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Radial Compressor Test Data Processing with Real Gas Equation of State

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Abstract. In this paper, a calculation of radial compressor characteristics from measured data is dealt with. The measured quantities as temperatures, pressures, working gas relative humidity, shaft rotational speed and torque are processed to obtain absolute and dimensionless parameters of the given machine. The methodology is generalized to both experimental and production machines and it deals with incorporation of any suitable equation of state. A modular and configurable in-house code is used for this purpose. It adopts the ASME performance test code for compressors scheme of measuring stations, stages, sections and machines. The theoretical background and the structure of the program are described. Benedict-Webb-Rubin equation of state for real gases is implemented. Gas humidity, orifice flowmeter, bearings mechanical losses and torquemeter measurements are covered. Various cases of measuring station calculations considering high speed and/or rotational flow based on total and static state parameters are elaborated. Polytrophic compression and the accuracy of evaluation of polytrophic efficiency are discussed in terms of various real gas corrections. Conversion of the results to comparable form is explained. The conversion involves standard gas intake state and composition and corrections to impeller rotational speed discrepancies.

INTRODUCTION

A complete computer program for measurement evaluation of radial compressor testing is presented. The required outputs are listed and then the types of machines and measurement setups are explained. The problematics of real gas behavior of the working fluid is introduced along with the approaches to deal with it. Then the algorithm is commented on as a whole, covering the main part of calculation process and omitting the data manipulation. The procedures carrying out the most interesting physical phenomena are described in detail.

The project was motivated by the needs of experimental single stage compressor test bed of Howden ČKD located in Pilsen, Czech Republic. The outcome of the standard measurement is the compressor stage working characteristics, which are used as an input for commercial radial compressor design process. Parts of this process have been handled manually in past few years. These parts are mentioned as the possible extensions to the presented software.

There are intentions to use the same code to analyse the measured data from the commercial machines in operation or under construction to have a consistent methodology. Some functionality is missing for this purpose, but the architecture allows various add-ins.

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PROBLEM DESCRIPTION

Calculation outputs

The most important outputs are the integral performance curves of the compressor. Typically, this involves discharge total pressure p_D, shaft power N and efficiency η versus gas mass flow rate G for given rotational speed n. Discharge temperature T_D may also be useful. These parameters can also be expressed in dimensionless manner (efficiency is dimensionless already): pressure ratio π and power coefficient σ versus flow coefficient ϕ for given reference Mach number Ma. There are two useful types of efficiency - polytrophic and isentropic based on what is considered to be the theoretical ideal of required power. Isothermal efficiency is usually not suitable for turbomachinery.

Besides the integral quantities, one may be interested in local state variables at various measuring stations. It is generally possible to measure (wall) static and total pressure and total temperature anywhere in the compressor excluding the impeller interior. To obtain complete information about the gas thermodynamic state, pressure and temperature needs to be measured in a single cutting surface perpendicular to the meridional component of flow velocity in order to be able to precisely correct the measurements and evaluate static parameters, density and flow velocities. This applies to the stations where high flow velocities occur, the flow is compressible and sources of heat and momentum are close. It is not always possible or feasible to equip such station completely, so corrections, simplifying assumptions or extrapolation may be applied. Flow properties as Reynolds or Mach number may also be expressed.

Experimental and production machines

The methodology was created and used for an experimental single stage radial compressor (Fig. 1), which is characterized by relative simplicity of the machine itself, but it is equipped with extensive number of sophisticated measurement devices in maximum reasonable number of measuring stations. The stations often incorporate many spatially distributed sensors to make sure a local phenomenon does not affect the value valid for the whole area of the station. It also allows a simple description of velocity and temperature profiles in the station and localization of the mentioned local anomalies. Only the station in front of the impeller or in front of the inlet guide vanes (if installed) and the station behind the straightening vanes at the exit from the stage are used for integral characteristics evaluation. Theoretically any pair of stations can be used to gain insight in the gas flow process parameters between them. The single stage forms the only one section of the machine.

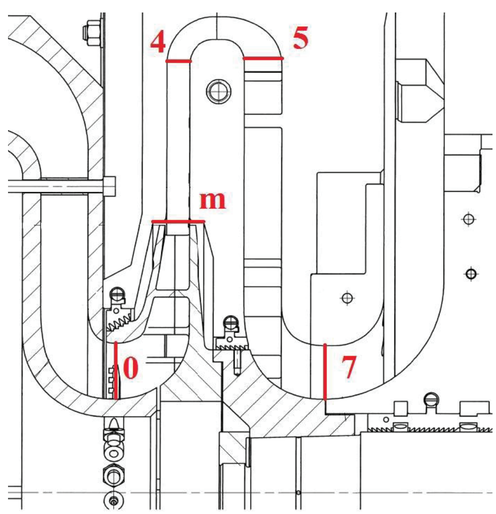

FIGURE 1. Experimental compressor meridional cross-section with measurement stations highlighted in red. Flow path description: 0 to m – impeller, m to 4 – diffuser (vaneless in this case), 4 to 5 – return bend, 5 to 7 return channel vanes. [9]

Besides the standard measuring campaigns determined to integral characteristics assessments, bearings or rotor seals were tested, flow angle in vaneless diffusers was measured with a three-hole probe. Therefor all measured data are collected whether or not are they a part of the standard set or not, whether or not are they required for the standard calculations. Only low sampling frequency sensors are eligible to be added to the dataset as only stabilized operation points are written out from data-acquisition system. A sliding average is even applied on those quantities. The test bed is equipped with fast response pressure transducers with proper accessories, but these transient data need to be handled separately.

The structure of the program is built while keeping in mind the possible future use for production machines measurements. Such machines may be complex multi-stage and multi-section machines with intercoolers, partial gas discharge between sections and even multi shaft machines with barrel assemblies or scroll diffuser design. They may be working with various gaseous media and almost never with air. Despite their complexity, their measuring stations are located usually in the connecting pipes where the flow velocity is low and also low number of redundant probes down to one is mounted. The code would need to be enriched by the algorithms handling cooling in the heat exchangers including condensation of gas humidity.

FIGURE 2. Production machine in factories large-scale test bed [10]

Real gas

The implementation of the gas state properties along with some important thermophysical properties dependent on them is universal. The limitation is that the input parameters for the equation of state (further EoS) are always pressure and temperature. This is the way most general equations of state used for real gases are explicitly written.

Current code includes perfect gas along with Sutherland formula for viscosity and a real gas equation of Benedict-Webb-Rubin (BWR). The variant with 11 substance constants is used. This equation is commonly used for hydrocarbon gases and their mixtures. The mixture is simple to account for with BWR. As long as constants for all its components are known, new constants for the mixture of given composition can be evaluated easily. This also allows incorporating the humidity of air or any other gas. Calculations near critical point may fail or lead to invalid results. In practice this have only caused problems with some individual carbon dioxide compressors.

The nature of the EoS explicit for pressure and temperature asks for iterative procedures at many instances in the program. This includes mainly the need of calculation of theoretical isentropic compression and another isentropic phenomenon - the static and total parameters of a working fluid are constrained by an isentropic curve. In this case, the iteration becomes nested, e.g. the static pressure is quantified based on the current guess of static temperature and total specific entropy, which remains constant and then the static temperature may change for the next outer loop iteration as the estimation of flow velocity gets adjusted.

ALGORITHMS DESCRIPTION

Top level workflow

The calculation takes place after data loading and averaging the spatial multiples of measured quantities. The averaging itself is not trivial. Cases are handled when total pressure and temperature is sensed by a number of multi sensor rake probes or a set of single sensor probes in a single measuring station. It balances the averaging weights of single values to cancel the influence of a sensor which was turned off due to defect causing irregular spatial distribution of remaining sensors. For static pressure wall taps, it is able to distinguish whether the tap is on the impeller hub- or shroud-side wall. Then the program offers two averaging weights distribution options for cases where the numbers of taps on both sides of flow channel are not equal – equal weight for all taps or equal weight for both sides, where taps are averaged a priori.

The sole calculation is split in four main parts.

First one is the calculation of gas streams involving every orifice flowmeter configured for calculation. In the beginning of this procedure, the correct humidity sensors are found for every orifice and the gas composition of the whole stream is determined, the BWR constants are loaded. The humid gas mass flow rate can then be determined from orifice plate differential pressure sensor using also the temperature and absolute pressure in front of the flowmeter. The process is not commented on later because the real gas EoS causes no difficulties here. The routine is built according to ČSN EN ISO 5167-2 standard on flat plate orifice flowmeters [1].

Second part is the determination of gas thermodynamic state in all configured measuring stations. A paragraph is devoted to the details of the procedure. The top level takes care of using the data of the correct gas stream - the composition of gas and already determined mass flow rate.

Third part is the assessment of power loss in rotor bearings. It is the only influence to the power measured by the torquemeter device which can be relatively reliably quantified and then subtracted from such gross measured shaft power. Then the result can be later compared to the power resulting from thermodynamic calculations. The procedure is a straightforward usage of calorimetric equation applied on the oil flow rate and temperature difference between injection piping and drain basin of the bearings box. Surprisingly, the compulsory thermophysical data of the oil are not fully supplied by the manufacturer and it is not easy to find a relevant source. The book [2] was used as a source, but the original source can be tracked back to the 1920's, where synthetic oils have not existed yet [3]. Reliability of such data is questionable without relevant measured values of the temperature dependent properties.

The bearings loss routine can be theoretically launched independent of the first two mentioned parts, but it has to be finished before starting with the last part.

The last step of data elaboration is the calculation of the compression process for all configured sections. The sections are defined by a pair of intake and discharge station and can be sequential, overlapping or any possible arrangement as the section results can also be a useful standalone output. It can be specified which sequence of section does form a complete machine, so their parameters add up.

The compression is considered adiabatic, although both experimental and production machines are not insulated usually. The mass throughput of the compressor is so large compared to the surfaces where heat transfer to the environment occurs, that this assumption is valid for most cases. After the calculation of the actual measured process, its parameters are scaled to standardized intake gas parameters and also a correction to standardized impeller rotational speed. A single paragraph is dedicated to every of these three sub-steps of section data elaboration.

General concepts

When iterative procedure is employed to express state variables implicitly, initial guess on pressure, temperature or both is made depending on whether any of them is fixed or not. An arbitrarily wide interval around those guessed values is generated and its boundary values are both used for expression of the state variable which is known to remain constant (usually specific enthalpy or entropy). Using linear interpolation, the pressure or temperature value is corrected. A residual of the constant quantity can be evaluated and if it has dropped since the last iteration, the new interval around the new value of pressure or temperature is made narrower. Fast convergence is usually experienced.

The wide range of extreme operating conditions of the experimental compressor motivated some robust implementation features. The conditions can involve problematic compressions where very low compression occurs, low flow rates cause the adiabatic dogma is false, low efficiency causes polytrophic exponent to be very large or even negative, and similar.

At some points in the code, limiting functions of minima and maxima were applied to the interpolated values. Also, every while loop break is triggered by any one condition from a set. The set usually includes relative residual and absolute residual and, in some cases, more trigger quantities are chosen.

Details of the humid mixture routine

The BWR EoS constants for mixtures of gases are expressed based on volume fractions of its components. This is a user input for a dry mixture or mixture with known humidity, but when the humidity is variable, it needs to be measured and the constants of the user-input mixture need to be adjusted to incorporate the right amount of water vapour. Relative humidity is sensed. It is necessary to calculate the water volume fraction from the relative humidity. The water vapour saturation pressure is stated using Antoine equation (1) for temperatures below 276 K [11] and for higher temperatures a proprietary equation of similar exponential type is used. The transition between those two expressions is step-free and practically smooth, although not mathematically smooth. The vapour partial pressure p_{pV} from saturation pressure p_{sat} and relative humidity φ (2) is used to get vapour volume fraction c_V (3). The user-supplier vector of dry mixture volume fractions is then normalized and new BWR constants are calculated for a humid mixture.

$$p_{sat} = 10^{6.44721 - \frac{2617.658}{\mathrm{T} - 273.15 + 270.716}} \cdot 10^6 \quad (1)$$

$$p_{pV} = \varphi \cdot p_{sat} \quad (2)$$

$$c_V = p_{pV}/p \quad (3)$$

Details of the station gas state routine

The primary purpose of this routine is to state the flow velocity in the station and fix static and pressure parameters by doing so. Considering high velocity in a station, there are three possible types of gas state analysis.

1. Non-swirling flow is assumed, total pressure and temperature is used.
2. Non-swirling flow is assumed, static pressure and total temperature is used.
3. Swirling flow, both static and total pressure and total temperature need to be used. Tangential velocity component is determined.

For non-swirling flow the absolute flow velocity equals the velocity normal to the station surface (meridional velocity component), which can be evaluated from mass flow rate, fluid density and surface area as $c_m = c_\rho = G/(S \cdot \rho)$. The density is a static parameter and in all the calculation types the static state is not fully defined.

Enthalpy h is converted to kinetic energy when increasing gas velocity c isentropically, as stated in (4). No further derivation of explicit expressions for dynamic pressure or temperature is possible in real gas. Such expressions valid for compressible flow of perfect gas (5) are not valid for real gas as the specific heat c_p and isentropic exponent κ are not constant and mean value usage is neither always feasible nor accurate enough.

$$\text{valid for real gas:} \begin{cases} h_s = h_c - \frac{c^2}{2} \\ s_s = s_c \end{cases} \quad (4)$$

$$\text{invalid for real gas:} \begin{cases} T_s = T_c - \frac{c^2}{2c_p} \\ p_s = p_c \left(\frac{T_s}{T_c}\right)^{\frac{\kappa}{\kappa-1}} \end{cases} \quad (5)$$

Therefor the missing total or static parameters (based on the type of algorithm enumerated above) need to be iterated with the p, T explicit EoS. The loop is stopped when $(c_\rho - c_h) < tolerance$.

Another phenomenon playing role here is the restitution factor of the total temperature probe. The thermocouple sensor together with specifically designed deflector are bringing the fluid to stagnation at sensing location with only partial success. The sensed value of temperature T_{c0} needs to be corrected and the correction based on restitution factor R is dependent on the unknown velocity of the flow. The corrected total temperature T_c has an effect to the quantification of the velocity, so the equation (6) needs to be a part of the iteration loop mentioned above.

$$T_c = T_{c0} + (1 - R) \cdot (T_c - T_s); \quad R = f(c) \tag{6}$$

Well designed static pressure wall tap values can be used directly as well as total pressure values if the probe deflector is directed to face the flow perpendicularly. The angle setting has a relatively large insensitivity margin of around $\pm 15°$.

The swirling flow procedure is similar, but the absolute velocity defined from dynamic enthalpy $h_c - h_s = c^2/2$ is not comparable to the meridional velocity component $c_m$defined from mass flow rate. Pressures are fixed from the measurement. The total temperature gets corrected with restitution factor in the iteration loop whose primary task is to reach correct static temperature T_s and absolute velocity based on $h_c - h_s$ and meridional velocity based on ρ which, of course, is also adjusted in the loop. The formerly mentioned threshold expression for the loop is not applicable here, so only the relative change of c between iterations is checked to be below threshold to stop the loop.

Details of the polytrophic compression routine

Main task of this procedure is to find ideal isentropic discharge temperature based on the fully determined states at beginning and end of the irreversible adiabatic process. The theoretical compression is defined by the same staring gas state and identical pressure at the end. The only unknown is the temperature at the end of the process, which is iteratively found based on the constraint of constant entropy. Again, the iterative process is necessary due to implicit nature of EoS to entropy. The isentropic discharge temperature allows stating the isentropic specific work and later the isentropic efficiency.

Finally, ideal polytrophic specific work is expressed in algebraic manner to later state the polytrophic efficiency. See [4] for explanation the advantages of using polytrophic efficiency along with the isentropic efficiency There are various attempts on how to account for the change of polytrophic exponent during the process in real gas. One from Schultz, dated 1962, is also encouraged by the ASME PTC 10 standard [5, 6]. An alternative can be the Mallen-Saville formula, Oldřich's polynomial approximation [7] or finally, the numerical solution of the differential equation he mentions in the cited article. First two alternatives are implemented at the moment. Schulz [8] uses parameters of the ideal adiabatic compression, see equations (7) to (10) while Mallen-Saville does not need such data - (11). The stability of the Oldřich's calculation seems questionable at first glance, but it may be interesting to try to implement the method in the future and see the results. The solution of the differential equation is not only tedious to implement, but also takes prohibitive computational costs when the calculation shall be used in live acquisitioned data evaluation and visualization.

$$n_{ise} = \frac{\log_{10}\left(\frac{p_{cD}}{p_{cI}}\right)}{\log_{10}\left(\frac{v_I}{v_{Dise}}\right)} \tag{7}$$

$$n = \frac{\log_{10}\left(\frac{p_{cD}}{p_{cI}}\right)}{\log_{10}\left(\frac{v_I}{v_D}\right)} \tag{8}$$

$$f_{corr} = \frac{h_{2ise} - h_1}{\frac{n_{ise}}{n_{ise}-1} \cdot (p_{cD} \cdot v_{iseD} - p_{cI} \cdot v_I)} \tag{9}$$

$$a_{tid\ Schultz} = f_{corr} \cdot \frac{n}{n-1} \cdot (p_{cD} \cdot v_D - p_{cI} \cdot v_I) \tag{10}$$

$$a_{tid\ M-S} = a_t - (s_D - s_I) \cdot (T_{cD} - T_{cI}) / \log_{10} \frac{T_{cD}}{T_{cI}} \tag{11}$$

When computing the regimes of experimental compressor at low reference Mach number (< ~0.5), the values of polytrophic work differ a lot. The comparison with the exact solution would be valuable tool for validation purposes. The low Mach numbers also occur in the production machines working with low molar mass like He, H_2.

Details of the intake gas state and the composition scaling routine

Similarity theory in thermodynamics gives tools to rescale the results obtained from the measured compression process in the machine section to any other state and composition of the gas in the section intake and possibly also different impeller outer diameter. It is necessary to process the measured data in this way to be able to compare the characteristics. A reverse process is applied to the datasets when designing a new machine for a specific application. Dimensionless numbers listed in the paragraph "Calculation outputs" are kept constant and so the absolute scaled quantities can be expressed as below (indices: r-rescaled, m-measured; D-section discharge, I-section intake):

- Total discharge pressure from constant pressure ratio π: $p_{cDr} = p_{cIr} \cdot \frac{p_{cDm}}{p_{cIm}}$
- Impeller rotational speed from constant reference Mach number Ma: $\Omega_c = \Omega_m \cdot \frac{a_{cIr}}{a_{cIm}}$
- Mass flow rate from constant flow coefficient ϕ: $G_r = G_m \cdot \frac{\rho_{cIr}}{\rho_{cIm}} \cdot \frac{a_{cIr}}{a_{cIm}}$
- Section specific work from constant power coefficient σ: $a_{tr} = a_{tm} \cdot \left(\frac{a_{cIr}}{a_{cIm}}\right)^2$

In the above equations, a_c, ρ_c denotes speed of sound and density evaluated from gas stagnation state and a_t stands for specific technical work done on the gas in the section

The rescaled discharge total temperature T_{cDr} needs to be searched for based on the enthalpy change in the section. Initial guess is made by perfect gas theory (12).

$$T_{cDr} = (T_{cDm} - T_{cIm}) \cdot \frac{T_{cIr}}{T_{cIr}} + T_{cIr} \tag{12}$$

The above described routine for polytrophic compression is reused in an iteration loop to correct the initial guess to conform to real gas behavior. Discharge enthalpy reached with reference intake gas state is defined in (13).

$$h_{cDr} = h_{cIr} + a_{tr} \tag{13}$$

T_{cDr} is then adjusted till it yields the correct discharge enthalpy h_{cDr}. All the parameters of the fictional reference intake state compression are determined in the process by the polytrophic compression routine – besides T_{cDr} and back-checked h_{cDr} those are the isentropic and ideal polytrophic specific work.

When there are more sections in a sequence, they need to be evaluated streamwise. The user prescribed reference intake state is applied to the intake of the first section and the scaled discharge parameters apply to the successive station.

Details of the rotational speed correction routine

An estimation is used to correct the discrepancies in the setting of impeller rotational speed to reach the correct reference Mach number during measurement. It also helps the machine design when data for exact required Mach number are not available and the designer needs to use the closest available performance curve.

Again, it is based on the similarity theory, but the assumptions are more simplifying here and the procedure is correct and accurate only for slight changes in Mach number. One should view it as a type of extrapolation.

The algorithm assumes the real compression power and flow coefficients and theoretical isentropic compression to remain constant with the change of impeller rotational speed. This yields the corrected polytrophic and isentropic specific work (a_{tc}, a_{tisec}) and corrected mass flow rate (G_c) based on ratio of corrected and reference intake rescaled impeller rotational speed (Ω_c, Ω_r), see equations (14) to (16). The gas intake state here is the same as for the rescaled quantities, so the density ratio is equal to unity and is not further mentioned.

$$a_{tc} = a_{tr} \cdot \left(\frac{\Omega_c}{\Omega_r}\right)^2 \tag{14}$$

$$a_{tisec} = a_{tiser} \cdot \left(\frac{\Omega_c}{\Omega_r}\right)^2 \tag{15}$$

$$G_c = G_r \cdot \frac{\Omega_c}{\Omega_r} \tag{16}$$

First, the isentropic discharge pressure and temperature is searched for iteratively. Loop break threshold is the close approach to a_{tisec}. The initial guesses of corrected discharge total pressure and temperature are derived again from the perfect gas theory, see equation (17) and (18). The total isentropic discharge pressure for actual rotational speed p_{c7c} is adjusted using the ratio of rotational speeds (Ω_g/Ω_c). The "g" subscript denotes parameters for reaching the exact desired reference Mach number. The "c" subscript denotes parameters resulting from the intake gas state correction routine described in the previous paragraph. The adiabatic exponent κ and specific heat c_p are evaluated from EoS in arbitrary suitable pressure and temperature as accuracy can be limited at this point.

$$p_{cDc} = \left\{\left[\left(p_{c7c}{}^{\frac{\kappa-1}{\kappa}}\right) - 1\right] \cdot \left(\frac{\Omega_g}{\Omega_c}\right)^2 + 1\right\}^{\frac{\kappa}{\kappa-1}} \tag{17}$$

$$T_{cDisec} = T_{cIc} + \frac{a_{tisec}}{c_p} \tag{18}$$

Both parameters are adjusted while ensuring constant entropy. Similar procedures were already described.

After the isentropic state change is defined, the corrected polytrophic compression can be accounted for, yielding the same discharge pressure as the isentropic compression and the enthalpy change equal to a_{tc}. The initial guess of temperature in (19) is analogical to the isentropic case above.

$$T_{cDc} = T_{cIc} + \frac{a_{tc}}{c_p} \tag{19}$$

Ideal polytrophic work for the corrected compression can be stated according to the approximative approaches commented in the paragraph "Details of the polytrophic compression routine".

RESULTS

An example of program graphic output is presented in Figures 3 to 6. As an input of calculations, a set of experimental data from the Howden testing facility in Pilsen were taken. The curves present compressor working characteristics at a constant value of reference Mach number ($Ma = 0.9$ [-]), which practically means almost constant impeller rotational speed. Every single curve represents a different setting of inlet guide vanes - from -15° (overloading) to +75° (partial load). The curve points remain discrete in the calculation, no regression or fitting is done.

The calculation goes as it was described in the previous chapter - the measured values are correctly averaged, gas streams are analysed taking gas humidity into account, then state of gas in the measuring stations 0 and 7 (see Fig. 1) is evaluated and the measured compression in the section is described based on these intake (0) and discharge (7) stations. Measured compression data are then rescaled for prescribed intake gas state and composition. After that, they are corrected for exact prescribed reference Mach number. In Figures 5 and 6, the dashed curves suffixed D in the legend denote the torquemeter measurement based data.

As the precise data are confidential, they were normalized by arbitrarily norms not to show absolute values used in the compressor development process of the Howden company.

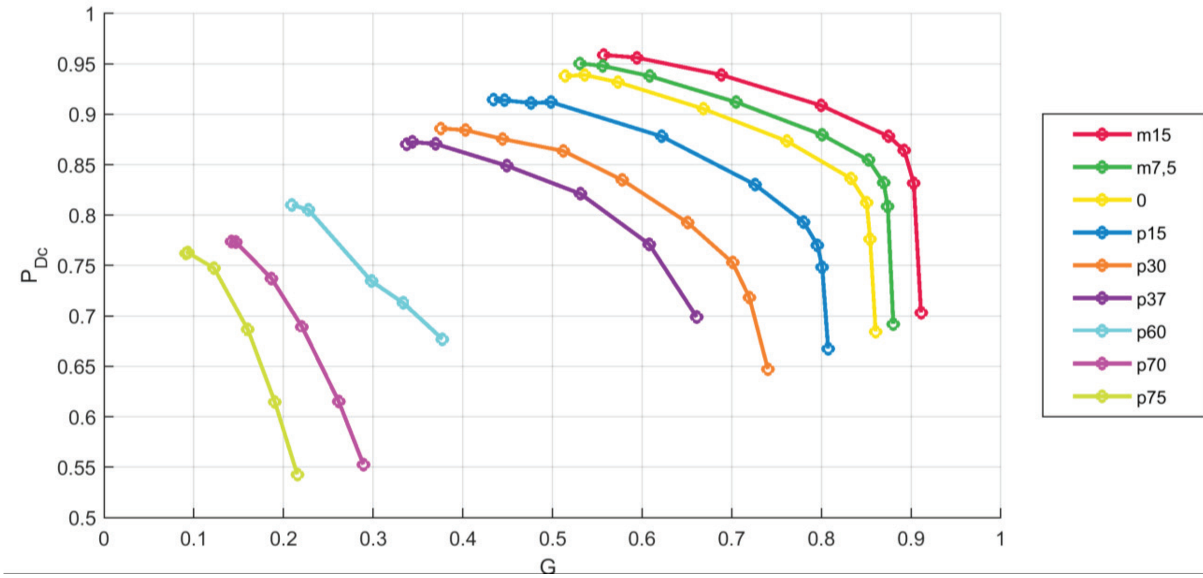

FIGURE 3. Normalised discharge pressure-mass flow rate performance curves for vaious IGV setting

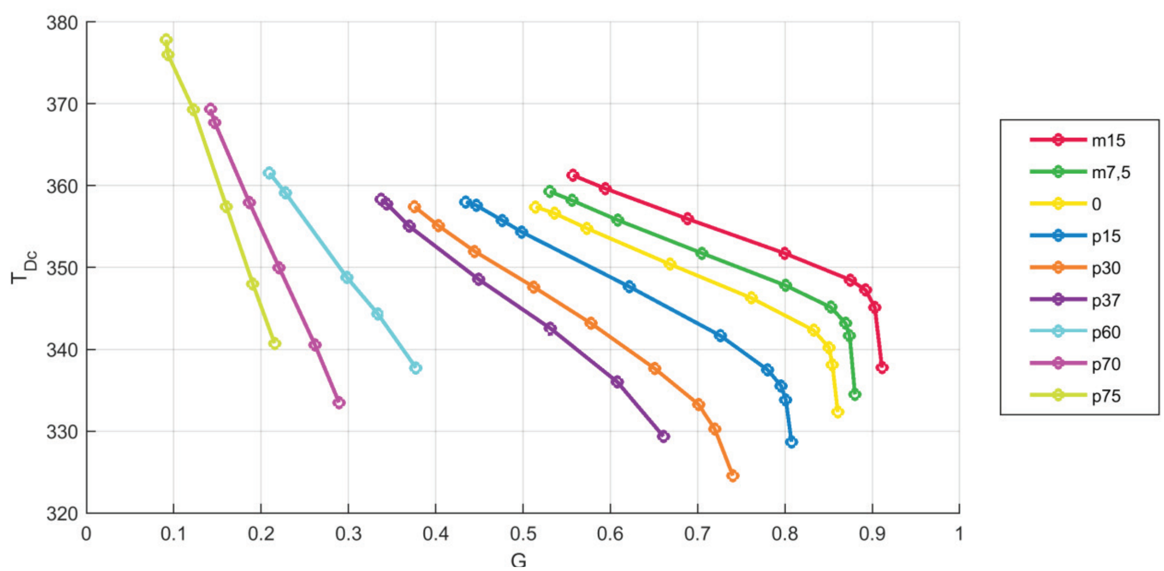

FIGURE 4. Normalised discharge temperature-mass flow rate performance curves for vaious IGV setting

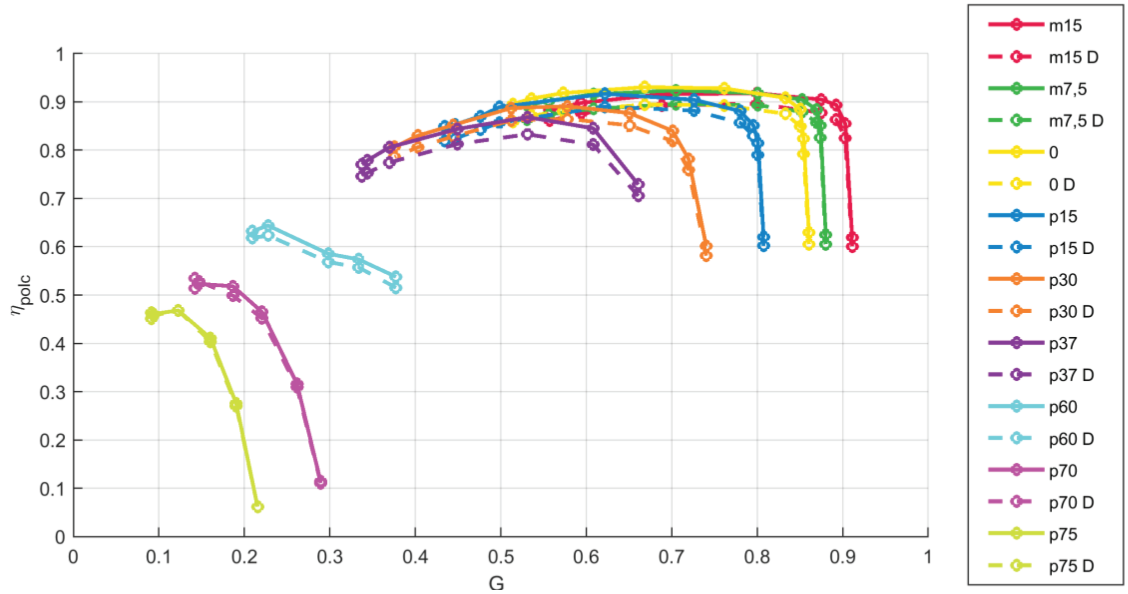

FIGURE 5. Normalised polytrophic efficiency-mass flow rate performance curves for vaious IGV setting

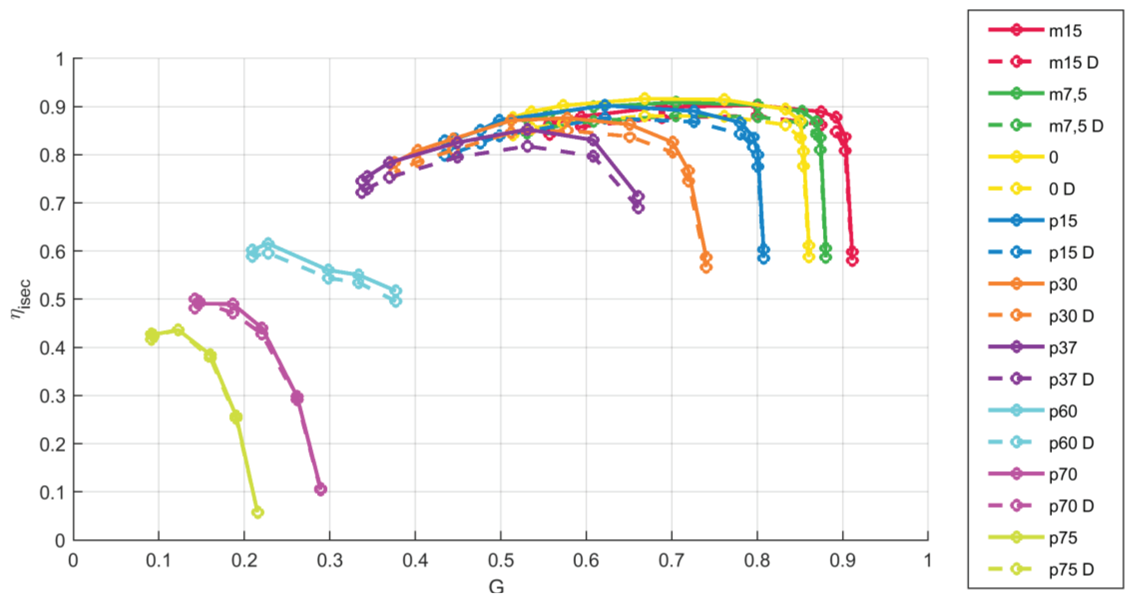

FIGURE 6. Normalised isentrophic efficiency-mass flow rate performance curves for vaious IGV setting

CONCLUSIONS

From the point of view of thermodynamic and fluid flow phenomena, the most important approaches to the processing of performance characteristics of radial compressors were described. All the calculations respect the real gas behavior of gases with no sacrifices. Most of the procedures are ready to be used for a production machine measurement data with certain level of user comfort.

Extensive work would be required to carry out the research to find out the best approach to express the ideal polytrophic work with the use of real gas. It is a problem which needs to be tackled very often and the investment would pay off according to authors' opinion.

A common approach to handling turbomachinery performance curves is polynomial fitting. This should be implemented in the future, but the all the calculations would remain discreet-point-wise till the end of the algorithm as presented in this work. A semiempirical model to cancel the influence of the used type of rotor seals on the experimental machine should be then used and after that a final output of the polynomial fitted curves could be generated.

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