Attenuation of Gas Pulsations Using A Perforated Tube

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INTRODUCTION

The presence of pressure pulsation in the piping of a reciprocating compressor results from the working process of compressor . Pulsation is produced by unsteady cyclically repeated suction or discharge and its influence manifests itself not only in the piping but also in the working process of the compressor. This influence is unfavourable, in a predominant measure, not only for the operation of the compressor but also for its service life and therefore the effort to limit pulsation to the lowest possible measure is quite naturally.

There are a number of possibilities how to reach a reduced level of pulsation but not all of them are equally suitable. Let us therefore determine some criteria for a judgement of them. The damping member is expected to

1) reduce the level of pulsation within a wide frequency band
2) not to obstruct the flow of gas and not to produce a major pressure loss
3) not to be costly and complicated.

Using a calculation a low level of pulsation can be reached by the choice of suitable lengths of pipes and volumes of connected equipment. However, such a solution is limited to a relatively narrow band of frequencies and it is not easy to comply with. Even though it is not customary to vary the speed of a compressor , a change of the temperature of the gas and, as a consequence, also a change of the speed of sound may considerably affect the tuning of the system of pipes. And such changes are common, for instance, with refrigeration compressors .

Although the often used orifice plate satisfies the requirement of damping within a wide frequency band it is usually a source of energy losses when it must be located in a pipe through which gas flows continuously. Fig.1a.

The criteria mentioned above are comparatively best satisfied by an orifice plate situated outside the direction of flow of the gas or by a tube with perforations.

The damping principle is the same in both cases. If the pressure inside the tube is greater the gas flows out from the tube, in the opposite case it flows in the reverse direction. By this alternation flow energy losses are caused, mostly at the detriment of the gas pulsation. The flow conditions of the tube are not substantially impaired thereby. Fig.1b.

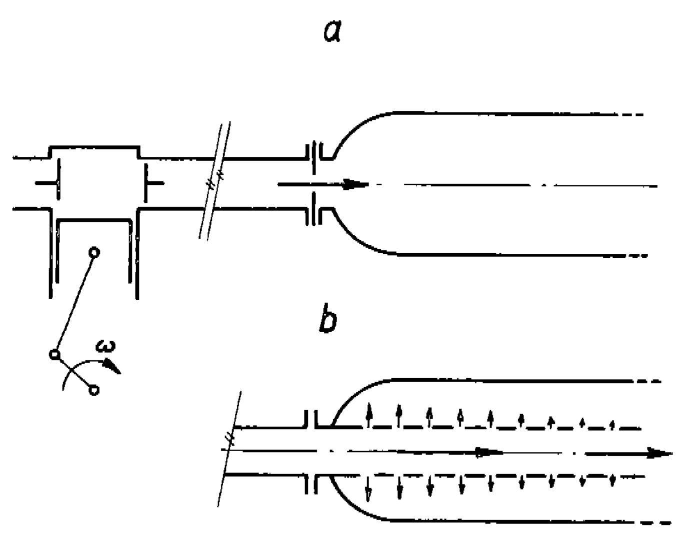


FIGURE 1

A characteristic representative of this type of damper is the perforated tube. Its properties, from the point of view of pulsation damping, can be expressed analytically. For that purpose it is assumed that the pressure on the outside of the tube is constant and equal to the mean pressure in the piping system.

PERFORATED TUBE

Let us have a tube of a constant cross section provided with a row of perforations. Fig. 2. When the wave equation is

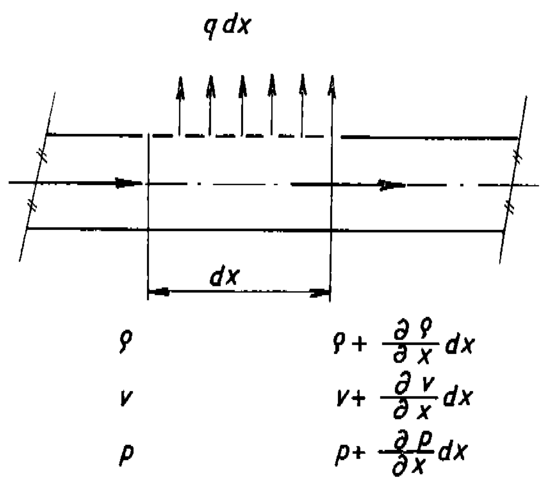


FIGURE 2

applicable, with a certain approximation, to a nonstationary flow through a tube, then an analogous equation can be derived for the tube with perforation in the shape

$$\rho_0 \frac{\partial v}{\partial t} + \frac{\partial p}{\partial x} = 0 \qquad \rho \doteq \rho_0$$

$$\rho_0 a^2 \frac{\partial v}{\partial x} + \frac{\partial p}{\partial t} = -\rho_0 \frac{a^2 q}{F} \qquad a^2 = \frac{dp}{d\rho}$$

The member $-\rho_0 \frac{a^2 q}{F}$ on the right side of the equation represents the pressure loss due to the outflow of the gas from the perforated tube.

The value q is determined on the assumption that the flow of gas through the perforations in the tube is governed by relations analogous to those of flow through an orifice plate. Thus the following will hold good for

$$q = \psi \cdot \varepsilon \cdot K \cdot \sqrt{2gRT} \cdot \sqrt{\frac{p - p_0}{p}}$$

during the flow of the gas from the tube and

$$q = \psi \cdot \varepsilon \cdot K \cdot \sqrt{2gRT} \cdot \sqrt{\frac{p_0 - p}{p_0}}$$

during the inlet of the gas into the tube.

A solution of the above relations for the given boundary conditions in the closed form is impossible. Therefore the method of characteristics was used, modified with regard to the quantity of gas which is flowing through the perforations of the tube. The solution is shown schematically in the p - Q diagram in Fig. 3. From the known conditions at points " $i-1$ " and " $i+1$ " a position of point " i " is sought with a view to satisfying, at the same time, also the condition for the outlet or inlet of gas through the perforations in relation to the pressure at that point. Obviously this is an approximate solution which will be the more accurate the shorter the step of the calculation /Δ x /.

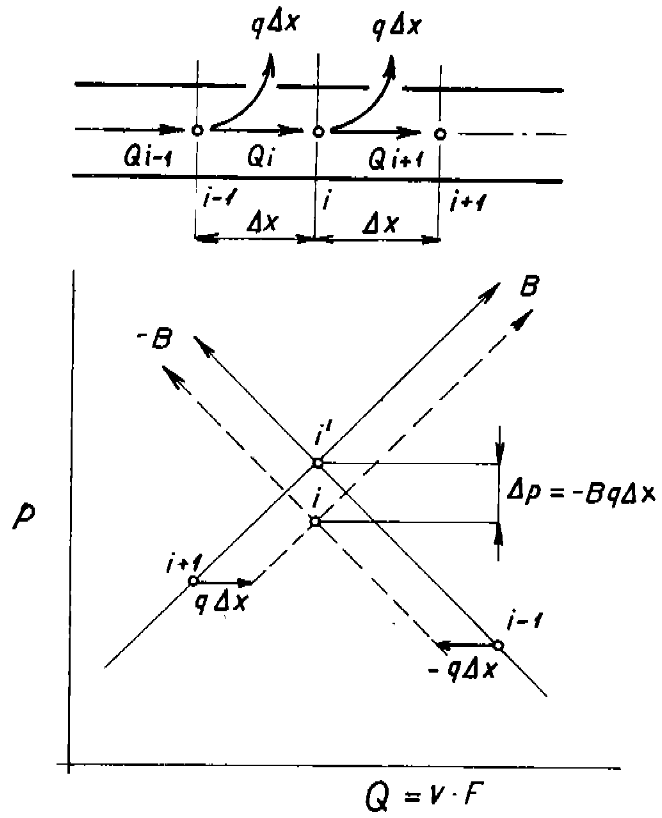


FIGURE 3

A perforated tube is an effective damper, though only as long as the perforations are situated at points where pronounced pressure changes take place inside the tube. When the perforation is at the end of the tube the final effect depends to a considerable degree on the length of the perforation. Its length should not be less than approximately one tenth of the wave length of the frequency which is still to be damped effectively.

ANALYTICAL MODEL

The present concept of piping systems of reciprocating compressors with dampers on both the suction and the delivery sides creates favourable conditions for the application of the perforated tube as a damper of the pulsation which originates in the space between the valve chamber and the damper and reaches considerable values in comparison with all other parts of the piping system.

In order to prove the efficiency of the damping of pulsation by a perforated tube and, at the same time to test the developed analytical method, an analytical model was built for the area between the working space and the dampers. The model is shown in Figs. 5 and 6. As regards the pressure in the dampers it is assumed to be constant or at least that its changes are negligibly small in relation to the pressure changes in the valve chambers.

The model is made up of relations for the following processes:

1) Adiabatic compression and expansion in the working space.
2) Relations for flow through valves. Included in the calculation is the pressure loss due to the gas flow through the valve but the dynamics of the valve plates and leakage of the valves are not considered.
3) Non-stationary flow in the tube with consideration of the outlet of gas from the tube and inlet of it at the points of the perforation.

The entire solution is programmed on a computer. Since every numerical solution must start from concrete values inputs were inserted into the calculation which correspond to the experimental equipment with a ČKD Type 2 SK 240 B compressor modified to operate as a single- acting unit. The results obtained by calculation as well as experimentally are shown in Figs. 5 and 6.

The values of pressure pulsation are presented here depending on the compressor speed. Regarding to the very different form of the pressure pulsation it would be very difficult to compare them mutually in their original form and a comparison according to peak values would not be correct. Therefore a substitution was used, according to Fig. 4. The plotted amplitudes of the pressure changes are the amplitudes of harmonic oscillations the mean values of which are identical with the mean values of the measured oscillations. For completion also a comparison is shown of the calculated behaviours of the pressure changes in the valve chambers with the measured behaviours.

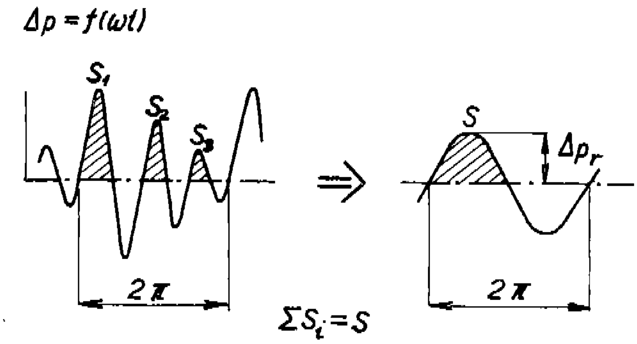


FIGURE 4

CONCLUSION

The identical results of the computation and the experiment prove that, provided the length of the perforation is adequate to the wave length of the frequencies to be damped, a perforated tube is an effective means with the capability of pulsation damping within a very wide frequency range. Also apparent, however, from the measurements carried out, is the fact that the damping effect of the tube is limited to only that part of the piping system to which the perforated tube is adjacent. From the opposite side its effect is small. For instance, in the case of the discharge piping the resonance amplification of oscillations at a speed of approximately 400 r.p.m. which corresponded to the tuning of the damper - receiver system manifested itself in an almost unreduced measure also in the valve chamber.

Even though the perforated tube cannot be considered a universal means for the damping of pulsation it has a number of advantages thanks to which it will doubtlessly be used to good advantage in many cases. Capable of contributing in no small measure to a correct choice and suitable utilization of it is also the mathematical solution.

REFERENCES

Bráblík J., Analysis of Movement of Valve Plate of Automatic Valve under Influence of Pulsation of Gas in Piping of Reciprocating Compressor. Candidate's Thesis. Czechoslovak Akademy of Sciences, Praha, 1969.

Notation

a Acoustic velocity

g Gravitational acceleration

p Pressure in tube

p_1 Pressure in valve chamber

p_2 Pressure in damper

p_0 Mean pressure in tube

q Volume of gas escaping from the tube per unit of time and on a unit legth

t Time

v Velocity of gas

x Coordinate of length

F Cross section of tube

K Total area of damping holes per unit of length of tube

Q Volume of gas per unit of time

R Gas constant

T Temperature

ε Expansion coefficient

ρ_0 Specific mass of gas at p_0

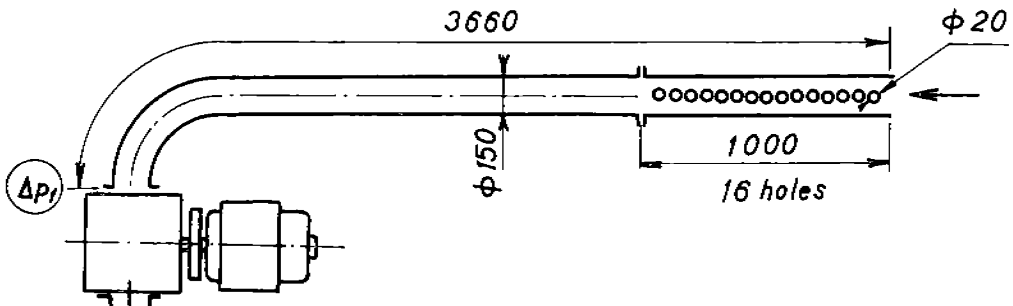
ψ Flow coefficient of holes in the tube

ω Angular velocity

$B = \frac{\rho_0 a}{F}$

$Q = vF$

SUCTION : Air, atm. pressure

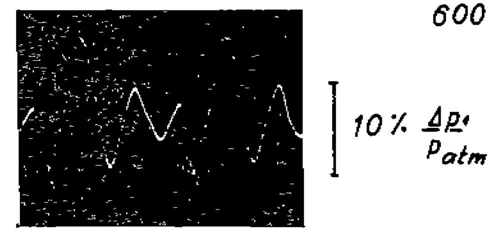


Single acting compressor
bore : 240 mm , stroke : 150 mm

FORM OF DAMPED PRESSURE PULSATION IN THE VALVE CHAMBER

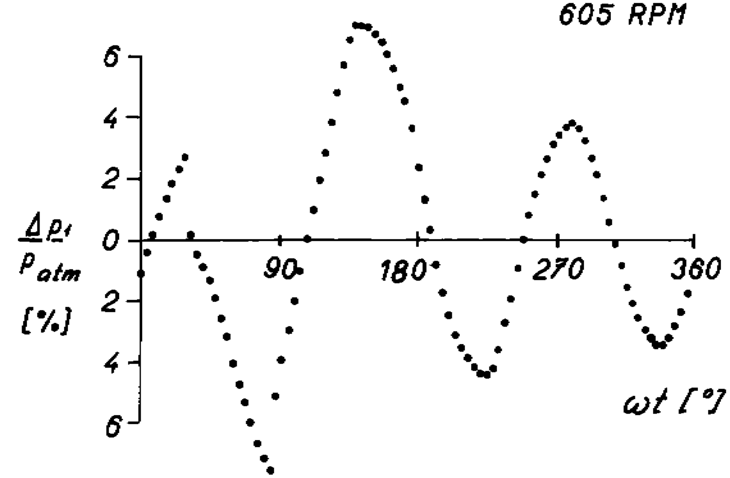
measured

600 RPM

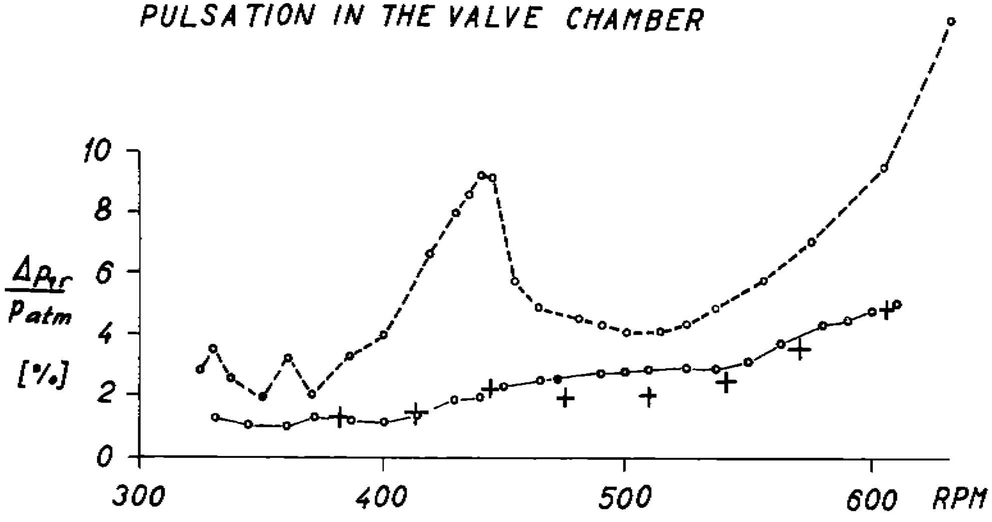


computed

605 RPM



PULSATION IN THE VALVE CHAMBER



Suction pipe : with perf. end : (measured) o——o——o (computed) + + +
without perfor: (—"—) o----o----o (pipe length 3660 mm)

FIGURE 5

DISCHARGE Air, pressure 0'392 MPa

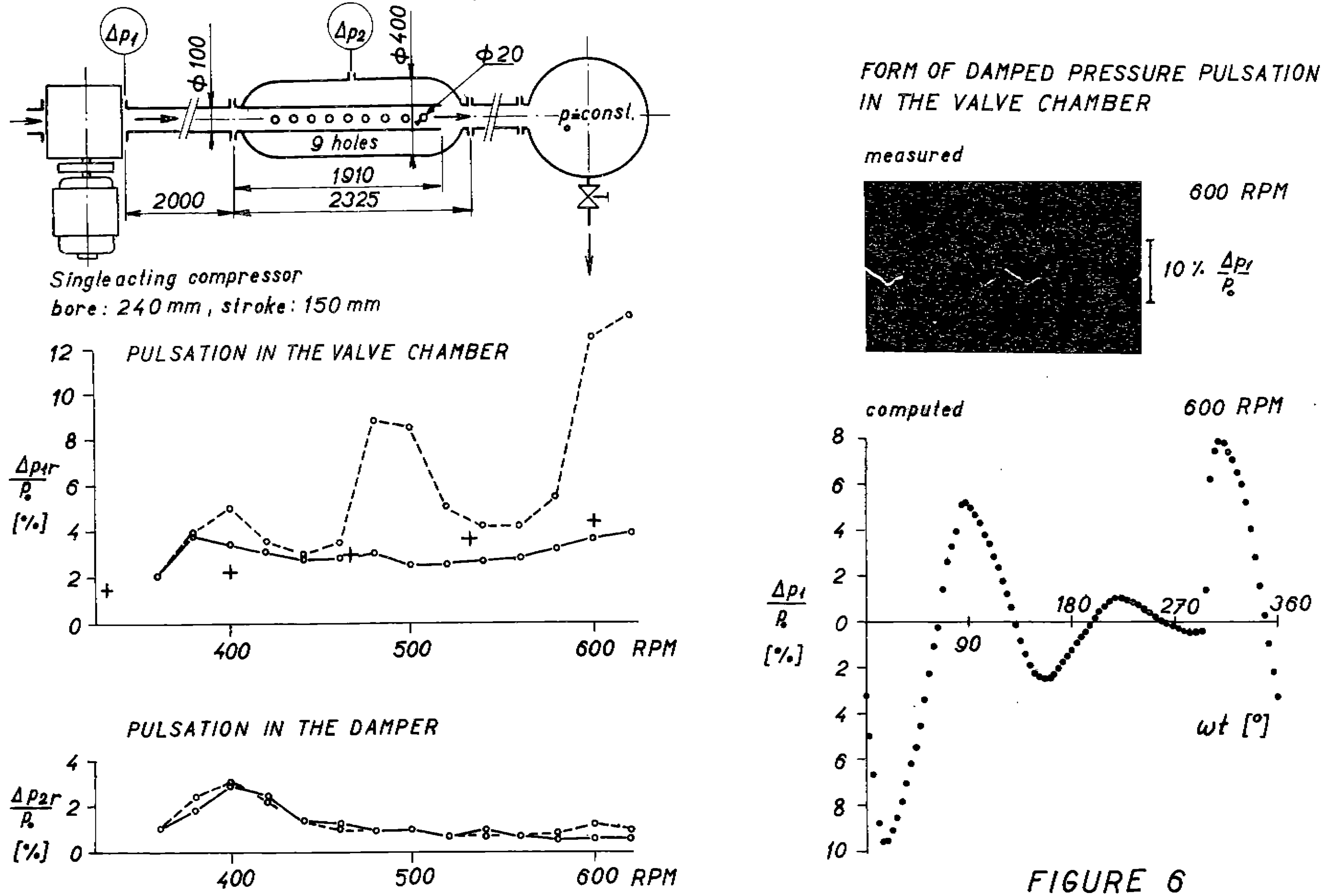


FIGURE 6

Discharge pipe: with perf. end: (measured) o—o—o (computed) + + +
without perf. end (—॥—) o- - -o- - -o (pipe length 2000mm)