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K. M. Ignatiev  
Technical University of St. Petersburg

I. B. Pirumov  
Technical University of St. Petersburg

B. S. Chrustalyov  
Technical University of St. Petersburg

M. M. Perevozchikov  
Technical University of St. Petersburg

V. B. Zdalinsky  
Technical University of St. Petersburg

See next page for additional authors

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Authors
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STUDY OF THE VALVE ELEMENT MOTION AND THE GAS FLOW IN THE STRAIGHT-FLOW VALVES

K.M.Ignatiev, I.B.Pirumov, B.S.Chrustalyov,
M.M.Perevozchikov, V.B.Zdalinsky, M.Esper.

Compressor Department, Technical University of St.Petersburg,
St.Petersburg, Russia

ABSTRACT

This paper describes the simulation method of gas flow and valve dynamics in the straight-flow valve of reciprocating compressor. This model determines the gas pressure distribution and deformations of valve plate as an elastic plate without using any empiric information. Simulation was tested by static gas flow measurements in valve and dynamic measurements of the valve plate end motion in experimental compressor. Comparisons are made between the computed and measured results.

INTRODUCTION

Nowadays most of the high flow rate reciprocating compressors, manufactured in Russia, are equipped with straight-flow valves. These valves have high effective flow area and work with relatively low losses with high crankshaft rotating speeds. At the same time, their reliability limited by the impact fatigue is not high enough. Improving the reliability is an actual work which couldn't be solved without more correct simulation methods determining both the gas pressure parameters in the valve flow channel and the valve plate representation as a system with several degrees of freedom.

MATHEMATICAL MODEL.

Cross-sectional view of the straight-flow valve is explained at Fig.1. The following hypothesis were used in mathematical model:

1. Gas pressure distribution in the valve flow channel is one-dimensional;
2. The mass flow of gas is the same in any cross-section of the valve flow channel;
3. The valve plate bending is cylindric;
4. The gas flow in the valve flow channel is quasistatic. In accordance with this hypothesis the equation of motion of the valve plate looks as follows:

\[ \frac{\partial^4 u}{\partial x^4} + \frac{g}{a} \frac{\partial^2 u}{\partial t^2} = \frac{b}{a} \frac{(P_1 - P_2)}{a} \]  
(1)

with boundary conditions:

\[ u|_{x=0} = 0; \quad \frac{\partial u}{\partial x}|_{x=0} = 0; \quad \frac{\partial^2 u}{\partial x^2}|_{x=L} = 0; \quad \frac{\partial^3 u}{\partial x^3}|_{x=L} = 0 \]

where: x - coordinate along the channel, changes from 0 (fixed end) to L (free end);
- u - displacement of the plate;
- g - cross section mass of the plate;
- a - cylindrical rigidity of the plate;
- b - plate width;
- P - pressure distribution under the plate;
- P2 - pressure over the plate.

More of that there are the contact conditions which occurs when the plate contacts the limiter or seat.

To solve this equation with the computer, it must be discretized and transformed to the matrix ordinary differential equation:

\[ AU + BU = P - P_2, \]  
(2)

where: A - matrix of rigidity (formed by finite differential or some other method);
- B - matrix of inertia;
- U - vector of displacements;
- P - vector of pressure under the plate;
- P2 - vector of pressure over the plate (In this model this pressure considered to be equal to the pressure after the valve).
Contact conditions can be represented in traditional form:

If \( u[i] > u[i] \) then \( u[i] - u[i] = 0 \); \( u[i] = z \cdot u[i] \);
if \( u[i] < 0 \) then \( u[i] = 0 \); \( u[i] = z \cdot u[i] \);
where \( z \) - jump coefficient (about 0.2);
\( u[i] \) - an element of the vector of limiter coordinates.

The gas flow is governed by adiabatic and Sent-Venam and Ventse equations:

\[
\nabla^2 \varphi = \frac{\kappa}{\kappa - 1} \frac{p_0}{\rho_0} \left[ 1 - \left( \frac{p}{p_0} \right)^{\kappa - 1} \right]
\]

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\]

\[
\frac{\rho}{\rho_0} = \left( \frac{\varphi}{\varphi_0} \right)^{\kappa}
\]

Mass flow equation:

\[
Q = \nabla \cdot d \cdot \rho = \text{Const}
\]

Where: \( v \) - gas velocity along the valve channel;
Where: \( p \) - gas pressure along the valve channel;
\( \rho \) - gas density along the valve channel;
\( d \) - cross sectional area of the valve channel (Can be determined as \( d = (d_0 + u) \cdot b \). where \( d_0 \) - thickness of valve channel when valve is closed);
\( k \) - adiabatic coefficient;
\( Q \) - mass flow of gas;
\( p_0, \rho_0 \) - gas pressure and density before the valve.

Mass flow of gas is determined by equation (6) and (4) at point \( x - 1 \) considering \( p - p_1 \) (pressure after the valve).

Because of relative pressure drop being not higher that 0.1; density linearisation can be used: \( p = p_0 - \varphi \), where \( \varphi < p_0 \); it could be determined easily:

\[
\varphi = \frac{\rho^2}{d^2 \rho_0^2} \frac{2 Q^2}{d^2 \rho_0^3 - (\kappa - 1) p_0}
\]

Where:

\[
C = \frac{2 \kappa}{\kappa - 1} \frac{p_0}{\rho_0}
\]

Pressure in the valve channel could be founded easily:

\[
p = p_0 \left( 1 - \frac{\varphi}{p_0} \right)
\]

For the discharge valve model cylinder pressure could be determined according to well-known differential equation in cylinder:

\[
\frac{dp_0}{dt} = -\frac{f'(y)}{f(y)} \cdot n_0 \cdot \frac{p_0 T_0 \cdot n \cdot \varphi}{3 \cdot \pi \cdot r_1} \cdot Q
\]

Where:
\( p_0 \) - pressure in cylinder;
\( f'(y) \) - crankshaft rotating angle, \( w \) - rotating speed;
\( f(y) \) - function of cylinder volume;
\( df(y) \)
\( f'(y) = \frac{d}{dt} \)
\( s \) - piston area;
\( r_1 \) - crankshaft radius;
\( n_1 \) - number of valve elements;
\( R \) - universal gas constant;
\( n \) - polytrope coefficient;
Temperature $T_{0}$ is determined as follows:

$$\frac{T_{0}}{T_{s}} = \left(\frac{p_{0}}{p_s}\right)^{-\frac{1}{n+1}}$$  \hspace{1cm} (11)

where $T_{s}, p_s$ - suction temperature and pressure.

Differential equations (10), (2) with the contact conditions and equations (9) and (11) simulating the discharge valve are integrated with computer. Starting conditions are: -180 grad, $p_0-p_s$; $T_{0}-T_{s}$; $U-0$. Time step calculated automatically as $1/100$ of the highest natural frequency of the system (2). Integration stops when the discharge valve has closed.

EXPERIMENTAL SETUP AND PROCEDURES.

Fig.2 presents a schematic general view of the static flow experimental setup. Compressed air from the industrial pipeline with the pressure of 6 bar flows through the control valve, rotating flow meter and through the valve model. Cross section of it is presented at Fig.3. It consists of the valve seat, valve limiter and the valve plate. By using the micrometric screw one can install a proper valve plate end displacement. Both the seat and the limiter has small holes connected with pipes to measure the gas pressure through the valve channel by using manometers. After installing the pressure drop on the valve gas flow and pressure distribution was measured.

Fig.4 presents a general principle of dynamic experimental unit. It consists of 1 cylinder DC-driven experimental compressor with variable rotating speed (it varies from 500 to 1500 rpm) with the cylinder diameter of 200 mm and the piston stroke of 75 mm. The upper piston of 75 mm. The upper piston position inductive transducer is installed in the crankshaft. The piston stroke. The combined (suction + discharge) straight flow valve was installed. The amount of valve elements could be changed. One of the valve elements has a differential displacement inductive transducer which controls the displacement of the end of the valve plate. Pressure inductive transducers measure gas pressure in the cylinder and in the suction volume. This volume was connected through the control valve and rotating flow meter with the atmosphere. It was equipped with the statical manometer and thermometer.

Signals from the inductive transducers are introduced in an amplifier bridge and through programmable commutator controlled with the upper piston transducer are transferred in the ADC to be registered by the computer. Digital frequency register also connected with the piston position transducer is used for operative rotating speed control. The relative mistake of valve displacement measurement is about 10%; The cylinder pressure measurement relative mistake is about 5%.

RESULTS.

To test the mathematical model of gas flow comparison between simulation and measurement was provided. Mathematical model was transformed to the conditions of statical experiments. The results are shown on fig. 5 and 6. The abscissa is the x-coordinate along the valve channel, $P_0$ - pressure before the valve, $P_{1}$ - pressure after the valve. The good agreement between the numerical solution and experimental results as shown in fig.5,6 lends evidence to the validity of the numerical solution.

To test the mathematical model of valve dynamics comparisons between simulation and laboratory compressor measurement has been done. Fig 7 and 8 represents valve plate end lifting and gas pressure measured experimentally and simulated by computer. Comparison between computed and measured results also demonstrate a good correlation.

Fig.9 presents the discharge valve plate lifting process visualization after 5 deg of crankshaft rotating since the cylinder pressure overcomes the discharge pressure. For better visual impression the vertical scale is greater that the horizontal one.

This simulation can be used to determine bending tensions, contact velocities for further fatigue predictions, and also for valve limiter and gas channel optimal design.
CONCLUSIONS.

This work presents the numerical results of gas flow and valve dynamics in straight flow valve of reciprocating compressor, determining gas pressure distributions and elastic plate deformations. Experimental results have been used for validation of the numerical model. Valve lifting process visualization is presented.

REFERENCES.


Fig. 1. Cross-sectional view of the straight-flow valve.

Fig. 2. General view of the static flow experimental setup.
Fig. 3. Test section for the static flow measurements.

Fig. 4. Experimental compressor unit and measurement equipment.
Fig. 5. Static comparison.

Fig. 6. Static comparison.

Fig. 7. Comparison of simulated and measured valve end dynamics. N=500 rpm.
Fig. 8. Comparison of simulated and measured valve end dynamics. $N=1000$ rpm.

Fig. 9. Visualization of discharge valve opening (computer simulation).