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NUMERICAL ANALYSIS OF THE DYNAMICS OF REED VALVES TAKING INTO ACCOUNT THE ACOUSTIC COUPLING WITH THE FLUID.
APPLICATION TO COMPRESSORS FOR DOMESTIC REFRIGERATION.

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ABSTRACT

The dynamics of reed valves of a reciprocating compressor is tightly coupled with the dynamics of fluid inside compressor ducts. A global model of a refrigerator compressor is developed taking into account the net flow through the ducts. Results show that reed valves vibrations influence the compressor cycle and can produce noise in the high frequency range.

INTRODUCTION

Most domestic refrigerators are equipped with reciprocating compressors. Reed valves are used to connect the compressor cylinder with both the suction and the discharge circuit. The opening and the closure of these valves are caused by the static and dynamic pressures which the refrigerating fluid exerts on their surfaces. Reed valves during their motion cyclically collide with the seat, which accomplishes the seal of the gas and with the opening limiter, therefore they vibrate and transmit impulsive forces to the compressor body. Since the flow of the refrigerating gas depends on the area of flow through reed valves, valves vibrations disturb the gas flow and cause pressure fluctuations; it is interesting to highlight that pressure fluctuations also affect valve motion, which is the primary source of vibrations.

For the above-mentioned reasons reed valves motion is one of the most important sources of excitation in domestic refrigerator compressors and, owing to collisions, it is able to generate noise and vibrations in the high frequency range.

As reed valves motion is tightly coupled with pressure fluctuations inside compressor ducts, the two phenomena have to be studied simultaneously by means of a global model of compressor [1][2]. In the first part of this paper a global dynamic model of a real compressor is developed; it comprises a model of reed valves, a model of gas compression inside the cylinder of the compressor and a model of compressor circuits.

The main features which differentiate this model from the models developed by other authors are the following. First the valves are simulated as systems with many degrees of freedom, whose modal parameters are calculated by means of a finite elements code [3]; this approach allows to simulate with good approximation the actual vibration of reed valves. Then, not only the suction ducts and the discharge ducts are simulated, but the whole refrigerating circuit is modelled; with this approach gas flow through compressor ducts can be simulated and the flow through reed valves depends on the pressure fluctuation caused by valves motion itself. In the second part of the paper the results of a series of numerical simulations are presented. They were aimed at showing the effect of valves motion on the compressor cycle and at pointing out the high frequency vibrations caused by valves motion.
MATHEMATICAL MODEL

In figure 1 the scheme of the refrigerating gas circuit of a GL 80 reciprocating compressor is represented, where CC is the cylinder volume. Downstream from the discharge valve there are three small chambers (C1, C2, C3) connected in series by means of two ducts (T1, T2). The compressor cavity (C5) and three small chambers (C6, C7, C8) are upstream from the suction valve, they are connected in series and in parallel by ducts (T6, T7, R1, R2, R3). Ducts T3 and T4 and chamber C4 simulate the impedance of the circuit through the condenser, the capillary tube and the evaporator. This "closed loop" model can simulate the net flow of the refrigerating fluid through the refrigerator. Rotation speed of the compressor is assumed constant (314 rad/s); the refrigerating fluid is R134A having the following properties at t=60°C and p=101325 Pa: density $\rho=3.73$ kg/m$^3$; dynamic viscosity $\mu=1.862\times10^{-5}$ kg/(m s)

**Compressor chamber**

With the assumptions of ideal gas and of polytropic process the following equation is derived

$$\frac{dp}{dt} - n \frac{q_{\text{suction}}(t) - q_{\text{discharge}}(t) - \dot{V}(t)}{V(t)} p = 0$$

where $p$ is the pressure, $q$ the arc gas flows and $n$ is the exponent of polytropic process. $V(t)$ is the cylinder chamber volume, which, owing to piston motion, varies with the laws:

$$V(t) = A_{\rho} \left[ r(1 - \cos(\Omega t)) + l \left( 1 - \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2(\Omega t)} \right) \right] + V_n$$

$$\frac{dV(t)}{dt} = A_{\rho} r \Omega \sin(\Omega t) \left[ 1 + \frac{r}{l} \cos(\Omega t) \right] \sqrt{1 - \left( \frac{r}{l} \right)^2 \sin^2(\Omega t)}$$

where $r$ is the crank length and $l$ is the coupler length and $V_n$ is the minimum cylinder volume.

**Chambers**

The equation derived for compressor chamber holds (it becomes simpler as $V(t)=$constant$=V$):

$$\frac{dp}{dt} - n \frac{q_{\text{input}}(t) - q_{\text{output}}(t)}{V} p = 0$$

It is useful to point out that, since $p$ is the pressure (not only pressure fluctuation), this equation is not the standard equation which is used to represent a chamber in a lumped element model of an acoustic circuit [4].

**Ducts**

The equation which describes the motion of gas inside the ducts is

$$l' \frac{dv}{dt} + \left( \xi_r \sqrt{|v|^{0.25} + \xi_{d,T}} \right) \sqrt{|v|^{0.75}} = \frac{p_1(t) - p_2(t)}{\rho}$$

where $l'$ is the effective length of the duct, $p_1$ is the inlet pressure, $p_2$ is the outlet pressure, $v$ is the velocity, $\rho$ is the mean density of the fluid and $v_L$ is the velocity of transition between laminar regime and turbulent regime. This equation describes not only the vibration of the fluid, but also the mean flow through the duct, therefore flow resistance is calculated by means of the classical equations of fluid mechanics. The first term which depends on velocity accounts for minor losses and $\xi_{r} = \sum \psi_i / 2$, where $\psi_i$ is the loss coefficient of the $i^{th}$ minor loss [5]. The second term depending on velocity accounts for dissipation along the length of the duct (Blasius formula) and $\xi_{d,T} = 0,1582 \mu^{0.25} l / \rho^{0.25} D^{0.25}$.

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Connection of three ducts

The equations of ducts R1, R2 and R3, which join in the same point, are the following:

\[
\begin{bmatrix}
\frac{dv_1}{dt} \\
\frac{dv_2}{dt} \\
\frac{dv_3}{dt}
\end{bmatrix} =
\begin{bmatrix}
A_1 & -A_2 & -A_3 \\
A_2 & V_1 & V_2 \\
A_3 & V_1 & 0
\end{bmatrix}^{-1}
\begin{bmatrix}
\rho \frac{dv_1}{dt} - A_1 \frac{dv_1}{dt} - A_1 \frac{dv_2}{dt} \\
\rho \frac{dv_2}{dt} - A_2 \frac{dv_2}{dt} - A_2 \frac{dv_3}{dt} \\
\rho \frac{dv_3}{dt} - A_3 \frac{dv_3}{dt} - A_3 \frac{dv_1}{dt}
\end{bmatrix}
\]

They were calculated on the basis of the equation of the three ducts and of continuity equation.

Reed valves

The valve is simulated as a n-degrees of freedom system by means of the finite element method and the first m modes of vibration are calculated by means of a computer code. Once performed this numerical analysis the equation of motion of the reed valve are:

\[
[m][\ddot{\eta}] + [c][\dot{\eta}] + [k][\eta] = [U]^T \{F\} = [U]^T \{F_{\text{hole}}\} + \{F_{\text{external}}\} + \{F_{\text{impact}}\}
\]

where \(\{\eta\}\) is the vector mx1 of modal displacements; \([m]\), \([c]\) and \([k]\) are the mxm modal matrices of mass, damping and stiffness respectively; \([u]\) is the modal matrix of the valve having dimension mxn. \([F]\) is the vector of the forces applied in the nodes of the finite elements mesh; these forces are equivalent to the distributed loads caused by refrigerating fluid pressure and by impacts.

As far as the suction valve is considered, the forces equivalent to pressure on the part of the valve over the suction hole can be expressed as \(\{F_{\text{hole}}\} = (p_{\text{CS}} - p_{\text{CC}} + \rho v_{\text{suction}} v_{\text{suction}}) \{D_{\text{hole}}\}\) where the first term accounts for static pressure on the upper surface of the valve, the second accounts for static pressure on the lower surface of the valve, whereas the third accounts for dynamic pressure on the upper surface of the valve caused by fluid motion with velocity \(v_{\text{suction}}\). Velocity \(v_{\text{suction}}\) is calculated with expression

\[
v = A \alpha (h_{\text{max}}) \frac{h_{\text{ave}}}{h_{\text{max}}} (p_{\text{CS}} - p_{\text{CC}}) \sqrt{\left(1 - \rho \frac{h_{\text{ave}}}{\rho_{\text{ave}} (p_{\text{CS}} - p_{\text{CC}})}\right)},
\]

where \(\alpha\) is a discharge coefficient which is assumed to depend linearly on the average valve displacement \(h_{\text{ave}}\). \(A\) is the area of the suction hole and \(\rho_{\text{ave}}\) is the average density of the refrigerating fluid. Pressure on the suction valve is not uniform, its distribution is taken into account in the model by introducing the distribution vector \(\{D_{\text{hole}}\}\) which is based on the experimental results presented in [6].

The forces equivalent to pressure on the rest of suction valve surface can be expressed as

\[
\{F_{\text{ext}}\} = (p_{\text{CS}} - p_{\text{CC}}) \left(1 - \frac{h_{\text{ave}}}{\rho \frac{h_{\text{ave}}}{e h_{\text{max}}} (p_{\text{CS}} - p_{\text{CC}})}\right) \{D_{\text{ext}}\}
\]

where \(\{D_{\text{ext}}\}\) is the distribution vector which represents pressure distribution on the surfaces of the valve which are not over the hole, and \(e\) is an experimental coefficient.

The forces acting on discharge valve are calculated in similar way.

The forces caused by the impacts of the valve with the seat and the opening limiter are calculated with the model represented in figure 2. The following equations hold:

\[
\{F_{\text{impact}}\} = \begin{bmatrix}
F_1 \\
\vdots \\
F_i \\
\vdots \\
F_n
\end{bmatrix},
F_i = \begin{cases}
-k_{\text{seat},i} h_s - c_{\text{seat},i} \dot{h}_i & h_i < 0 \\
0 & 0 \leq h_i \leq h_{\text{stop},i} \\
-k_{\text{stop},i} (h_s - h_{\text{stop},i}) - c_{\text{stop},i} \dot{h}_s & h_i > h_{\text{stop},i}
\end{cases}
\]

Fig. 2: Model of impact

where \(h_i\) is the vertical displacement of the \(i^{\text{th}}\) node of the mesh, \(k_{\text{seat}}\) and \(c_{\text{seat}}\) are the stiffnesses and the damping coefficients of the seat, whereas \(k_{\text{stop}}\) and \(c_{\text{stop}}\) are the stiffnesses and the damping coefficients of the
opening limiter. The damping coefficients values are chosen in order to simulate the experimental coefficient of restitution.

NUMERICAL RESULTS

The drawings of the two reed valves with their meshes are represented in figure 3; the frequencies and the shapes of the modes calculated by means of the FEM code are summarized in the table. The suction valve modes can be divided into the classes of cantilever modes (C) and transverse modes (T). The cantilever modes are like those of a cantilever and they show a flexural deformation of the reed; the transverse modes are two-dimensional and show a torsional deformation of the reed. The first mode of the discharge valve can be considered a cantilever mode, whereas the others, which show periodic deformations of valve contour are named periodic (P).

The simulations of compressor behaviour were carried out taking into account five modes of vibration in the valve model (5 d.o.f model), for comparison other calculations were carried out considering only the first mode of vibration of the two valves (1 d.o.f model).

<table>
<thead>
<tr>
<th>Mode</th>
<th>Suction valve</th>
<th>Discharge valve</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>274.54 C</td>
<td>530.66 C</td>
</tr>
<tr>
<td>2</td>
<td>1690.03 T</td>
<td>1472.55 P</td>
</tr>
<tr>
<td>3</td>
<td>1715.40 C</td>
<td>3001.46 P</td>
</tr>
<tr>
<td>4</td>
<td>4621.00 C</td>
<td>3563.92 P</td>
</tr>
<tr>
<td>5</td>
<td>5516.32 T</td>
<td>5088.60 P</td>
</tr>
<tr>
<td>6</td>
<td>8677.20 C</td>
<td>6477.05 P</td>
</tr>
<tr>
<td>7</td>
<td>10076.19 T</td>
<td>7716.69 P</td>
</tr>
<tr>
<td>8</td>
<td>14502.13 C</td>
<td>9829.74 P</td>
</tr>
</tbody>
</table>

Tab. 1: Natural frequencies

Compressor cycle

A series of results is presented, which shows the effect of valves vibrations on the compressor cycle. Figures 4 and 5 show the cycles which were calculated with the 5 d.o.f model and with the 1 d.o.f model respectively. The large differences between suction and discharge flow rates are caused by the variation of the density during compressor cycle. The wide suction flow rate fluctuations which are foreseen by the 5 d.o.f model are the main difference between the two cycles. These fluctuation are caused by large vibrations of the suction reed valve, which is more flexible than in the 1 d.o.f model. The 5 d.o.f model shows also the back flow phenomenon: at the end of the suction phase, owing to the rebound of the suction valve from its seat, some refrigerating gas flows from the compressor chamber to chamber C8. The peak pressure foreseen by the
5 d.o.f model is a bit lower than the peak pressure foreseen by the 1 d.o.f model; the corresponding variation of thermodynamic cycle efficiency is negligible.

![Pressure time history](image)

Figure 6b shows pressure time history inside chamber C8. A series of oscillations having a frequency of about 1100 Hz is highlighted by the 5 d.o.f model. Since these oscillations were not foreseen by the 1 d.o.f model, they are related to the higher order modes of the reed valve.

The time history of pressure inside chamber C1 is represented in figure 6c; the pressure fluctuations, which have a frequency of about 450 Hz, were predicted also by the 1 d.o.f model, therefore they are probably related to acoustic vibrations of the fluid or to the first mode of vibration of the discharge valve.

**Valves vibrations**

As far as suction valve motion is considered (whose nodes are depicted in figure 3), fig. 7a shows that the tip of the reed (node 76) rebounds repeatedly from the limiter. The frequency of suction valve vibrations is about 1100 Hz; it is different from the natural frequencies calculated considering the reed valve alone, because valve vibrations are coupled with fluid motion and because, when the valve tip touches the limiter, there is an additional constraint. Figure 7b shows that three nodes located in a transverse section of the reed vibrate in the same way, therefore fluid forces and impact forces do not excite the transverse modes. This assumption is confirmed by figure 7c, which shows large modal amplitudes of cantilever modes (1,3,4) and negligible amplitudes of transverse modes (2,5).

![Suction pressure](image)

![Discharge pressure](image)

**CONCLUSIONS**

The 5 d.o.f model points out important aspects of suction valve vibrations which were not highlighted by the 1 d.o.f model. The valve vibrates at high frequency and causes high frequency oscillations in the suction circuit; these phenomena are related with high frequency noise emission. The gas flow through the suction hole, owing to valve vibrations, is rather irregular and there is a backflow phenomenon at the end of the suction phase. Discharge valve motion is rather regular and there are only low frequency oscillations in the discharge circuit.
Fig. 7: Suction valve motion

Fig. 8: Discharge valve motion

REFERENCES