Model of a Reversible Heat Pump for Part Load Energy Based Optimization Design

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SUMMARY

With the increase of air conditioning needs in commercial sectors, reversible heat pumps represent a smart solution for energy consumption. Design and sizing of reversible air conditioning / heating systems are often based on the full load performance, in order to respect thermal comfort at extreme conditions. However, the HVAC industry is switching to designs based on improved seasonal performance at part load.

In this context, this article represents a modeling of a 60 kW reversible heat pump. Detailed models of each component of the system have been developed: a plate heat exchanger (R410A/water), finned tube heat exchanger (R410A /air), two parallel scroll compressors, and an expansion valve. The model simulates the whole cycle after simulating each component separately. Results obtained are validated through tests done on a reference bench machine. Then simulations covering all the climatic conditions and regions of France for a whole year are achieved, helping to model a design of a new reversible heat pump energetically optimized at part load.

This paper will describe the main elements of the model, which has an objective to reduce the annual energy consumption of the reference commercial reversible heat pump by 50%.

INTRODUCTION

Heat pumps have got great interest during the last two decades, due to the increasing attention of the international community towards global environment and energy savings. Therefore, heat pumps are promising efficient solution for buildings air conditioning / heating compared to conventional devices, especially now when electricity production has reached efficiency about 60%.

The paper presents a model developed to assume the steady state characteristics of a commercial reversible air-to-water heat pump. In cooling mode, heat source is the ambient air and heat sink is the water cycle inside the building. Conversely, in heating mode, heat sink is the ambient air and heat source is the water cycle inside the building. Results of the simulations are validated by experimental measures in climatic chamber. Initial validation allows simulating several cases by changing the model parameters, and extrapolating their results to annual conditions in order to design an annually optimized solution. This is coherent with the tendency to introduce energy performance of buildings and HVAC systems not only based on rating conditions but also extended to whole annual building load. This means to calculate a seasonal energy efficiency ratio SEER and seasonal coefficient of performance SCOP.
MODEL DESCRIPTION

The structure of the global model is an image of the vapor compression refrigeration cycle. It is divided into modules representing the different cycle stages for different configurations.

![Diagram](image)

Figure 1: Layout of the vapor compression refrigeration system in heating and cooling modes

![Diagram](image)

Figure 2: R410A Vapor compression cycle on T-s diagram

Each module takes its inlet properties from the precedent module, calculates its outlet properties and transmits them to the next one. Details of the equations used are explained below.

The refrigerant used in this study is the R410A (mass fraction: 50% R32 and 50% R125). Calculation of the refrigerant, air, and water thermodynamic and transport properties is performed by correlations formulas written as computer code functions, using thermodynamic properties coming from REFPROP [1] data base. These functions are grouped in a particular module included in the overall source code.

Compressor model

A scroll compressor model is developed from the data given by the manufacturer and measurements carried out in climatic chamber on the whole heat pump. Performance curves are given using two equations fitting the compressor energy suction and the mass flow rate as a function of the evaporating temperature \( T_e \) and the condensing temperature \( T_c \),

\[
W_{\text{comp}} = C_0 + C_1 T_e + C_2 T_c + C_3 T_e T_c + C_4 T_e^2 + C_5 T_c^2 + C_6 T_e + C_7 T_c \cdot T_c + C_8 T_e^2 T_c + C_9 T_c^3
\]  

(1)
\[ m = M_0 + M_1 T_e + M_2 T_c + M_3 T_e T_c + M_4 T_e^2 + M_5 T_c^2 + M_6 T_e^3 + M_7 T_c^3 + M_8 T_e T_c^2 + M_9 T_c^3 \]  
\[ \text{where C}_1 - C_9 \text{ and } M_1 - M_9 \text{ are constant coefficients.} \]

Compressor calculating module inputs and outputs are:

**Inputs**
- Evaporating Temperature \( T_e [K] \)
- Condensing Temperature \( T_c [K] \)
- \( \eta_{global}, \eta_{vol} \)

**Outputs**
- \( W_{comp} [W] \)
- \( m_{comp} [kg/s] \)
- \( (T, h)_{discharge} \)

Compressor isentropic efficiency expresses the ratio between the isentropic and real compression work:

\[ \eta_s = \frac{W_{is}}{W_{real}} \]

where \( W_{is} = m(h_{2is} - h_1) \) is the real energy rate absorbed by the refrigerant during compression. Isentropic efficiency is needed to calculate the fluid properties (enthalpy, temperature...) at the compressor outlet. Compressor mechanical efficiency, assumed to be constant, relates real energy rate \( W_{real} \) transmitted to the refrigerant to power \( W_{comp} \)

\[ \eta_{mec} = \frac{W_{real}}{W_{comp}} \]

Compressor global efficiency expression

\[ \eta_{global} = \frac{W_{is}}{W_{comp}} = \eta_s \times \eta_{mec} \]

Compressor volumetric efficiency

\[ \eta_{vol} = \frac{V_{real}}{V_{th}} = \frac{v_{asp} \cdot m}{V_{th}} \]

where \( v_{asp} \) is fluid specific volume at the compressor aspiration \([m^3/kg]\), \( V_{th} \) is displacement volume given by manufacturer \([m^3]\).

**Expansion device model**

A thermostatic expansion valve (TXV) is considered in the model. TXV is used as rate flow control device, which feeds back the superheat \( \Delta T_{SH} \), and adjusts the mass flow to the evaporator in order to maintain a constant superheat preventing liquid refrigerant from entering the compressor.

Inputs are the inlet enthalpy and pressure, the outlet pressure and the superheat \( \Delta T_{SH} \). The expansion is assumed to be adiabatic procedure without heat exchange neither work drawn or produced, thus it is an isenthalpic expansion: \( h_2 = h_1 \).

Mass flow rate governing equation depends from the difference of pressure between the condenser and the evaporator \( \Delta P = P_c - P_e \), and from the superheat \( \Delta T_{SH} \):

\[ m_{TXV} = k \cdot \Delta T_{SH} \cdot \sqrt{\rho \cdot \Delta P} \]

where \( k \) is a coefficient specific to the expansion valve considered.
Heat exchangers models

The air-to-water heat pump studied in this article involves two heat exchangers (HEs): a plate heat exchanger (R410A/water), and a finned tube heat exchanger (R410A/air). These exchangers play the role of condensing or evaporating unit according to the case of cooling or heating mode.

Three types or levels of detail exist for modeling HE depending on the required accuracy on the results: global, zonal or finite elements. In this work, HR are modeled through zonal for every type of HE wherever its working mode: parallel flow, counter flow, cross flow, in a sequence of superheat – condensation – subcooling or evaporation – superheat. However, due to space limitations, only heating mode will be considered to illustrate the different heat exchange stages of the condenser and the evaporator and the calculation procedure.

Discrete elements considered calculate the outlet temperatures \( T_{c,o}, T_{h,o} \), starting by knowing their inlet temperatures \( T_{h,i}, T_{c,i} \). The calculation is based on LMTD approach:

\[
Q = F \cdot UA \cdot LMTD
\]  

(8)

where

- \( Q \) is the heat flux exchanged [W].
- \( UA \) is global heat transfer coefficient [W/K], got from the correlations applied on both sides of the heat exchange process.
- \( F \) coefficient for flow type.
- \( LMTD \) is the log mean temperature difference defined by:

for parallel flow HE:

\[
LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln \left( \frac{\Delta T_2}{\Delta T_1} \right)} = \frac{(T_{h,i} - T_{c,j}) - (T_{h,o} - T_{c,o})}{\ln \left( \frac{T_{h,i} - T_{c,j}}{T_{h,o} - T_{c,o}} \right)}
\]  

(9)

for counter flow HE:

\[
LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln \left( \frac{\Delta T_2}{\Delta T_1} \right)} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,j})}{\ln \left( \frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,j}} \right)}
\]  

(10)

Another expression for the heat flux exchanged between the two fluids:

\[
Q = C(c) (T_{c,o} - T_{c,i}) = C(h) (T_{h,i} - T_{h,o})
\]  

(11)

where \( C = c \cdot m_s \) and \( c \) is the specific heat [kJ/(kg.K)].

The procedure to solve the set of equations is as follows. An estimate of one of the outlet temperature \( T_{c,o} \) or \( T_{h,o} \) is taken as initial guess. Then the other outlet temperature and heat flux \( Q \) are found with equation (11). Next, a new value of \( Q \) is determined using equations (8). If the newly calculated heat flux value matches the previous one, the element computation is complete. Otherwise, \( Q \) becomes the new estimate of equation (11) and another computational sequence is performed.

Plate heat exchanger

Since the heat sink in heating mode is the water cycle inside the building, the plate heat exchanger PHE will play the role of condensing unit. During the condensing process, refrigerant passes through three regions: superheated vapor, condensation and subcooled liquid (figure 2). Therefore, modeling of the PHE differs between these three regions through refrigerant quality \( x \). Indeed, PHE is divided to superheated zone where \( x=1 \), subcooled zone \( x=0 \), and a two-phase zone subdivided to 10 elements where vapor quality \( x \) is decremented by a value of 0.1 from 1 to 0.
Condenser model inputs and outputs are:

<table>
<thead>
<tr>
<th>Inputs</th>
<th>Outputs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant ((T, P, x, m))</td>
<td>Refrigerant ((T, P))</td>
</tr>
<tr>
<td>Water ((T_{in}, m))</td>
<td>Water ((T_{out}))</td>
</tr>
<tr>
<td>(x=1)</td>
<td>(x=0)</td>
</tr>
<tr>
<td>(x=1, 0.9, \ldots, 0.1)</td>
<td>(PHE \text{ Capacity})</td>
</tr>
</tbody>
</table>

The specified geometry of the PHE includes the number of plates, plate spacing, length and width of the HE. Water side heat transfer coefficient was first taken from the single-phase water to water pre-tests done by Kim [5, 6]. First results lead to use a simple correlation function of Reynolds number and Prandtl number:

\[
Nu = 0.386 \cdot Re^{0.623} \cdot Pr^{0.369} \tag{12}
\]

where Nusselt number is equal \(Nu = \frac{h_w \cdot D_h}{k_w}\), \(k_w\) represents water thermal conductivity, and \(D_h\) hydraulic diameter.

Heat transfer for refrigerant side in PHE [7] depends on refrigerant state: for single-phase flow most correlations that have been proposed by literature are of the form:

\[
Nu = C \cdot Re^n \cdot Pr^m \tag{13}
\]

while for boiling flow, correlations are more complicated like Cooper boiling correlation [6]. Thus, the model uses correlations similar to equation (13) after identifying its parameters by measurements. The parameters \(C, m\) and \(n\) vary with the vapor quality \(x\).

**Finned tube heat exchanger**

In heating mode, the finned tube heat exchanger FTHE is considered as evaporating unit. As shown in figure 2, refrigerant passes through two regions: evaporation and superheated vapor. Evaporator model inputs and outputs are:

<table>
<thead>
<tr>
<th>Inputs</th>
<th>Outputs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant ((T, P, x, m))</td>
<td>Refrigerant ((T, P))</td>
</tr>
<tr>
<td>Air ((T_{in}, \phi, m))</td>
<td>Air ((T_{out}, \phi))</td>
</tr>
<tr>
<td>(x=1)</td>
<td>(x=1)</td>
</tr>
<tr>
<td>(x=0)</td>
<td>(FTHE \text{ Capacity})</td>
</tr>
</tbody>
</table>

The geometry of the FTHE is specified also, it includes the number of tubes heightwise \(N\), the number of tube rows in the direction of the air flow plate \(n\), the tube spacing in the direction of air flow \(S_l\) and in normal to the direction of air flow \(S_r\), tube inside and outside diameter, spacing between fins \(s\), tube length \(l\), HE height. Some of these dimensional parameters are shown on figure 3, which is a side cutaway view of the FTHE.

**Convergence criteria - Main computational loop**

First, prior to cycle calculation, initial conditions (air, water) must be specified. Besides, a degree of subcooling must also be specified. Using the ambient temperature, the program...
generates its initial estimate of evaporating and condensing pressures. A flow chart outlines the model structure and the order of computation in figure 4. The subcooling is compared to the required one. According to the difference between the two values, condensation pressure is adjusted. Then, the mass flow rate calculated by the expansion device model is compared to the mass flow rate obtained by the compressor model. According to the difference, evaporation pressure is adjusted.

Figure 4: Overall model flowchart

MODEL VALIDATION

Results obtained by the program are validated through tests done on a reference heat pump machine. This test machine is mounted in a climatic chamber where ambient air is maintained at specified conditions of temperature and humidity. Moreover, compensation cycles are conceived to consume the hot and chilled water produced by the system.

The main components of the test machine are: PHE with 64 plates, 2 scroll compressors, FTHE with 24 circuit 3 tube rows and louvered fins, and two thermostatic expansion valves. Temperature and pressure measurement instruments are well distributed on the inlet and outlet of the heat pump components. Compressor mass flow rate is also measured. In addition, flowmeter and air velocity measurement devices are used to determine water and air flow rates respectively.

![Graph a)](image)

![Graph b)](image)
Figure 5 shows comparison between the model and the experiments results. The comparison covers partial load and full load charge for a range of ambient temperature. The first plot (5.a) shows results for the power required by the compressor, which is predicted within 5% of error. Compressor mass flow rate (5.b) is perfectly computed by the model with a difference of 2%. COP and EER are predicted within 8% of error.

EXAMPLE OF RESULTS

As an example of the simulations that could be performed, the following figure shows effect of increasing air flow on the FTHE working as a condenser in cooling mode. The simulation was completed for partial load (1 compressor) and full load (2 compressors), it covers a range of temperature from 35°C to 10°C. A benefit of around 7% is noticed at full load and 3% at partial load (6.a), this is due to the decrease of condensation pressure which reduces the power required by the compressor (6.b).

In addition, results of simulations serve as tool for evaluating seasonal loads. Figure 7 illustrates reduced load curve for a commercial building in Nice (south east France) function of compression ratio, load and weighting factor. Weighting factor is a percentage expressing the number of functioning hours annually. The curve is divided between partial load 50% (one compressor) and full load 100% (two compressors).
CONCLUSION

A vapor compression cycle model has been developed to simulate the performance of an air-to-water R410A based reversible heat pump. The global model is based on first principles and thermodynamics. It includes detailed submodels of the system components: compressor, heat exchangers, and expansion device. The model was successfully tested on a real prototype, it predicts characteristics of the cycle with only a difference of about 5% with the experimental measures. It is able to calculate a complete cycle in a few seconds.

The coupling with the building simulations enables to optimize the reversible when energy consumption has the larger impact, part load and reduced temperature as compared to standard conditions.

Further work will consist in studying different optimization paths for different building and climate conditions.

ACKNOWLEDGEMENT

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