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NUMERICAL STUDY OF A DYNAMIC BEHAVIOUR
OF SIMPLE REED VALVE

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Introduction
Reed valves are the common type of valves used in small air and refrigeration compressors. Dynamic analysis of reed valve even for simple shape is rather complicated [1]. The progress of the finite element method gives new possibility to solve this problem in relatively simple way [2, 3, 4, 5, 6]. But the majority of known results [2, 4, 5, 6] were obtained with the help of rather big computers and specialised software (SUPERB, ANSYS, NASTRAN). The present paper describes a rather simple method of dynamic analysis (based on the FEM techniques) for rectangular reed. Calculation can be carried out even on the personal computer and additional software is not necessary.

Model of valve
Valve under consideration contain a seat with set of ports, rectangular reed clamped on one side and stops of different shape. The dynamic loading of the reed is due to the differences between the cylinder pressure and discharge or suction pressures.

Mathematical model of reed motion
The equation of motion in matrix form of a finite element model of considered structure are given by [7]

(1) \[ [K] \{\ddot{u}\} + [C] \{\dot{u}\} + [M] \{u\} + \{F\} = \{0\} \]

where
- \([K]\) - the system stiffness matrix
- \([C]\) - system damping matrix
- \([M]\) - system mass matrix
- \([F]\) - vector of outside forces
- \([\{u\}]\) - vector of displacements.

In our case analysis is limited to reed of constant thickness and rectangular shape

Element stiffness matrix
The beam type of element are used [8]

![Element stiffness matrix diagram]

Vector of displacement has a form
\[ \{d\} = [\{w_1, y_1, w_2, y_2\}] \]

Deflection is described as
\[ w = a + bx + cx^2 + ex^3 \]

Loading (forces and moments) are applied only in ends. Taking boundary conditions
\[ w(0) = w_1; \quad \dot{w}(0) = \dot{y}_1; \quad w(1) = w_2; \quad \dot{w}(1) = \dot{y}_2 \]

we can obtain constants \(a, b, c, e\).

Later calculating bending energy
\[ V = \frac{1}{2} \int 4EJ(y_{1} + 3e x^2) \left( \frac{dy}{dx} - y_{2} \right)^2 dx \]

after integration in matrix form we have
\[ V = \frac{1}{2} \{d\}^T \{f\} \]

and
\[ \{f\} = 2EJ \begin{bmatrix} 6 & 31 & -6 & 31 \\ 31 & 21^2 & -31 & 1 \\ -6 & -31 & 6 & -31 \\ 31 & 1^2 & -31 & 21^2 \end{bmatrix} \]

System stiffness matrix \([K]\) is a sum of element matrix \([k]\).
Damping matrix

Damping matrix has a form of diagonal matrix

\[
\begin{pmatrix}
c1 & 0 \\
0 & c2
\end{pmatrix}
\]

where \(c1\) - damping coefficient of motion
\(c2\) - damping coefficient of rotation

\(c1\) and \(c2\) was equal 0.801

Mass matrix

Here also we have diagonal matrix

\[
\begin{pmatrix}
m & 0 \\
0 & mb
\end{pmatrix}
\]

where \(m\) - mass of element \((9.591)\)
\(mb\) - moment of inertia \((9.591^3)/96\)

Vector of forces

The reed is loaded by forces which are results of pressure differences on both sides of reed.

It was assumed that force is a product of pressure difference and port area. Forces were applied in nodes with weights depending on the relative position of ports and nodes.

In described model the forces are also unknown. So the model of compression and flow through the valves were added.

Model of compression and valves flow

Have been used simple compression model described by

\[
\frac{dP_c}{dt} = \frac{k\ P_c}{V} \left( Q_s - Q_t - \frac{dV}{dt} \right)
\]

where \(V = Vsk \cdot 2(1 + \cos \omega \ t) + Vm\)

\[
\frac{dV}{dt} = \frac{Vsk}{2} \omega \sin(\omega t)
\]

\(Q_s = \alpha Ls \cdot h(t) \frac{2}{(Ps-Pc)^2} \cdot (Ps-Pc) / (Ps-Pc)\)

for \((Ps-Pc) \neq 0\)

\[
Qt = \alpha Lt \cdot h(t) \frac{2}{(Ps-Pc)^2} \cdot (Ps-Pc) / (Ps-Pc) \]

for \((Ps-Pc) = 0\)

Values \(\alpha Ls \cdot h(t)\) and \(\alpha Lt \cdot h(t)\) represents an effective area of flow dependent on time, port position, deflection of reed.

Solution

Solution of system of equation (1) and (2) can be obtained by different ways (f.e. by numerical integration by R-K or Wilson-theta method \([4,6]\)).

In our case (for small matrix) the method based on finite element in time sense to be more effective \([7]\).

If we consider first the set of equation (1) we can assume that the initial condition \(\{\delta\}^0\) and \(\{\dot{\delta}\}\) are known. We describe \(\{\delta\}\) in time period \(\Delta t\) as

\[
\{\delta\} = \begin{bmatrix} H_{00} & H_{01} & H_{02} & H_{03} \\ H_{10} & H_{11} & H_{12} & H_{13} \\ H_{20} & H_{21} & H_{22} & H_{23} \\ H_{30} & H_{31} & H_{32} & H_{33} \end{bmatrix} \begin{bmatrix} \{\delta\}^0 \\ \{\dot{\delta}\}^0 \\ \{\ddot{\delta}\}^0 \\ \{\dot{\delta}\}_1 \end{bmatrix}
\]

As a shape function we take hermitian Polynomials

\[
H_{00} = 1 - 3s^2 + 2s^3
\]
\[
H_{40} = s - 2s^2 + s^3
\]
\[
H_{04} = 3s^2 - 2s
\]
\[
H_{44} = -s + s^2
\]

Writing residual equation for \(t=\Delta t\) we obtain the requeential equation

\[
\begin{bmatrix} H_{00} & H_{10} & H_{20} & H_{30} \\ H_{01} & H_{11} & H_{21} & H_{31} \\ H_{02} & H_{12} & H_{22} & H_{32} \\ H_{03} & H_{13} & H_{23} & H_{33} \end{bmatrix} \begin{bmatrix} \{\delta\} \\ \{\dot{\delta}\} \\ \{\ddot{\delta}\} \\ \{\dot{\delta}\}_1 \end{bmatrix} \begin{bmatrix} \{\delta\}^0 \\ \{\dot{\delta}\}^0 \\ \{\ddot{\delta}\}^0 \\ \{\dot{\delta}\}_1 \end{bmatrix} \begin{bmatrix} 1 \\ 1 \\ 1 \\ 1 \end{bmatrix} = 0
\]

After integration we have

\[
\begin{bmatrix} P_{14} & P_{24} & \frac{1}{2} \{\delta\}_4 & \frac{1}{2} \{\dot{\delta}\}_4 \\ P_{15} & P_{25} & \frac{1}{2} \{\delta\}_5 & \frac{1}{2} \{\dot{\delta}\}_5 \end{bmatrix} \begin{bmatrix} E_{14} & E_{24} & \{\delta\}_4 \\ E_{15} & E_{25} & \{\dot{\delta}\}_5 \end{bmatrix} \begin{bmatrix} C_4 \\ C_5 \end{bmatrix}
\]

or

\[
\{u_1\} = - [E] \{u_0\} - [c]
\]
And finally after left side multiplication by \([A]^{-1}\) we obtain
\[
(3) \begin{bmatrix} u_1 \end{bmatrix} = \begin{bmatrix} A \end{bmatrix}^{-1} \left( -\begin{bmatrix} B \end{bmatrix} \begin{bmatrix} u_0 \end{bmatrix} - \begin{bmatrix} c \end{bmatrix} \right)
\]
the set of linear equation describing \([\bar{d}]_t\) and \([\bar{e}]_t \end{bmatrix} \begin{bmatrix} \bar{f} \end{bmatrix}
Matrices \(A, B, C\) have a form \([9]\)
\[
\begin{align*}
A_{44} &= -504/\Delta t [\bar{c}] + 216 [\bar{f}] + 156\Delta t [k] \\
A_{22} &= 56 [\bar{d}] - 4\Delta t [\bar{c}] \\
E_{44} &= 504/\Delta t [\bar{c}] - 216 [\bar{f}] + 54\Delta t [k] \\
E_{22} &= 54/\Delta t [\bar{d}] - 22\Delta t [\bar{c}] + 22\Delta t [k] \\
R_{24} &= -21\Delta t [\bar{f}]_0 - 63\Delta t [\bar{f}]_0 \\
C_{2} &= -21\Delta t [\bar{f}]_0 - 14\Delta t [\bar{f}]_0
\end{align*}
\]
with assumption that \( [\bar{f}]_e = [\bar{f}]_0 + ([\bar{f}] - [\bar{f}]_0) t/\Delta t \)

Described method was tested by comparison of obtain first natural frequency of beam with analytical solution. The agreement was satisfying.

Solution of (1) and (2) in our case was obtain in following way.
In every time step \(\Delta t\) equation (2) was independently integrated numerically basing on the known values of \(\bar{f}, \bar{f}_{\Delta t}\). So as a result pressure in cylinder at the end of period \(\Delta t\) was calculated. Now load of reed at the beginning and at the end of period \(\Delta t\) can be predicted and from equation (3) new values of \(\bar{f}, \bar{f}_{\Delta t}\) obtained.

Addition of boundaries
When a reed node violated the limits of displacement imposed by the valve seat or stop the node velocity were declared to be the negative product of this velocity and restitution coefficient (equal .3 ) and the displacement of this node was declared equal to displacement limit.

Results
Presented method has been used for study of dynamic behaviour of an air compressor valves.

Fig 1 shows the scheme of discharge and suction valve geometry.
For reference purposes all obtained results were presented in similar form. Each figure contains information about the pressure differences (PC-P1) or (PC-PA), displacements of different crossovers of reed, time scale (in ms), cutting force and moment near the clamping, symbolic cylinder volume. Also the deflection lines of reed in considered period of time are included.

Basic information about compressor are:
- discharge pressure 9-10 \(5\) kPa
- volume 107 cm\(^3\) clearance volume 6.78 cm
- nominal compressor speed 2800 rev/min.

The discharge valve reed geometry (length l=0.062 m, width \(s= 0.015 \) m and thickness \(g= 0.0004 \) m) was taken as a standard and the influence of different changes were observed.

Fig 2 shows the results obtained for standard configuration. In this case the assumption that the reed is perfectly clamped has been taken.

One can notice reed oscillation and strong bending of reed near the clamping. When the reed is thicker (\(g=0.0065 \) m) (Fig 3) valve is closing faster but force and moment near clamping are higher.

Of course the real clamping is not perfect. Such situation was simulated by increase of reed length (6% increase).

Fig 4 shows results of computation. Value is closing to late and the reverse flow can be observed.

The change of the stop geometry (now stop is oblique) significantly reduce forces and moments. Valve is closing faster but also the reverse flow appears.

The influence of pulsation of the discharge pressure was also considered.

Fig 5 and Fig 6 shows two examples of pulsating discharge pressure. Amplitudes of pulsation (0.1 Pc) and frequency (300 Hz - corresponding to 6-7 harmonic) are the same but the phases are different (\(\pi/2\) and \(\pi/4\)).

The reed motion is slightly different, the same is with time of closing and maximum values of forces and moments.

Considering the operation of the suction valve an acoustic model of noise emission similar to model presented in [1] has been used.

In following figures the typical examples of suction valve configuration are presented.
Fig 8 shows results of calculation of the suction valve without any stop. Fig 9 presents corresponding spectrum of emitted noise (in far field - 2m from the compressor). Very strong reed deflection can be observed.

The use of stop on the reed tip (1mm tip movement limit) increase the volumetric efficiency about 1% in comparison with previous configuration but emitted noise is a little higher (1 dB). Fig 10 and Fig 11 present obtained results.

Reduction of the reed tip stop high to 0.1 mm results in reduction of volumetric efficiency and increase of noise (6 dB higher) Fig 12 and Fig 13.

Conclusions:

Use of presented method has given some interesting information about the simple geometry valve operation. Reed stop with constant high causes a strong bending of reed and reed oscillation. Also caused pressure pulsation in compressor cylinder can negatively influence the dynamic load of crank mechanism.

The oblique reed stop seems to be much better solution. The use of oblique stop reduce forces and moments, reduces time closing and impact velocity, and there are no reed oscillation.

Not perfect clamping can significantly change operation of valve so this problem should be taken into account.

The discharge planum pressure pulsation can change the valve operation but it seems that the phase of pulsation is more important than the amplitude of pulsation.

The parameters of the suction valve have great influence on the compressor's volumetric efficiency and emitted noise. Increase of efficiency is always connected with increase of noise. The irregularities in reed motion reduce emitted noise. Reed with stop, changes stiffness during motion, so it seems that by careful investigation of relation between reed stiffness and stop position the optimal solution can be found.

References:


Fig 1 Scheme of valves geometry
Fig 2 Dynamic of standard geometry valve

Fig 3 Dynamic of the thicker valve reed

Fig 4 Results of not perfect clamping simulation
Fig 5 Dynamic of valve with oblique stop

Fig 6 Dynamic of valve with discharge pressure pulsation (Phase=π/2)

Fig 7 Dynamic of valve with discharge pressure pulsation (Phase=π)
Figure 8 Dynamic of suction valve without stop

Figure 9 Far field emitted sound spectrum

Figure 10 Dynamic of suction valve with stop (1 mm high)

Figure 11 Emitted sound spectrum

Figure 12 Dynamic of suction valve with stop (0.1 mm high)