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Non Adiabatic Centrifugal Compressor Gas Dynamic Performance Definition

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NON ADIABATIC CENTRIFUGAL COMPRESSOR GAS DYNAMIC PERFORMANCE DEFINITION

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ABSTRACT

Compression process for most turbo compressors is adiabatic, i.e. heat transfer to compressed gas is negligible. It gives a possibility to define mechanical work input and efficiency as functions of gas total temperature rise. Compressed gases in centrifugal compressors of small GT and of turbochargers get sufficient heat energy from a very hot turbine and pass a part of heat energy to ambience. Temperature rise does not correspond to mechanical energy of a driver if compression is non adiabatic. Performance modeling computer programs of TU SPb were applied to small turbocharger test data presented by colleagues from Hanover University (Germany). Several suppositions were formulated that made possible to predict performance curves of efficiency and pressure ratio of the compressor wide range of RPM corresponding to periphery Mach numbers within the range of 0,73 - 1,44.

Key words: centrifugal compressor, heat transfer, efficiency, pressure ratio, work coefficient, periphery Mach number, modeling.

NOTATION

c – absolute velocity; c_r - radial velocity; c_u – tangential velocity; C_p - specific heat; D_0 - impeller eye diameter; D_1 - impeller inlet diameter; D_2 - impeller diameter; γ - isentropic coefficient; $\dot{m}$ - mass flow rate; M - Mach number; $M_U = \frac{U}{\sqrt{\gamma R T_{inl\,t}}}$; p - pressure, Pa; R - gas constant; R_{bl} - radius of blade curvature; Re - Reynolds number, $\mathrm{Re}_u \frac{u_2 D_2}{\mu_{inl}} \frac{p_{inl\,tot}}{R T_{inl\,tot}}$; T - temperature, K; U - impeller periphery speed; w - relative velocity; z - number of blades; Φ -flow rate coefficient; η_t - total polytropic efficiency; π_t - total pressure ratio; $\psi_T = c_{u2}/u_2$ - Euler work coefficient.

Subscripts

1 - impeller blade inlet condition; des – design regime; ex – compressor exit; h – hub; in – compressor inlet; s – blade suction side, shroud; t – total parameters, tr - transition.

An aim of gas dynamic compressor test is to define its performance curves such as $\pi_t, \eta_t = f(\dot{m})$. There is no problem to measure total pressure and mass flow rate. But an efficiency definition is a problem if compression process is not adiabatic, i.e. in case when heat transfer is sufficient.

Most centrifugal compressors operate in conditions with negligible heat transfer (adiabatic compression). Their plant tests conditions are similar or close to it. Test regulations establish measures to diminish influence of a heat transfer “compressor body – atmospheric air”. The energy conservation in a compression process is:

$$N_i + N_{tr} = C_p\left(T_{ex\,t} - T_{int}\right)\dot{m}, \quad (1) \quad \text{or} \quad H_i + H_{tr} = C_p\left(T_{ex\,t} - T_{int}\right). \quad (1a)$$

If $H_{tr} \approx 0$ a temperature rise in a compressor can be correctly used to calculate work input. Then the total efficiency [11] is defined by the equation:

$$\eta_t = \frac{\ln\left(p_{ext} / p_{inlt}\right)}{\frac{\gamma}{\gamma-1}\ln\left(T_{ext} / T_{inlt}\right)}. \quad (2)$$

Heat transfer cannot be neglected though in some cases – centrifugal compressors of small GT units and of superchargers in particular. The small turbocharger view with partially opened compressor flow path is shown in the Fig. 1. It is evident that direct measure of power input N_i is impossible for this kind of machinery. The investigation presented at [12] demonstrated strong heat transfer "turbine – compressor" and "compressor – ambience". So there is no way to measure efficiency more or less accurately, because $H_i \neq C_p\left(T_{ext} - T_{inlt}\right)$.

Fig. 1. A small turbocharger view with partially opened compressor flow path [14]

Test data for the compressor with the impeller diameter 48 mm at different RPM were provided to the Author by Prof. J. Seume (Director of Institute of Turbo machines, Hanover University, Germany) in a course of cooperation with TU SPb. Pressure ratio and efficiency by Eq. (2) as for adiabatic compression is presented graphically in the Fig. 3 as function of mass flow at 72 000-202 000 RPM (M_U = 0,735-1,442).

The experience with adiabatic compression shows that efficiency is diminished at partial RPM due to impeller – stator mismatch. But the level of decrease cannot be as big as in the Fig. 2 – about 20%. Unrealistic influence of rotation speed on efficiency points at non adiabatic process indirectly. Measured mass flow and pressure ratio must be treated as reliable, but an efficiency level is quite indefinite.

The Author's idea was to apply TU SPb modeling technology to reduce non adiabatic test data with an aim to estimate its efficiency more realistic. The Universal modeling method of Prof. Y. Galerkin is well presented in Russian periodicals, in monographs [8, 10] and at international conferences [2, 3, 4, 5, 6, 13]. The new version of the Method is presented at this Conference too•. Therefore the necessary details are only touched below.

The proposed solution is based on following suppositions.

Supposition 1. The problem of heat transfer in compressors as in the Fig. 1 was studied in [12]. The opinion was formulated that there is a balance of heat transfer from very hot turbine to the compressor and from hot compressor to ambience at the highest RPM. It is assumed that the performance curves $\pi_t, \eta_t = f\left(\dot{m}\right)$ at 202000 RPM are reliable as the measured temperatures T_{int}, T_{ext} are reliable too. These performance curves were modeled by Universal modeling, 6th generation computer programs. The empirical coefficients for big "adiabatic" compressors were changed to some extent for better modeling.

• "New version of the Universal modeling for centrifugal compressor gas dynamic design. Y.B. Galerkin, K.V. Soldatova, A.A. Drozdov"

Supposition 2. Long-standing practice of Universal modeling demonstrated that calculated curves $\eta_t = f\left(\dot{m}, RPM\right)$ are sufficiently correct in all range of rotation speed.

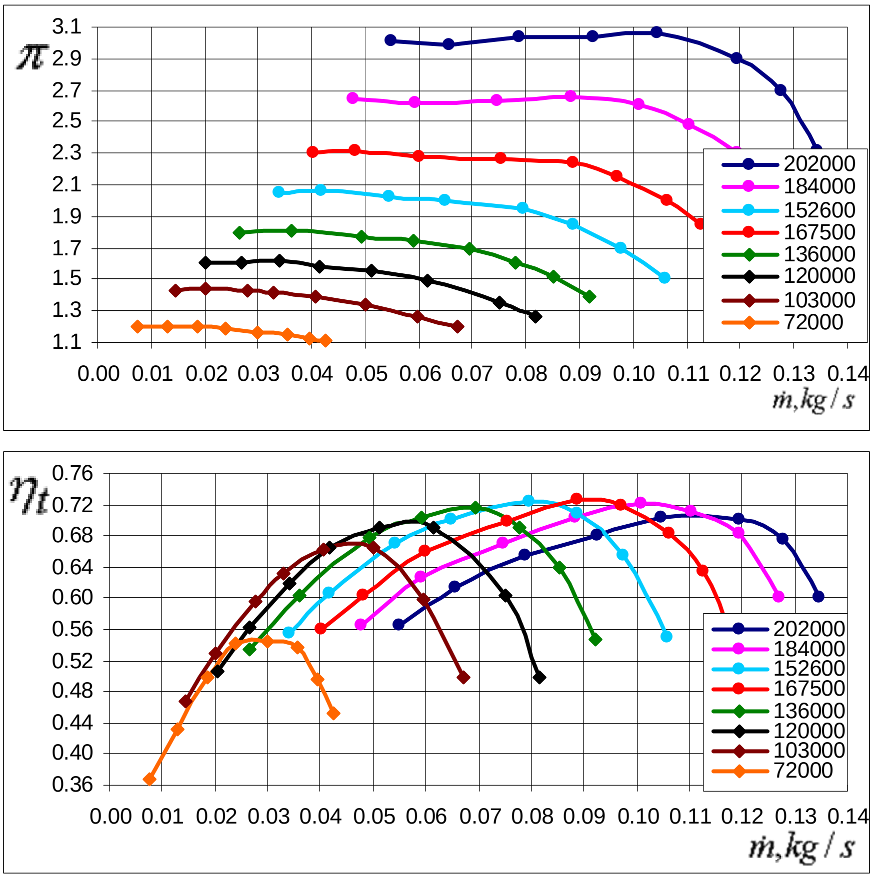

Fig. 2. Compressor performance curves. Efficiency is calculated by Eq. (1)

Supposition 3. It is experimentally proved [9] that Euler coefficient $\psi_T = c_{u2}/U$ is linear function of a tip flow coefficient φ_2 for subsonic impellers and is independent of Mach numbers for a given impeller – Fig. 3. The supposition is that for the supersonic impeller under analysis the Euler work coefficient function is linear too but can be different for different Mach numbers: $\psi_T = f\left(\varphi_2, M_U\right)$. Three values: $\varphi_{2des}, \psi_{T\,des}, \psi_{T0}$ are used to establish linear function $\psi_T = f\left(\varphi_2\right)$, fig. 3.

Modeling of performances at 202000 RPM. The new version of Universal modeling computer programs was applied. The set of 60 empirical coefficients in two dozens of algebraic equations guarantees accuracy of efficiency calculations inside 0,6% at design regime for subsonic stages in a wide range of design parameters $\Phi_{des}, \psi_{Tdes}, M_{Udes}$.

The "IDENT" program was applied to model compressor performance curves. Normalized stage parameters are calculated at the same flow rates with measured parameters. Values φ_{2des} = 0,30, $\psi_{T\,des}$ = 0,60, ψ_{T0} = 0,77 were chosen to match measured data of work input coefficient. The result is shown in the Fig. 4 as function $I = f(\text{F})$. Matching is quite good in the practically important part of the performance.

Surface roughness value participates in efficiency calculation process. The value 25 micrometers for the impeller and the diffuser surfaces were applied arbitrary as an example.

The first calculation of efficiency was made with standard set of 6th generation empirical coefficients. The matching was qualitatively good but calculated values were by several percents higher. For better matching the empirical coefficient that controls surface friction losses was increased by 40% in comparison with a value applied to big industrial compressors. The result of modeling is presented in the Fig. 4. The correlation of performance curves is rather good in a range $\Phi = 0{,}10\text{-}0{,}125$. It is important that only one empirical coefficient was changed for it. Three dozens of other empirical coefficients that participated at this exact calculation are the same as in well-proven calculations of industrial compressors.

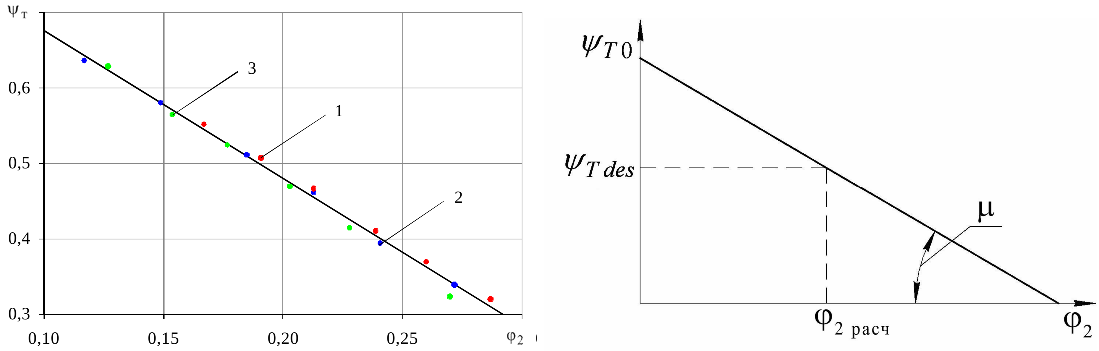

Fig. 3. Typical Euler coefficient function for a subsonic industrial impeller (left) and linear modeling of $\psi_T = f(\varphi_2)$

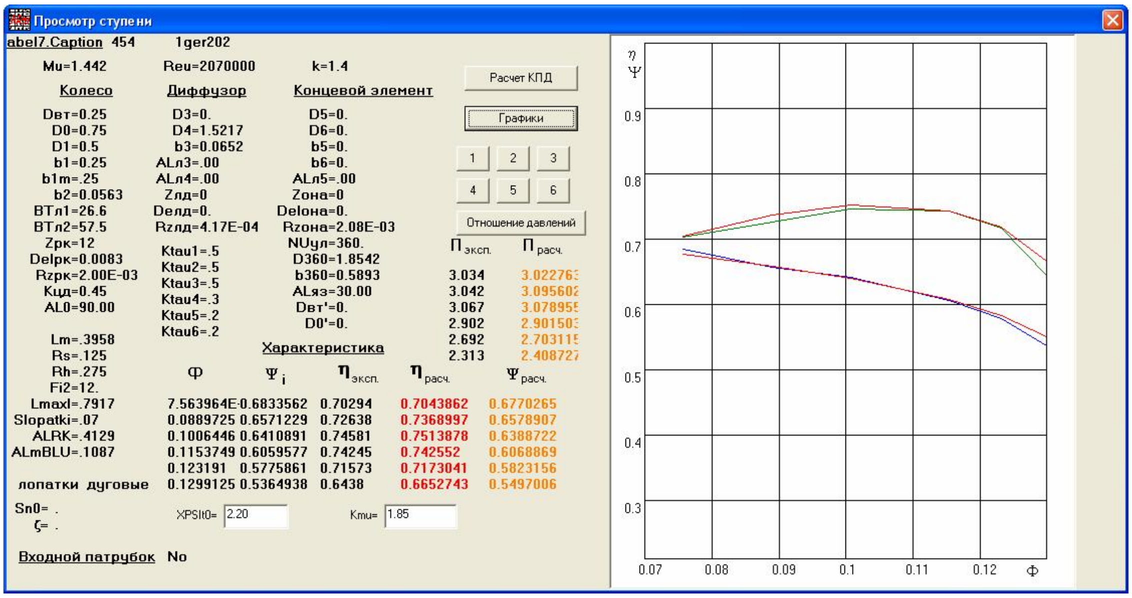

Fig. 4. Test and calculated performances as presented at the “IDENT” program display. $\eta_t, I = f(\Phi)$

Matching of pressure ratio curves repeats matching of efficiency. Mismatch at maximum flow rate is usual in modeling practice. It is not very important as these regimes are seldom used. Modeling is visibly more optimal at small flow rates where pressure ratio starts to decline.

Let us notice that calculated curve $I = f(\Phi)$ is not linear at 202 000 and 184 000 RPM (Fig. 4) and practically linear at 102000 RPM. It demonstrates compressibility effect, as connection of φ_2 and Φ depends on impeller exit density ρ_2 that depends on RPM:

$$\varphi_2 = \frac{\Phi}{4\frac{b_2}{D_2}} \frac{\rho_{inlt}}{\rho_2} \cdot \qquad (3)$$

Modeling of performances at 72 000-184 000 RPM. Efficiency curves were calculated by the set of empirical coefficients that was applied at 202000 RPM. Normalized parameters were calculated firstly by "IDENT" program with internal head defined as $H_i = C_p\left(T_{ext} - T_{inlt}\right)$ - measured temperatures. As result, calculated Pressure ratio based on measured T_{ext}, T_{inlt} does not match to measured one:

$$\pi_t = \left(\frac{T_{ext}}{T_{inlt}}\right)^{\frac{\gamma}{\gamma-1}\eta_t} . \qquad (4)$$

The temperature ratio depends on work coefficient and Mach number:

$$\frac{T_{ext}}{T_{inlt}} = 1 + \frac{I \cdot U^2}{c_p T_{inlt}} = 1 + I \cdot M_U^2 \cdot \qquad (5)$$

Linear functions $\psi_T = f\left(\varphi_2\right)$ corresponding to the best pressure ratio matching were defined in series of calculations. Proper values of work coefficient I were calculated by an empirical equation in the computer program.

It can be noticed that at 184 000 RPM calculated work coefficient exceeds measured value. It means that the compressor transfers to ambience more heat that it receives from its turbine. The situation is opposite at 103 000 RPM.

It appeared that the independence of the empirical function $\psi_T = f\left(\varphi_2\right)$ of Mach criteria (4, 5) does not take place in modeling of non – adiabatic tests. To match pressure patio at different RPM individual values of K_μ and ψ_{T0} coefficients were found for different RPM, i.e. for different Mach numbers.

MODELING METHOD PRINCIPLES

To model compressor performance following items are necessary: a model of mechanical work input, a model to calculate head loss in a flow path and an algorithm of gas parameter calculation in control planes of a flow path.

It was shown in the Fig. 3 that two values of an Euler work coefficient define a work input performance $\psi_T = f\left(\varphi_2\right)$. To calculate ψ_{Tdes} the scheme and formulae presented at [6] are applied –Fig. 5. In accordance with the scheme:

$$\psi_T = 1 - \varphi_2 ctg\beta_{bl2} - \Delta\bar{c}_{u2} . \quad (6) \qquad \Delta\bar{c}_{u2} = K_\mu \frac{\psi_T}{z \cdot \bar{l}_{m2}} , \quad (7)$$

$\bar{l}_{m2}$ - normalized meridian distance from a gravity center of a velocity diagram to an impeller exit, $K_\mu > 1$ - an empirical coefficient.

- a value at zero flow rate $\psi_{T0} < 1$. There are empirical correlations to calculate both parameters in case of big subsonic impellers. In case of the studied small compressor these values were defined for each RPM by matching process of pressure ratio curves.

Loss calculation procedure includes definition of friction drag force coefficients on all surfaces of a flow path, mixing loss coefficients where flow separation occurs and incidence losses at off- design regimes. Several dozens of empirical coefficients correlate loss coefficients with velocity level and gradients and with similarity criteria. To extend the method to transonic and supersonic stages inductive losses calculation was added and negative influence of a choke on a boundary layer was taken into account. More detailed description of 3D impellers and several improvements of iterative processes in thermodynamic calculations are added too.

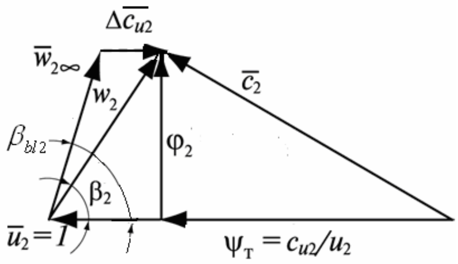

Fig. 5 Velocity triangle at an impeller exit

Several problems arouse in calculation of small-size turbocharger compressors due to simplified description of stage geometry in applied programs. The problems and ways of solution:

- the modeled impeller has split blades. The applied program calculates impellers with all full length blades. For approximate modeling 12 full length blades were applied with the half of a real thickness;

- there is no open impellers calculation in the programs at the moment as open impellers are practically not used in industrial compressors. Calculations are made for closed impellers. The influence of disc friction and labyrinth seal leakage is rather small as the impeller is of high flow rates ($\Phi_{des} \approx 0{,}13$);

- 1D calculation of flow at an impeller inlet is most important to define non-incidence flow rate. There is no problem for a 2D impeller where a blade inlet angle β_{bl1} is close to constant along the leading edge height. There is a method to choose necessary value of β_{bl1} for impellers designed by CD SPbTU method. Influence of blade blockage and blade load on critical streamline direction is calculated by formula presented in [7]. For presented compressors the values were found by series of calculation with different β_{bl1} . The value with better matching between calculated and measured flow rates was chosen for final modeling process.

MODELING RESULTS FOR SMALL COMPRESSOR

The 6th generation program was applied. As it was shown above, the surface roughness 20 -25 micrometers for the impeller and the diffuser was accepted and only one of empirical coefficients was modified to model efficiency curve at 202000 RPM. The measured temperatures were accepted as corresponded to adiabatic compression at that RPM. For other RPM efficiency was calculated by the same set of empirical coefficients as at 202000 RPM.

The linear functions $\psi_T = f(\varphi_2)$ were defined for each RPM individually on the principle of the best correlation of pressure ratio curves. Corresponding values of K_μ and ψ_{T0} are presented at the table 1.

For systematization of results the individually chosen values of $K_\mu, \psi_{T0} = f(M_u)$ were approximated by the equations:

$$K_\mu = 3(1{,}45 - M_u)^{3,2} + 1{,}7, \quad (8) \qquad \psi_{T0} = 0{,}5 + 3{,}5(M_u - 0{,}735). \quad (9)$$

Graphic representation of the table 1 content is shown in Fig. 7. The empirical coefficients are calculated for test data in a range of M_u =0.735-1.442.

Table 1

Empirical values of K_μ and Ψ_{T0} for different RPM

1	2	3	4	5	6
$n \cdot 10^{-3}$ RPM	M_u	K_μ	ψ_{T0}	$K_{\mu\ approx}$	$\psi_{T0\ approx}$
202	1.442	1.800	2.100	1.700	2.100
185	1.313	1.650	2.100	1.705	2.100
167	1.192	1.800	2.100	1.739	2.100
154	1.089	1.900	1.700	1.815	1.739
137	0.971	2.200	1.200	1.985	1.326
121	0.857	2.200	1.200	2.264	0.927
104	0.735	2.700	0.500	2.725	0.500

The result of modeling is presented in Fig. 6.

The independence of an empirical function $\Psi_T = f(\varphi_2)$ of Mach criteria (4, 5) does not take place in modeling of non – adiabatic tests. To match pressure ratio at different RPM it was necessary to choose individual values of K_μ and Ψ_{T0} coefficients for different RPM, i.e. for different Mach numbers. The set of empirical coefficients was the same that was used for modeling of the compressor at highest RPM.

The results are presented in Fig. 7 (above). The individually chosen values of K_μ and Ψ_{T0} are presented in columns 3, 4 in Table 1.

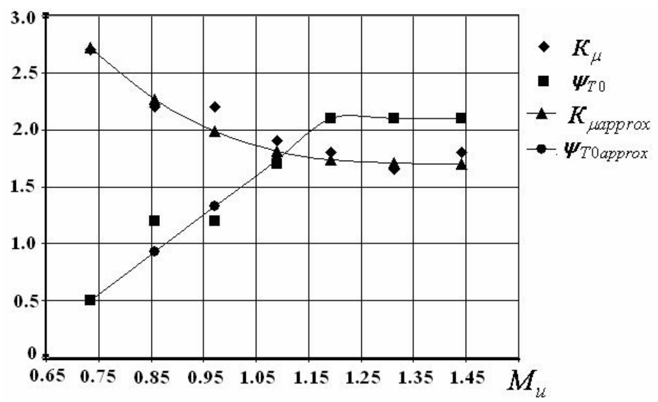

Fig. 6. Graphic representation of $K_\mu, \psi_{T0} = f(M_u)$ by formulae (8), (9)

Pressure ratio curves prediction with approximated values of $K_\mu, \psi_{T0} = f(M_u)$ in Fig. 7 demonstrates acceptable result. Let us notice that $K_\mu, \psi_{T0} = f(M_u)$ values are practically constant for range of $M_u > 1{,}15$. It means that performance $\Psi_{T0} = f(\varphi_2)$ is independent of M_u at high Mach numbers – as it is independent in case of subsonic stages tested adiabatically. It is possible to propose that deviation of the independence rule does not reflect flow character under different M_u but is due to strong heat transfer processes at non-adiabatic tests.

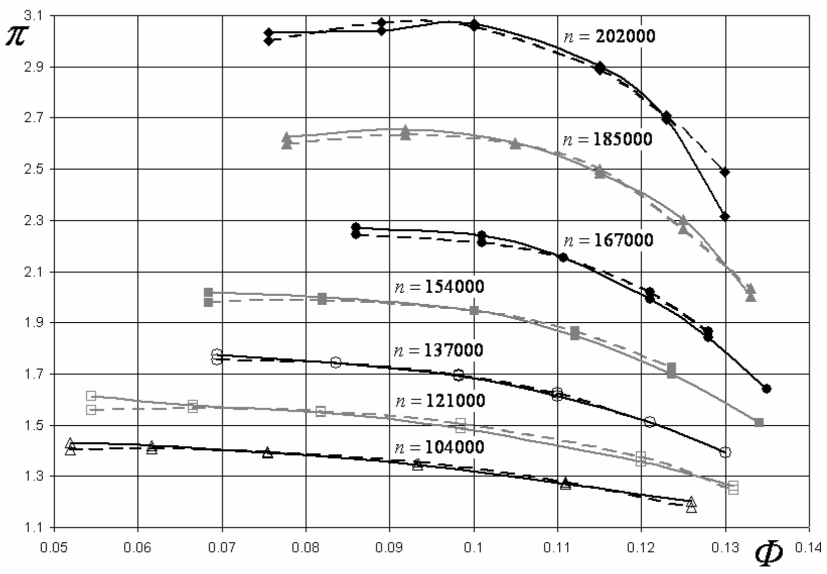

Fig. 7. Pressure ratio performances of TC-1 compressor in a range of RPM. Solid – test, stroke – modeling. Above – individual, below – approximated values of K_μ and Ψ_{T0}

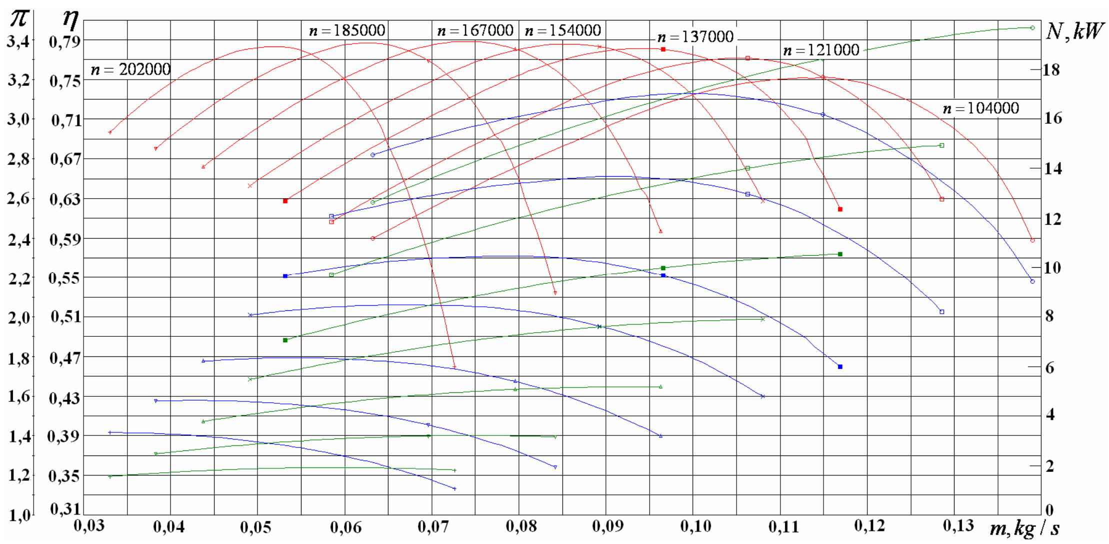

Fig. 8. Performance map of the studied compressor calculated with modified set of empirical coefficients and approximated values of $K_\mu, \psi_{T0} = f(M_u)$

The performance map for different RPM of the compressor was calculated by the computer program CSPM-G6E with the mentioned above set of empirical coefficients and approximated $K_\mu, \psi_{T0} = f(M_u)$ values from the column 5, 6 of Table 1. The performance map is shown in Fig. 8.

The configuration of performance curves seems quite logical. Up to the time when more universal ways of modeling would be available the described above methods of modeling could be recommended for practical use.

CONCLUSION

Turbine – compressor heat exchange influences exit temperature at different level at different RPM in turbochargers. Compressor performance is especially difficult to model because of indefinite mechanical work input that is measured by temperature rise in a compressor. If there are test data for a supercharger compressor at wide range of RPM the described modeling method can be applied for more or less reliable performance modeling. The Universal modeling computer programs [4] serve as a basic tool for modeling. The following should be done:

- the empirical coefficient that controls friction losses must be increased by 40% of the value in a standard set of Universal modeling method,
- the pressure ratio performance curves matching must be achieved by variation of function $\psi_T = f(\varphi_2)$ for each RPM,
- the empirical formulae (7), (8) are the key matter to define function $\psi_T = f(\varphi_2)$ for each RPM. For any other compressor the exact equations can be different.

The application of the modeling procedure to other compressor test data could demonstrate validity or inconsistence of the proposed routine.

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