Steady State and Transient Validation of Heat Pumps Using Alternative Lower-GWP Refrigerants

Paper #2266

Viren Bhanot*, Daniel Bacellar, Jiazhen Ling, Abdullah Alabdulkarem, Vikrant Aute (vikrant@umd.edu), Reinhard Radermacher

15th International Refrigeration And Air Conditioning Conference at Purdue, July 14-17, 2014, Purdue, Indiana
Contents

- Introduction
- Test Setup
- Modeling Details
- Simulation Results
- Conclusions
Introduction

- Greenhouse gases are on the rise
- AHRI: industry-wide initiative to examine low-GWP refrigerants
- Drop-in testing of 3-ton nominal capacity residential heat pump unit with ‘soft’ optimization
- Baseline: R410A, Alternatives: D2Y60 and R32
  - D2Y60: binary mixture of 40% R32, 60% R1234yf (by mass)
- Steady-state and transient simulations performed
Test Setup

- Charge optimization for A-test. Maximum COP selected
- TXV adjustments for D2Y60

<table>
<thead>
<tr>
<th>Test</th>
<th>Indoor Dry Bulb</th>
<th>Indoor Wet Bulb</th>
<th>Outdoor Dry Bulb</th>
<th>Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>Extd.*</td>
<td></td>
<td></td>
<td>46.1 °C</td>
<td>Steady</td>
</tr>
<tr>
<td>A</td>
<td></td>
<td>19.4 °C</td>
<td>35 °C</td>
<td>Steady</td>
</tr>
<tr>
<td>B</td>
<td></td>
<td>26.7 °C</td>
<td>27.8 °C</td>
<td>Steady</td>
</tr>
<tr>
<td>C</td>
<td>26.7 °C</td>
<td>≤ 13.9 °C</td>
<td>27.8 °C</td>
<td>Steady Dry Coil</td>
</tr>
<tr>
<td>D</td>
<td></td>
<td></td>
<td>27.8 °C</td>
<td>Cyclic, Dry Coil</td>
</tr>
</tbody>
</table>

* in-house test condition

MODELING DETAILS
Two modeling approaches typically employed for transient simulations:

<table>
<thead>
<tr>
<th></th>
<th>Moving Boundary</th>
<th>Finite Control Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat exchanger</td>
<td>Dynamic - MB</td>
<td>Dynamic - FCV</td>
</tr>
<tr>
<td>Compressor</td>
<td>Static – Adiabatic</td>
<td>Static - Adiabatic</td>
</tr>
<tr>
<td>Valve</td>
<td>Static - Adiabatic</td>
<td>Static - Adiabatic</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>Static</td>
<td>Static/Dynamic</td>
</tr>
<tr>
<td>Speed</td>
<td>Faster</td>
<td>Slower</td>
</tr>
<tr>
<td>Stability</td>
<td>Higher</td>
<td>Lower</td>
</tr>
<tr>
<td>Accuracy</td>
<td>Lower</td>
<td>Higher</td>
</tr>
</tbody>
</table>

Finite Control Volume
- Conservation equations on control volumes to get differential equations
- Coupled DEs solved at each time step

Heat Exchanger Model

- Fin-tube heat exchanger
- 10 segment model
- Uniform pressure
- Simplified HTC computation

$$HTC = \left( \frac{\dot{m}}{\dot{m}_0} \right)^{0.8} HTC_0$$

$$\dot{Q}_{air} = HTC_{air} \cdot A_o \cdot (T_w - T_{air})$$

$$\dot{Q}_{ref} = HTC_{ref} \cdot A_i \cdot (T_w - T_{ref})$$

$$\frac{dT_w}{dt} = \frac{-\dot{Q}_{ref} - \dot{Q}_{air}}{M \cdot C_{p,w}}$$

Continuity Equation:

$$V_i \left[ \left( \frac{\partial \rho}{\partial P} \right)_{h,i} \frac{dP}{dt} + \left( \frac{\partial \rho}{\partial h}_{p,i} - \rho_i \right) \frac{dh_i}{dt} \right] = \dot{m}_{i-1} h_{i-1} - \dot{m}_i h_i + \dot{Q}_{ref,i}$$

if \( \dot{m}_i > 0, \text{and}, \dot{m}_{i-1} > 0 \)

$$V_i \left[ \left( \frac{\partial \rho}{\partial P} \right)_{h,i} \frac{dP}{dt} + \left( \frac{\partial \rho}{\partial h}_{p,i} - \rho_i \right) \frac{dh_i}{dt} \right] = \dot{m}_{i-1} h_{i-1} - \dot{m}_i h_i + \dot{Q}_{ref,i}$$

if \( \dot{m}_i < 0, \text{and}, \dot{m}_{i-1} < 0 \)
Pipe and Accumulator Model

Accumulator

- Lumped volume
- Uniform pressure

\[ V_{\text{acc}} \frac{d\bar{\rho}_{\text{acc}}}{dt} = \dot{m}_{\text{in}} - \dot{m}_{\text{out}} \]

\[ V_{\text{acc}} \left( \bar{\rho}_{\text{acc}} \frac{d\bar{\rho}_{\text{acc}}}{dt} - \frac{dP_{\text{acc}}}{dt} \right) = \dot{m}_{\text{in}} (h_{\text{in}} - \bar{h}_{\text{acc}}) - \dot{m}_{\text{out}} (h_{\text{out}} - \bar{h}_{\text{acc}}) \]

\[ h_{\text{out}} = \begin{cases} h_g - \left( \frac{H_{\text{liq}} - H_{\text{out}}}{d_{\text{out}}} \right) (h_g - h_f), & \text{if } H_{\text{out}} + d_{\text{out}} \geq H_{\text{liq}} > H_{\text{out}} \\ h_f, & \text{if } H_{\text{liq}} > H_{\text{out}} + d_{\text{out}} \\ h_g, & \text{if } H_{\text{liq}} < H_{\text{out}} \end{cases} \]

\[ h_{\text{out}} = \bar{h}_{\text{acc}} \]
TXV and Compressor Model

Closing:
- Spring
- Offset
- Suction Pr.

Opening:
- Bulb

\[ \frac{dT_b}{dt} = \frac{T_{suc} - T_b}{\tau_b} \]

\[ y = \frac{P_b - P_{in} - \text{offset}}{G} \]

\[ G = \frac{k_{spring}}{A_{diaphragm}} \]

Compressor

\[ \frac{dM}{dt} = 0, \frac{dU}{dt} = 0 \]

\[ h_{out}, P_{out} \]

\[ h_{in}, P_{in} \]

\[ m_{comp} = \frac{\rho_{in} \cdot RPM \cdot \text{Disp} \cdot \eta_{vol}}{60} \]

\[ h_{out} = h_{in} + \frac{h_s - h_{in}}{\eta_s} \]
Model Setup

- Property calls through in-house enhancement of NIST’s REFPROP 9.1 database
- HTC obtained from CoilDesigner™
- Compressor efficiencies from measured data

VALIDATION RESULTS
Steady-State Results

- Results within ±7% of measured data
- $COP = \frac{Q_{evap}}{P_{comp}}$

![Graph showing pressure and capacity results for various refrigerants.](image-url)
Steady-State Results

- R32
  - Higher capacity than baseline
  - Lower COP
- D2Y60
  - Lower capacity
  - Lower COP
R32 D-test

**Graphs:**
- **Refrigerant Mass (kg):**
  - Time (s) range from 0 to 1800.
- **Capacity (kW):**
  - Time (s) range from 0 to 1800.
- **Pressure (bar):**
  - Time (s) range from 0 to 1800.
D2Y60 D-test

Graphs showing the refrigerant mass and capacity over time, with legends indicating.Cond, Pipe, Evap, Accu, Total for refrigerant mass, and Experimental, Simulated for capacity. Similarly, graphs for pressure over time showing Discharge [E], Suction [E], Discharge [S], Suction [S].
Conclusions

- Transient library developed for heat pump simulations
- Library contains fin-tube heat exchanger, TXV, compressor, pipe and accumulator models
- Validations carried out for R410A, R32 and D2Y60 refrigerants
  - Steady-state results match within 7% of measured values
  - Transient trends well reproduced, but slightly faster
The authors would like to thank the Center for Environmental Energy Engineering of the University of Maryland, College Park for enabling this research work to be undertaken.
BACKUP SLIDES
Model Setup

Nozzle flow rate equation

\[ V = \left[ (C_d A_6 Y \left( \frac{2\Delta P}{\rho} \right)^{0.5} \right] / \left(1. E \beta^4 \right)^{0.5} \]

Solver: ode23tb

- For stiff systems, with crude error tolerances
- Implicit Runge-Kutta, 1\(^{\text{st}}\) stage: trapezoidal, 2\(^{\text{nd}}\) stage: 2\(^{\text{nd}}\) order backward differentiation formula
- Rel. Tolerance: 10\(^{-8}\)
- Max. Time Step: 0.5 s
Simulation Details

- **R410A D-test:**
  - Simulation Time: 3600 s
  - Run Time: 41398.08
  - RTF: 11.50

- **R32 D-test:**
  - Simulation Time: 3600 s
  - Run Time: 55,609 s
  - RTF: 30.89

- **D2Y60 D-test:**
  - Simulation Time: 3600 s
  - Run Time: 55683.7
  - RTF: 30.94
Heat Exchanger Model

Finite Control Volume

- Transient conservation equations on control volumes to get differential equations
- Coupled DEs solved at each time step

Tank Tube

- Alternating “tank” (heat exchange) elements with friction “tube” (pressure drop) elements

Development of a component based simulation tool for the steady state and transient analysis of vapor compression systems, Winkler, J., PhD Thesis, Department of Mechanical Engineering, University of Maryland, 2009
References