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Abstract

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Dynamic Charge Management for Vapor Compression Cycles

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ABSTRACT

While the mass of refrigerant contained in a vapor compression cycle has a significant effect on the cycle's power consumption, conventional cycle architectures cannot optimize energy efficiency by varying the mass as ambient conditions and operational requirements change. This paper proposes a new system architecture that allows the refrigerant mass circulating in the system to be modulated over time in response to the operating conditions. This new architecture is developed with a set of dynamic cycle simulations that facilitate the specification of the mass in the cycle, and which eliminate numerical errors that can cause nonphysical fluctuations in the total mass. Controls that optimize the mass in the cycle as a function of the operating conditions are also incorporated into the overall system, allowing it to be constructed without the addition of new sensors. Gains in the coefficient of performance for this new system exceeding 10% over cycles with a fixed refrigerant charge are demonstrated.

1. INTRODUCTION

One common trait of many distributed parameter multiphysical systems, such as vapor compression cycles, is that their performance is dependent on characteristics that are determined over a wide range of system-level design, installation, and operation activities. The energy efficiency of vapor compression cycles, as measured through the coefficient of performance (COP), is a prime example of such a trait; this performance metric is dependent on a host of geometric parameters (e.g., heat exchanger geometry), material parameters (e.g., refrigerant properties), physical parameters (e.g., type of compression process), installation parameters (e.g., pipe length), and operational parameters (e.g., control algorithms). Engineers with an interest in improving the performance of such systems thus have the daunting task of identifying optimal values for this large set of parameters to achieve their objective.

Further complicating the task of the system designer, especially for such systems as reversible heat pumps, is the presence of significant uncertainty in many important operational variables affecting the system performance. For example, one impact of globalization in the heat pump market is that particular system configurations will be installed in many different climates for many different applications. There is also significant variability in the specifics of the system installation, due to local building practices and the expertise of contractors. Finally, these systems must also satisfy numerous operational constraints, such as refrigerant pressures or temperatures, to meet safety and regulatory requirements. As a result, the collection of uncertainties in aggregate makes it impractical or impossible to choose a set of parameter values during the system design process that will optimize the energy performance of the installed system.

The total refrigerant mass, or system charge, in most conventional vapor compression cycles is an excellent example of a parameter, the value of which is specified at the design stage and remains constant during the machine operation, that affects many different aspects of the system performance and whose optimal value is dependent upon conditions that are largely unknown at design time. Because the system charge affects the energy consumption, it is often optimized to meet specifications set by the manufacturer or regulatory agencies. For example, the specific value of the system charge affects the COPs of the cycle in heating and cooling modes, the refrigerant-side temperatures and pressures over the range of operating conditions, and is also subject to government regulations concerning the use of chemicals which contribute to global warming. Unfortunately, such an effort to pick a single optimized value of the refrigerant charge is often confounded by the uncertainty in the installed machine's climate or application. Challenges in assessing and

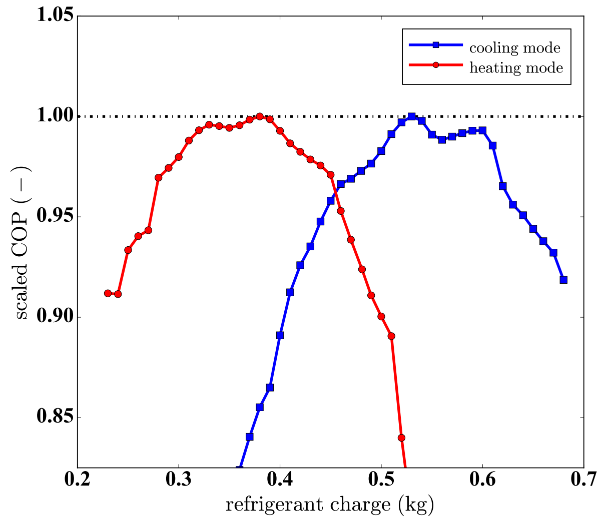

Figure 1: Steady-state scaled COP as charge is varied from 200 g to 680 g for a standard vapor compression cycle in both heating and cooling modes.

guaranteeing the installed system charge as a function of pipe length and the installer's proficiency, not to mention the possible presence of refrigerant leaks that arise over time, also contribute to significant uncertainty in the system charge over time. Consequently, the impact that the refrigerant charge has on the system performance has been extensively studied in the literature (Kim and Braun, 2012; Corberan et al., 2011; Poggi et al., 2008).

The effect of refrigerant charge on the COP of a controlled cycle coupled to an lumped room with a load, in both heating and cooling modes, can be clearly seen by considering the data illustrated in Figure 1, which was generated by using the models subsequently discussed in this paper. This figure displays the cycle COP, normalized to the peak COP in each mode for the ease of visual comparison, for two different operating conditions: heating mode with a -2000 W load, 0 °C ambient temperature, and 23 °C room air temperature setpoint, and cooling mode with a 2000 W load, 35 °C ambient temperature, and 27 °C room air temperature setpoint.

Figure 1 clearly demonstrates the sensitivity of the operating mode to the refrigerant mass, as the COP is reduced by more than 10% if the charge is optimized for one operating mode but the system is then operated in the opposite mode. It is also evident that the choice of any fixed value of the refrigerant charge will result in suboptimal performance for other modes; for example, a system charge of 0.45 kg, where the COP of both modes is equal, will result in a reduction in COP from the peak value for both heating and cooling modes of 4%. As this plot illustrates the performance for only one set of conditions, it is also expected that variations in the ambient temperature, room air temperature, and heat load will give rise to corresponding variations in optimal value of the cycle charge.

We propose the incorporation of a new component into the cycle which we call a "dynamic receiver," as seen in Figure 2, which modulates the charge in the cycle to address the challenges presented by the observed system behavior as discussed above. This new component injects refrigerant into or removes refrigerant from the main vapor compression cycle to improve system-level energy efficiency by using two electronic expansion valves under active control to control the amount of refrigerant in a storage tank, thereby regulating the active charge circulating through the components of the main vapor compression cycle. Such a component can be viewed as adding an additional control variable to the cycle, similar to earlier efforts to replace components with fixed system parameters with components that use variable actuation to modulate and improve the system performance. This new component also has the benefit of being continuously coupled to the system to improve oil distribution issues and avoid connecting and disconnecting refrigerant hoses.

Other researchers have studied methods of adjusting the distribution of refrigerant mass in a cycle. Since the differences between the optimal system charge in heating and cooling modes are well known, prior researchers have proposed systems which trap a fixed volume of refrigerant in a tank with solenoid valves for the duration of the heating operation, and then release this refrigerant into the active cycle during cooling operation (Lifson et al., 2006). While this method

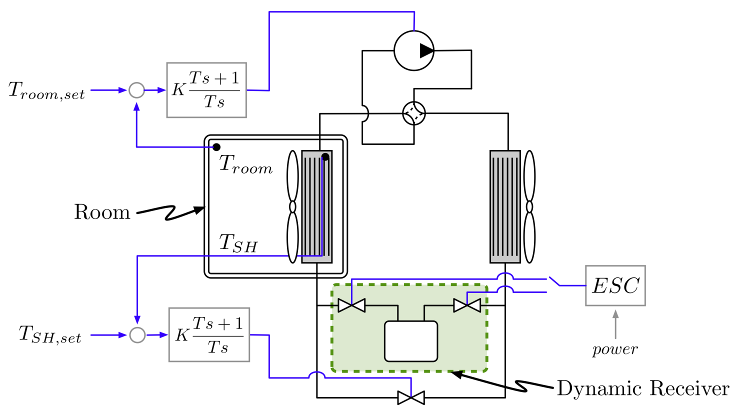

Figure 2: Architecture of proposed cycle for dynamically managing refrigerant charge.

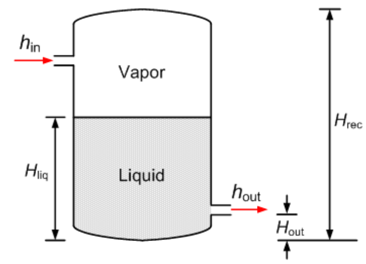

Figure 3: Schematic of receiver model.

does change the distribution of refrigerant, it does not do it in a very controlled manner; in addition, it is possible that refrigerant oil will accumulate in the sequestered volume over time, causing long-term lubrication problems. Yoo et al. (2014) also proposed a variable system for adjusting the refrigerant mass in the cycle, but this methodology essentially constructed a variable size receiver with discrete changes in the volume, rather than the capability of continuously varying the active cycle charge.

A vapor compression cycle in which the active refrigerant charge is managed via a dynamic receiver that is controlled by an optimizing control method will have a number of performance advantages over a conventional cycle. The most immediate of these is that the system COP will be improved over a wide range of operating conditions. In addition, a system with actively managed charge will have a greater robustness of its energy performance to installation and commissioning errors, and will relieve the burden on system installers due to the difficulty of calculating and charging the optimal amount of refrigerant for even one condition. Similarly, such a system could be installed in many different climate zones without tuning of refrigerant charge because the dynamic receiver will effectively provide a site-specific tuning of the charge for every installation. Finally, a system with an actively managed charge level could also relax some of the geometric constraints on system design, allowing development engineers to optimize the components under the assumption of different charge levels, rather than relying upon the fixed charge assumption.

In this paper, we describe and explore the operation of the proposed system through the use of extensive simulation using a set of dynamic models developed in Modelica (Modelica Association, 2014). In section 2, we describe the dynamic models of the basic vapor compression cycle; innovative modeling strategies were needed to specify the value of the total system charge for these models and maintain a constant refrigerant charge throughout the duration of these simulations. Following the exposition of the models, we describe the construction and operation of the dynamic receiver, as well as the development of the control and optimization method for the overall system. In section 3, we briefly present a set of results illustrating the performance of the proposed system in comparison to a basic cycle without the dynamic receiver for a reversible cycle operating in both cooling and heating modes, and then discuss conclusions and further work in the final section. A nomenclature section describing the variables used in this paper is also included after the last section as a reference.

2. CYCLE MODEL

Because the operation of the dynamic receiver is tightly coupled to that of the main vapor compression cycle, this paper first reviews and describes the behavior of the main vapor compression cycle (i.e., compressor, expansion valve, and two heat exchangers) before describing the behavior of the dynamic receiver and the structure of the control system. The control method is particularly important for the successful implementation of the overall system concept, as the refrigerant stream injected into the cycle from the dynamic receiver can potentially destabilize the system operation if the controller is improperly designed.

2.1 Basic Cycle Models

The models used to study the dynamics of the vapor compression cycle with an actively managed refrigerant charge must satisfy two main requirements: spatial and temporal fidelity for the behavior of the heat exchangers, and the ability to specify and maintain a fixed refrigerant inventory in the overall system. Finite volume models (Li et al., 2014) are used to describe the heat exchangers' behavior because they have the highest level of spatial and temporal fidelity among the popular modeling methodologies. Algebraic models of the compressor and expansion valve are also used because the time scales at which these components evolve are much faster than the dominant time constants for the system, which are governed by the heat exchangers. These different component models are integrated into a model of a vapor compression cycle, which is implemented as a spatially discretized set of differential algebraic equations (DAEs) requiring solution at each time step. The solution of this set of DAEs represents a more challenging problem than the solution of an analogous set of ordinary differential equations (ODEs), due to the fact that the state variables must not only be sufficiently well behaved to integrate forward in time, but must also satisfy a set of algebraic equations at each time step.

As the core of this work is focused on the behavior of the refrigerant mass in the cycle, we placed a particular emphasis on modeling the dynamics of the system charge. While the objectives of specifying and maintaining a constant refrigerant charge in the cycle are conceptually straightforward, these characteristics are not common among most extant dynamic models; the total system charge of many dynamic models of vapor compression cycles has been found to fluctuate over time (Cecchinato and Mancini, 2012). The work in this paper builds upon the authors' previous research on techniques to maintain a constant mass in the refrigerant cycle (Laughman and Qiao, 2015) by extending them to allow the simulation user to specify the total system charge, which is then maintained throughout the duration of the simulation. Because of the important role played by these modeling methodologies, we discuss these approaches after describing the heat exchanger models.

We employed a number of assumptions to simplify and expose relevant features of the dynamics of both refrigerant and air flow through channels. These assumptions include one dimensional fluid flow, no axial heat conduction in the direction of the fluid flow, a homogeneous flow field in the two-phase region, thermodynamic equilibrium in each control volume, dry air composition neglecting humidity, and negligible gravitational forces. These assumptions facilitated the discretization of the standard partial differential equations describing the conservation of mass, momentum, and energy. We use a staggered-grid discretization (Patankar, 1980) of these equations to avoid numerical instabilities, which can be represented as

$$\frac{d(\rho_j V_j)}{dt} = \dot{m}_k - \dot{m}_k \tag{1}$$

$$\frac{d(\dot{m}_i l)}{dt} = \rho_j v_j^2 A_j - \rho_{j+1} v_{j+1}^2 A_{j+1} + \frac{A_j + A_{j+1}}{2}(P_{j+1} - P_j) + F_{f,i} \tag{2}$$

$$\frac{\partial(\rho_j u_j A_j)}{\partial t} = \dot{H}_k - \dot{H}_k + v_j A_j (P_{j+1} - P_j) + v F_{f,i} + \dot{Q}_j, \tag{3}$$

where

$$\dot{H}_k = \dot{m}_k h_{upstream,j} \tag{4}$$

and the set of ODEs corresponds to the number of volumes used to subdivide the length of the refrigerant pipe. The indices in these equations refer to the grid under consideration; the i indices are referred to the momentum grid, the j indices are referred to the thermal grid, and the $k = j + 1$ indices refer to the boundaries of the thermal grid. Additional details can be found in Laughman et al. (2015).

While the above equations describe the basic conservation relations, additional closure equations are needed to describe both the heat transfer between the fluid and the tube wall and the frictional pressure losses within the fluid. Because of the nonlinear form of the usual experimentally-derived correlations and the fact these behaviors can often be modeled by simple relations for particular working fluids and geometries, we use simplified closure relations, e.g., constant phasic heat transfer and

$$\Delta P = K \frac{(\Delta P)_0}{\dot{m}_0^2} \dot{m}^2, \tag{5}$$

where the nominal values of K, $(\Delta P)_0$, and $\dot{m}_0$ are set by the simulation user as constants in the model.

Heat transfer through the tube wall is governed by

$$(M_w c_w + M_{fin} c_{fin}) \frac{dT_w}{dt} = \alpha_{ref} A(T_r - T_w) + \dot{m}_{air}\left[c_{p,air}(T_{air,in} - T_{air,out})\right]. \tag{6}$$

while the heat transfer equation on the air side is given by

$$\dot{m}_{air} c_{p,air} \frac{dT_{air}}{dy} \Delta y = \alpha_{air}(A_{o,tube} + \eta_{fin} A_{o,fin})(T_w - T_{air}), \tag{7}$$

where Δy is the discretized length of the tube in the direction of the air flow, η_{fin} is the fin efficiency, and the other symbols are given in the nomenclature section in the appendix. The fin efficiency is calculated using the equation proposed by Hong and Webb (1996).

In addition to these models of the conservation equations and the closure relations for the system, a set of equations of state for the fluid are also needed. The equations of state for a pure fluid enforce the relationships between the different thermodynamic properties of the fluid, and depend on at most two variables (Bejan, 2006). While there are many potential choices for these two variables, most conventional modeling approaches use pressure and mixture specific enthalpy to describe the fluid, since these variables are continuous throughout the entire phase space. Unfortunately, the large magnitude of the density derivatives in the region close to the saturated liquid line can cause numerical fluctuations in the system charge. This is evident by considering the following expression for the total mass of refrigerant of the system, written in terms of the state variables (p, h) and the numerical error ε that is introduced at every step by the numerical integration routine, such as DASSL or Radau IIa (Cellier and Kofman, 2006),

$$M_{total} = \sum_k \rho_k(P + \varepsilon, h + \varepsilon) V_k. \tag{8}$$

When the refrigerant state is within ε of the saturated liquid line, these integration errors can affect the total mass inventory because the error will result in the refrigerant density being calculated for the wrong side of the saturated liquid line. The large magnitude of these erroneous density derivatives will cause a small error in pressure and specific enthalpy to result in a large deviation for the total refrigerant mass.

We address this problem through the alternate selection of pressure and density as state variables, as this new choice of state variables allows the errors in the density to be directly controlled. Having switched the choice of state variables, the constitutive relations for the fluid become

$$\frac{dM}{dt} = V \frac{d\rho}{dt} \tag{9}$$

$$\frac{dU}{dt} = \frac{d\left(\rho u(P,\rho) V\right)}{dt} \tag{10}$$

$$= V\left[\left(\rho \left.\frac{\partial h}{\partial P}\right|_{\rho} - 1\right)\frac{dP}{dt} + \left(\left.\frac{\partial h}{\partial \rho}\right|_{P} + h\right)\frac{d\rho}{dt}\right]. \tag{11}$$

Laughman and Qiao (2015) demonstrated that this alternative choice of state variables effectively eliminates any variation in the system charge, though this choice of state variables can make the integration routine slow because the integrator needs to take small steps when the refrigerant state is close to the saturated liquid line. To speed up the integration method in these situations, we made a slight modification to the selection of state variables by expanding the number of state variables to include pressure, specific enthalpy, and density. While this approach does result in a larger number of state variables, it has the advantage of simultaneously minimizing the variations in system charge while enabling the use of P and h for calculating other refrigerant properties, thereby achieving a commensurate increase in speed over the (P, ρ) models. Such a method uses the same differential equations as the (P, ρ) model, but also includes the additional ODE

$$\frac{dh}{dt} = \left.\frac{\partial h}{\partial P}\right|_{\rho} \frac{dP}{dt} + \left.\frac{\partial h}{\partial \rho}\right|_{P} \frac{d\rho}{dt}. \tag{12}$$

While this approach in theory relaxes the constraint forcing the state variable being integrated to be equal to the computation of the specific enthalpy as a function of the other state variables, e.g., $\hat{h}(P, \rho)$, evidence provided in Laughman and Qiao (2015) suggests that these deviations are negligible in practice.

The above method is successful at maintaining the system charge at a given level, but does not allow the user of the simulation to directly set the initial mass of refrigerant in the system. Conventional approaches to cycle simulation for these system representations often require the user to specify the initial values for the state variables in each control volume, and then use a nonlinear equation solver to determine a consistent set of values for all of the other state variables to ensure that the set of algebraic equations for the DAE system is satisfied at $t = 0$. Within this framework, the total system charge is effectively specified when the user specifies the state variables for each of the control volumes because the density is a function of the other state variables. The problem of specifying the initial value of the cycle charge is thus equivalent to the problem of solving for the values of the state variables in the control volumes such that the sum of the products of the density and the control volume sizes are equal to the desired total system charge.

Because of the limitations of the existing initialization procedure, we developed an alternative approach to system initialization that facilitates the specification of a numerical value of the system charge within a defined range. The simulation user first specifies the pressures in each volume by directly prescribing the inlet and outlet pressures for each heat exchanger, after which a linear approximation to the pressure is used to determine the pressures in all volumes. This simulation user then specifies the inlet and outlet refrigerant enthalpies for the evaporating heat exchanger and again uses a linear approximation to determine the value of the specific enthalpy in each cell; this effectively specifies the amount of mass in the evaporating heat exchanger. Because the condensing heat exchanger will contain the largest amount of subcooled liquid, the mass in that heat exchanger is set to the difference between the desired system mass and the mass in the evaporator, and this remaining mass is evenly distributed between all of the control volumes and used to calculate the density in each control volume. While this will result in a somewhat nonphysical initial distribution of refrigerant mass, a simulation that conserves mass will only undergo an initial transient as this initial guess dies out, and the total system mass will remain constant.

An isenthalpic model of the expansion valve, with neither mass nor energy storage, is used to describe the expansion process. The mass flow rate through the expansion device is related to the valve orifice size θ, the inlet refrigerant density ρ_{in}, and the pressure drop across the valve $P_{in} - P_{out}$, e.g.,

$$\dot{m}_{EEV} = \theta\sqrt{\rho_{in}(P_{in} - P_{out})}. \tag{13}$$

A simple compressor model, also with no mass or energy storage, is also used. This model describes the mass flow rate of refrigerant through the compressor and the change in the specific enthalpy of the refrigerant as it travels from the suction port to the discharge port, e.g.,

$$\begin{aligned} \dot{m} &= \eta_v \rho_{suc} VN && (14)\\ \eta_v &= a_1 \frac{P_{dis}}{P_{suc}} + a_2 && (15)\\ h_{dis} &= h_{suc} + \frac{h_{dis,is} - h_{suc}}{\eta_{is}} && (16)\\ h_{dis,is} &= f(p_{suc}, h_{suc}), && (17) \end{aligned}$$

where the constants a_1 and a_2, as well as the maps for the volumetric efficiency η_v and the isentropic efficiency η_{is}, are determined through a compressor performance test and are built from polynomial expansions of the compressor speed and the pressure ratio.

A room model is also required to test the dynamics of the closed-loop heat pump system, since the system cannot be tested under load without the use of feedback controllers to regulate the room air temperature and other salient internal process variables, such as the evaporator superheat. A lumped adiabatic model is used to describe the room; the sensible heating or cooling capacity of the machine is matched to the sensible load imposed in the room in steady-state, while the controller regulates the transient behavior of the machine to achieve the specified process variable setpoints.

2.2 Dynamic Receiver

The dynamic receiver is primarily designed to regulate a mass of refrigerant, which is sequestered from the active cycle, to maximize the cycle COP. This component must be able to operate in both heating and cooling modes, and control the contained mass of refrigerant over a wide range of values in response to the cycle's operating conditions. While these requirements could be realized by a number of possible architectures, one attractive configuration is illustrated in Figure 2, which contains three main components: two electronic expansion valves and one storage tank. This architecture allows the pressure in the storage tank to be modulated between the condensing pressure and the evaporating pressure in both heating and cooling modes, and its flow-through design avoids the entrapment of oil in the storage tank. Each of these expansion valves, the model of which was identical to that stated in Equation 13, performs a distinct function: the EEV connected to the evaporating heat exchanger is used to regulate the flow of refrigerant into the system, while the EEV connected to the condensing heat exchanger is used to control the quality and flow rate of the refrigerant entering the storage tank. This ensures that the fluid in the storage tank is neither entirely in the low-density portion of the two-phase region, nor entirely liquid.

The dynamic model of the storage tank consists of a lumped control volume with an inlet and an outlet, as is illustrated in Figure 3. The refrigerant dynamics are governed by

$$\begin{aligned} V_{tank}\frac{d\bar{\rho}_{tank}}{dt} &= \dot{m}_{in} - \dot{m}_{out} && (18)\\ V_{tank}\left(\bar{\rho}_{tank}\frac{d\bar{h}_{tank}}{dt} - \frac{dp_{tank}}{dt}\right) &= \dot{m}_{in}\left(h_{in} - \bar{h}_{tank}\right) - \dot{m}_{out}\left(h_{out} - \bar{h}_{tank}\right), && (19)\end{aligned}$$

where the specific enthalpy of the outlet stream depends upon the state of the outlet port. In one case, in which the refrigerant state at the outlet port is a two-phase fluid, the enthalpy of the outlet stream is determined as

$$h_{out} = \begin{cases} h_f & \text{if } H_{liq} > H_{out} + d_{out} \\ h_g - \left(\dfrac{H_{liq} - H_{out}}{d_{out}}\right)\left(h_g - h_f\right) & \text{if } H_{out} + d_{out} \geq H_{liq} \geq H_{out} \\ h_g & \text{if } H_{liq} < H_{out} \end{cases} \tag{20}$$

where the liquid height is determined as

$$H_{liq} = \frac{\bar{\rho}_{tank} - \rho_g}{\rho_f - \rho_g} H_{tank}. \tag{21}$$

In the opposite case, in which the refrigerant state is in either the single-phase liquid or vapor regions, the enthalpy of the outlet stream should be equal to the mean enthalpy of the refrigerant in the storage tank, and the outlet mass flow rate is given by

$$\dot{m}_{out} = \frac{\sqrt{(p_{tank} - p_{out})}}{f_{tank}}. \tag{22}$$

2.3 Controllers

Because the behavior of the dynamic receiver is tightly coupled to the behavior of the main vapor compression cycle, proper controller design is essential to the successful operation of the complete system. In particular, the injection of the refrigerant stream from the dynamic receiver will act as a disturbance to the other control loops that regulate process variables. Different controllers regulate and optimize the behavior of the overall cycle, as is evident from Figure 2. Proportional-integral regulators are used to control the process variables of interest, and these feedback loops are tuned to improve their ability to reject disturbances; the fastest control loop uses the main EEV position to regulate the evaporator superheat error signal, while the next fastest control loop uses the compressor frequency to regulate the room temperature error signal.

In comparison, an extremum-seeking controller (ESC) is used to control the expansion valves in the dynamic receiver. While a PI controller was evaluated in the course of this work and found to be effective at regulating the refrigerant mass in the receiver to a fixed or time-varying setpoint, such an approach would be imprudent in practice because of the difficulty of measuring the mass of refrigerant in the storage tank, the difficulty of determining the optimal active charge for the system over the different installed system configurations and conditions, and the uncertainty in the value for the initial amount of refrigerant charged into the system. Extremum seeking control, in comparison, uses a model-free approach to determine the control inputs that achieve an extremum in a performance variable by estimating the gradient of the performance variable and driving it to zero. A time-varying extremum-seeking controller (Burns et al., 2015) is used to identify the position of the dynamic receiver EEV that is connected to the evaporating heat exchanger that minimizes the system's electrical power consumption, due to the fact that this algorithm converges to the optimum faster than standard perturbation-based extremum seeking algorithms.

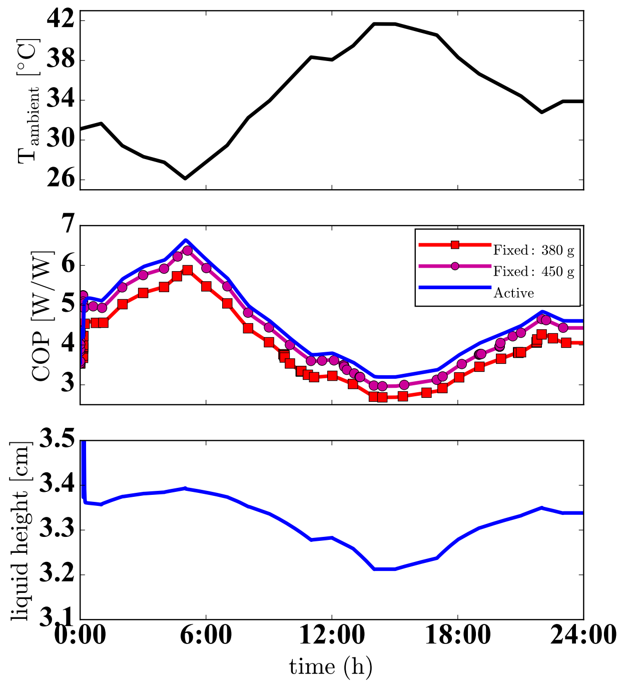

Figure 4: Mass control for the cooling cycle in Phoenix on 6/23/2013.

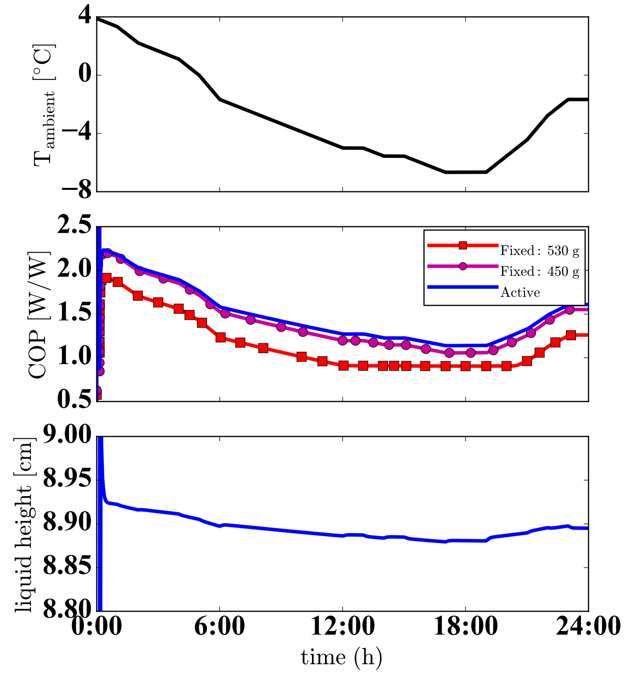

Figure 5: Mass control for the heating cycle in Washington, D.C. on 1/1/2014.

3. RESULTS

The models described in §2 were implemented in Modelica and used to evaluate the efficacy of the vapor compression cycle with dynamically managed charge. Simplified identical geometries were used for both of the heat exchangers in the system, which had two layers, four tubes per layer, and a total tube length of 11.2 meters. The refrigerant R410a was used as the working fluid, and the heat transfer coefficients were set to 500 W/m^{2}K, 2000 W/m^{2}K, and 700 W/m^{2}K for the vapor, two-phase, and liquid phases respectively. Initial simulations of the system were run in both cooling and heating modes with a load of +/- 2000 W; in cooling mode, the nominal ambient temperature was 35 °C and the indoor air temperature setpoint was 27 °C, while in heating mode, the nominal ambient temperature was 0 °C and the room air temperature was 23 °C. In all cases, the evaporator superheat setpoint was set equal to 2 °C. These nominal conditions were used to generate the data illustrated in Figure 1. These Modelica models were compiled using Dymola 2016, and executed on a computer that had an i7-6700K CPU with 32 G of RAM.

A PI controller was used to directly control the sequestered mass in the storage tank during the first set of simulations, the results of which are displayed in Figures 4 and 5, because it facilitated the assessment of the performance of the dynamic receiver in different heating and cooling environments without the added dynamics of the extremum-seeking controller. We chose a representative ambient temperature profile over one day for each mode, and simulated the performance of the system over that full day. Figure 4 illustrates the COP of the system under three different charge levels for the ambient temperature recorded on 23 June 2013 in Phoenix, AZ, USA. Two of these cycle simulations, corresponding to 380 g system charge and 450 g system charge, were performed on a cycle without the dynamic receiver; in comparison, the final cycle simulation, labeled "active," used the cycle with the dynamic receiver with the active charge setpoint for the PI controller set to the value of 530 g. This value of the optimal refrigerant charge was determined from the data illustrated in Figure 1. It is clear that the cycle with the dynamically managed charge has the highest COP of the three; the cycle with 450 g of charge has a COP that is is 5.3% lower than that of the charge management cycle, while the cycle with 380 g of charge has a COP that is 10.7% lower.

In addition, the bottom plot in Figure 4 illustrates the variation in the liquid height in the storage tank for the active cycle over the course of the day, demonstrating the fact that this new component can dynamically shift mass in and out of the main vapor compression cycle. It is notable that the height of the liquid in the tank varies over the course

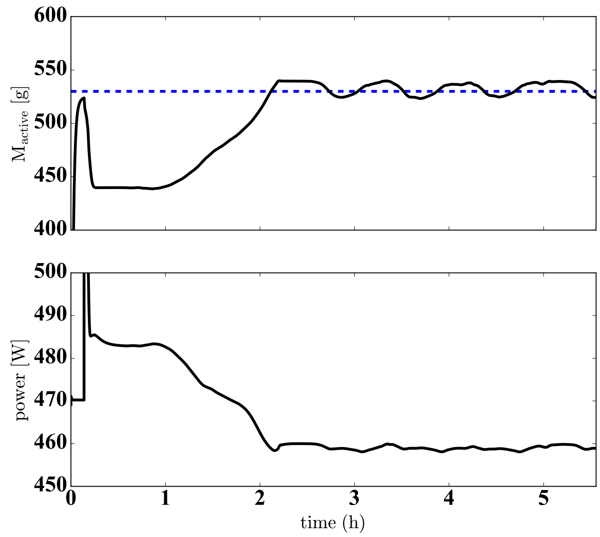

Figure 6: TV-ESC minimization of power by modulating the evaporator-side EEV in the dynamic receiver, thereby changing the amount of active charge in the cycle.

of the day though the mass of refrigerant in the dynamic receiver remained the same, because the density of the liquid experiences small changes as the ambient temperature varies. This suggests that simple control strategies that use liquid height as a surrogate for refrigerant mass may have errors from neglecting such density changes.

Similar results were obtained for the cycles in heating mode, seen in Figure 5, which used the ambient temperature recorded on 1 January 2014 in Washington, D.C., USA. In these simulations, the COP of the cycle with 450 g of charge is 4.4% lower than that of the dynamically managed charge, while the cycle with 530 g of refrigerant has a COP that is is 13.0% lower. These results are consistent with the intuition obtained from Figure 1, suggesting that the cycle with the dynamically managed charge experiences a substantial increase in energy efficiency.

The impracticality of making direct measurements of the refrigerant mass in the dynamic receiver also led us to conduct simulations of the dynamics of the system with the time-varying extremum seeking controller, as shown in Figure 6. In this final simulation, which ran the system in cooling mode with an ambient temperature of 35 °C, a room air temperature of 27 °C, and a total system charge of 650 g, the system is initialized with 200 g of refrigerant in the dynamic receiver. The TV-ESC algorithm was then turned on after 45 minutes to determine the optimal value of the refrigerant mass in the dynamic receiver by varying the orifice size of the low-side EEV. The data in both subplots of this figure make it clear that the TV-ESC method is able to find the minimum power consumption after about 75 minutes, as the power is reduced until it oscillates around a steady-state value and the active charge circulating through the cycle reaches 530 g.

4. CONCLUSIONS & FUTURE WORK

These initial results obtained by simulating the behavior of a modified vapor compression cycle, which incorporates new components to dynamically manage the amount of active refrigerant, suggest that appreciable performance gains might be attained by implementing such a system. Simulated performance gains suggest that the increase in COP may exceed 10% in some situations, providing ample incentive to continue these investigations. Such a system could also allow the installation of these systems in the field to be much more robust to installation, effectively enabling site-specific refrigerant charge tuning and maintaining the system's energy performance despite refrigerant leaks.

This work could be extended in a variety of directions to further understand the potential performance benefits of dynamically managing refrigerant charge. Further work in simulation could include work on multi-indoor unit systems with simultaneous heating and cooling, which may yield further increases in energy efficiency, while experimental and field testing of this and other related system architectures would doubtlessly lend new insights into the realized performance gains of such a system over more conventional approaches.

NOMENCLATURE

A	cross-sectional area	a_i	polynomial coefficients	θ	expansion valve orifice size
COP	coefficient of performance	c_p	specific heat capacity at constant pressure	κ	isothermal compressibility
	$= (\dot{Q}_c/\dot{W}$ or $\dot{Q}_h/\dot{W})$	h	specific enthalpy	v	specific volume
F_f	frictional pressure drop	$\bar{h}$	average specific enthalpy	ρ	density
H	height	l	control volume length	$\bar{\rho}$	average density
$\dot{H}$	total enthalpy transfer rate	$\dot{m}$	mass flow rate		
M	mass	q	heat flow input	**Subscript**	
N	compressor speed	t	time	f	liquid
P	pressure	u	specific internal energy	g	gas
$\dot{Q}$	heat transfer rate	v	velocity	dis	compressor discharge port
U	internal energy	α	heat transfer coefficient	liq	liquid
T	temperature	β	isobaric coefficient of expansion	suc	compressor suction port
V	volume	η	efficiency	is	isentropic
$\dot{W}$	compressor work				

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