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# STUDY OF PNEUMATIC DAMPING BEHAVIOUR FOR RECIPROCATING COMPRESSOR'S VALVES

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## ABSTRACT

Based on the theoretical analysis and experimental study of pneumatic damping behaviour for the valves in the reciprocating compressor, this paper presents a simple mathematical model of pneumatic damping valves, in which the non–viscous damping force is introduced. The non–viscous damping coefficient is obtained by the experimental determination. The simulation results of this simple model for pneumatic damping valves are in good agreement with the experimental results.

## INTRODUCTION

Many studies of compressor's valves show that valve damages result mainly from excessive stresses of the valves striking on the valve stop or valve seat. Thus, apart from the cause of the material used, reduction of the striking velocity of valve plate on the valve stop or seat will prolong the service life of the valves remarkably.

Based on the idea mentioned above, a valve with pneumatic damping structure has been designed and put into production recently. Because of its advantages of low noise, long service life and reliable performance, the pneumatic damping valve is welcomed by consumers and has a bright prospect for its application.

In the related literature about pneumatic damping valves in the compressor, the mathematical model is modified either by the introduction of the viscous damping term into the valve movement equation, or by the addition of the gas pressure loss equation to the pneumatic damping pocket, both of them are satisfactory with computed results. In this paper, the valve movement equation is modified by introducing the non–viscous damping term, and the characteristic curve of the non–viscous damping coefficient is obtained by experiments, therefore the calculation for the valve would be convenient.

## MOVEMENT EQUATION

A common ring valve is generally assumed to be a one–dimensional and single–particle system (Fig.1), and its movement equation is:

$$m\frac{d^2h}{dt^2} + K(h_0 + h) = \beta A \Delta p \tag{1}$$

where $\Delta p$ is the pressure difference between the outlet and inlet of valve: for the suction valve, $\Delta p = p_s - p_c$, for the discharge valve, $\Delta p = p_c - p_d$, the subscript $s$ is suction, $d$ is discharge, $c$ is cylinder;

$\beta$ is the draging force coefficient;

$A$ is the exit area of valve seat;

$h$ is the displacement of valve plate;

$K$ is the spring stiffness;

$h_0 = L_f - L_1$, $L_f$ is the free length of spring, $L_1$ is the compressed length of spring when the valve closes;

$m = m_V + \frac{1}{3} m_S$, $m_V$ is the mass of valve plate, $m_S$ is the mass of spring.

Since the pneumatic damping pocket is designed within the valve stop, the valve can only move in the ring channel when it opens or closes (see Fig.2). Flowing of gas into and out of the pneumatic damping pocket through a small hole and the matching clearance between the valve plate and the ring channel leads to gas pressure variations in the pneumatic damping pocket, thus having buffering effect on the valve movement.

Take a discharge valve for example, the pressure difference between the inner and outer parts of the pneumatic damping pocket is $\Delta p_p = p_p - p_d$, while the pressure difference between the inlet and outlet of the valve is $\Delta p = p_c - p_p$, and this pressure difference $\Delta p$ is resolved as follows :

$$\Delta p = (p_c - p_d) - (p_p - p_d)$$

where $(p_c - p_d)$ is the same as the pressure difference between the inlet and outlet of the valve for a common ring valve, and $(p_p - p_d)$ is the pressure difference between the inner and outer parts of the pneumatic damping pocket.

When the valve opens, its velocity $\frac{dh}{dt} > 0$, and the varying rate of the pocket volume $\frac{dV_p}{dt} < 0$, thus the gas is compressed in the pocket ($p_p > p_d$). Therefore in the pocket a gas force against the valve movement direction is produced, which relaxes the valve striking force on the valve stop. On the contrary, when the valve closes, $\frac{dh}{dt} < 0, \frac{dV_p}{dt} > 0$, and $p_p < p_d$, so a gas force against the drop of valve takes place, which buffers the drop of valve to some degree. Therefore the gas force produced is always against the valve movement, and this force $F_D$ is equal to $\beta A \Delta p_p$. When the concept of "damping" in vibration theory is introduced, and its definition is :

$$F_D = C_D' \cdot \theta(v) \tag{2}$$

where $C_D'$ is the damping coefficient, $\theta(v)$ is a function of valve plate velocity $v$.

The valve system is simplified into a one–dimensional and single–particle vibration system with the damping (Fig.3). Thus the movement equations are as follows:

for a suction valve,

$$m\frac{d^2 h}{dt^2} + C_D' \cdot \theta(v) + K(h_0 + h) = \beta A(p_s - p_c) \tag{3}$$

for a discharge valve,

$$m\frac{d^2h}{dt^2} + C_D' \cdot \theta(v) + K(h_0 + h) = \beta A(p_c - p_d) \tag{4}$$

## EXPERIMENTAL INVESTIGATION

According to the damping theory of mechanics, the damping force is defined as a linear relation to velocity or a non–linear relation to velocity, and has a clear physical meaning. The former is viscous damping, and the latter is non–viscous damping. This theory is then introduced into our investigation. For viscous damping force,

$$F_D = C_D^* v = \beta A \Delta p_p = \beta A(p_p - p_d)$$

*or*

$$C_D^* = \beta A(p_p - p_d) / v \tag{5}$$

for non–viscous damping force,

$$F_D = C_D v^2 = \beta A \Delta p_p = \beta A(p_p - p_d)$$

*or*

$$C_D = \beta A(p_p - p_d) / v^2 \tag{6}$$

$C_D^*$ and $C_D$ are defined as viscous damping and non–viscous damping coefficients respectively, and their valves can be determined by experiments. Our experiments are performed in an air compressor, model II ZA–1.5 / 8, with a pneumatic damping valve. With the change of valves parameters and operating conditions (leakage area, volume of pneumatic damping pocket and compressor rotation speed, etc.), the effect of these parameters on the damping performance of valve is investigated, and the displacement curve of valve and the variation curve of pressure inside the pocket are measured.

The experimental results show that when the valve opens, the pressure inside the pocket varies remarkably, and a large damping pressure difference $\Delta p_p$ is formed due to the greater velocity of valve plate; when the valve closes , the value of pressure difference $\Delta p_p$ in the pocket is smaller due to the smaller velocity of valve plate.

With the increase of the leakage area in the pocket, the damping pressure difference of the pocket is decreased when the valve opens and closes, and the valve striking velocity on the valve stop and valve seat is decreased.

When the leakage area approaches to a certain degree, the damping pressure difference $\Delta p_p$ cannot be formed in the pocket owing to excessive leakage of the pocket.

The movement velocity curve of the valve can be obtained by the differential treatment of the displacement curve of the valve plate (see Fig.4). In the experiment the application of microcomputer–control to the high–speed data collecting system requires a certain precision. Fig.4 shows that during the rise course of the valve plate, when the valve plate rises to a lower stage, its moving velocity increases more rapidly because the pressure difference $\Delta p_p$ required by the damping force in the pocket has not yet formed; and after a period of time, its moving velocity becomes slower because the pressure difference $\Delta p_p$

has formed; finally when the plate reaches the valve stop, its moving velocity is the maximum value.

The three curves in Fig.4 express the velocity curves with different leakage areas. With the increase of the leakage area ratio $\delta$, the damping becomes weaker and the valve plate velocity faster.

## EXPERIMENTAL RESULTS

The experimental data are treated separately with equations (5) and (6). The curve of viscous damping coefficient and that of non–viscous damping coefficient are obtained as shown in Fig.5 and Fig.6 respectively. From the figures, the variation of viscous damping coefficient $C_D^*$ with valve displacement $h$ is larger, while the variation of non–viscous damping coefficient with valve displacement $h$ is smaller. Hence the introduction of non–viscous damping coefficient will facilitate the computation of the valve.

The experiment also shows that non–viscous damping coefficient is unrelated to the compressor rotation speed and the volume of pneumatic damping pocket. Although the striking velocity of valve plate on the valve stop incresses with the increase of the compressor rotation speed, yet the damping pressure difference $\Delta p_p$ in the pocket also increases, and the damping effect strengthens, thus leading to basic constant of non–viscous damping coefficient.

Fig.7 shows the variation of non–viscous damping coefficient with the leakage area ratio $\delta$.

## VERIFICATION

In order to verify the correctness of the movement equation of the pneumatic damping valve with a non–viscous damping term and the practical use of non–viscous damping coefficient curve, the operating process of the compressor's valve is simulated by computer and verified by experiment, and the results are satisfactory (see Fig.8). Fig.8 shows the comparison between the computed and experimental valve plate displacement $h$, cylinder pressure $p_c$ and valve plate velocity $v$. The operating conditions are $p_d \approx 0.5MPa$ and $n=510rpm$. The experiment and computation are performed with a compressor, type Π ZA–1.5 / 8.

## CONCLUSIONS

Modification of the movement equation by introducing the non–viscous damping term makes the mathematical model simpler and its physical meaning clearer.

The damping effect of pneumatic cushion depends mainly on the structure parameters of the pneumatic pocket and the movement velocity of the valve plate. The non–viscous damping coefficient may be considered as relevant to the leakage area ratio only.

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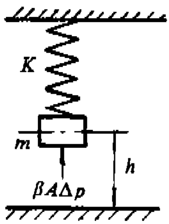


Fig.1 One–dimensional and single–particle system.

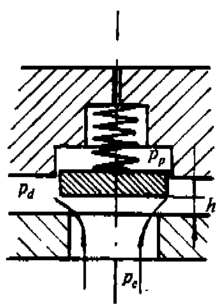


Fig.2 Pneumatic damping valve model.

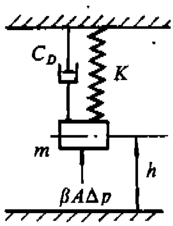


Fig.3 Vibration system with damping.

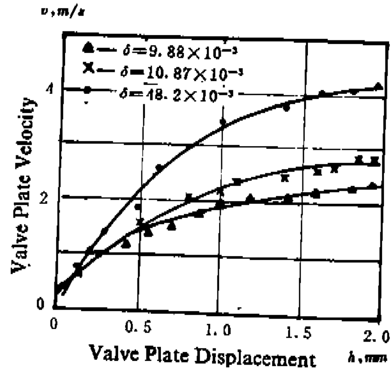


Fig.4 Velocity curves of the valve plate while it opens.

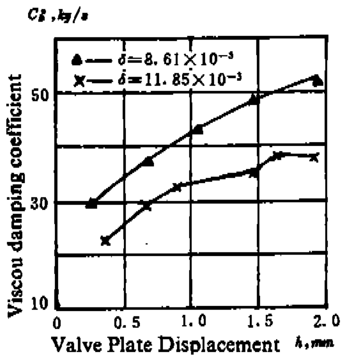


Fig.5 The curves of viscous damping coefficient.

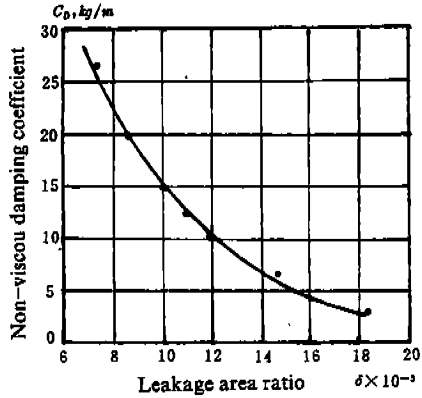


Fig.7 . The variation of non–viscous damping coefficient with the leakage area ratio.

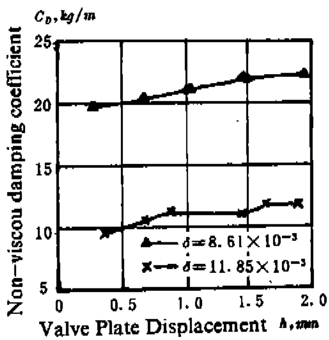


Fig.6 The curves of non–viscous damping coefficient.

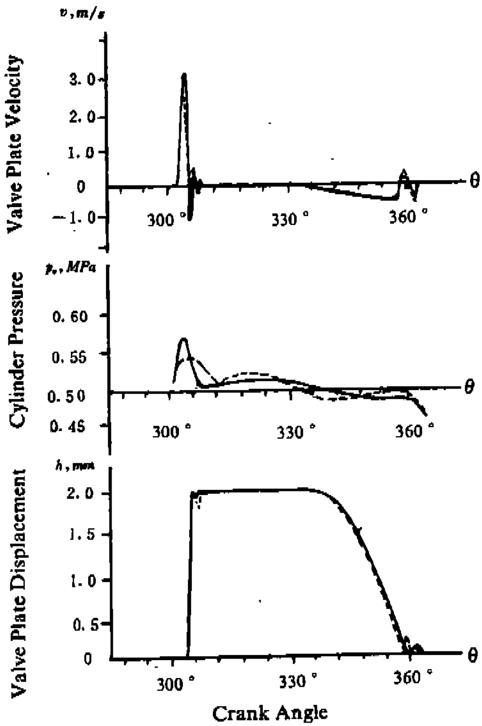


Fig.8 Experimental verification curve of the aircompressor, —computed value, ----exp.value.