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ANALYSIS OF THE WORKING CYCLE OF SINGLE-STAGE REFRIGERATION COMPRESSORS USING DIGITAL COMPUTERS

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INTRODUCTION

After 1950 Costagliola [1] had reported first about the basic relations to calculate the dynamic working cycle of a reciprocating compressor in the Sixties because of the increasing application of digital computers, complex calculation models for compressor simulation have been developed, including the important dynamic valve behaviour of the compressor valves (for example: [2] [3]). In order to gain a better description of the whole compressor system, the basic models have been extended by integrating certain sub-systems in the last years. Till today the most important extension consists in incorporating the instationary gas pulsations in the valve chambers and the connected pipes of the compressor (for example [4]). Recently investigations have been made, treating further problems as for example heat transfer in a compressor [5] and its insertion in the calculation model [6]. Despite regarding of all these additional aspects most of the investigators apply perfect gas laws. The general validity of this assumption has to be proved, especially if a realistic p, V - diagram is needed for other calculations concerning compressor design, as for example the predetermination of the sliding bearings [7] and the lubrication conditions at the piston [8]. Comparing estimations show that deviations between corresponding computations using either real or ideal gas laws reach values up to 10% and more. More exact statements about their influence on the calculation of the working cycle of the compressor can only be made by applying a generally valid compressor model, which has to be established in a structure allowing the use of either the ideal or different real gas equations of state for the working fluid.

This given situation was the reason to develop at the Technical University of Hannover a compressor model, which is able to use up by choice either the ideal gas equation or some different complicated real equations of state. Further the model was constructed in such a manner that the heat transfer in the cylinder and also the instationary gas pulsations in the valve chambers can be included.

The aim of the project was, to clear up special problems of compressor modeling in order to see, how much it is necessary to take into account the mentioned model extensions. For this purpose the influence of several parameters on the compressor working cycle and its valuation factors as for instance the adiabatic and volumetric efficiency, the specific working power etc. have to be stated by means of variations. It is necessary to study the possibility of simplifications. Especially this question is very important concerning a practical economic application of the model for further development of compressors. Combined with these basic questions there is coupled a great complex of many individual problems which can only be solved by a direct application of the extended model. Some important questions shall be answered in the following chapters by a short description of the structure and the application of the mathematical model.

PROBLEMS IN MODEL DEVELOPING

The problem of model developing involves the general mathematical description of the working mechanism of a reciprocating compressor. In order to use different equations of state it is necessary to develop a mathematical description which is independent from the working fluid itself and its special form of equation of state. In order to avoid unnecessary iterations, only has to be taken into account, which value is the dependent variable of the gas equation. Because most of the complicated equations of state express the pressure p in terms of temperature T and specific volume v, p = f(T, v), it was demanded here to use this type of equation with the secondary condition to reduce time-consuming iterations.

In doing this, it is important to take care, that the describing differential equations are constructed in such a way that by numerical integration the independent variable T has suitable to be computed. The conservation theorem of energy applied to the compressor system (Fig.1) yields the following equation (see Nomenclature)
The intention to compute a thermodynamic state of the working fluid leads to the state equation of the corresponding gas process and the definition of the thermodynamic values. For this purpose different heat transfer coefficient correlations were used, as for instance the formulae of Nusselt [10], Prisman [11], Pflaum [12], Woschni [13] ... in order to estimate the possible spectrum of the heat transfer influence on the working cycle. The principal interplay of the different variables influencing the heat transfer is shown in Fig. 2, qualitatively. The quantitative heat transfer effect on the compressor process is very small because of the small differences of temperatures between gas and walls in a refrigerating compressor. Variations of the volumetric efficiencies compared with corresponding adiabatic process calculations could not be stated and the p, V diagrams did not show significant differences — also no significant influences on the whole compressor process could be observed by including the computation of pressure pulsation by means of the acoustic theory in the model when using realistic sizes for the dimensions of the valve chambers and the diameters of the connected pipes. The reason for this result is the assumption of anechoic termination of the lines, which has been made because they end in either the evaporator or the condenser where a phase-change of the refrigerant occurs.

When comparing the result of real gas simulation with ideal gas simulation important differences concerning the thermodynamic values could be stated. While only the plots of cylinder pressure, computed with both equations of state, show relative small deviations, there results more significant differences by regarding the trace of curves of temperature, density and mass of gas in the cylinder. Fig. 2 shows for example the plots of temperature either computed by ideal and real equation of state. Such differences influence the volumetric efficiency which is given for example in Fig. 3 as function of the volume-element with a connected anechoic pipe.

APPLICATION OF THE MODEL

The first aim in applying the model was, to study the influence of the heat transfer in the cylinder and the gas pulsations in the valve chambers on the whole working cycle. For this purpose different heat transfer coefficient correlations were used, as for instance the formulae of Nusselt [10], Eichelberg [11], Pflaum [12], Woschni [13] ... in order to estimate the possible spectrum of the heat transfer influence on the working cycle. The principal interplay of the different variables influencing the heat transfer is shown in Fig. 2, qualitatively. The quantitative heat transfer effect on the compressor process is very small because of the small differences of temperatures between gas and walls in a refrigerating compressor. Variations of the volumetric efficiencies compared with corresponding adiabatic process calculations could not be stated and the p, V diagrams did not show significant differences — also no significant influences on the whole compressor process could be observed by including the computation of pressure pulsation by means of the acoustic theory in the model when using realistic sizes for the dimensions of the valve chambers and the diameters of the connected pipes. The reason for this result is the assumption of anechoic termination of the lines, which has been made because they end in either the evaporator or the condenser where a phase-change of the refrigerant occurs.

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of the initial spring force. Parallel simula-
tion by using the ideal gas equation and a real
gas equation by varying the description para-
eters of the valves represents a possibility
to find out whether a model with real gas equa-
tions is necessary or not for valve optimization.
By plotting the valuation variable, chosen for
quality criterion, as function of the variation
parameter, there is only interest to know the
location of the extrem value and not its magni-
tude. The valve parameter have such a great
influence on the closed compressor – working
cycle, that the applied type of equation of
state is of secondary importance. That's why
for valve optimization the more simple and less
computer-time consuming ideal gas – calculation
is sufficient. For this, Fig. 4 illustrates
that the time depended valve displacements show
- except a certain delay of phase - no signi-
ficant deviations.

The analysis of the working cycle of recipro-
cating compressors is especially then very
clear and evident, if the results can be given
in a graphical manner. Studying for example the
effect of compressor speed variations the
results can be plasticly demonstrated in
a three dimensional p, V-diagram, plotted
with the speed as third dimension (Fig.5). Con-
templating this diagram great pulsations of
the cylinder pressure can be registred in the
lower speed range. These pulsations are based
on the behaviour of the dynamic self-acting
compressor valves which are not designed for
this range of compressor speed. The alterna-
ting intersection line of reexpansion-curves
shows obviously, that the volumetric efficiency
will alternate too in this speed range because of
the corresponding shortening of the suction
line. A continuous decrease occurs finally
with higher speeds. This result can be seen in
Fig.6 in which the graph for the volumetric
efficiency is determined by a greater number of
calculated points. Especially the volumetric
efficiencies have a close correlation to the
valve forces and displacements which may effect
remarkable backflow through the valves at
special speeds by an unfavorable interplay of
the valve parameters. The valve displacement
can be demonstrated clearly in Fig.7:
In the lower speed range the valves never reach
the resting position of the full opened valve
and the occurring valve flutter causes the men-
tioned pressure pulsations in the cylinder.
Normal valve displacement diagrams can only be
stated at higher speeds. By plotting in an
analog manner the relative pressure difference,
defined as the relation of the absolute pressure
difference to the actual stationary pressure at
the valve, pressure plots as in Fig.8 can be
gained: Those points where the pressure curves
are cutting the zero-plane the backflow, caused
by a negative pressure difference, begins and
the backflow ends when the valve has closed
finally. This is given by the projection of the
final points of the pressure-curves on the zero-
plane. In contemplating this diagram it can be
seen, that, when increasing the speed, the time
of the not closed valves increases too. At
a medium speed a "cut" can be seen where
practically no backflow occurs. Up to
higher speeds a continuous increase of the
time of backflow can be stated again. Besides
of the volumetric efficiency in Fig.6 further
valuating factors of the working cycle of a
refrigerating compressor are plotted as a
function of the speed. It can be seen that
the average temperature behind the discharge
valve increases with higher speeds, leading to
a lower adiabatic efficiency. The cycle
work, given by the included area of the corre-
sponding p, V-diagrams shows a maximum value
at a medium speed which results from the
superposition of two influences:

1.) When increasing the speed work,losses
are increasing too and with them the
needed cycle work.

2.) Because of the increasing pressure drops,
combined with increasing period of back-
flow, the periods of compression and
re-expansion are starting with time delay
which effects a smaller included work
area.

At higher speeds the secondary influence is
more important leading to a decrease of the
cycle work. But with regard to the specific
energy per unit of mass it can be seen clearly,
that the smallest specific power-consumption
occurs at lowest speed.

CONCLUSION

A complex calculation model for refrigerating
compressors had been presented. By applying
this it was shown, that heat transfer and
instationary pressure pulsations with the
chosen acoustic systems have only unimportant
influences on the shape of the whole working
process. The most important parts of the
system are the dynamically working compressor
valves, which influence the working cycle in
such a great amount, that in order to optimize
the valves it is allowable with good accuracy
to use the simpler computation model with the
ideal gas equation.

For the determination of the valuation factors
of the working cycle as for instance the volu-
metric efficiency and the specific compression
work the real gas behaviour should not be
neglected. In order to solve special problems
for compressor design, therefore, a using
of simulation models including the real gas
behaviour is favourable.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>AD&lt;sub&gt;eff&lt;/sub&gt;</td>
<td>Effective force area (m&lt;sup&gt;2&lt;/sup&gt;)</td>
</tr>
<tr>
<td>AQ&lt;sub&gt;eff&lt;/sub&gt;</td>
<td>Effective flow area (m&lt;sup&gt;2&lt;/sup&gt;)</td>
</tr>
<tr>
<td>A&lt;sub&gt;W&lt;/sub&gt;</td>
<td>Area for heat transfer (m&lt;sup&gt;2&lt;/sup&gt;)</td>
</tr>
<tr>
<td>C&lt;sub&gt;D&lt;/sub&gt;</td>
<td>Damping coefficient (N·s·m&lt;sup&gt;-1&lt;/sup&gt;)</td>
</tr>
<tr>
<td>C&lt;sub&gt;1&lt;/sub&gt;</td>
<td>Constant value (-)</td>
</tr>
<tr>
<td>F</td>
<td>Spring force (N)</td>
</tr>
</tbody>
</table>
h \text{ Enthalpy (N·m·kg}^{-1}\text{)}

H_v \text{ Valve lift (m)}

m \text{ Mass (kg)}

m \text{ Exponent (-)}

m_v \text{ Valve mass (kg)}

n \text{ Exponent (-)}

Nu \text{ Nusselt number (-)}

p \text{ Pressure (N·m}^{-2}\text{)}

Pr \text{ Prandtl number (-)}

Q_w \text{ Heat energy (N·m)}

Re \text{ Reynolds number (-)}

s \text{ Entropy (N·m·kg}^{-1}\text{K}^{-1}\text{)}

T \text{ Temperature (K)}

T_w \text{ Temperature of walls (K)}

u \text{ Specific internal energy (N·m·kg}^{-1}\text{)}

v \text{ Specific volume (m}^{3}\text{·kg}^{-1}\text{)}

V \text{ Cylinder volume (m}^{3}\text{)}

y \text{ Relative valve displacement (-)}

\alpha \text{ Heat transfer coefficient (N·m}^{-1}\text{·s}^{-1}\text{·K}^{-1}\text{)}

\varphi \text{ Crank angle (rad)}

\omega \text{ Crank speed (s}^{-1}\text{)}

Indices

EIN \text{ Mass flow into the cylinder}

AUS \text{ Mass flow out of the cylinder}

1 \text{ Upstream condition}

2 \text{ Downstream condition}

REFERENCES


WANDWAERME-DIAGRAMM PVD-NR. 35/2/2

Fig. 1

TEMPERATURVERLAUF ÜBER DEN KURBELWINKEL (VERGLEICH: IDEALES-IDEALES GAS)

Kältemittel: R13
tc = 0 °C
t0 = -40 °C
Δt0h = 5 °C

Fig. 2
Fig. 3

Fig. 4
$p, V = f(n)$

Fig. 5
Kreisprozeß – Bewertungsgrößen bei Änderung der Kompressordrehzahl

Kondensationstemperatur $t_c = 30^\circ\text{C}$  
Verdampfungstemperatur $t_v = -20^\circ\text{C}$  
Ansaugüberhitzung: $\Delta t_0 = 5^\circ\text{C}$

Fig. 6

Drosselventil

Relative Druckdifferenz – Drosselventil

Verlauf der relativen Druckdifferenz bei Änderung der Drehzahl

Saugventil

Relative Druckdifferenz – Saugventil

Fig. 7

Fig. 8