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COMPUTATION OF A COOPERATION BETWEEN
RECIPROCATING COMPRESSOR AND COMPLEX
PIPING SYSTEM INCLUDING MUFFLERS

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ABSTRACT

Proposed scheme of computation in a general outline is following. In the first stage by use of a method based on the wave theory pressure spectra for the whole installation are predicted. As a boundary condition a standard expansion of piston velocity into Fourier series is used. Then by the inverse Fourier transformation pressure in the outlet section of the compressor muffler is computed. This pressure now becomes a boundary condition in the method of numerical simulation. This simulation comprises only a small part of the installation, i.e. between the compressor and the outlet of a muffler. Such a simulation is well known, but usually the influence of the rest of pipeline is neglected. This simplification is not always a proper one.

INTRODUCTION

The main engineering problem of reciprocating compressor plant design lies in junction of a compressor with a technological installation in such a way which simultaneously fulfils all technological requirements, lightens the influence of pipeline pressure pulsation on the run of the compressor and excludes mechanical resonance. Chemical installations are always rather complicated. They are composed of many volumes like irrigation coolers, separators, dampers and also many branches and connections.

Designers need two basic information i.e. about work of compressor (power consumption, volumetric efficiency, valve plate impact velocity etc.) and about the pipeline (pulsating pressure spectrum at basic points of the system).

This problem may be solved by means of electroacoustical analogy using special analog computers [1], which are not everywhere available. Digital computers however, are now easy of access, thus a numerical method should be also taken into account. There are many well known methods of numerical computation of nonstationary flow. The most exact is a method of characteristics [2] or Lax-Wendroff method. But when the installation is complicated or contains some volumes the solution loses the accuracy. So in the complicated cases simpler methods achieve almost the same accuracy. The problem of accuracy is less connected with a method but rather with transmission of particulars of a real installation into a model.

OBJECT OF INVESTIGATION

A simple one-cylinder compressor joined to installation shown in Fig. 1 was an object of investigation. For simplicity suction valve chamber was opened directly to the atmosphere. Geometry of the pipeline was not optimised. The stand was supposed to be a test bed for checking methods of predicting a pulsating flow. In Fig. 2 a computed damping characteristic (transmission loss) of the applied attenuator has been shown.

WAVE THEORY


We have used a solution obtained by Chen [4] and have built a numerical program based on the method proposed by Abe, Fujikawa and Ito [5]. The following set of known solutions of wave equations
has been used with different boundary conditions, e.g., volumes, branches and abrupt change in cross section. As an excitation function Fourier series expansion of piston velocity, during time interval of valve opening, has been used. Pressure spectra at some points of the investigated valve chamber have been calculated and measured. In Fig. 3 spectra corresponding to the valve chamber (point 1 in Fig. 1) are shown. Predicted and measured values of higher harmonics (10-20) exhibit a significant discrepancy. This results from oversimplified boundary conditions (piston velocity) and three-dimensional phenomena in a damper at high frequencies. The spectrum computed for the part downstream of the muffler (a closed end of the branch, point 2 in Fig. 1) shows a better agreement with experimental data (Fig. 4).

From our experience with this method, it is effective particularly in the case of a complicated configuration. The method, however, has two serious disadvantages: nonlinear elements of the installation need an iteration procedure and it seems not possible to take directly into account a dynamics of the valve system. But the influence of the leakage or opening delay time have been investigated using the wave theory [6]. Elson and Soedel [7] have applied a wave theory with iterative procedure for predicting the interaction of valve with pulsating flow in long duct.

**METHOD OF SIMULATION**

For computation of an interaction between a valve and pulsating flow the method of numerical simulation may be used. A set of ordinary differential and algebraic equations describing the work of all components of the system must be collected. For the nonstationary flow in a pipe of constant cross section, the following equations have been used:

\[
\begin{align*}
\frac{\Delta p}{\Delta x} &= 8 \left( \frac{du}{dt} + \frac{\lambda u^2}{2d} \right) \\
\frac{\Delta u}{\Delta x} &= \frac{1}{8 \alpha^2} \frac{dp}{dt}
\end{align*}
\]

It is an lumped parameter approach to simplified equations of motion, continuity and state with density change being neglected (low Mach number) and convective derivatives of pressure and velocity omitted. Similar equations in linearised form but in different ways have been used by Grover [8] and Łuczczcycki [9]. For the simulation of a thermodynamic process in a cylinder a polytropic compression has been used:

\[
\frac{dp}{dt} = \frac{n-1}{V} \left( Q_s + Q_d - \frac{dV}{dt} \right)
\]

The leakage was not considered and the heat transfer through the walls has been included in the polytropic exponent. The change of the cylinder volume has been described by:

\[
V = V_0 ((1-\cos \omega t) + r/4(1-\cos 2\omega t))/2 + V_m
\]

For describing flow through the valves the following simple equations have been applied:

For suction

\[
Q_s = \alpha_s \cdot F_s \cdot \sqrt{2 \Delta p/\rho_s}
\]

and for discharge

\[
Q_d = \alpha_d \cdot L \cdot h \cdot \sqrt{2 \Delta p/\rho_d}
\]

where \( h \) is a valve plate lift. The discharge valve was considered as having one degree of freedom:

\[
\frac{d^2h}{dt^2} + \frac{dh}{dt} \cdot c + h \cdot k = A_p \cdot \Delta p
\]

The force acting on the valve plate is a product of pressure difference \( \Delta p \) across the valve times an effective force area \( A_p \). The coefficients \( A_p, \alpha_s \) and \( \alpha_d \) have been determined experimentally by steady flow tests and then assumed constant. A solution of such a set of equations might be performed by an analog computer [10], but in this case a digital computer with a simulating language MIMIC has been used. If the method of simulation is joined to the method of characteristics it may be noticed that the required time for programing is significantly shortened and computer time consumption enlarged. The method of simulation is very effective but only for not too complicated systems.

**PROPOSED MIXED METHOD**

The proposed method comes out from an idea that a strong interaction between valve and pulsating flow depends primarily on the flow between the compressor and the
damper. The valve plate oscillation may cause the flow pulsation with high frequencies only. The period of oscillation is shorter than the time of valve opening. But the influence of the rest of installation on the pressure pulsation in a muffler not always can be neglected. In the mixed method both previously described methods (a wave theory method and a method of simulation) in two steps have been used. First pressure spectra, for some points of the installation, by the wave theory, have been obtained. Then by the invers Fourier transformation a pressure pulsation in the muffler has been found. This pressure as a boundary condition in the method of numerical simulation is applied. This simulation comprises only a small part of the installation between the compressor and the muffler. Eventually the Fourier transformation of flow through the valve may be used as an improved boundary condition in the wave theory method applied again. In Fig. 5 and 6 pressure pulsation at point 1 has been shown. The curve of small amplitude in Fig. 5 corresponds to a partial simulation of the system, based on the common assumption that the pressure pulsation downstream of a muffler may be neglected. The smooth curve comes from wave theory, so without taking into account valve motion. The agreement with an experimental curve is rather poor. Fig. 6 shows results obtained by full simulation of the installation, by proposed mixed method and by experiment. Full simulation and experiment agree very well. Generally mixed method shows greater discrepancy than full simulation but much smaller than wave theory or partial simulation. Also the information about operation of valves was secured. In the case of this simple geometrical configuration a computer time consumption by full simulation is more than 50% higher than by mixed method. The time increases rapidly when the installation becomes more complicated. The applied muffler was not very effective and this has strengthen the influence of the downstream part of the installation on pressure pulsation. In the case of a proper muffler the influence may be less significant.

CONCLUSION

Method of simulation considered above is very simple and now many simulating languages as MIMIC or CSMPL are available. But for simplicity in programing must be paid with greater computer time consumption, so application of this method is limited to simple installations only. Method based on wave theory is fast and effective in the case of complicated configurations but no information about the operation of valving system is obtained. Proposed scheme gives full information and the accuracy of the mixed method seems sufficient for engineering applications.

NOTATION

- speed of sound
- pulsation of speed of sound
- area
- damping coefficient
- pipe diameter
- valve lift
- spring coefficient
- length of the valve slit
- valve plate mass
- polytropic exponent
- rate of flow through the valve
- time
- particle velocity
- pulsation of particle velocity
- cylinder volume
- distance
- flow coefficient
- friction coefficient
- density
- rotational velocity

SUBSCRIPTS

- cylinder
- suction
- discharge
- valve
- mean value
- dead space
- static

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Fig. 1 Scheme of installation

Fig. 2 Damping characteristic of the muffler (transmission loss)

Fig. 3 Spectra of pressure in valve chamber (point 1 in Fig. 1)
Fig. 4 Spectra of pressure at closed end of the branch (point 2 in Fig. 1)

Fig. 5 Cylinder pressure $p_c$, discharge valve chamber pressure $p_v$, discharge valve flow rate $Q_d$, valve lift $h$
Fig. 6 Cylinder pressure $P_c$, discharge valve chamber pressure $P_v$, discharge valve flow rate $Q_d$, valve lift $h$.