor system.

The operational characteristics of the compressor needed to be verified at operating points corresponding to engine speed regimes of n = 1300; 1800; and 2000 rpm, at full power, /6/. According to the developed method described here, the quality assessment procedure involves forming compressor characteristics at constant rotation velocity corresponding to the mentioned shaft rotation speeds of the prime mover. Characteristics are determined using four operating points. Boundary points at maximum flow rates correspond to a fully open throttle valve, while boundary points in the zone of maximum pressures are achieved by closing the valve and locating it near the ‘surge limit’. The other two points are within the compressor's operating range. Based on the relative position of reference points compared to the recorded characteristics and the condition of technical correctness, an assessment is provided as to whether the compressor meets the requirements for installation on the engine.

In the development of the testing method described here, being entirely original, similar research on centrifugal compressors applied in various technical fields was analysed to confirm the proposed mathematical model, select appropriate measurement equipment, and configure the test bench. The wide range of centrifugal compressors with micro and macro differences can be somewhat mysterious for the typical user during their selection. In /10/, dimensional and dimensionless similarity parameters are defined which can be used successfully for more precise compressor selection, providing a reliable assessment when choosing an appropriate compressor. In /11/, it is shown that in the development of turbo-compressors aimed at increasing flow capacity above the usual level, a reliable high-flow compressor that retains high performance could be developed by a suitable combination of CFD and FEM analysis, and optimisation approaches. Recognizing the fact that flow, efficiency, and pressure rise within the compressor are the three most important parameters used in defining compressor performance and selection, /12/ provides a comparative analysis of the advantages of centrifugal compressors over other types of compressors. Overpressure is an unstable operating mode in the compression system that occurs at mass flows below the so-called ‘overpressure line’. Instability is characterised by large fluctuations in the compressor flow boundary cycle and pressure rise, which reduce compressor performance. High thermal and mechanical loads can also compromise safe operation of the compression system. The analysis described in /13/ confirms the significance of the so-called ‘stability parameter’ that motivated the development of two approaches to identify this parameter in the tested compression systems. The designed active overpressure control system, consisting of pressure sensors, a vent valve control on the compressor discharge pipe, and a linear quadratic Gaussian controller, provided noticeable stabilization at 95 % of compressor mass flow at low speeds. A case study /14/ describes factors contributing to increased vibration problems after three centrifugal compressors at a pipeline station were retrofitted with larger impellers. Disturbances between the natural frequencies of the pipeline in compressor mode and the casing, and potential excitation sources such as variable flow and blade-pass-induced pulsations, were taken into account. Authors of /15/ emphasize that while the goal of ensuring correct discharge pressure at the right flow rate can be achieved using different control methods, some are much more efficient than others, and the impact of the proposed control method on compressor efficiency should be carefully assessed. /16/ discusses the effects of inlet recirculation on compressor characteristics using a 1-D model. The analysis shows that an increase in inlet recirculation associated with reduced flow can cause the compressor characteristics to have a positive gradient, but this destabilizes compressor operation. As a constructive modification, the installation of ribs at the inlet is proposed, initiating the growth of pre-swirl as the main factor enabling inlet recirculation.

EXPERIMENT

Test bench

A schematic representation of the test bench with indicated measurement points is provided in Fig. 1, /17/. The individual labels in he Fig. 1 indicate the following: 0 - measurement cross-section 0; 1 - measurement cross-section 1; 2 - measurement cross-section 2; 3 - direct current electric motor with independent excitation; power P_{EM} = 85-170 kW in the shaft speed range n_{EM} = 1500-3000 rpm; 4 - torsiometer with tachometer; 5 – multiplier; 6 - centrifugal compressor N-46-6; 7 - compression pressure regulation valve, slide type, nominal diameter DN = 100 mm and nominal pressure PN = 16 bar; 8 - air flow measurement inlet; 9 - oil reservoir; 10 - gear pump marked MZN 2, driven by a three-phase asynchronous electric motor with rated shaft speed n = 1445 rpm and power P_{EM} = 3 kW; 11 - overflow valve for oil pressure regulation.

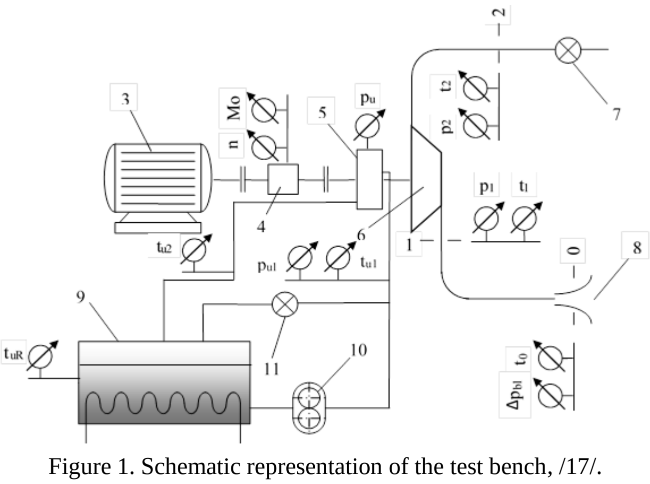

Figure 1. Schematic representation of the test bench, /17/.

The lubrication device is also shown in Fig. 1, which maintains required lubrication pressure values (8 ± 2 bar) and oil temperature (inlet min 55 °C and outlet max 115 °C) during testing. Oil pressure is measured at the inlet to the multiplier, and the temperature is measured at both the inlet and outlet of the multiplier, as well as in the oil reservoir. Inside the reservoir is a heat exchanger used to heat or cool the oil as needed. A physical representation of the test bench is provided in Fig. 2.

Figure 2. Physical representation of the test bench.

Methods

Measured variables, measurement methods and equipment used are given in Table 1.

Table 1. Measured parameters.

Measured parameter	Symbol	Measurement method
barometric pressure	p_{bar} (mbar)	portable device
shaft speed of the prime mover	n_{EM} (rpm)	tachometer - T30FN
pressure at the inlet for measuring air flow	Δp_0 (mbar/mmH$_2$O)	differential pressure transmitter / ‘U’ pipe with water
air pressure before the compressor	p_1 (mbar/mm H$_2$O)	pressure transmitter HBM with measuring tapes / ‘U’ pipe with water
air pressure downstream of the compressor	p_2 (mbar/mm Hg)	pressure transmitter HBM with measuring tapes / ‘U’ pipe with mercury
temperature before the inlet for measuring air flow	t_0 (°C)	thermocouple type J
air temperature before the compressor	t_1 (°C)	thermocouple type K, diameter 3 mm, protected
air temperature after the compressor	t_2 (°C)	thermocouple type K, diameter 3 mm, protected
engaged torque of the prime mover	M (Nm)	torsiometer T30FN
oil temperature in the reservoir	t_{uR} (°C)	thermocouple type K, diameter 3 mm, protected
oil pressure at the inlet to the compressor multiplier	p_u (bar)	manometer with Bourdon tube
oil temperature at the inlet to the compressor multiplier	t_{u1} (°C)	thermocouple type K, diameter 3 mm, protected
oil temperature at the outlet of the compressor multiplier	t_{u2} (°C)	thermocouple type K, diameter 3 mm, protected

Electrical signals from the sensors are introduced into the CompactDaq National Instruments® acquisition device, connected to a PC computer. Voltage signals from thermocouples are directly input into the acquisition system, while signals from pressure transducers are brought into this device through the measuring bridge. Signals from the tachometer and torsiometer are also routed into the acquisition system through the measuring bridge.

Procedure for determining the total state variables at reference sections

Since velocities exceeding 100 m/s occur at the reference sections in the installation, it is necessary to include the factor representing kinetic energy when processing data and forming the energy equation. In the calculation of turbomachinery parameters, kinetic energy is included through total state variables. In the data processing procedure, a computational method for determining total (denoted by *tot*) and static (denoted by *s*) variables from measured pressure and temperature values is applied. The desired variables are obtained by solving the system of basic equations (1), (2), (3), and (4). Equations (1) and (3) are derived from the laws of conservation of energy and deal with the change in air stream enthalpy in the flow process over the temperature probe. Equations (2) and (4) are the state equation and continuity equation for the given measurement section.

$$T_{tot} = T_s + K\frac{c^2}{2c_p}\,, \tag{1}$$

$$p_s = \rho RT\,, \tag{2}$$

$$p_{tot} = p_s + \frac{\rho c^2}{2}\,, \tag{3}$$

$$\dot{m} = cA\rho\,. \tag{4}$$

Standard temperature probes of ‘K’ type are applied in the installation with a 3 mm external diameter of the protective tube. According to data from /7, 8/, such probes in addition to static temperature, register approximately 50 % of dynamic temperature, thus the adopted value for the coefficient in Eq.(1) is $K = 0.5$.

Procedure for reducing effective characteristics to reference state

The reduction of compressor effective characteristics to reference state is carried out according to original technical conditions of the licensing provider /6/. The reference state represents ambient parameters: $T_0 = 288.152$ K and $p_0 = 101.325$ kPa.

The corrected parameters which according to /6/ represent the primary criterion for evaluating the operational quality of the tested compressor, expressed in SI units, are:

- corrected pressure rise:

$$\Delta p_{k0}\ [\text{Pa}] = p_{2tot} - p_{1tot}\,, \tag{5}$$

- corrected temperature rise:

$$\Delta T_{tot}\ [\text{K}] = T_{2tot} - T_{1tot}\,, \tag{6}$$

- corrected mass flow:

$$\dot{m}_{k0}\left[\frac{\text{kg}}{\text{s}}\right] = k\dot{m}\,, \tag{7}$$

where correction factor k is calculated according to:

$$k[-] = \sqrt{\frac{T_{1tot}}{T_0}}\cdot\frac{1000\,p_0}{p_{1tot}}\,. \tag{8}$$

Engaged power $$P_{k0}\ [\text{kW}] = M\frac{n_{EM}\,\pi}{30}\,. \tag{9}$$

Calculation procedure for operating characteristics of the centrifugal compressor

Constants used in the calculation:
d_0 [m] = 0.10512 - diameter of air flow measurement inlet
g [m/s^2] = 9.81 - gravitational acceleration
R [J/kgK] = 287.02 - gas constant for air
κ = 1.4 - adiabatic exponent for air
c_p [J/kgK] = 1004.57 - specific heat of air at constant pressure
ρ_{Hg} [kg/m^3] = 13545.82 - density of mercury
ρ_{H2O} [kg/m^3] = 1000 - density of water
α = 0.9920 [-] - flow coefficient of the pipe for flow measurement
A_0 [m^2] = $d_0^2\pi/4$ - flow section of pipe for flow measurement (measuring section 0)
K [-] = 0.5 - sensitivity coefficient of the transmitter to dynamic temperature
d_1 [m] = 0.147 - internal diameter of inlet to the compressor
A_1 [m^2] = $d_1^2\pi/4$ - area of measuring section 1
a [m] = 0.147 mm - major axis of the elliptical cross- section at the outlet of the compressor
b [m] = 0.128 mm - minor axis of the elliptical cross-section at the outlet of the compressor
A_2 [m^2] = $ab\pi/4$ - area of measuring section 2
T_{0R} [K] = 288.152 - temperature at reference atmospheric conditions
p_{0R} [kPa] = 101.325 - pressure at reference atmospheric conditions
ρ_{0R} [kg/m^3] = 1.22514 - air density at reference atmospheric conditions

Mass flow rate of air through the compressor $\dot{m}_{k0}$ [kg/s] is calculated according to the following equation:

$$\dot{m}_{k0}\left[\frac{\text{kg}}{\text{s}}\right]=k\alpha\varepsilon A_0\left[\text{m}^2\right]\sqrt{2\rho_{0R}\left[\frac{\text{kg}}{\text{m}^3}\right]\cdot\Delta p_0[\text{mm}_{\text{H}_2\text{O}}]\cdot g\left[\frac{\text{m}}{\text{s}^2}\right]},\quad(10)$$

where: ρ_{0V} [kg/m^3] density of air in front of the flow measurement inlet:

$$\rho_{0V}\left[\frac{\text{kg}}{\text{m}^3}\right]=\frac{p_{bar}[\text{mbar}]\cdot 100}{R(273.15+t_0[^\circ\text{C}])},\quad(11)$$

ε [-] expansion coefficient of air in the inlet according to the empirical equation:

$$\varepsilon=1-0.55\frac{\Delta p_0[\text{mm}_{\text{H}_2\text{O}}]\cdot g\left[\frac{\text{m}}{\text{s}^2}\right]}{p_{bar}[\text{mbar}]\cdot 100}.\quad(12)$$

Fluid parameters at measurement section 1 before the compressor are determined according following:
p_1 [Pa] - measured pressure before the compressor, if ‘U’ pipes with water are used:

$$p_1[\text{Pa}]=\frac{p_1[\text{mm}_{\text{H}_2\text{O}}]\cdot\rho_{\text{H}_2\text{O}}\left[\frac{\text{kg}}{\text{m}^3}\right]\cdot g\left[\frac{\text{m}}{\text{s}^2}\right]}{1000},\quad(13)$$

if HBM pressure transmitter with measuring tapes is used:

$$p_1[\text{Pa}]=p_1[\text{mbar}]\cdot 100\quad(14)$$

p_{1as} [Pa] - absolute static pressure before the compressor:

$$p_{1as}[\text{Pa}]=p_1[\text{Pa}]+p_{bar}[\text{mbar}]\cdot 100\quad(15)$$

T_1 [K] - read temperature before the compressor:

$$T_1[\text{K}]=t_1[^\circ\text{C}]+273.15\quad(16)$$

T_{1s} [K] - static temperature before the compressor:

$$T_{1s}=\frac{-E_1+\sqrt{E_1^2-4D_1E_1}}{2D_1}.\quad(17)$$

Equation (17) requires an explanation, considering Eq.(1) a relation can be established:

$$T_{1tot}-T_{1s}=K\frac{c_1^2}{2c_p}.\quad(17a)$$

Based on the general equations of state ($p\dot{V}=\dot{m}RT$) and continuity ($\dot{V}=cA$), the air flow velocity can be obtained from the expression:

$$c=\frac{\dot{m}RT}{pA},\quad(17b)$$

and its square, for cross section 1, is:

$$c_1^2=\frac{\dot{m}_{k0}^2R^2T_{1s}^2}{p_{1s}^2A_1^2}.\quad(17c)$$

Now, a quadratic equation can be set up:

$$\frac{K\dot{m}_{k0}^2R^2}{2c_p\,p_{1as}^2A_1^2}T_{1s}^2+T_{1s}-T_1=0,\quad(17d)$$

whose real solution is Eq.(17), where:
D_1 - coefficient of quadratic equation with quadratic term:

$$D_1=\frac{K\dot{m}_{k0}^2R^2}{2c_p\,p_{1as}^2A_1^2},\quad(18)$$

E_1 - coefficient of quadratic equation with linear term:

$$E_1=1,\quad(19)$$

F_1 - free term of the quadratic equation:

$$F_1=-T_1.\quad(20)$$

Other quantities are obtained from parameters whose calculation and measurement units are explained.
c_1 [m/s] - air velocity in front of the compressor:

$$c_1=\frac{\dot{m}_{k0}RT_{1s}}{p_{1as}A_1},\quad(21)$$

ρ_1 [kg/m^3] - air density in front of the compressor:

$$\rho_1=\frac{p_{1as}}{RT_{1s}},\quad(22)$$

p_{1tot} [Pa] - total air pressure in front of the compressor:

$$p_{1tot}=p_{1as}+\frac{\rho_1c_1^2}{2},\quad(23)$$

T_{1tot} [K] - total air temperature in front of the compressor:

$$T_{1tot}=T_{1s}+K\frac{c_1^2}{2c_p}.\quad(24)$$

Fluid parameters at measurement section 2 behind the compressor are determined according to the following:
p_2 [Pa] - measured pressure after the compressor, if mercury ‘U’ pipes are used:

$$p_2[\text{Pa}]=\frac{p_2[\text{mm}_{\text{Hg}}]\cdot\rho_{\text{Hg}}\left[\frac{\text{kg}}{\text{m}^3}\right]\cdot g\left[\frac{\text{m}}{\text{s}^2}\right]}{1000},\quad(25)$$

and if the HBM pressure transmitter with measuring tapes is used:

$$p_2[\text{Pa}]=p_2[\text{mbar}]\cdot 100,\quad(26)$$

p_{2as} [Pa] - absolute static pressure after the compressor:

$$p_{2as}[\text{Pa}] = p_2[\text{Pa}] + p_{bar}[\text{mbar}] \cdot 100\,, \tag{27}$$

T_2 [K] - read temperature in measuring section 2 after the compressor:

$$T_2[\text{K}] = t_2[°\text{C}] + 273.15\,, \tag{28}$$

T_{2s} [K] - static temperature in measuring section 2 after the compressor:

$$T_{2s} = \frac{-E_2 + \sqrt{E_2^2 - 4D_2E_2}}{2D_2}\,. \tag{29}$$

Equation (29) is derived according to the same procedure as Eq.(17), explained earlier, and the specified parameters are calculated from the following equations:

D_2 - coefficient of the quadratic equation with the quadratic term:

$$D_2 = \frac{Km_{k0}^2R^2}{2c_p\,p_{2as}^2A_2^2}\,, \tag{30}$$

E_2 - coefficient of quadratic equation with the linear term:

$$E_2 = 1\,, \tag{31}$$

F_2 - free term of the quadratic equation:

$$F_2 = -T_2\,. \tag{32}$$

Remaining quantities are obtained from parameters whose calculation and measurement units are already explained:

c_2 [m/s] - air velocity in measuring section 2 after the compressor, i.e., its square:

$$c_2^2 = \frac{\dot{m}_{k0}^2R^2T_{2s}^2}{p_{2s}^2A_2^2}\,, \tag{33}$$

ρ_2 [kg/m^3] - air density in measuring section 2 after the compressor:

$$\rho_2 = \frac{p_{2as}}{RT_{2s}}\,, \tag{34}$$

p_{2tot} [Pa] - total pressure in measuring section 2 after the compressor:

$$p_{2tot} = p_{2as} + \frac{\rho_2c_2^2}{2}\,, \tag{35}$$

T_{2tot} [K] - total air temperature in measuring section 2 after the compressor:

$$T_{2tot} = T_{2s} + K\frac{c_2^2}{2c_p}\,. \tag{36}$$

Effective characteristics of the compressor

Δp_{K0} [Pa] - air pressure rise in the compressor:

$$\Delta p_{K0} = p_{2tot} - p_{1tot}\,, \tag{37}$$

ΔT_{tot} [K] - air temperature rise in the compressor:

$$\Delta T_{tot} = T_{2tot} - T_{1tot}\,, \tag{38}$$

P_{K0} [kW] - engaged power of the drive motor:

$$P_{K0} = M\frac{n_{EM}\pi}{30}\,, \tag{39}$$

degree of air pressure rise in the compressor:

$$\pi_{K0} = \frac{p_{2tot}}{p_{1tot}}\,, \tag{40}$$

Y_{is} [kJ/kg] - unit isentropic work of air flow:

$$Y_{is} = \frac{\kappa}{\kappa-1}RT_{1tot}\left[\pi_{k0}^{\frac{\kappa-1}{\kappa}} - 1\right], \tag{41}$$

Y_k [kJ/kg] - actual work of the air flow (circuit):

$$Y_k = c_p(T_{2tot} - T_{1tot})\,, \tag{42}$$

η_{is} [-] - isentropic compressor level of efficiency:

$$\eta_{is} = \frac{Y_{is}}{Y_k}\,, \tag{43}$$

P_{is} [kW] - isentropic compressor power:

$$P_{is} = Y_{is}\dot{m}_{k0}\,, \tag{44}$$

η_u [-] - total level of efficiency:

$$\eta_u = \frac{P_{is}}{P_{k0}}\,, \tag{45}$$

η_m [-] - mechanical efficiency:

$$\eta_m = \frac{\eta_u}{\eta_{is}}\,. \tag{46}$$

RESULTS AND DISCUSSION

Key diagrams obtained from the presented calculations are provided below, as follows:

- pressure rise diagram (Δp_{K0} [Pa]) in Fig. 3,
- temperature rise diagram (ΔT_{tot} [K]) in Fig. 4,
- engaged power diagram (P_{K0} [kW]) in Fig. 5.

The reference values to be satisfied according to /6/ are marked in Figs. 3, 4, and 5 as follows: a square ( ) for 1300 rpm, a circle (○) for 1800 rpm, and a rotated square (◊) for 2000 rpm.

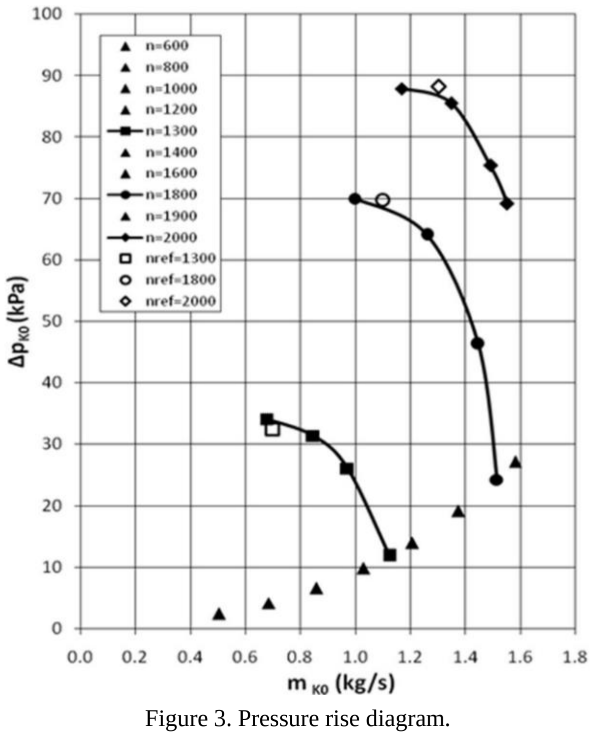

Figure 3. Pressure rise diagram.

Other important parameters of the compressor operation are presented in:

- Fig. 6, diagram of pressure ratio (π_{K0}), and
- Fig. 7, diagram of efficiency ratio (isentropic η_{is}, mechanical η_m, and total η_u).

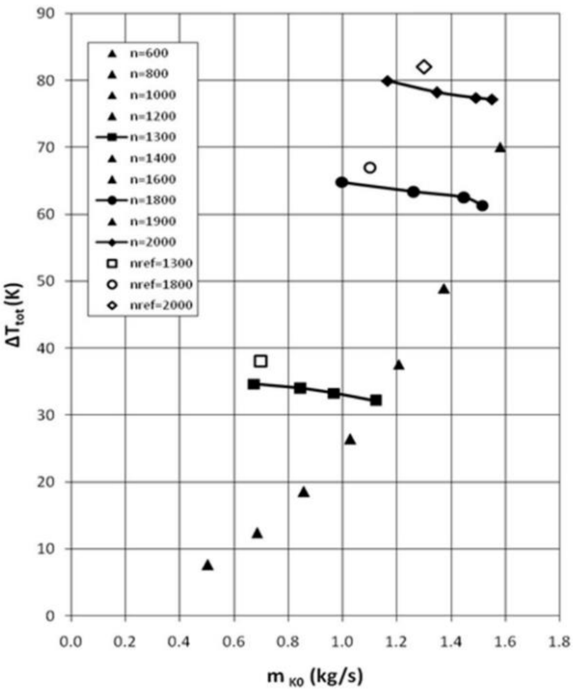

Figure 4. Temperature rise diagram.

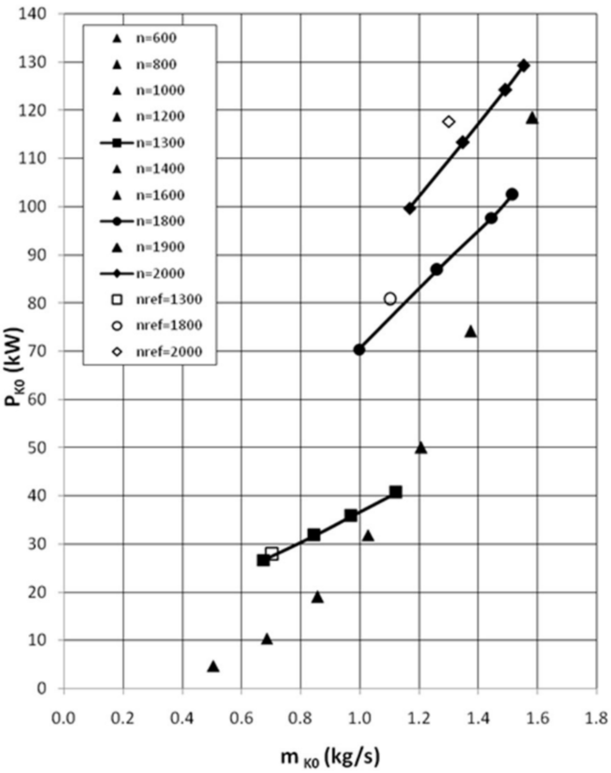

Figure 5. Engaged power diagram.

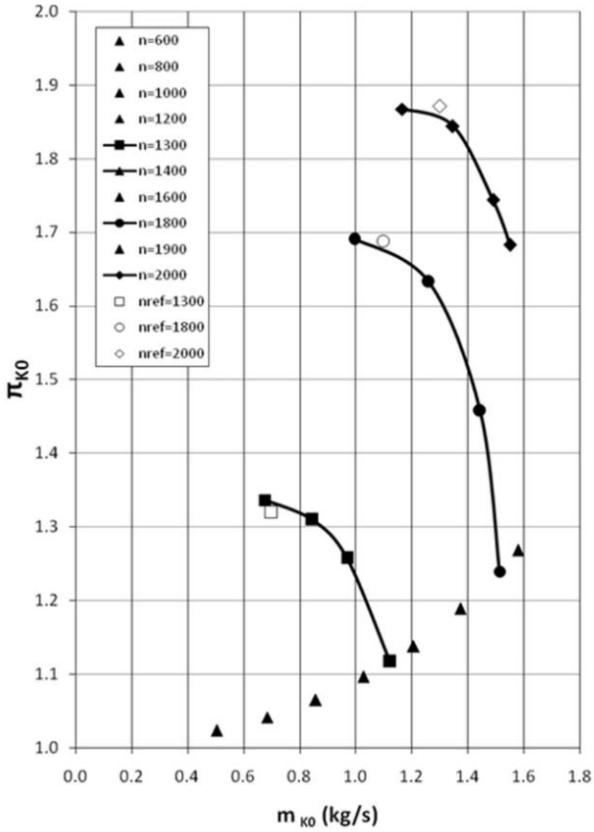

Figure 6. Diagram of pressure ratio.

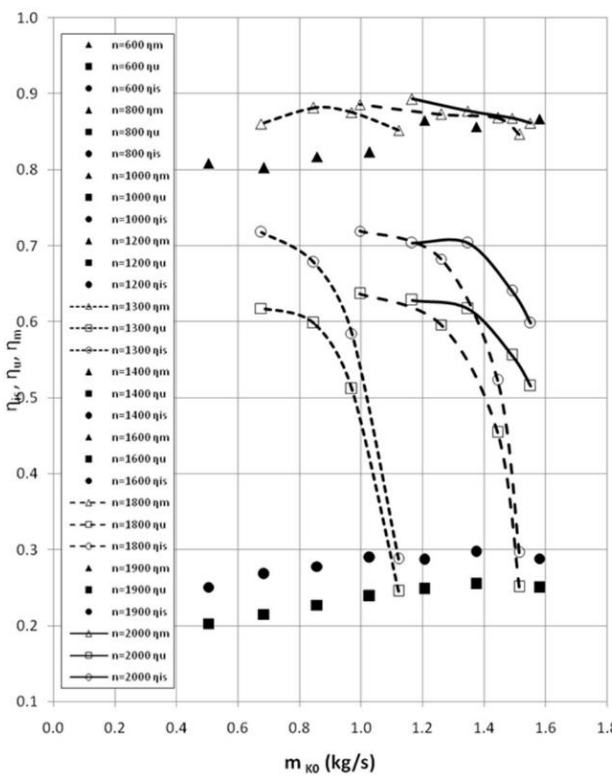

Figure 7. Diagram of efficiency ratio.

DISCUSSION

The compressor quality assessment is based on the following criteria:

- achieved pressure rise for the reference mass airflow, with a tolerance for deviation from the reference value of less than 3 %,
- achieved temperature rise for the reference mass airflow, with a tolerance for deviation from the reference value greater than 3 %,
- engaged power for the compressor drive for the reference mass airflow, with a tolerance for deviation from the reference value greater than 3 %,
- any damages to the compressor impeller, drive gear, external parts of the spiral, and housing, as well as any traces of potential oil leaks, which are not allowed,
- potential localised overheating in the areas of the multiplier bearings, or increased temperatures and flows of lubricating oil for the sliding bearings of the multiplier, which are not allowed, and
- potential increased noise levels during compressor operation, which are not allowed.

If the aforementioned criteria for effective characteristics and overall technical correctness are met, the compressor can be installed on the engine.

CONCLUSION

The paper describes a procedure for functional testing and determining the operating characteristics of mechanically driven centrifugal compressors intended for high-power engines on a specially designed test bench for this purpose. The developed mathematical model, based on fundamental knowledge in the fields of fluid mechanics, thermodynamics, and turbomachinery, with only nine input parameters, enables a quality analysis of the operating characteristics of these units, which along with an additional four parameters, provides a complete picture of the suitability for compressor installation on the engine and its technical correctness.

In support of the validation of the applied method, it is noteworthy that during several years of compressor testing according to the presented procedure, there have been no complaints lodged, nor has a motor with an installed compressor rated as satisfactory failed to meet the set requirements for nominal torque, power, or efficiency due to poor operational (functional) characteristics of the compressor.

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Nomenclature

a	major axis of elliptical cross-section at the outlet of the compressor
A_0	flow section of the pipe for measuring flow (measuring section 0)
A_1	area of measuring section 1
A_2	area of measuring section 2
b	minor axis of elliptical cross-section at the outlet of the compressor
c	air flow rate
c_p	specific heat of air at constant pressure
c_1	air velocity in front of the compressor
c_2	air velocity after the compressor
D_1	coefficient of the quadratic equation with quadratic term
D_2	coefficient of the quadratic equation with quadratic term
E_1	coefficient of the quadratic equation with linear term
E_2	coefficient of the quadratic equation with linear term
F_1	free term of the quadratic equation
F_2	free term of the quadratic equation
d_0	diameter of the pipe for measuring air flow
d_1	internal diameter of the inlet to the compressor
g	gravitational acceleration
k	correction factor according to DIN 70020
K	sensitivity coefficient of the transmitter to dynamic temperature
$\dot{m}_{k0}$	corrected mass flow rate
m_{ref}	reference air flow
M	engaged torque moment of the drive motor

n_{EM} number of revolutions per minute of drive motor shaft
p_{bar} barometric pressure
p_u oil pressure at the inlet to the compressor multiplier
p_{0R} pressure at reference atmospheric conditions
p_1 air pressure before the compressor
p_{1as} absolute static pressure before the compressor
p_{1tot} total air pressure before the compressor
p_2 air pressure after the compressor
p_{2as} absolute static pressure after the compressor
p_{2tot} total air pressure after the compressor
P_{k0} engaged power
P_{ref} reference engaged power
R gas constant for air
t_0 temperature before the pipe for measuring air flow
t_{u1} oil temperature in the tank
t_{u2} oil temperature at the inlet to the compressor multiplier
t_{uR} oil temperature at the outlet of the compressor multiplier
t_1 air temperature before the compressor
t_2 air temperature after the compressor
T_{0R} temperature at reference atmospheric conditions
T_1 temperature before the compressor
T_{1s} static temperature before the compressor
T_{1tot} total temperature before the compressor
T_2 temperature after the compressor
T_{2s} static temperature after the compressor
T_{2tot} total temperature after the compressor
Y_{is} unit isentropic work of air flow
Y_k actual work of the air flow (circuit)

Greek symbols

α flow coefficient of the air flow measurement inlet
ε air expansion coefficient in the flow measurement inlet
η_{is} isentropic efficiency of the compressor
η_m mechanical efficiency
η_u overall efficiency
κ adiabatic exponent for air
Δp_0 pressure at the air flow measurement inlet
Δp_{K0} pressure rise of air in the compressor
Δp_{ref} reference pressure rise
ΔT_{ref} reference temperature rise
ΔT_{tot} temperature rise of air in the compressor
ρ_{Hg} density of mercury
ρ_{H2O} density of water
ρ_{0R} density of air at reference atmospheric conditions
ρ_{0V} density of air in front of the flow measurement inlet
ρ_1 density of air in front of the compressor