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THEORETICAL AND EXPERIMENTAL RESEARCHES OF UNSTEADY GAS FLOW IN THE PIPELINE OF THE RECIPROCATING COMPRESSOR

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ABSTRACT

This paper investigates the unsteady gas flow in the reciprocating compressor communications. The numeric method for nonlinear differential equations system is presented.Differential equations for the unsteady one-dimensional gas flow in pipeline is resolved by finite differences numeric method. The experimental techniques are also presented. The results of the numeric and experimental researches are compared and analyzed.

INTRODUCTION

It is well known that the geometrical dimensions of the suction and discharge systems and interstage communication essentially influence on compressor performances. Therefore the special attention should be paid to the modelling of the unsteady flows in these parts when considering the working processes of the reciprocating compressor [1].

Over the last 40 years the so called acoustic models were mainly used. Those models allow exact solutions in several simple cases. For more complicated systems the electrical simulations are used. The weak sides of these methods are: the speed of sound has to be assumed constant, impossibility of valve-pipeline mutual influence estimation, static and dynamic leakages through the valve are out of concern, the gas is assumed to be perfect without heat exchange. These aproaches are suitable for low amplitudes of pulsations only. The compressor performance analysis, eigher theoretical and experimental shows that pulsation amplitudes can reach 20% and more of the mean value. Practice also shows that those models can be used for resonant frequencies calculations.

Starting from 70's in Russia the method of characteristics has been widely used [2,3]. However this method is also not free from weaknesses: the speed of sound is also assumed constant to ensure even-spaced calculation network, otherwise variable speed of sound would require complicated proportional network that is not suitable for gas flow calculations under high pressure gradients [4].

MATHEMATICAL MODEL

It most cases the unsteady gas flow in reciprocating compressor communications can be considered one-dimensional and described by corresponding gasdynamic equations based on the laws of preserving mass, energy and momentum. The generalized divergent form of equations of preserving is:

$$\frac{\partial F}{\partial t}+\frac{\partial (cF)}{\partial x}=-f \qquad F=\begin{pmatrix}\rho\\ \rho c\\ \rho E\end{pmatrix} \quad f=\begin{pmatrix}0\\ \dfrac{\partial p}{\partial x}+B\\ \dfrac{\partial (pc)}{\partial x}+q\end{pmatrix} \qquad (1)$$

$$E=u+\frac{c^2}{2} \qquad B=(\lambda+\xi\frac{D}{\Delta x})\frac{c|c|}{2D} \qquad q=\frac{\alpha b(T_w-T)}{\rho A} \qquad (2)$$

This way of equations representation makes it possible to use similar calculating schemes for gas parameters in particular nodes of time and space of the calculation network. These equations do not exsplicitly comprise the speed of sound. Therefore the numeric method should not necessarily use this parameter as the basic one when selecting calculation scheme. To select the method the following rule must be taken into consideration: the calculation scheme should not comprise the state equation differentiation. This rule makes both model and method universal and the form of the state equation can be much simplified.

The equations system (1) is resolved by finite differences method. We have studied and analysed various methods of obtaining discrete analogues [5,6]. We suggest the calculating schemes as follows:

$$\begin{aligned}
&\frac{\partial \rho_i}{\partial t}=-\frac{(\rho_{i+1}c_{i+1}+\rho_{i-1}c_{i-1})}{2\Delta x}\\
&\frac{\partial c_i}{\partial t}=-c_i\frac{(c_{i+1}-c_{i-1})}{2\Delta x}-\frac{1}{\rho_i}\frac{(p_{i+1}-p_{i-1})}{2\Delta x}+B \qquad (3)\\
&\frac{\partial u_i}{\partial t}=q+c_iB-c_i\frac{(u_{i+1}-u_{i-1})}{2\Delta x}+\frac{p_i}{\rho_i^2}\left[\frac{\partial \rho_i}{\partial t}+c_i\frac{(\rho_{i+1}-\rho_{i-1})}{2\Delta x}\right]\\
&i=1\ldots N-1
\end{aligned}$$

The equations of mathematical model for the compressor are used as the bounary conditions [4]. The calculated parameters of gas should be considered as periodical. However the problem is to prove the existance of the stable periodical solution for such complicated nonlinear system. The final solution is obtained by sequential solutions for the vast number of cycles under various initial conditions. The similar approach has been used for mathematical model analysis and for several other compressor designs.

EXPERIMENTAL RESEARCH

The experimental investigations were aimed at confirming the adequacy of mathematical model to the real physical processes in compressor communications. The instant values of pressure and velocity in nonsteady flow in suction pipe, instant pressure in cylinder and dynamic characteristics of valves were measured simultaneously. The instant velocity in the pipes were measured by pneumometric method. The stagnation pressure was measured by L-form tube. The instant static pressure and instantpressure in pneumometric passage and in cylinder were measured by tensometric method, using modified pressure transducers manufactured by ENDEVCO. Fig.1 represents the scheme of measurements. The diagrams of valve plate movements were registered by inductive transducers, developed at the Compressor Department.L-form tube had been dynamically calibrated in the shock tube installed at Compressor Department.

RESULTS

The theoretical and experimental investigations of the suction pipeline (Ø 50 mm) for the compressor stage with the cylinder diameter 200 mm have been carried out using the above presented model and methodics.

Fig.2 shows the results of the numeric estimations of the resonant and near-resonant behavior of the system under investigation. It is obvious that the model is able to work in the resonance-trasient zones. The frequency of the parameter pulsations practically coincide with the values obtained by the well-known linear theory formulae.

The advantage of the presented model for the compressor is well represented by fig.3. The calculation scheme, used in the model allows to determine the integral efficiency characteristics even under resonant conditions.

The comparison of the experimental and theoretical data (fig.4) for the wide variety of the suction system parameters proves both qualitative and quantative correspondence: valve diagrams, pressure diagrams for the different sections of the pipeline.

CONCLUSIONS

1.The experimental and theoretical methodics for the investigations of the pressure pulsations in the compressor communications are developed.

2.The mathematical model adequacy to the physical process is proved experimentally.

3.The numeric experiment shows the calculation scheme convergence even for resonant conditions.

4.It is experimentally and theoretically proved that the valve dynamics and the parameters of the unsteady flow in the pipelines are mutually dependent.

At present at Compressor Department the wide variety of the interstage communications schemes undergoes the thorough experimental and theoretical analysis.

NOMENCLATURE

- p pressure (Pa)
- T temperature (K)
- ρ density (kg/m3)
- E energy (J)
- $\overline{M}$ mass flow rate (kg/min)
- φ crank angle (deg)
- n rotation speed (rpm)
- t time (s)
- L pipe length (m)
- b pipe perimeter (m)
- D pipe diameter (m)
- u specific internal energy (J/kg)
- q specific heat flow (W/kg)
- α heat transfer coefficient (W/m2*K)
- c gas velocity (m/s)
- A heat transfer square (m2)
- B dissipative function
- ξ hydraulic resistance coefficient
- λ local resistance coefficient
- F transfered value vector
- f coresponding vector of sources.

SUBSCRIPTS

w wall; d discharge

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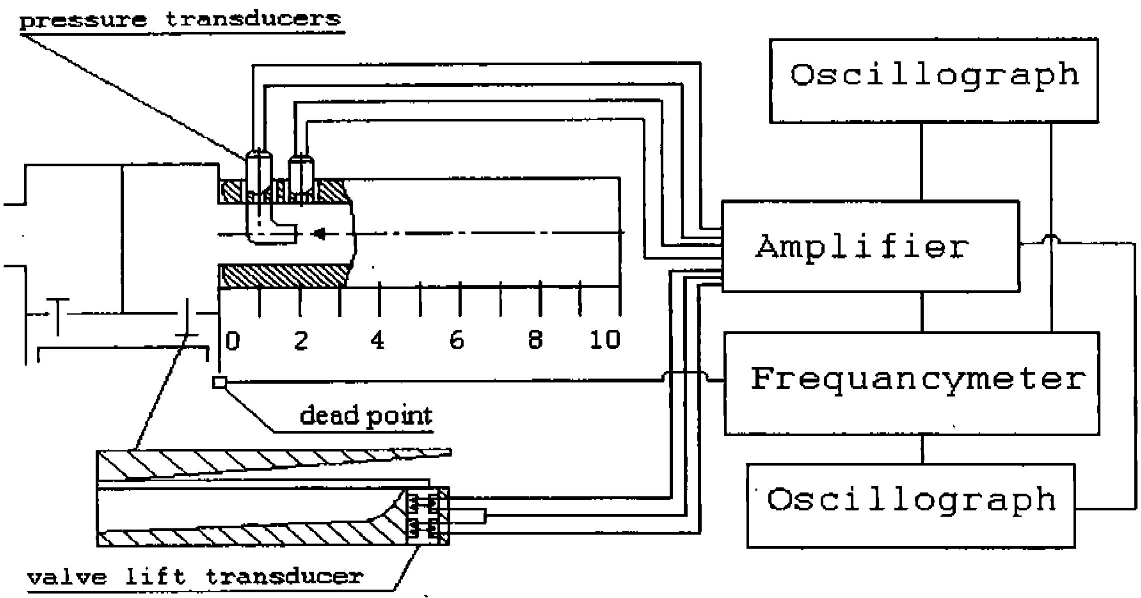

Fig.1 Scheme of experimental installation

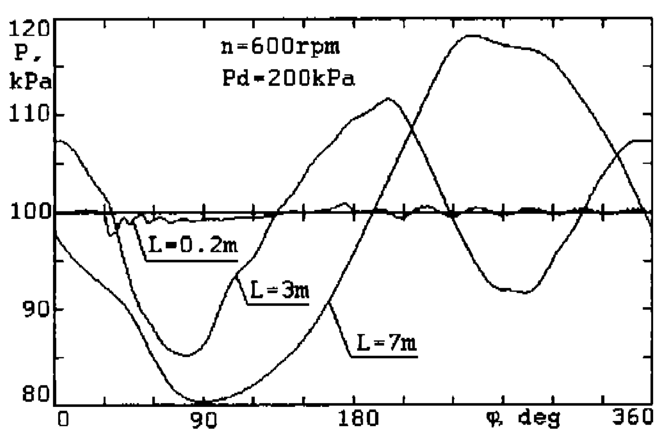

Fig.2 The pressure pulsations at the frequencies (calculated)

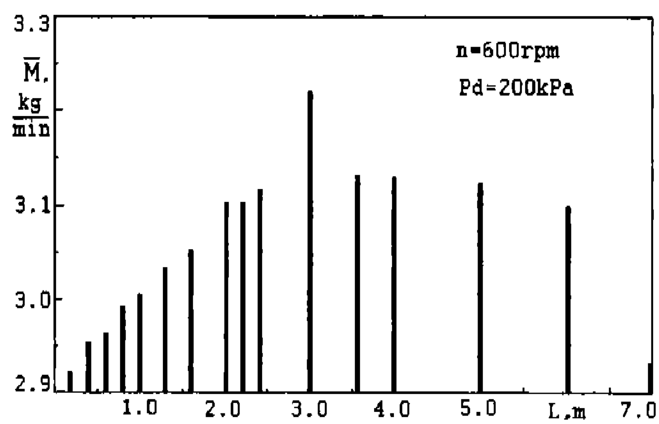

Fig.3 Influence of length of suction pipe on the mass flow rate (calculated)

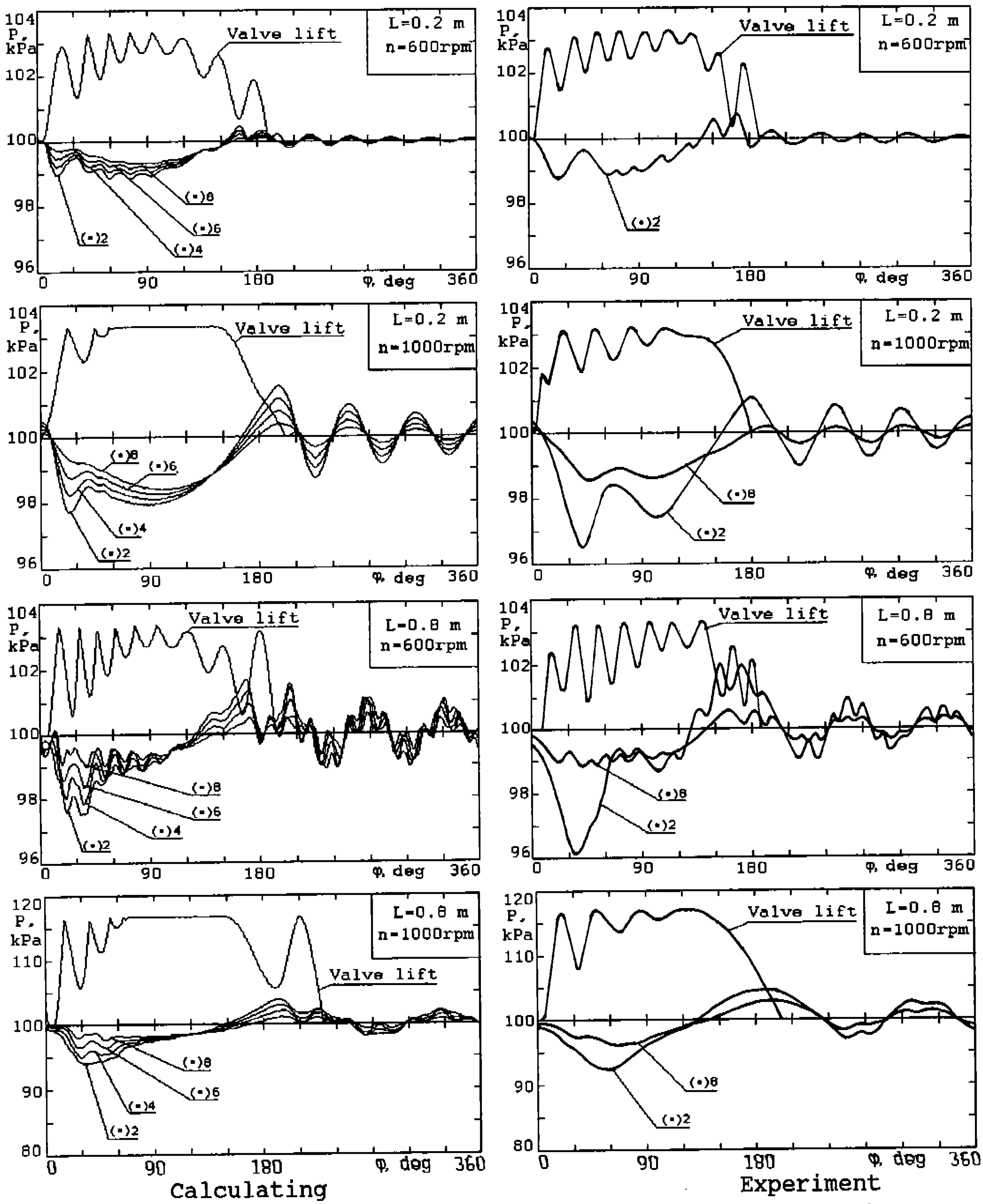

Fig.4 The theoretical and experimental reserch of processes in suction system.