2014

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ABSTRACT

Due to the relatively high global warming potential (GWP) value of R410A, much effort has been devoted to the exploration of potential refrigerants to replace R410A in the heat pump applications. Those studies involving natural refrigerants, which are of zero or single-digit GWP values, have not yet demonstrated the readiness for the substitution because of various reasons such as low cycle efficiency, toxicity and flammability. Henceforward, some synthetic refrigerants whose GWP values are significantly lower than R410A such as R32 and some R32-based blends such as D2Y60, are getting more attention. To evaluate the transient performance of those low-GWP refrigerants, a Modelica-based heat pump model is developed to simulate a heat pump cycle during both steady state operations and transient operations. The model includes an efficiency-based compressor model, two segmented heat exchanger models, a control volume-based valve model and segmented pipe models. The heat exchanger model is capable of simulating multi-bank multi-circuit tube-fin heat exchangers with an arbitrary number of segments. The pipe model is developed based on the segmented heat exchanger model, and its implementation provides for better charge prediction compared to single lump model. In order to speed up the simulations, accelerated refrigerant property routines were developed for R32 based on REFPROP. System-level steady-state and transient simulations for several alternative low-GWP working fluids, R32 and D2Y60, are conducted. The simulation results are compared with the published experimental data obtained from the Alternative Refrigerant Evaluation Program (AREP). The data includes steady-state operation data based on ASHRAE A, B and C tests, and transient data from ASHRAE D cyclic operation test. The validation of R32 and D2Y60 steady-state data shows a maximum deviation of 7.5% and an average deviation of 4%. The transient simulation well captures the dynamic performance of vapor compression cycle during start-up and shut-down.

1. INTRODUCTION

In the residential heat pump application, R410A has been widely used as a replacement for R22 since the late 1990's due to its zero ozone depletion potential (ODP). However, R410A has a high global warming potential (GWP) value of around 2100 (IPCC, 2007). In order to reduce the global warming impact, Air-conditioning, Heating and Refrigeration Institute (AHRI) initiated a low GWP alternative refrigerant evaluation program (AREP) to investigate possible candidates to replace those high-GWP refrigerants such as R410A. As a part of the current study, baseline tests were conducted for an R410A heat pump, followed by drop-in tests for same system with alternative refrigerants such as R32, D2Y60 and L41a. D2Y60 has a composition of 40% R32 and 60% R1234yf. A brief introduction of the test facility and test matrix will be provided in Section 2 of this paper. Interested readers are referred to Alabdulkarem et al. (2013) for additional information.

The performance comparison of the two refrigerants against R410A were mainly conducted by following test conditions defined in the ASHRAE standard 116-1995. Those comparisons include system COP, cooling/heating capacities and cycle state points (pressure and temperature). However, not all the performance characteristics of any vapor compression system can be captured by steady state tests. This is because during real operation, the heat pump systems are seldom running at steady state conditions. Systems typically experience frequent start-ups and shut-downs to maintain the required room temperature. Consequently, heat pump systems, at most of the time, operate in transient mode. In light of this, a good understanding of transient behavior of heat pump system is equally important, if not more, as that of steady state behavior. In addition, the design of control components in the heat pump system such as valves is more effective based on transient operating data.

However, compared with steady state experiments, transient tests take much more time and effort. Thus, transient simulations of heat pump system become attractive to engineers. Rasmussen B. (2012) and Rasmussen and Shenoy B. (2012) wrote two excellent review papers regarding transient simulation of vapor compression systems. In his review, he compared different modeling methods for components as well as system behaviors under different modeling approaches. Hermes and Melo (2008) developed a first-principles methodology for modeling and simulating the dynamic behavior of domestic refrigerators. Pfafferott and Schmitz (2004) developed a Modelica-based model to simulate transient performance of CO_2 refrigeration system. The objective of this paper is to introduce a Modelica-based transient heat pump model which can be used to simulate the dynamic performance of vapor compression system using REFPROP (NIST, 2013) supported refrigerants and user defined mixtures. The validity of the model is demonstrated by comparing the simulation results against experimental data for both R32 (predefined pure refrigerant) and D2Y60 (user defined mixture) under different ASHRAE cooling conditions.

2. EXPERIMENTAL FACILITY

Figure 1 shows the experimental facility schematic for the AREP project. The test facility was installed in one environmental chamber and one wind tunnel. The environmental chamber is used to simulate various ambient conditions defined in Table 1. The indoor conditions are simulated in the wind tunnel where a heater and a humidifier are used to balance the indoor heat exchanger capacity. An R410A residential heat pump system was tested as per ASHRAE test conditions (Table 1) were first completed. Those test results served as baseline for the later comparison. The R410A refrigerant was then recovered from the system and new refrigerant was charged into the system. Both R32 and D2Y60 were tested in the same heat pump unit following the same ASHRAE test conditions. The system COP, capacities, cycle state points were measured and compared against their R410 counterpart.

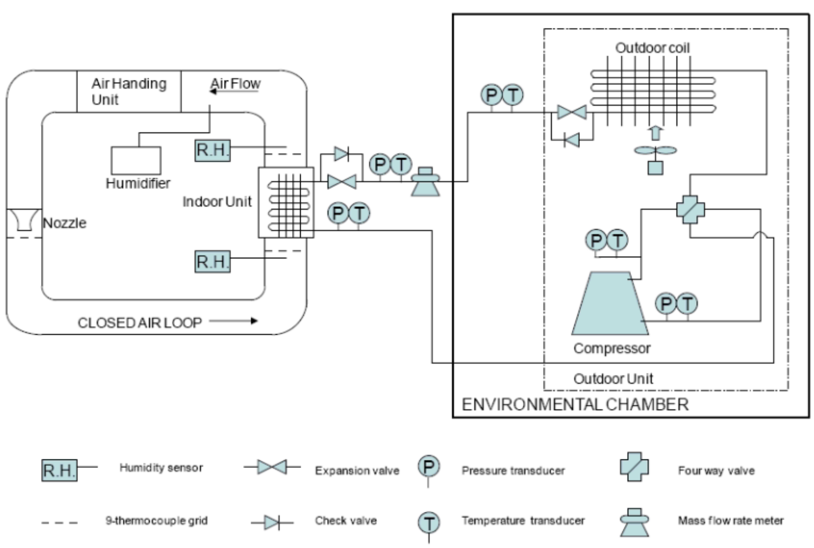

Figure 1: AREP test facility schematic diagram

Table 1: AREP test conditions for cooling application

<table>
<tr><th rowspan="2">Test</th><th colspan="2">Indoor</th><th colspan="2">Outdoor</th><th>Operation</th></tr>
<tr><th>DB</th><th>WB</th><th>DB</th><th>WB</th><th></th></tr>
<tr><td>Extended condition</td><td rowspan="5">26.7°C</td><td rowspan="3">19.4°C</td><td>46.1°C</td><td rowspan="5">NA</td><td>Steady state</td></tr>
<tr><td>A</td><td>35.0°C</td><td>Steady state</td></tr>
<tr><td>B</td><td>27.8°C</td><td>Steady state</td></tr>
<tr><td>C</td><td rowspan="2"><=13.9°C</td><td rowspan="2">27.8°C</td><td>Steady state</td></tr>
<tr><td>D</td><td>Cyclic</td></tr>
</table>

3. MODELICA TRANSIENT COMPONENT LIBRARY

To simulate the AREP cycle, a set of components was first modeled using the Modelica Language. Since year 2000, Modelica has been developed and adopted for many applications, especially in the transient simulation. The Modelica language has several attractive features such as equation-based text input and “acausal” modeling approach. These features make Modelica a useful and convenient modeling platform. The modeling details for individual components are described in the following sections.

3.1 Compressor Model

Even in transient modeling, the compressor is often treated as a quasi-steady state component because the timescales associated with the variation of the compressor mass flow rate are very small compared to those associated with heat exchangers and charge distribution. The compressor is modeled by using three efficiencies: isentropic efficiency, volumetric efficiency and motor efficiency. Equations. (1) to (3) are the definitions of the three efficiencies. However, steady-state models are not adequate to capture the important transient characteristics of the compressor, such as the refrigerant mass and energy storage within the suction and discharge chambers. Meanwhile, internal heat transfer between the refrigerant and pertinent elements of the compressor plays an essential role in simulating the start-up and shut-down transients. To address the issue, a free volume in the compression chamber filled with refrigerant is added as a lumped control volume (CV) upon which the mass and energy conservation is imposed.

$$h_{out} = h_{in} + \frac{h_{out,s} - h_{in}}{\eta_{isen}} \tag{1}$$

$$\dot{m} = \eta_{vol} \rho_{in} D \frac{RPM_{motor}}{60} \tag{2}$$

$$\frac{\dot{m}\left(h_{out} - h_{in}\right)}{\eta_{motor}} = \dot{P}_{motor} \tag{3}$$

3.2 Expansion Device (TXV) Model

The thermostatic expansion valve (TXV) is comprised of two portions: the throttling portion which regulates the refrigerant mass flow through the valve, and the sensor bulb portion which monitors the refrigerant temperature leaving the evaporator and converts the change in temperature into the change in pressure on the diaphragm, causing the needle to move upward or downward, as shown in Figure 2.

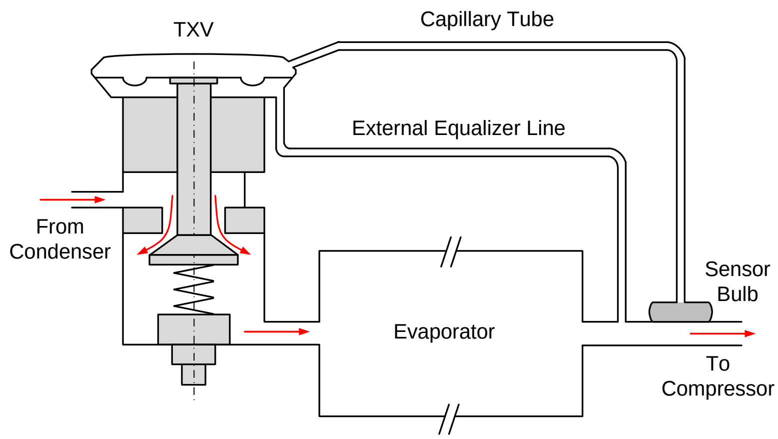

Figure 2: Schematic of a thermostatic expansion valve

Since the superheat is sensed by the bulb attached on the suction line, there is a delay between the sensed superheat and the actual superheat due to the thermal inertia of the bulb and the heat transfer resistance between the substance in the bulb and the refrigerant flowing in the suction line. The sensor bulb is modeled as a lumped section in the present analysis and its temperature variation with time is given by

$$M_b c_{p,b}\frac{dT_b}{dt}=\frac{T_{amb}-T_b}{R_{ab}}+\frac{T_w-T_b}{R_{wb}} \tag{4}$$

where M_b is the mass of the sensor bulb, $c_{p,b}$ is the specific heat, T_b is the temperature, T_w is the temperature of the tube wall to which the sensor bulb is attached, R_{ab} is the thermal resistance between the ambient and the bulb, R_{wb} is the thermal contact resistance between the tube wall and bulb. The details of the TXV model can be found in Qiao et al. (2012)

3.3 Heat Exchanger Model

The heat exchanger is modeled using finite volume method. The heat exchanger is divided into a pre-defined number of segments such as 3, 5 or 10. For each segment, mass and energy conservation equations (equations (5) and (6)) are applied. Each refrigerant volume transfers heat to the corresponding air volume through wall element and the equation is represented in equation (7). The air side sensible and latent heat transfer is calculated based on equations (8) and (9). To simplify the model, only heat transfer calculation is conducted in each volume. The control volume diagram is described in Figure 3.

$$V_i\left[\frac{\partial\rho}{\partial P}|_{h,i}\frac{dP}{dt}+\frac{\partial\rho}{\partial h_i}|_P\frac{dh_i}{dt}\right]=\dot{m}_{i-1}-\dot{m}_i=\frac{dM_i}{dt} \tag{5}$$

$$V_i\left[(h_i\frac{\partial\rho}{\partial P}|_{h,i}-1)\frac{dP}{dt}+(h_i\frac{\partial\rho}{\partial h_i}|_P-\rho_i)\frac{dh_i}{dt}\right]=\dot{m}_{i-1}h_{i-1}-\dot{m}_i h_i-\dot{Q}_{ref,i}=\frac{dU_i}{dt} \tag{6}$$

$$(MC_p)\frac{dT_w}{dt}=Q_{rw}-Q_{wa} \tag{7}$$

$$T_{a,o}=T_w-(T_w-T_{a,i})\exp\left(-\frac{\eta_f\alpha_o A_t}{\dot{m}_a c_{p,a}}\right) \tag{8}$$

$$X_{a,o}=X_{sat,w}-(X_{sat.w}-X_{a,i})\exp\left(-\frac{\eta_f\alpha_o A_t}{\dot{m}_a c_{p,a}Le^{2/3}}\right) \tag{9}$$

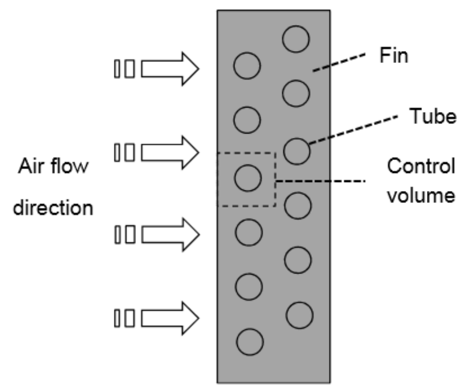

Figure 3: Control volume of HX model

3.4 Other Components Models

Besides the four major component models, two auxiliary components, an accumulator model and a pipe model, are also required. The accumulator is modeled using a lumped parameter approach. The exit refrigerant enthalpy is determined by both the exit port height (H_{out}) and liquid level (H_{liq}) inside the tank which is illustrated in equation (10). The piping model is modeled similarly as the heat exchanger model, but only one lumped volume is used.

$$h_{out} = \begin{cases} h_f & \text{if } H_{liq} > H_{out} + d_{out} \\ h_g - \left(\dfrac{H_{liq} - H_{out}}{d_{out}} \right) \left(h_g - h_f \right) & \text{if } H_{out} + d_{out} \geq H_{liq} \geq H_{out} \\ h_g & \text{if } 0 < H_{liq} < H_{out} \end{cases} \tag{10}$$

4. SIMULATION RESULTS AND VALIDATIONS

Figure 4 shows the schematic of the heat pump system model using a commercially available Modelica simulation platform (Dassault System, 2014). It includes one compressor, one outdoor unit, one expansion device, one indoor unit, three pipes and one accumulator. As mentioned in the previous sections, the simulations were conducted according to ASHRAE test conditions A to D and extended conditions.

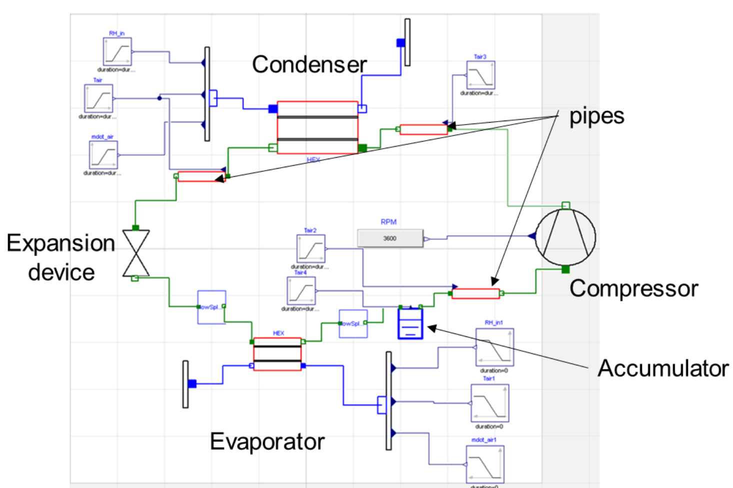

Figure 4: Schematic of the tested heat pump system

4.1 Steady-state Results and Validations

The model was first simulated for baseline R410A system according to the ASHRAE test conditions and achieved a good agreement with experimental data. Due to the page limit, the authors would like to focus on presenting results and validations for alternative refrigerants R410A and D2Y60. Figure 5 demonstrates the R32 system capacities and pressures under the extreme test condition. Under the extreme condition, the ambient temperature is 46°C. Therefore, the steady state condensing pressure is over 3.4 MPa and the evaporator capacity reduces from the rated 10 kW to only 8.9 kW. The simulation shows a good agreement with experimental data. The maximum deviation came from the condenser capacity prediction which the simulation under-predicts the results by about 5%. Table 2 and 3 show simulation results and validations for R32 at the rest of conditions and D2Y60 at all steady state conditions. For both refrigerants, the average deviations between simulation and experimental data are around 4%. The maximum deviation for R32 is around 7.5% while the one for D2Y60 is around 7%. Improvements can be made especially through the development of a more accurate compressor model. However, the current models show reasonable agreement with experimental data.

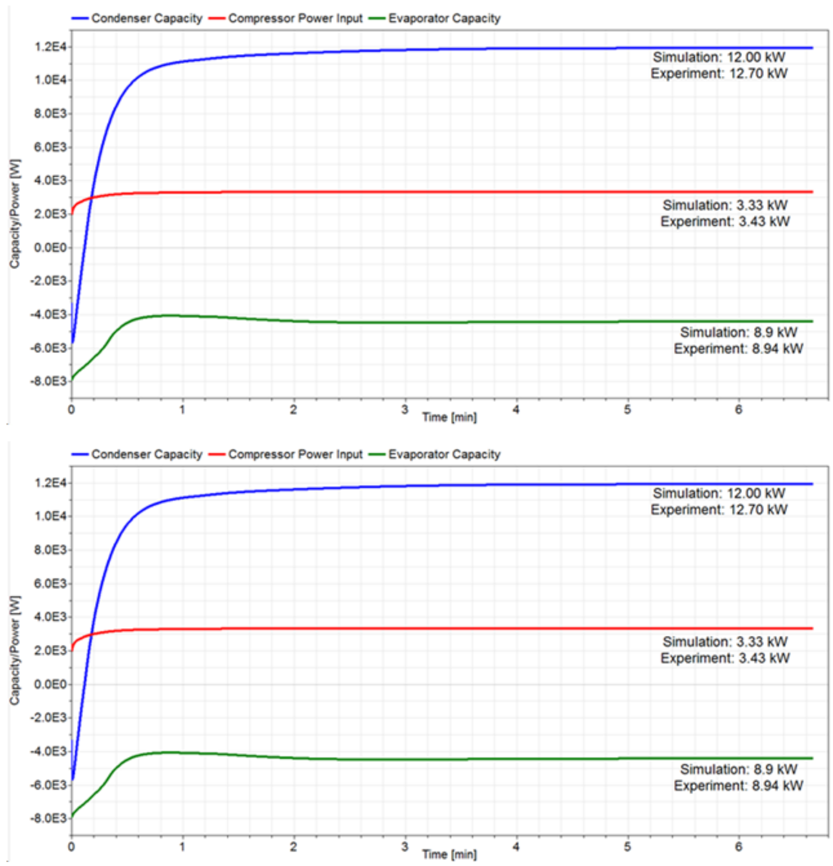

Figure 5: Dymola capacity and pressure outputs at extreme conditions

Table 2: R32 steady state simulation results and validations

R32	Test condition A			Test condition B			Test condition C		
	Sim.	Exp.	Error	Sim.	Exp.	Error	Sim.	Exp.	Error
Q_{evap} (kW)	10.0	10.2	1.96%	10.4	10.9	4.59%	10.0	9.6	-4.17%
Q_{cond} (kW)	12.4	12.5	0.88%	12.4	13.4	7.46%	12.1	11.9	-1.68%
COP (compressor power only)	4.0	3.9	-0.77%	5.0	4.9	-2.04%	4.8	4.5	-6.67%
$P_{discharge}$ (Bar)	28.0	26.1	-7.28%	23.7	22.6	-4.87%	2.3	2.2	-4.55%

$P_{suction}$ (Bar)	10.9	11.0	0.91%	10.5	11.0	4.55%	10.0	10.0	0.00%

Table 3: D2Y60 steady state simulation results and validations

D2Y60	Test condition A			Extended condition		
	Sim.	Exp.	Error	Sim.	Exp.	Error
Q_{evap} (kW)	8.14	8.11	-0.37%	7.19	6.85	-4.96%
Compressor Input (kW)	2.05	1.98	-3.54%	2.6	2.55	-1.96%
COP (compressor power only)	3.97	4.10	3.06%	2.77	2.69	-2.94%
$P_{discharge}$ (Bar)	20.1	20.62	2.52%	25.7	26.97	4.71%
$P_{suction}$ (Bar)	9.2	8.76	-5.02%	9.02	9.09	-0.77%
	Test condition B			Test condition C		
	Sim.	Exp.	Error	Sim.	Exp.	Error
Q_{evap} (kW)	8.15	8.58	5.01%	8.07	7.99	-1.00%
Compressor Input (kW)	1.75	1.74	-0.57%	1.83	1.73	-5.78%
COP (compressor power only)	4.66	4.93	5.55%	4.41	4.62	4.52%
$P_{discharge}$ (Bar)	18.5	17.81	-3.87%	16.78	17.56	4.44%
$P_{suction}$ (Bar)	8.77	8.67	-1.15%	8.77	8.2	-6.95%

4.2 Dynamic Simulation and Validations

The transient simulation results are compared with experimental data for the purpose of validating the model under dynamic conditions. The simulation time for the R32 system for the 30-minute cyclic test (6 minutes compressor-on and 24 minutes compressor-off) is 90 seconds on a 3.7 GHz Xeon CPU using in-house accelerated refrigerant property routines based on NIST REFPROP (Aute and Radermacher, 2014). Figure 6 compares the simulation results with the experimental data. Both pressures match the experimental data well during the start-up. The simulation shows that the TXV closes slightly faster than the real one does in the shut-down. Since not all the valve information, such as the spring constant and pin angle, is available, it is almost impossible to exactly match the real valve characteristics. The simulated pressures at compressor-off period is 3% off compared to measurement data. This deviation mainly results from the refrigerant charge difference. The accumulator inner volume was not measurable and the data was approximated from outer dimension and therefore may not be accurate enough. The temperature profile comparison also shows that improvements to the simulation can be made once the entire system charge prediction is improved. The actual system charge is 4.2 kg while the simulation shows a total charge of 3.8 kg. Figure 7 details the charge migration during the D test (cyclic test). As the compressor starts, it draws the refrigerant from the evaporator and discharges it to the condenser. Therefore, the charge in the condenser increases and the charge in the evaporator decreases. The charge distribution is then stable as the cycle becomes stable. During the compressor-off period, the refrigerant flows due to the pressure difference. Therefore, the refrigerant charge in the condenser migrates towards the evaporator until the valve is almost closed. The total charge in the system is conserved during the simulation.

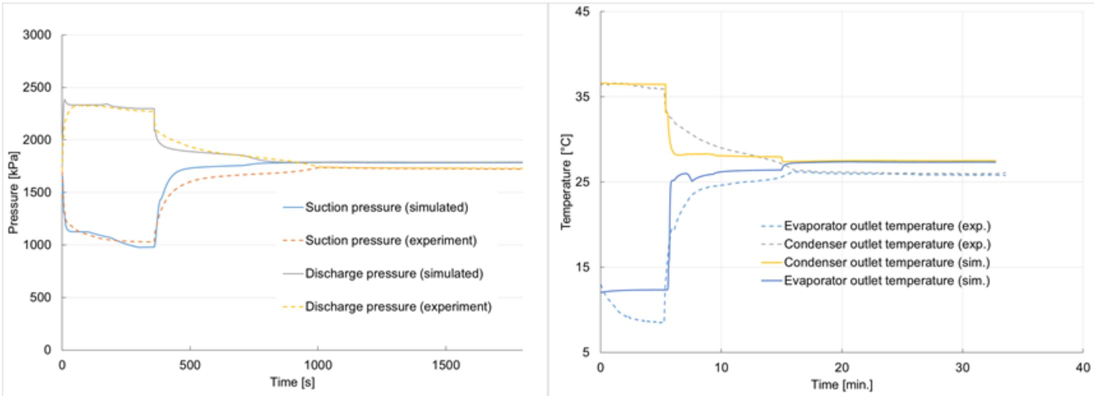

Figure 6: R32 Cycle pressure and temperature comparisons under the cyclic condition

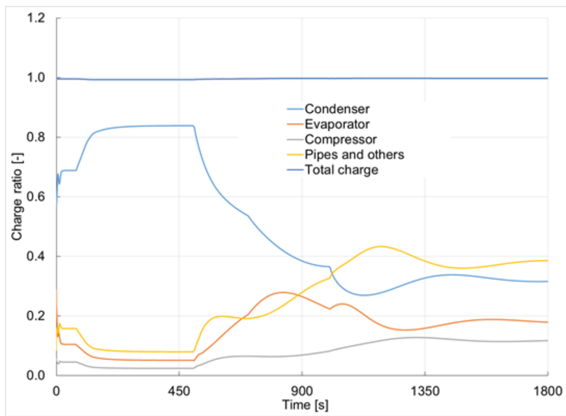

Figure 7: R32 charge migration under the cyclic condition

Since D2Y60 is a relatively new refrigerant, its accelerated property routines have not been developed. The calculation speed is therefore much slower than the R32 case. A full cyclic simulation takes hours and therefore only a 100-second start-up and shut-down cyclic simulation is presented in the paper. Figure 8 shows the capacities of heat exchangers and refrigerant charge migration. The charge migration follows a trend similar to R32 case: the evaporator loses charge during system start-up and gain charge during system shut-down. Figure 9 shows the simulated compressor suction and discharge pressures versus measured values. A direct overplay of data is impossible because the simulation, due to computation cost, is only conducted for 100 seconds while the experiment, following ASHRAE D test condition, lasted up to 30 minutes. Nevertheless, the simulation results show a good agreement with experimental results in terms of the pressure magnitudes and trend during start-up and shut-down.

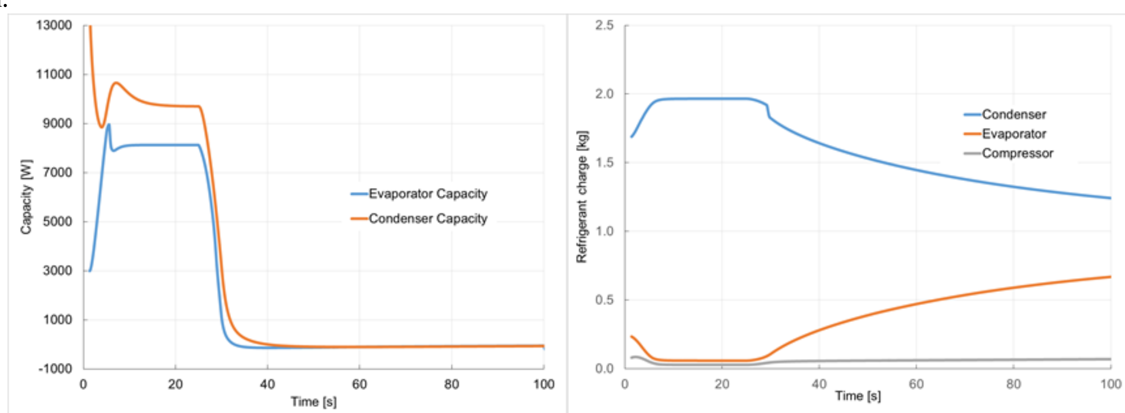

Figure 8: D2Y60 cycle capacities and charge migration under the cyclic test (100 seconds)

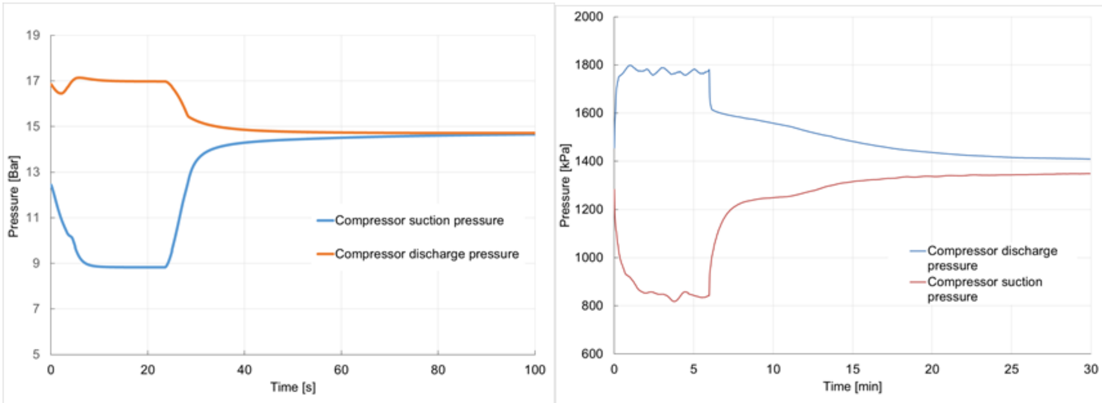

Figure 9: D2Y60 cycle pressure comparison under the cyclic condition (left: simulation 100 seconds, right: experiment 30 minutes)

6. CONCLUSIONS

This paper introduces a Modelica-based transient model to simulate heat pump systems using low-GWP alternative refrigerants. To validate the model, both R32 and D2Y60 systems are simulated and compared against experimental data under both steady state conditions and cyclic conditions. The steady state simulation shows that most of the parameters are predicted within 5% difference of experimental data. The maximum deviation of the R32 system is 7.46% and the maximum deviation of D2Y60 is 6.95%. The transient simulation was conducted according to the ASHRAE D test. R32 model predicts the system cyclic operation and the results match the experimental results well. For D2Y60 model, due to the relatively slow property calls, the model simulates the system during compressor start-up and shut-down up to 100 seconds. The Modelica-based transient model has been proven to be able to predict both steady-state and transient performance of low GWP refrigerant systems with good accuracy.

NOMENCLATURE

A	area	(m^2)
C_p	specific heat	(kW/kgK)
Cv	flow constant	(-)
D	displacement volume	(m^3)
h	enthalpy	(kJ/kg)
H	accumulator height	(m)
Le	Lewis number	(-)
M	mass	(kg)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(kPa)
Q	capacity	(kW)
R	thermal resistance	(k/W)
RPM	rev. per minutes	(-)
t	time	(s)
T	temperature	(K)
U	internal energy	(kW/kg)
X	humidity	(kg/kg)
η	efficiency	(-)
ρ	density	(kg/m^3)

Subscript

a	air
b	bulb
i, i-1	segment i and i-1
in	inlet/inner
isen	isentropic
o	outer
ref	refrigerant
sat	saturated
t	tube
vol	volumetric
w	water/wall

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ACKNOWLEDGEMENT

We gratefully acknowledge the support of this effort the sponsors of Center for Environmental Energy Engineering (CEEE) at the University of Maryland.