Compressor Modeling Technique for Realistic and Broad Range Simulation

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COMPRESSOR MODELING TECHNIQUE FOR REALISTIC AND
BROAD RANGE SIMULATION

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ABSTRACT
The realistic simulation of refrigerant compressors for heat pump applications is difficult because the needed physical parameters like clearance volume, blow-by, valve losses, polytropic coefficient(s), ohmic and core losses, jacket losses, etc. are not readily available.

We have developed a simple yet very effective approach to this problem for our HFROST heat pump simulation program, which only requires a reasonably low amount of computer time. It consists of a mathematical model of the physical performance of a compressor, including the above unknown parameters, which is tailored to a particular, real compressor by using measured data points as published by standard compressor performance maps and determining the unknown parameters by solving the corresponding system of linear and non-linear equations.

The achieved successful simulations cover a wide range of evaporating/condensing temperatures, and have been used extensively in heat pump simulations. Major difficulties encountered in this work involved: 1. Experimental errors in available compressor data, 2. Simulation of compressor jacket losses (up to 60% of input power) and 3. The feasibility to derive one set of compressor parameters that can be used for heating and cooling operation.

INTRODUCTION
Heat pump and refrigerant cycle simulation capability is one of many process simulation capabilities which have been developed at Honeywell in order to provide a good understanding of the process under both normal and abnormal operating conditions, to define control strategies and set points, and identify implications of equipment design, load, weather, controls, etc. on the system performance and energy consumption. In order to answer the broad range of questions mentioned above, we have built our simulation models on the basis of the physical principles of the involved processes (vapor compression in this case), and augmenting these with empirical relationships as needed.

We designed the model to be simple, yet predictive, using input parameters that can be derived experimentally and providing a simulation output that can be validated by comparison with measured data.

Manufacturer's compressor data typically consist of plots of input power, heating/cooling capacity, and/or mass flow of the refrigerants such as R22, measured under steady state conditions for various evaporating and condensing temperatures. These data may contain some experimental and extrapolation errors, since one compressor may be used to generate the basic data and to extrapolate to other models, which results in the provided smooth performance curves.

Data obtained for four commercially available compressors are discussed here, for which we were able to achieve good agreement between their experimental performance and simulated curves.

DISCUSSION
Compressor Model Assumptions
The compressor model is based on an ideal Rankine cycle and comprises the following assumptions:

1. The equation of state of R22 gas in the pressure and temperature range of interest was derived from Dupont's data(1) into the form
\[
v(P, T) = f_1(P) + RT/f_2(P) \quad (1)
\]
where \(f_1\) and \(f_2\) are functions of pressure. See Table 1 for all other nomenclature.

2. The integrated effect of the deviation from isentropic compression work, \(PV^0\), can be accounted for by the introduction of a polytropic coefficient, \(K_1\):

\[
PK^0K_1 = PVK = \text{constant} \quad (2)
\]
to represent the real gas behavior during compression, where \(K_0\) and \(K_1\) are constant for a given compressor. For FREON 22 \(K_0 = c_v/c_p = 1.18\) at 60°F, and atmospheric pressure.

3. The clearance volume, \(V_0\), is expressed as a fraction of the displacement volume \(V_1\). Its effect reduces the mass flow by a factor \(V_0(P_2/P_3)^{1/2}\).

4. Blowby, \(B_0\), expressed as a fraction of \(V_0\); reduces the mass flow by a factor \(V_0(1 - P_2/P_3)^{1/2}\) of that reduced by the clearance volume.

5. The actual values for mass flow, \(M_0\), and input energy, \(W\), can be represented by a sum of the ideal value and a term that depends on the total energy loss, \(L\). This is tied to the further assumption that \(L\) causes the suction gas temperature to increase from \(T_7\) to \(T_3\); and that \(M_0\) and \(W\) can be specified as functions of \(P_2\), \(P_3\) and suction gas temperature:

\[
M_0(P_3, P_2, T_7) - M_0(P_3, P_2, T_9) = A(P_3, P_2, T_7)XL \quad (3)
\]

\[
W(P_3, P_2, T_9) - W(P_3, P_2, T_7) = L \quad (4)
\]
where \(A(P_3, P_2, T_7)\) is a function that converts \(L\) to the mass flow increment.

6. In going through the restrictor the liquid refrigerant undergoes ISENTHALPIC expansion.

Compressor Parameter Estimation

Jacket Losses. Energy conservation requires that all energy inputs into the compressor also exits from it. From the known input and output refrigerant enthalpy flow and compressor input electric power, we derived the jacket losses by difference. They are plotted for compressor C in Fig. 1. As shown, they range from about 21 to 41% of the compressor input power and generally are highest at conditions of low evaporating temperatures and suction gas mass flows. Since we lacked data on jacket temperatures, we fitted the jacket loss data to a heuristic function of the evaporating and condensing temperatures. The comparison with

\[
L_J = a_0 + a_1 T_3 + a_2 T_3^2 \quad (5)
\]
is shown in Fig. 2 for several compressors.
(A,B,C,E) and one heat pump under test. The fit only improved marginally after incorporating also $T_2$ to form

$$L_{ij} = a_0 + a_1 T_3 + a_2 T_3^2 + a_3 T_2 T_3 \quad (6)$$

The coefficients for $L_{ij}$ are presented in Table 2.

Displacement and Clearance Volumes, Blowby and Polytropic Coefficient

From the above equations (3) and (4), we derived expressions that were not dependent on $L$ and in general would involve functions of the type

$$F(M_0, W, V_h) = G(T_3, T_2, T_7, T_9, B_0, V_0, V_1, K) \quad (7)$$

We then found regression analysis solutions for $V_1$, $V_0$, and $B_0$ that would best fit many operating points of $M_0$ and $W$ with trial and error inputs of the polytropic coefficient, $K$. Values for $V_1$, $V_0$, $B_0$, and $K$, found for commercial compressors A, B, C and D ranging from 2 to 3.5 tons are listed in Table 3.

Internal Compressor Losses are due to the following:
- Non negligible DC resistance of the motor windings (ohmic losses).
- Hysteresis and magnetic resistance of the iron core of the motor (magnetic losses).
- Mechanical and fluidynamic friction (friction losses).

All were lumped into one category of power losses, $L$, which we computed on the basis of actual (experimental data) vs. adiabatic compression. Fig. 3 shows that the switch from heating to cooling mode operation of the compressor (higher and variable superheat and environmental temperature), while causing a discontinuity in the data, still provided a reasonable fit by one set computer generated parameters.

The error margin of the obtained results is determined by
- Uncertainty in the experimental data and in the used averaging techniques
- Systematic errors introduced by our solution approach and approximations.

Independent of the above, we also checked relations between available input power and current. The results for compressor C are shown in Fig. 4. They show that under some conditions the power factor was greater than unity, reflecting inconsistency in the experimental data.

Of the various relations we have tried to find between current and power, the simplest one
\[ I = A e^{B} \]  
(8)

where \( I \) = current in amps and A, B are constants that adopt different values for the two modes of operation of the heat pump gives an adequate description of the current.

We also made an attempt at quantifying the magnetic and friction losses by subtracting out the ohmic losses. Assuming a nominal resistance of 2 ohms for the compressor motor, the friction losses for the 3 ton compressor "C" are plotted in Fig. 5 against the evaporating temperature and gas density.

For computer simulation purposes we have adopted only a fit to the total losses in terms of the evaporating and condensing temperatures and four coefficients \( F_1 - F_4 \):

\[ L = F_1 + (F_2 + F_3 T_2 / 273.15) [(T_3/273.15)^2 + F_4] \]  
(9)

which were plotted in Fig. 3. As with the total power losses, we found the friction losses to increase with increasing power consumption, evaporating temperature but, surprisingly, with decreasing condensing temperature.

RESULTS

Compressor and Heat Pump Simulations

With the derived model, as described above, we made comparisons between experimental (manufacturer's data sheets) and HFROST-computed results.

Figs. 6 and 7 show the comparison for heating and cooling mode of operation of Compressor "C". The points represent the experimental data we used as inputs to derive the compressor parameters, and the curves were then generated with HFROST. As shown, the agreement is very satisfactory, and even extrapolation to lower evaporating temperatures indicates that the model is behaved.

We used this compressor model to simulate the energy performance of heat pumps. Fig. 8 shows one example of a successful comparison between experimental (small numbers) and HFROST-simulated (curves) over a range of outdoor temperatures.
CONCLUSIONS

The derived compressor model is well suited to simulate the performance of the compressors on the market today. We demonstrated its use by computing the performance of heat pumps with HFROST.

The model could be improved by improving the accuracy of the experimental input data. However, we found that the compressor's overall performance could be well described by only the following set of parameters:

- Displacement volume
- Clearance volume
- Blowby
- Polytropic coefficient

Empirical fits were used for total internal power losses and jacket losses.

These parameters could be derived from inputs of:

- Input Power
- Mass flow
- Capacity and temperatures for superheat, subcooling, condensation and evaporation.

The influence of suction and discharge valve operation, internal heat exchange from the discharge line, refrigerant-lubricant interaction, and return of liquid refrigerant were not included in this compressor model.

Jacket losses were found to correlate somewhat with evaporating and condensing temperatures. The level of these losses were higher than expected: While those shown here did not exceed 42% of the compressor input power at low evaporator temperatures, we found other cases with losses of over 60%.

The chosen modeling approach based on physical parameter estimation served us well when we extended this technique to simulate the operation of complete heat pumps, as demonstrated by the shown comparison between model and experimental results.

REFERENCES

Table 1: Compressor parameters and other symbols appearing in the compressor model.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>B_0</td>
<td>Blowby</td>
<td>fraction of V_0</td>
</tr>
<tr>
<td>F_1, F_2, F_3</td>
<td>Losses determining parameters</td>
<td>watts</td>
</tr>
<tr>
<td>F_4</td>
<td>Losses determining parameters</td>
<td>dimensionless</td>
</tr>
<tr>
<td>K_0</td>
<td>Ratio of specific heats</td>
<td>dimensionless</td>
</tr>
<tr>
<td>K_1</td>
<td>Polytropic coefficient</td>
<td>dimensionless</td>
</tr>
<tr>
<td>L</td>
<td>Total compressor losses</td>
<td>watts</td>
</tr>
<tr>
<td>M</td>
<td>Mass flow</td>
<td>mole/h</td>
</tr>
<tr>
<td>M_h</td>
<td>Mass flow if L were zero,</td>
<td>mole/h</td>
</tr>
<tr>
<td>P_2</td>
<td>Discharge pressure - Saturation vapor pressure at T_2</td>
<td>atm</td>
</tr>
<tr>
<td>P_3</td>
<td>Suction pressure - Saturation vapor pressure at T_3</td>
<td>atm</td>
</tr>
<tr>
<td>R</td>
<td>Universal gas constant</td>
<td>atm liter/(°K/mole)</td>
</tr>
<tr>
<td>T_2</td>
<td>Condensing Temperature</td>
<td>K</td>
</tr>
<tr>
<td>T_3</td>
<td>Evaporating Temperature</td>
<td>K</td>
</tr>
<tr>
<td>T_6</td>
<td>Discharge superheated gas temp.</td>
<td>K</td>
</tr>
<tr>
<td>T_7</td>
<td>Evaporator superheated gas temp.</td>
<td>K</td>
</tr>
<tr>
<td>T_8</td>
<td>Liquid subcool temp. in condensor</td>
<td>K</td>
</tr>
<tr>
<td>T_9</td>
<td>Average suction gas temperature</td>
<td>K</td>
</tr>
<tr>
<td>V_0</td>
<td>Clearance volume</td>
<td>fraction of V_1</td>
</tr>
<tr>
<td>V_1</td>
<td>Displacement volume</td>
<td>liter</td>
</tr>
<tr>
<td>v(P, T)</td>
<td>Specific volume of gas at pressure P and temp T</td>
<td>liter/Mol</td>
</tr>
<tr>
<td>W</td>
<td>Input power</td>
<td>watts</td>
</tr>
<tr>
<td>W_h</td>
<td>Input power if L were zero</td>
<td>watts</td>
</tr>
</tbody>
</table>

Subscripts
- 2: Condenser
- 3: Evaporator
- 6: Discharge

Table 2. Correlation coefficients for compressor jacket losses:

\[ L_j = L_j - 100 \times (a_0 + a_1 T_3 + a_2 T_3) \]

<table>
<thead>
<tr>
<th>Compressor</th>
<th>A. 2 tons</th>
<th>B. 2.5 tons</th>
<th>C. 3 tons</th>
<th>D. 3.5 tons</th>
</tr>
</thead>
<tbody>
<tr>
<td>a_0</td>
<td>0.5064</td>
<td>0.3501</td>
<td>0.4068</td>
<td>0.2377</td>
</tr>
<tr>
<td>a_1</td>
<td>-0.0102</td>
<td>-0.0108</td>
<td>-0.0110</td>
<td>-0.0107</td>
</tr>
<tr>
<td>a_2</td>
<td>0.9729x10^{-4}</td>
<td>0.1832x10^{-3}</td>
<td>0.2057x10^{-3}</td>
<td>0.2411x10^{-3}</td>
</tr>
</tbody>
</table>

Table 3. Values of the parameters for four analyzed compressors.

<table>
<thead>
<tr>
<th>Compressor</th>
<th>A. 2 tons</th>
<th>B. 2.5 tons</th>
<th>C. 3 tons</th>
<th>D. 3.5 tons</th>
</tr>
</thead>
<tbody>
<tr>
<td>V_1</td>
<td>0.026</td>
<td>0.031</td>
<td>0.038</td>
<td>0.052</td>
</tr>
<tr>
<td>V_0</td>
<td>0.083</td>
<td>0.102</td>
<td>0.092</td>
<td>0.103</td>
</tr>
<tr>
<td>B_0</td>
<td>0.948</td>
<td>0.094</td>
<td>0.000</td>
<td>0.342</td>
</tr>
<tr>
<td>K_1</td>
<td>1.194</td>
<td>1.130</td>
<td>1.073</td>
<td>1.046</td>
</tr>
</tbody>
</table>