Selection of design parameters of automatic ring valves of reciprocating compressors

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Abstract. One of the main tasks at designing and operation of reciprocating compressors is provision of energy efficient and reliable operation of the automatic valves. Imposed to the automatic valves is a number of requirements (low gas-dynamic impedance to valve flow, timeliness of valve closing, etc.) the fulfillment of which requires multi-variant calculations in order to select efficient valve parameters for the specified operation modes. Given in the study are the results of valve design analysis depending on the compressor operating conditions (at change in the crankshaft speed) and valve parameters (at change in spring tension and maximum valve stroke).

1. Introduction

Provision of efficient and reliable operation of the automatic valves of reciprocating compressors is one of the important issues at designing, manufacture and operation of the compressor equipment. The automatic valves belong to the critical assemblies of the reciprocating compressors to which high requirements are imposed. These requirements include [1, 2, 3]: low gas-dynamic impedance to valve flow, low pressure drop necessary for valve opening, absence of moving elements vibration, timeliness of valve closing, high reliability of valve, etc. Power consumption required to overcome resistance in the valves of the modern reciprocating compressors is approx. 10% of the nominal value. In the mobile and special compressors such consumption may reach 20-30% [1]. Decrease of such consumption is a critical task. The valve efficiency can be increased through optimization of its structural elements.

Purpose of the study is to increase in efficiency of reciprocating compressors through optimization of the automatic valve parameters.

Object of the study is an automatic valve of the reciprocating compressor. Considered in the study is calculation and optimization of the ring valve parameters (see Figure 1).
Figure 1. Ring valve.

The valves of such design are used at the low, medium and high pressure stages. The valve shut-off device is one or several ring concentric plates with a thickness of 1-5 mm located on narrow beads of the seat. Usually all plates have the same thickness, radial width and lift height, which is adjusted by the corresponding setting of a restrainer. The plates are lifted from the seat by force from the pressure difference upstream and downstream the valve, and their seating is carried out by springs [4, 5].

2. Material and methods of operation

The calculation method of the automatic valves dynamics is based on the mathematical model described in [6, 7, 8, 9]. The simplified mathematical model of the valve represents two systems of non-linear differential equations [10-13]

- for suction valve

\[
\frac{d\beta_{bc}}{d\phi} = -\frac{1}{f(\phi)} \left[ \frac{2\sqrt{2k}}{\pi M_{bc}} f(\beta_{bc}) - k(1 - \beta_{bc}) f'(\phi) \right],
\]

\[
\frac{d^2\chi}{d\phi^2} = B_{bc} \xi \beta_{bc} - Z_{bc}^2 (\chi + \chi_0) - \eta \frac{d\chi}{d\phi},
\]

- for discharge valve

\[
\frac{d\beta_{nar}}{d\phi} = -\frac{1}{f(\phi)} \left[ \frac{2\sqrt{2k}}{\pi M_{nar}} f(\beta_{nar}) + k(1 + \beta_{nar}) f'(\phi) \right],
\]

\[
\frac{d^2\chi}{d\phi^2} = B_{nar} \xi \beta_{nar} - Z_{nar}^2 (\chi + \chi_0) - \eta \frac{d\chi}{d\phi},
\]

where $\beta_{bc}$ ($\beta_{nar}$) – relative pressure losses in the suction (discharge) valve; $\phi$ – crankshaft rotation angle; $f(\phi)$ – dimensionless function of piston movement; $k$ – adiabatic exponent; $\chi$ – relative valve stroke; $f(\beta_{bc/nar})$ – pressure function of suction (discharge) valve; $M_{bc}$ ($M_{nar}$) – dimensionless complex characterizing losses in the process of suction (discharge), $B_{bc/nar} = \frac{F_{int} P_{bc/nar}}{\omega^2 x_{max} m}$ – dimensionless complex characterizing gas and inertial forces acting to the shut-off device; $\xi$ – pressure coefficient; $Z^2 = c/m\omega^2$ – relative natural oscillation frequency; $x_0$ – relative preliminary compression of elastic elements; $\eta_0$ – damping factor [14, 15].
Impact of the shut-off element with the seat and guard shall be simulated by the model of partially elastic impact [15]. After impact, the speed value shall be calculated by formula

\[ V'' = -V'K_{otc}, \]

where \( K_{otc} \) – bounce coefficient (for ring valves shall be taken in the range of 0.25 to 0.35); \( V' \) – shut-off element speed before impact.

The optimization task is formulated, if the following is set: optimality criterion the change of which makes it possible to influence the process efficiency; mathematical model of the process; limitations connected with economical and structural parameters, etc. The task of calculating the optimum valve parameters can be solved by the following methods.

1. Mathematical simulation. The method is based on the calculation of valve dynamics with the use of the mathematical model of working process taking place in the reciprocating compressor stage. Complexity of this method consists in that even if a single-mas model of valve dynamics is used a number of variables determining the compressor and valve stage operation can reach 78 parameters. Large number of variables characterizing valve operation makes it impossible to compose a target function.

2. Target function. In order to simplify the optimization task the determining parameter should be selected, and the rest parameters should be transformed to limitations. Selected as the target function for automated valves shall be the function connected with the stage operation efficiency, as a rule, necessary for gas forcing through the valve. However, as shown in [1, 2, 6], this operation may have no minimum value with the optimal valve parameters.

Thus, the choice of operation for gas forcing through the valve as a target function is indefensible. Proposed in the study [1] is application of a function determining the dimensionless loss of pressure as a target function, which shall be determined by the following formula

\[ \beta_{cp} = \frac{\int [\beta] f^i(\varphi) d\varphi}{\varphi_1 \int \left( f(\varphi_2) - f(\varphi) \right) d\varphi}, \]

where \( f(\varphi) \) – piston stroke function; \( \varphi_0 \) and \( \varphi_2 \) – valve opening and closing angles, respectively.

The experience of valve designing shows that actually a small zone of valve parameter change is available for each certain compressor. That is why the task of valve parameter optimization can be solved with the use of the excellence criterion. At that, several calculation variants are prepared the comparison of which helps to select rational valve parameters. If the calculation program is available, the task can be solved within a short time.

The task of the automated valve rational parameters calculation consists also in the fact that in the majority of cases the compressor being designed is intended for work in different operation modes. Taken as an example can be the evaluation results of valves operation for the compressor 2BM10-60/8. The compressor is opposite, two-stage with process valves. The compressor operation analysis was performed for the modes with the crankshaft variable speed within \( n = 300 \ldots 600 \) rpm. Figure 2 shows diagrams of valve movement for the first stage. It is seen from the Figure, that as the cycling is decreased the flutter takes place and the valve closing angles change. Presence of self-oscillations leads to speed increase at seating to the inadmissible limits: more than 1.5 m/s. The same picture is observed at the second stage. The flutter has an impact on the compressor operation efficiency factors [16].
Thus, the task of the automatic valve optimal designing is quite difficult. Let us consider the task of rational valve parameters determination through the example of a ring valve.

### 3. Calculated analysis and initial data

The study was carried out according to the developed software program the description of which is given in [6]. The program makes it possible to analyze diagrams of the valve working plates movement, to evaluate pressure losses in valves, valve opening and closing angles, speed of working plate fitting onto a seat and a guard, etc.

In the course of the study the following parameters varied: stiffness of elastic elements, valve maximum stroke and crankshaft speed.

**Table 1.** Initial data on compressor stage.

| Parameter                  | Value |
|----------------------------|-------|
| Piston diameter, mm       | 210   |
| Piston stroke, mm         | 100   |
| Rotational frequency, rpm | 977   |
| Clearance space, %        | 8     |
| Expansion polytropic index| 1.3   |
| Compression polytropic index| 1.35 |
| Initial pressure, MPa     | 0.1   |
| Final pressure, MPa       | 0.32  |
| Initial temperature, °C   | 20    |
| Final temperature, °C     | 140   |

**Figure 2.** Diagrams of valve movement for the first stage: a) \( n=600 \) rpm; b) \( n=300 \) rpm.
Table 2. Geometrical parameters of valves.

| Valve                                      | Plate 1 | Plate 2 | Plate 3 |
|--------------------------------------------|---------|---------|---------|
| Outer diameter, mm                         | 63      | 93      | 123     |
| Internal diameter, mm                      | 47.2    | 77.2    | 107.2   |
| Ring thickness, mm                         | 3.2     | 3.2     | 3.2     |
| Width of the channel in the valve seat, mm | 5       | 5       | 5       |
| Preliminary spring compression, mm         | 1.3     | 1.3     | 1.3     |
| Number of springs per ring                 | 4       | 6       | 8       |
| Nominal valve stroke, mm                   | 2       | 2       | 2       |
| Nominal preliminary compression, mm        | 1.5     | 1.5     | 1.5     |
| Nominal stiffness of elastic elements, N/m | 400     | 400     | 400     |

Carried out in the study [6] was investigation of spring stiffness and valve stroke influence on the valve operation at variation of values for all plates and for one plate. The investigation has shown that minimum relative pressure losses are not indicative of the valve optimal parameters, and it is necessary to observe identical trajectory of all working plates. Within the frames of this study it is necessary to obtain a function of the valve parameters versus different values of variable parameters.

4. Calculation results and discussion
1. Change in spring stiffness at different values of the valve maximum stroke. Let us consider the spring stiffness influence on the valve operation. The spring stiffness will be changed within 100 to 1000 N/m for three values of the maximum valve lift height equal to 1 mm, 2 mm and 3 mm. Figures 3-6 show the calculation results of the performance characteristics of suction and discharge valves.

As shown by the given graphs, increase in stiffness leads to the growth of relative pressure losses (see Figure 3); at that, it is more expressed for the suction valve. This is due to that at some moment the valve goes into flutter mode, i.e. the valve plate opens and closes several times over the course of the process. At that, the average relative stroke of the valve becomes less (see Figure 4), and thus the total passage area will be less over the process of suction or discharge, which shall lead to increase in pressure losses at gas forcing through the valve. As shown in the study [6] the flutter mode occurs at values $x_{cp}=0.6$. Increase in maximum stroke leads to decrease in relative pressure losses, since the passage area becomes larger. However, considering that the valve goes into flutter mode at higher values of $x_{max}$, pressure losses start increasing sharply from a certain moment (see Figure 3a).
Valve closing angles become less with increase in the spring stiffness; at that, variation of a function may have a minimum. However, as shown by Figure 5, there may be several minimums. For the cases considered the valve closing takes place with delay (for suction valve the value is more than 180 degrees, for discharge valve - more than 360 degrees). Figure 6 shows maximum speeds of valve plates seating. This parameter determines the valve performance reliability. With increase in stiffness the speed of the suction valve seating decreases; this is more obvious for a case with $x_{\text{max}}=3$ mm. The pattern for the discharge valve is different. With increase in the spring stiffness the speed starts growing sharply to a certain value, after that its decrease takes place.
Thus, the rational parameters for the valves considered are: for suction valve - $c=400 \text{ N/m}$ and $x_{\text{max}}=2 \text{ mm}$, for discharge valve - $c=700 \text{ N/m}$ and $x_{\text{max}}=2 \text{ mm}$.

2. Change in crankshaft speed with different values of shut-off device stiffness. The crankshaft speed was changed within 300 to 977 rpm for three values of stiffness - 400 N/m, 700 N/m and 1000 N/m. Figures 7-9 show the calculation results.

It can be seen from the given diagram of valves movement that decrease in the crankshaft speed leads to flutter occurrence (see Figure 7). With increase in the crankshaft speed the pressure losses become higher, which is connected with speed increase of gas passing through the valve. With decrease in the crankshaft speed the angles corresponding to the valve closing become smaller.
However, in certain cases this takes place too soon (see Figure 7a). The valve plate seating speed in the considered range of the crankshaft speed variation was changed insignificantly, the change range was 0.5-1 m/s.

![Figure 7. Diagram of suction (a) and discharge (b) valves movement.](image)

![Figure 8. Functions of relative pressure losses in the valve versus change in the crankshaft speed for suction (a) and discharge (b) valves.](image)

5. Conclusions
The study undertaken shows that decrease in the valve elastic element stiffness leads to decrease in operation required for gas forcing through the valve. However, further decrease in stiffness shall lead to delay in the valve closing and, consequently, to gas back flowing through the valve. At designing
the compressor intended for operation in different modes, in particular, at change in the crankshaft speed, it is necessary to select the valve parameters thoroughly. The flutter not only influences the valve operation, but also has a significant impact on the performance characteristics of the whole compressor. The results obtained are not rejected by the data obtained earlier and described in the literature by other authors.

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