Dynamic Charge Management for Vapor Compression Cycles

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Motivation & Objective

Motivation:

- The energy consumption of vapor compression cycles is related to the mass of refrigerant contained in the system (refrigerant charge).
- The optimal refrigerant charge is dependent upon the operating conditions.
  - Mode (heating or cooling)
  - Ambient conditions
  - Installation (pipe lengths)
- Cycles with fixed charge can only be optimized for one set of conditions.
  - Which condition is best?

Objective:

- Develop and simulate a modified cycle architecture that enables the online optimization of the refrigerant mass circulating in the cycle.
Methodology

We propose adding a *dynamic receiver*, consisting of a reservoir and two expansion devices, that allow the *active charge* in the cycle to be modulated.

This requires the development of the following items:

- Modelica-based dynamic models for the system components that conserve refrigerant mass when connected as a cycle.
- Control method to optimize the cycle charge for a set of conditions that does not require expensive or unreliable sensors.
- Testing metrics to evaluate the performance of this approach.
Models: Heat Exchangers

Mass balance:
\[
\frac{\partial (\rho A)}{\partial t} + \frac{\partial (\rho A v)}{\partial x} = 0
\]

Momentum balance:
\[
\frac{\partial (\rho v A)}{\partial t} + \frac{\partial (\rho v^2 A)}{\partial x} = -A \frac{\partial p}{\partial x} - F_f
\]

Energy balance:
\[
\frac{\partial (\rho u A)}{\partial t} + \frac{\partial (\rho v h A)}{\partial x} = vA \frac{\partial p}{\partial x} + vF_f + \frac{\partial Q}{\partial x}
\]

Air-side heat transfer:
\[
m_{a} c_{p,a} \frac{dT_a}{dy} \Delta y = \alpha_a \left( A_{o,t} + \eta_{fin} A_{o,fin} \right) (T_w - T_a)
\]

Energy balance: Tube wall
\[
\left( M_t c_{p,t} + M_{fin} c_{p,fin} \right) \frac{dT_w}{dt} = q_r + q_a
\]
Models: Dynamic Receiver, Compressor, EXV

Variable-speed compressor model:

\[ \eta_v = c_0 + c_1 \phi + c_2 \phi^2 + c_3 \left( p_{\text{dis}} - p_{\text{suc}} \right) \left( 1 + c_4 p_{\text{suc}} \right) \]

\[ W = z_1 p_{\text{suc}} \dot{V}_{\text{suc}} \left( \phi^{z_2} - z_3 \right) + z_4 \]

\[ \phi = \frac{p_{\text{dis}}}{p_{\text{suc}}} \]

Linear expansion valve (LEV):

\[ \dot{m} = C_v A \sqrt{\rho_{\text{in}} \Delta p} \]

\[ C_v A = c_0 + c_1 \varphi \]

Refrigerant reservoir model:

Mass balance:

\[ V \frac{d \bar{\rho}}{dt} = \sum \dot{m} \]

Energy balance:

\[ V \left( \bar{\rho} \frac{d \bar{h}}{dt} - \frac{d p}{dt} \right) = \sum \dot{m} \bar{h} - \bar{h} \sum \dot{m} + \sum q \]
Observed Refrigerant Mass Behavior

Observation:
- The total mass in dynamic simulations of vapor compression cycles can vary over time.

This can be problematic:
- The mass should be conserved in the simulation as in the real system.
- Dynamics from changing refrigerant mass affect the dynamics of many system variables and can increase the model execution time.
- Unrealistic to control the refrigerant mass in a dynamic cycle simulation if it varies from numerical artifacts.
Models: Standard Property Relations

Property relations are needed to relate the thermodynamic variables to the discretized PDEs describing the system.

- Select the specific thermodynamic coordinates for integration (state variables)

Conventionally, pressure and specific enthalpy are selected as the thermodynamic coordinates:

\[
\begin{align*}
\frac{dM}{dt} &= V \frac{d\rho(P,h)}{dt} \\
&= V \left[ \frac{d\rho}{dP} \frac{dP}{dt} + \frac{d\rho}{dh} \frac{dh}{dt} \right] \\
\frac{dU}{dt} &= V \frac{d(\rho(P,h)u(P,h))}{dt} \\
&= V \left[ \left( h \frac{d\rho}{dP} - 1 \right) \frac{dP}{dt} + \left( h \frac{d\rho}{dh} + \rho \right) \frac{dh}{dt} \right]
\end{align*}
\]

Variations of refrigerant mass are related to this selection of thermodynamic coordinates:

\[
M_{\text{total}} = \sum_k \rho_k(P,h)V_k
\]

\[
M_{\text{total}} = \sum_k \rho_k(P + \varepsilon, h + \varepsilon)V_k
\]
Models: Alternative Property Relations

Explanation: Small errors in the pressure and specific enthalpy near phase transitions map to large errors in the density.

Alternate approach: Select density as a state variable, so that the integration errors won’t be amplified and mass will be conserved.

$$\frac{dM}{dt} = V \frac{d\rho}{dt}$$

$$\frac{dU}{dt} = V \frac{d(\rho u(P, \rho))}{dt}$$

$$= V \left[ \left( \rho \frac{dh}{dP} - 1 \right) \frac{dP}{dt} + \left( \rho \frac{dh}{d\rho} + h \right) \frac{d\rho}{dt} \right]$$

This approach is not conventionally used because the integration can be slow in regions near phase transitions where the density derivatives are high.

$$M_{total} = \sum_k \rho_k (P + \varepsilon, h + \varepsilon)V_k$$

The computation time can be reduced by calculating the specific enthalpy as an extra state variable:

$$\frac{dh}{dt} = \frac{dh}{dP} \frac{dP}{dt} + \frac{dh}{d\rho} \frac{d\rho}{dt}$$
Mass-Conservative Simulations

- There are significant changes in system charge for all three tolerances.

- These changes in the charge are highly correlated with very small variations in the static quality $x$.

- Refrigerant mass dynamics can have a significant effect on other system dynamics and can affect the integration time.

<table>
<thead>
<tr>
<th>State Variables</th>
<th>Max Error</th>
<th>% Error</th>
<th>Sim Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>$(P, h)$</td>
<td>-19.3 g</td>
<td>12.8 %</td>
<td>1925 s</td>
</tr>
<tr>
<td>$(P, \rho)$</td>
<td>2.0e-4 g</td>
<td>1.3e-4 %</td>
<td>1374 s</td>
</tr>
<tr>
<td>$(P, h, \rho)$</td>
<td>2.0e-4 g</td>
<td>1.3e-4 %</td>
<td>450 s</td>
</tr>
</tbody>
</table>

Solver tolerance: 1e-6
Initial System Simulations: Cooling Mode

This architecture was evaluated via:

- Cycle with the dynamic receiver
- 3 PI controllers:
  - Room temperature to compressor frequency
  - Suction superheat to LEV position
  - Mass in dynamic receiver to optimal setpoint
- Conditions in Phoenix, AZ on 6/23/2013

Compared to a cycle with fixed charge optimized for only one mode.

<table>
<thead>
<tr>
<th>Active Charge</th>
<th>COP Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>530 g</td>
<td>-</td>
</tr>
<tr>
<td>450 g</td>
<td>5.3%</td>
</tr>
<tr>
<td>380 g</td>
<td>10.7%</td>
</tr>
</tbody>
</table>
Initial System Simulations: Heating Mode

This architecture was also evaluated in heating mode using conditions in Washington, D.C. on 1/1/2014.

These results suggest that properly modulating the refrigerant charge can significantly reduce power consumption.

<table>
<thead>
<tr>
<th>Active Charge</th>
<th>COP Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>530 g</td>
<td>13.0%</td>
</tr>
<tr>
<td>450 g</td>
<td>4.4%</td>
</tr>
<tr>
<td>380 g</td>
<td>-</td>
</tr>
</tbody>
</table>
Extremum Seeking Control

- Direct measurements of mass in the receiver will be difficult and/or expensive.

Alternative strategy: *Extremum seeking*

- Optimize system performance by driving to zero the gradient of the power consumption with respect to the position of an expansion valve in the dynamic receiver.
- Does not require any measurement of refrigerant mass
- Decoupled with the other process control loops (room temperature and evaporator superheat temperature control)
- We used a time-varying (TV) extremum seeking algorithm because of the slow convergence rate of conventional perturbation-based methods.
Results: Extremum Seeking Control

A TV extremum seeking controller was implemented on the system.

- Controlled the position of the LEV between the receiver and the condenser outlet to minimize the total power consumption.

This controller successfully brought the system to the optimal active charge in approximately 2 hours.

Demonstrated the potential for optimizing the active charge in a vapor compression cycle without directly measuring or estimating charge.
Conclusions & Future Work

Conclusions:

• We developed a set of dynamic models of the vapor compression cycle that conserve refrigerant mass.
• We demonstrated the reduction in power consumption that may be obtained by modulating the refrigerant mass.
• We incorporated a modern extremum-seeking method that can optimize the refrigerant mass without intrusive measurements.

Future work:

• Further refinement of both the dynamic receiver (design methodology) and the control methods.
• Experimental evaluation over a range of conditions.
Thank you for your attention.

Questions?

Contact info: laughman@merl.com
Backup Slides
Evaluation Methodology

**Goal:**

Compare the performance of the dynamic test system model using both conventional and alternative property relations.

**Methodology:**

Choose input waveforms that excite dynamics in the refrigerant mass.

- Pump speed
- Heat input

Evaluate changes in the refrigerant mass.

- Changes in the property relation
  - \((P, h), (P, \rho), (P, \rho, h)\)
- Changes in the solver tolerance
  - Tol = 1e-4, 1e-5, 1e-6
Refrigerant Reservoir Model

- Lumped parameter control volume
  
  Mass balance:
  \[ V \frac{d\bar{\rho}}{dt} = \sum \dot{m} \]

  Energy balance:
  \[ V \left( \bar{\rho} \frac{d\bar{h}}{dt} - \frac{dp}{dt} \right) = \sum \dot{m} h - \bar{h} \sum \dot{m} + \sum q \]

- Account for the variation in liquid level

- Refrigerant leaving enthalpy is determined based on whether the outlet is fully submerged, partially submerged or not submerged

\[
h_{out} = \begin{cases} 
  h_g & \text{if } H_{out} > H_{liq} > H_{out} + d_{out} / 2 \\
  h_g \left( \frac{H_{out} + d_{out} / 2}{d_{out}} \right) (h_g - h_f) & \text{if } H_{out} + d_{out} / 2 \geq H_{liq} \geq H_{out} - d_{out} / 2 \\
  h_f & \text{if } 0 < H_{liq} < H_{out} - d_{out} / 2 
\end{cases}
\]