Theoretical study of gas parameters influence on reciprocating compressor performance intended for NGV-refueling compressor station

A A Kotlov* and L G Kuznetsov

Peter the Great St.Petersburg Polytechnic University, Polytechnicheskaya 29, St. Petersburg, 195251, Russian Federation

* E-mail: kotlov_aa@spbstu.ru

Abstract. Ensuring the reliable and efficient operation of the reciprocating compressors in the wide range of the operating parameters is one of the important issues at designing, manufacturing and operating the compressor equipment. The quality of any equipment is determined by the improvement of calculating methods used at its designing. While designing the reciprocating compressors with a high compression degree the issue arises of determining the actual properties of the medium being compressed. The theoretical investigation of the gas initial parameters influence on the reciprocating compressor operation process is represented in this study. The results of the comparison of the reciprocating compressor operation for compression of ideal and actual gas are given in this study.

1. Introduction
As far back as early 1980s, while developing the requirements to NGV-refueling compressor stations (NGVRCS), the designers faced the fact that it is impossible to develop the universal scheme of NGVRCS. Such scheme cannot be suitable for all operating conditions. The station parameters highly depend on suction pressure which may vary within a significant range from 0.15 to 12.0 MPa subject to the station location. For example, suction pressure for small garage-type stations is equal to pressure in the intra-urban gas network. For stations located not far from the main pipelines such pressure may vary from 2.5 to 6 MPa. Naturally, the higher the suction pressure, the less the costs for production of the compressed gas unit. Consequently, at designing the compressors making part of the NGV-refueling compressor stations it is necessary to solve the following tasks:
- ensure the compressor serviceability in wide ranges of operation parameters by suction and discharge parameters (pressure and temperature);
- ensure reliable and efficient operation of the automatic valves in variable modes;
- determine the compressed gas practicability parameters.

The joint solution of these tasks requires multi-variant calculations in order to find the best compressor parameters. For this purpose it is necessary to implement modern methods of designing at production, namely, the CAD technique. Application of CAD systems makes it possible to shorten the time, reduce costs and errors in design works, and increase quality of products. Mathematical simulation, as early as the design stage, makes it possible to forecast compressor performance in different operation modes.
**Objective** is to improve the efficiency of the reciprocating compressor operating as part of the NGV-refueling compressor station by implementation of digital design methods.

**The research object** is the four-stage reciprocating compressor intended for use in the gas-transmission and gas-filling compressor stations. Figure 1 contains a general view and movement mechanism of the compressor under consideration. The given compressor is developed by Compressor JSC and made with a crankshaft vertical configuration. The compressor is intended for compression of methane or natural gas to the final pressure of 24.6 MPa, dried and purified to the required parameters.

![Figure 1. Scheme of compressor with vertical shaft: a) general view; b) motion work.](image)

The unique vertical design of the compressor is specially developed for the gas compression systems and for installation at facilities with a restricted area for equipment arrangement, which makes it possible to place it in small block-boxes and industrial premises with a restricted area, small motor-transport enterprises and on motor vehicles.

Such design makes it possible to reach high balance of the compressor and significantly decrease dynamic loads on foundation, which allows equipment arrangement without additional strengthening of foundations. Consequently, this ensures a low vibration level.

The compressor is designed for different inlet pressures. The range by the absolute initial pressure varies from 0.4 to 0.7 MPa. Number of stages may vary from two to four. The optimum values of the pressure increase stages in cylinders provide low heat release and increased service life of the cylinder-piston groups and valves.

2. **Mathematical model**

The mathematical model (MM) includes: equation of the first law of thermodynamics in notation of calculus, equations of mass flow, equation of state, caloric equations, motion dynamic equation and valve dynamic equation. Thermodynamic equation of variable-mass body is written in the energy form, i.e. the equation of internal energy variation is selected as the main one; at that, pressure and temperature values are deduced from the gas state equation.

The initial system of equations for description of gas properties in the considered cavity is as follows [2-6]

\[
\frac{dU}{dt} = \alpha F_c (T_c - T) - \frac{PdV}{dt} + \sum_{j} i \bar{m}_j - \sum_{j} i \bar{m}_j
\]
\[ \frac{dM}{dt} = \sum_j \tilde{m}_j - \sum_l \tilde{m}_l \]

\[ \rho = M / V; \quad u = U / M; \quad T = f(u, \rho) \]

\[ z = f(T, \rho); \quad P = z\rho RT; \quad i = u + P / \rho \]

where \( U \) — internal energy; \( t \) — time; \( \alpha \) — heat transfer coefficient; \( F_{ct} \) — radiation surface; \( T_{ct} \) — wall temperature; \( T \) — temperature; \( P \) — pressure; \( V \) — cavity space; \( i_j, i_l \) — specific enthalpy of inflowing and outflowing gas; \( \tilde{m}_j, \tilde{m}_l \) — mass flow rate of inflowing and outflowing gas; \( M \) — gas mass; \( \rho \) — density; \( u \) — specific internal energy; \( z \) — compressibility coefficient; \( R \) — gas constant.

The equation of valve shut-off device motion as a function of time is as follows [6-9]

\[ m_{np} \frac{d^2 h}{dt^2} = \xi_p F_c \Delta P - c(h + h_0) - \eta \frac{dh}{dt} + m_{np} g \cos \beta, \]

where \( m_{np} \) — reduced mass of valve moving elements; \( h \) — movement of valve shut-off device; \( \xi_p \) — pressure coefficient; \( F_c \) — passage area in valve seat; \( \Delta P \) — pressure drop at valve; \( c \) — stiffness of valve elastic components; \( h_0 \) — preliminary compression of valve elastic components; \( \eta \) — damping coefficient; \( g \) — acceleration of gravity; \( \beta \) — angle between motion axis and direction of gravity.

Method of intermediate pressure calculation is based on the assumption of mass flow rate equality at all stages. In order to determine intermediate pressure it is necessary to have capacities of all stages as a function of both suction pressure and discharge pressure. Somewhat different approach is used in the study, which makes it possible to simplify and accelerate the calculation process. Let us consider the calculation algorithm through the example of graphical method of intermediate pressure determination by stages of the four-stage compressor (see Figure 2). Functions of mass flow rates of the first and the last stages shall be set up for the given initial and final pressure. Then, an arbitrary value of the compressor mass flow rate is set up, and discharge pressure of the first stage and suction pressure of the last stage are determined by graphs. The obtained values are used for plotting the characteristics of the neighbor stages, etc. At that, when setting a new capacity value at the next iteration step it is necessary to keep in mind that with the discharge pressure increase the stage capacity decreases, and with the suction pressure increase the stage capacity increases. The target values of intermediate pressure will be found when the functions of mass flow rates for the neighbor staged have the common point at the specified capacity. The graphical method of calculation is given for descriptive reasons; the calculation in MM is carried out analytically with automation of the target value search process.
The mathematical model is intended for the analysis of the multi-stage reciprocating compressor operation with account of actual gas properties. The algorithm of actual gas calculation is based on the processing of the known information on gas properties and obtaining the polynomial functions for two dependencies: 1) temperature versus specific internal energy and density; 2) compressibility coefficient versus temperature and density. In order to calculate natural gas parameters the methods of VNIIGAZ is used [10]. Figure 3 shows the compressibility coefficient values for methane with different pressure and temperature values.

Created on the basis of the developed MM was the automated software complex (ASC) intended for calculation and analysis of RC working processes. Figure 4 shows the ASC window in the mode of
variants calculation. The calculation results are represented in numerical and graphical form and contain different performance indicators of both individual stages and the compressor on the whole (indicated power, volumetric capacity, valve efficiency indexes, etc.). The ASC makes it possible, as early as the design stage, to take into account and correct possible adverse consequences of the compressor operation in variable conditions, optimize the performance parameters of compressor valves and assemblies, calculate the forces acting to the motion mechanism and select limit pressures for the given reciprocating force.

3. Calculation results and discussion
As can be seen in Figure 3, the methane compressibility coefficient in the compressor operation zone varies significantly enough. So, this must be taken into account at compressor calculation. Let us analyze the actual gas parameters influence on the working process of the compressor under consideration. Given in Figure 5 are calculation indicator diagrams and valve motion diagrams for the first and the fourth compressor stages with and without account of the gas actuality. As can be seen in the given figures, the actual gas compressibility differs from the ideal gas compressibility. In the first place, the influence is exerted on redistribution of intermediate pressure. Table 1 gives the results of intermediate pressure $P_m$ calculation depending on the initial pressure $P_n$. At that, such influence is more noticeable at the high pressure stages. Conspicuous is the fact that at compression of ideal gas with the initial pressure of 0.7 MPa the mode of the fourth stage "blow-down" takes place, i.e. gas is completely compressed in three stages. This fact is illustrated in Figure 6, where indicator diagrams of the fourth stage are given with different initial pressure values.

![ASC window in the mode of variants calculation](image)

**Figure 4.** ASC window in the mode of variants calculation.

| $P_n$, MPa | Ideal gas | Actual gas |
|------------|-----------|------------|
|            | $P_{m1}$, MPa | $P_{m2}$, MPa | $P_{m3}$, MPa | $P_{m1}$, MPa | $P_{m2}$, MPa | $P_{m3}$, MPa |
| 0.4        | 1.438     | 5.362      | 15.797      | 1.366        | 4.571        | 13.706       |
| 0.55       | 1.953     | 7.183      | 20.124      | 1.839        | 6.053        | 17.557       |

**Table 1.** Distribution of intermediate pressure by stages.
Table 2 contains some calculation results at compression of ideal and actual gas, which characterize the performance of stages. The important difference between compression of ideal and actual gas is observed in the compression and expansion processes. This is clearly visible at comparison of the compression and expansion polytropic coefficients. As can be seen in the table, the compression and expansion polytropic coefficients for high pressure stages differ significantly from the methane adiabatic exponent equal to 1.32.

Figure 5. Indicator diagrams and motion diagrams of the first and fourth stage valves.
Figure 6. Indicator diagrams of the fourth stage at compression of ideal (a) and actual (b) gas.

Table 2. Performance indicators of compressor stages at $P_n=0.4$ MPa.

| Parameter                        | Ideal gas | Actual gas |
|----------------------------------|-----------|------------|
| Stage 1                         | 1         | 2          |
| $n_c$                            | 1.32      | 1.32       |
| $n_p$                            | 1.32      | 1.32       |
| $\pi_{ct}$                       | 3.594     | 3.729      |
| $N_c$, kW                        | 14.42     | 14.03      |
| $T_k$, °C                        | 135.9     | 139.6      |
| $c_{bc}$, m/s                    | 3.86      | 1.303      |
| $c_{nag}$, m/s                   | 2.34      | 2.228      |
| Stage 2                         | 3         | 4          |
| $n_c$                            | 1.32      | 1.32       |
| $n_p$                            | 1.32      | 1.32       |
| $\pi_{ct}$                       | 2.947     | 3.414      |
| $N_c$, kW                        | 10.84     | 4.133      |
| $T_k$, °C                        | 116.7     | 13.69      |
| $c_{bc}$, m/s                    | 1.82      | 3.82       |
| $c_{nag}$, m/s                   | 2.01      | 0.89       |
| Stage 3                         | 4         | 1          |
| $n_c$                            | 1.32      | 1.277      |
| $n_p$                            | 1.32      | 1.279      |
| $\pi_{ct}$                       | 1.557     | 3.414      |
| $N_c$, kW                        | 4.133     | 13.69      |
| $T_k$, °C                        | 60.9      | 130.9      |
| $c_{bc}$, m/s                    | 1.227     | 3.82       |
| $c_{nag}$, m/s                   | 0.89      | 2.30       |
| Stage 4                         | 5         | 6          |
| $n_c$                            | 1.277     | 1.294      |
| $n_p$                            | 1.279     | 1.295      |
| $\pi_{ct}$                       | 3.414     | 3.347      |
| $N_c$, kW                        | 4.133     | 13.69      |
| $T_k$, °C                        | 130.9     | 128.9      |
| $c_{bc}$, m/s                    | 3.82      | 1.199      |
| $c_{nag}$, m/s                   | 2.30      | 1.958      |

When analyzing the calculation results it is necessary to mention that the error at calculation of the compressor parameters with the assumption of gas ideality can be significant enough. So, in the dependency on operation mode, the error at determination of the stage indicated power can be quintuple; this is mainly subject to the high pressure stages. When determining the final temperature the maximum difference is 35%; when determining the intermediate pressure - 15%; when determining the capacity - up to 10%. The analysis carried out confirms once again the necessity in taking into account the actual gas properties at designing the HP compressors, in particular, the compressors used in NGV-refueling compressor stations.

Further on, we shall carry out the analysis of the compressor performance in different operation modes. The compressor under study is proposed to be used with the initial pressure varying from 0.4 to 0.7 MPa. It is necessary to determine the compressor characteristics depending on the initial pressure and its influence on the valves performance and on the working process taking place in the cylinders.

Figure 7 shows the compressor characteristics (dependency of capacity and indicated power of the compressor on the initial pressure). Table 3 gives the main parameters characterizing the performance of stages with different values of initial pressure.
Figure 7. Design characteristics of compressor: indicated power (a) and volumetric capacity (b).

Table 3. Design parameters by stages with different values of $P_n$.

| Parameter | $P_n$, MPa | Stage 1 | Stage 2 | Stage 3 | Stage 4 |
|-----------|------------|---------|---------|---------|---------|
| $\pi$     | 3.41       | 3.34    | 3.31    | 3.35    | 3.29    | 3.29    | 3.00    | 2.90    | 2.82    | 1.79    | 1.40    | 1.14    |
| $N_i$, kW | 13.69      | 18.50   | 23.35   | 12.43   | 16.58   | 20.87   | 9.80    | 13.36   | 17.42   | 5.14    | 4.00    | 2.11    |
| $\lambda$ | 0.816      | 0.818   | 0.819   | 0.776   | 0.782   | 0.779   | 0.697   | 0.728   | 0.758   | 0.875   | 0.941   | 0.979   |
| $T_k$, °C | 130.9      | 128.8   | 127.9   | 128.9   | 127.1   | 127.1   | 118.4   | 115.4   | 112.6   | 72.6    | 52.4    | 36.8    |
| $n_p$     | 1.28       | 1.28    | 1.28    | 1.29    | 1.30    | 1.32    | 1.39    | 1.45    | 1.54    | 1.89    | 2.14    | 2.40    |
| $n_c$     | 1.28       | 1.28    | 1.28    | 1.29    | 1.30    | 1.32    | 1.39    | 1.46    | 1.55    | 1.91    | 2.16    | 2.41    |
| $c_c_{bc}$, m/s | -3.82    | -2.84   | -2.99   | -1.20   | -1.34   | -1.36   | -2.07   | -1.52   | -1.27   | -1.46   | -0.97   | -1.00   |
| $c_c_{nag}$, m/s | -2.30    | -2.22   | -2.88   | -1.96   | -2.46   | -2.52   | -1.73   | -1.47   | -1.74   | -1.09   | -0.76   | -0.73   |

The given data indicate that the compressor capacity and the power consumed by the compressor increase with the initial pressure rise. Notwithstanding that redistribution of the intermediate pressure takes place in different modes, the pressure ratio in the first three stages varies insignificantly. The main influence is exerted on the fourth stage. We take notice of the fact that selection of valve parameters for satisfactory performance in all operation modes represents a quite difficult task. Depending on the operation mode the motion diagrams of the shut-off components of the valves may change significantly. This is clearly seen in Figure 8, where motion diagrams of the fourth stage valves are given for two design variants: variant 1 (preliminary) - preliminary compression of elastic components is equal to 3.5 mm; variant 2 (final) - 1.5 mm.
It can be seen in the given figure that such phenomenon as flutter may occur for the suction valve design with the preliminary compression of elastic components equal to 3.5 mm in the mode when the initial pressure is equal to 0.4 MPa. The similar pattern was observed for valves installed at the other stages. Upon selection of valve parameters the admissible performance of the compressor at all operation modes was achieved.

4. Conclusions
The working process analysis has shown that the actual gas compressibility differs significantly from the ideal gas compressibility. This circumstance cannot be neglected at designing the compressor equipment for NGV-refueling compressor stations, since a gross error is possible at calculation. Provision of the compressor serviceability in all operating modes is a difficult task, and especially this concerns the performance of valves. So, in order to create reliable and efficient equipment it is necessary to implement the digital design methods. This will make it possible, as early as the equipment design stage, to carry out the analysis of structural parameters in order to optimize the working process and to obtain the best final results with the specified operating conditions and modes, and to simplify the task for selection of valves, compressor base, its actuator, etc.

References
[1] http://gmt-gazprom.ru.
[2] Khrustalyov B S 1999 Mathematical modeling of working processes in volumetric compressors for solution of automated design: thesis Doctor of Engineering Science (Saint Petersburg) p 377
[3] Kotlov A A 2011 Mathematical model of MP pneumatic reciprocating compressor for solution of energy audit tasks: thesis Candidate of Technical Sciences (Saint Petersburg) p 138
[4] Mistry H, Bhakta A, Dhar S, Bahadur V and Dey S 2017 Capturing valve dynamics in reciprocating compressors through computational fluid dynamics International Compressor Engineering Conference at Purdue 1210
[5] Kalekin V S Kalekin D V, Nefedchenko A N 2013 Mathematical model of reciprocating pneumatic engine with automatic valves Omsk scientific bulletin 3 72–6
[6] Kondratyeva T F and Isakov V P 1983 Valves of reciprocating compressors (Leningrad: Machine-building, Leningrad subdivision) p 158
[7] Makoveyeva A S, Prilutsky A I, Prilutsky A A and Ganzha V Yu 2018 Simulation of automatic multi-component valves with reduction in plates at analysis of reciprocating compressor stages operation Compressor engineering and pneumatics 1 21–6
[8] Mistry H, Bhakta A, Dhar S, Bahadur V and Dey S 2017 Capturing valve dynamics in reciprocating compressors through computational fluid dynamics International Compressor Engineering Conference at Purdue 1210
[9] Sarmanayeva A F, Mustafin T N, Chekushkin G N 2015 Calculation and experimental investigation of automatic valves operation in reciprocating compressors Compressor engineering and pneumatics 4 17–20
[10] OAO “Gazprom” 1999 Methodical guidelines for thermo-technical and gas-dynamic calculations at tests of gas-turbine gas-compressor units (Moscow: VNIIGAZ)