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COMPRESSION MODEL FOR OPEN RECIPROCATING COMPRESSOR.
APPLICATION TO CYLINDER WALL COOLING STUDY.

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ABSTRACT

A compression model for open reciprocating compressor is elaborated. Relevant literature has been analyzed, assumptions and equations related to gas flow through valves, characteristic of valves, choice of gas-wall heat transfer correlation are given. A numerical resolution is performed and results are compared to experimental measurement. Intrusive instrumentation has been set in the compressor and a pressure sensor has been introduced in one cylinder. The model running deals with wall cooling during compression.

NOMENCLATURE

c_v [J.kg⁻¹.K⁻¹] specific heat of fluid at constant volume
C_am [N.m⁻¹.s] damping coefficient
C_p [m] cylinder bore
h [J.kg⁻¹] specific enthalpy
h_conv [W.m⁻².K⁻¹] heat transfer coefficient
k [N.m⁻¹] spring stiffness
L [m] connecting rod length
m [kg] mass
P [Pa] pressure
Q [J] heat
R_c [m] crank radius
r [J.kg⁻¹.K⁻¹] gas constant
S [m²] area
T [K] temperature
t [s] time
V [m³] volume
V_cl [m³] clearance volume
V [m³.mol⁻¹] molar volume
v [m³.kg⁻¹] specific volume
x [m] valve lift
x_o [m] uncharged valve lift
ω [rad.s⁻¹] angular speed
y entropic exponent
Re REYNOLDS number
Nu NUSSELT number

Indices : w wall up upstream suc suction
dam damping dw downstream val valve

INTRODUCTION

Open reciprocating compressors are widely used in refrigeration and especially in food industry where most of systems are run with ammonia. Heat pump applications are possible since a range of compressors which accept a design working pressure of 4 MPa has been developed. This compressor has been studied and tested ([4], [8]).

Numerical models with various degrees of complexity elaborated ([7],[9],[12],[13]). This paper presents a simple model which describes the compression chamber and valve motion. This model permits studying the heat transfer
impact on energy performance during compression. It can be extended to the study of cylinder wall cooling. Because of this extension, some values hardly available by experimentation can be calculated and compared with other data ([2], 3),([10],[11]).

The system

The study deals with the compression chamber of an open reciprocating compressor. For a complete cycle, four stages of simulation are considered: compression and expansion work with closed valves and, suction and discharge as soon as the respective valves start to open. A complete cycle is from expansion to discharge (from 0 to 360° for crank angle).

The assumptions

Assumptions are as follows:
• the rotational speed of the compressor is constant,
• the flow is one-dimension,
• the potential and kinetic energies are neglected,
• the conduction heat flux in flow direction is neglected,
• system is uniform and gas is perfect,
• lubricating oil effects are neglected.
Concerning the mass flow through valves, the upstream conditions are the stagnation conditions, and the gas flow is assumed to be adiabatic so the fluid dynamic equations can be used.

SYSTEM EQUATIONS

> Mass flow equation : \( dm = \sum j dm_j \) (1)

> Mass flow rate through valves

Generally the gas flow through a valve is assumed to be a flow through an orifice for a compressible gas.

For a perfect gas :

\[ \frac{dm}{dt} = P_{up} \cdot S_{val} \cdot \sqrt{\frac{2 \gamma}{(\gamma-1) \cdot r \cdot T_{up}}} \cdot \sqrt{\frac{2}{\tau^2 - \tau^\gamma}} \]

where \( \tau \) is the ratio between downstream and upstream pressures.

This equation is adapted for a subsonic flow [1].

> Energy conservation equation : \( mc_v dT = -mT \left( \frac{\partial P}{\partial T} \right)_v dv + \delta Q + \sum_j h_j dm_j - hdm \)

(3)

For example at suction port \( mc_v d\dot{T} = -Pd\dot{V} + \dot{Q} + (h_{suc} - h) d\dot{h} + rTd\dot{m} \)

(4)

> Convective heat transfer equation

A bibliographic study has been performed [4] for correlation choice, we use ANNAND’S correlation.

\[ \dot{Q} = h_{conv} \cdot S \cdot (T_{wall} - T_{fluid}) \quad \text{where} \quad Nu = 0.7 \cdot Re^{0.7} \]

The flow speed is chosen equal to the mean piston absolute speed for the REYNOLDS number calculation.

> Valve motion equation

\[ m_{val} \cdot \ddot{x} + C_{dam} \cdot \dot{x} + k \cdot (x + x_0) = C_g \cdot S_{val} \cdot (P_{up} - P_{gw}) \]

(6)
Cylinder volume equation

Cylinder volume variation depends on crank angle \( \omega t \):

\[
V(t) = V_0 + \frac{\pi D^2}{4} R_c \left[ 1 + \cos \omega t + \frac{L}{R_c} \left( 1 - \sqrt{1 - \frac{R_c^2}{L^2} \sin^2 \omega t} \right) \right]
\]

EXPERIMENTAL PROCESS

The compressor has eight cylinders and the swept volume is 0.054 m\(^3\)/s. The loop sensors allow to perform tests with respect to ISO917 standard (refrigerating compressor testing standard). The mass flow rate is obtained by applying the standard methods. A piezoelectric sensor has been introduced in one cylinder and measurements are compared to results obtained with the model. A crankshaft position sensor is also added.

RESULTS

Coefficients used in equation (6) are in accordance with the real refrigerating compressor which performance has been tested [4]. Experimental data allow fitting of coefficients for spring stiffness and damping coefficient.

Figure 1: Motion and Speed of Suction and Discharge Valves.

Figure 1 presents the opening and the speed of the suction and discharge valves. The valve oscillations and collisions between the plate and the valve stop or seat can be seen.

The model mass flow rate is 1.4% higher than experimental mass flow rate. Experimental input power is 6.5% higher than the power calculated by the model. Discharge temperatures differ from 2 K.
When comparing experimental measurement to model results (figure 2), the model shows better stability for the discharge valve. Since gas behavior has not been simulated in the discharge chamber, this could explain the lack of pressure pulsations in the model results.

On the other hand, the slope of the model compression curve is higher: experiment shows higher pressure at the end of the suction phase, and heat exchanges occurring during real operation (gas-wall heat transfer) are probably more significant.

Figure 2: Comparison Between Model and Experiment Watt Diagrams.

Gas leakage's (neglected in the model) can happened at the closed valves or at piston-cylinder clearance; also the pressure is lower.

The comparison between diagram areas obtained from the model and from experiment shows a difference of 12%. This results is in agreement with the variation of the input power given previously.

Study of the heat transfer correlation

<table>
<thead>
<tr>
<th>$h_{conv}$</th>
<th>Input power (%)</th>
<th>Heat transfer (kW)</th>
<th>Discharge temperature (°C)</th>
<th>Mass flow rate (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$2 \cdot h_{conv}$</td>
<td>+ 0.1</td>
<td>-2.2</td>
<td>128.2</td>
<td>- 1.0</td>
</tr>
<tr>
<td>$h_{conv}$</td>
<td>- 0.1</td>
<td>-0.6</td>
<td>128.2</td>
<td>+ 0.7</td>
</tr>
</tbody>
</table>

Table 1: Convective Heat Transfer Coefficient Variation. Results are given in relation to calculation with $h_{conv}$

Table 1 presents some results obtained from the model. These results can be related to Figure 3 showing the gas-wall heat transfer during a complete cycle.

Heat transfer during suction phase (this stage is equal to half a cycle) is greater than during other stages and particularly when compared to discharge.

This preponderance of heat transfers during suction has a direct effect on heat transfer during a complete cycle.

Figure 3: Instantaneous Heat Transfer Between Gas and Wall During a Complete Cycle.

Results obtained by the correlation fitting $2 \cdot h_{conv}$ are more in the same range as experimental measurements than results obtained with the reference working test $h_{conv}$, especially the mass flow rate. Moreover, this remark would come to an agreement with other works in the literature [10] performed on heat transfers during compression.
As literature review on convective heat transfers between gas and wall supposed, the correlations chosen for "simple" models underestimate these transfers. The correlation has been fitted and model results are in the same range as experimental tests.

**COMPRESSION COOLING STUDY**

In most of the models, the wall temperature is usually chosen equal to the mean fluid temperature at the inlet and the outlet of the compression chamber. Table 2 presents various parameters as a function of the wall temperature evolution.

<table>
<thead>
<tr>
<th>Input power (%)</th>
<th>Heat transfer (kW)</th>
<th>Discharge temperature (°C)</th>
<th>Mass flow rate (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_{\text{moy}} - 20\text{K})</td>
<td>-0.6</td>
<td>-4.5</td>
<td>123.9</td>
</tr>
<tr>
<td>(T_{\text{moy}} - 10\text{K})</td>
<td>-0.3</td>
<td>-2.2</td>
<td>128.3</td>
</tr>
<tr>
<td>(T_{\text{moy}} + 10\text{K})</td>
<td>+0.1</td>
<td>+2.3</td>
<td>137.3</td>
</tr>
</tbody>
</table>

Table 2: Cylinder Wall Temperature Variation. Results in relation to simulation with \(T_{\text{moy}}\).

Figure 4 indicates that the lower the wall temperature, the smaller the heat transfer during suction (the superheating of the gas is low during suction in this case) and the greater it is during discharge. For the temperature called mean temperature \((T_{\text{moy}})\), the global heat transfers over a complete cycle is 0 kW. The combining of an endothermic and then an exothermic transfer during compression results in a nil global heat transfer for one cycle.

The cylinder wall cooling allows improvement of compressor energy performance: mass flow rate is increasing and input power and discharge temperature are decreasing. Cooling is more efficient when temperature discharge decreases than when obtained by cylinder-heads cooling during experiment.

On the other hand, the wall temperature variation shows that the choice of the wall temperature for simulations is important. However the heat transfer is low compared to the input power.

**CONCLUSIONS**

A compression model has been developed and its results have been compared to experimental measurements. This model has been applied to the evaluation of heat transfer during the four phases of the compression cycle. Comparison between experimental and simulation results shows that usual calculations underestimate heat transfers.

The model indicates that cooling of the cylinder wall implies improvement of the compressor energy efficiency. This improvement is due to the increase of the mass flow rate and to the decrease of the input power.
REFERENCES


