2018

Model-Based Air Flow Path Optimization for Heat Pump Clothes Dryer

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ABSTRACT

A heat pump clothes dryer (HPCD) uses a vapor compression system to dry clothes. The condenser heats air, which passes through the drum to evaporate moisture out of the clothes, and the evaporator condenses water out of the air stream. As a result, the HPCD can achieve 50% energy savings compared to a conventional electric resistance dryer. In this work we developed a physics-based, quasi-steady-state HPCD system model with detailed heat exchanger and compressor models. The system model can simulate the inherently transient HPCD drying process, to size components, and to reveal trends in key variables (e.g. compressor discharge temperature, power consumption, required drying time, etc.) The system model was calibrated using experimental data from a prototype HPCD. Air leakages, in and out, along the closed air circulation path of HPCD cause varied effects on the performance. Understanding the location, magnitude, and direction of air leakage of the heat pump clothes dryer is critical for accurately characterizing the performance and developing a high-performance design. The system model was used to reveal the impacts. In addition, model-based parametric optimizations were conducted to design the HPCD charge inventory, flow rate, air path and leak points for optimum performance.

1. INTRODUCTION

In the United States, most residential clothes dryers use electric resistance heaters with a capacity of approximately 4 kW for clothes drying. US dryers typically use a tumble-type drum with a blower to push air and dry clothes. Most existing electric products are electric resistance with once-through airflow, with some condensing dryers using closed-loop airflow. Starting in late 2014, vapor-compression (VC) heat pump dryers have been available. A heat pump clothes dryer uses the evaporator to condense water and recovers condenser energy to heat clothes load. It doesn't require a venting duct through a building wall, also eliminating fire hazards from lint accumulation. The heat pump clothes dryer has the potential of lowering the energy consumption by 50% as compared with the conventional resistance heaters. These energy savings equate to 25.8 TWh per year when the new technology is fully deployed nationwide (Ling and Muehlbauer, 2013).

A schematic of a HPCD is shown in Figure 1, while detailed schematics and the operation principle are illustrated in Figure 2. A heat pump is a mechanical vapor compression refrigeration system consisting of primarily four main components, namely, an evaporator, a compressor, a condenser, and an expansion valve. The processed air is recirculated in the cycle continuously until drying is complete. The heated and dried air enters the clothes drum. It extracts the moisture from the wet clothes in the drum, where its temperature decreases and its relative humidity increases. At the drum exit, the air is almost saturated, at least in the initial stages of drying. The warm (i.e., relatively cooler) and moist air proceeds through the lint screen and flows over the evaporator of the vapor compression refrigeration system. Due to the low-temperature refrigerant flowing inside the evaporator coil, a significant amount of moisture from the air is condensed out and is collected in a tray, while the drier and cooler air is blown by the fan over the condenser of the vapor compression system. The drier air gets heated due to the hot refrigerant flowing inside the condenser and is fed back to the drum. Thus, the net effect is that moisture evaporated from the wet clothes is condensed at the evaporator. This cycle continues until full drying is accomplished.

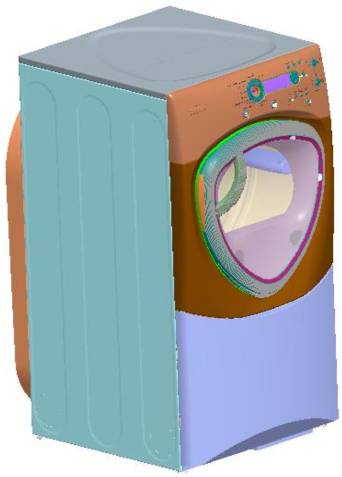

Figure 1. View of a heat pump dryer.

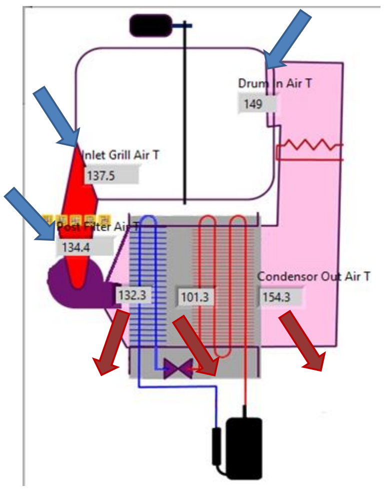

Figure 2. Schematics of a heat pump clothes dryer.

The HPCD is a closed system. Energy is added to the control volume as electric power to the compressor, circulation fan, and the drum rotator, and energy leaves the control volume by the condensate water, heat loss to the surrounding air, and air leakages in and out of the flow path. In the beginning of a drying process, the energy leaving the system is lower than the energy added. As a result, the internal air heats up, and the compressor suction and discharge pressures increase. Later in the drying process, as the drier clothes provide less of an evaporative cooling load in the drum, compressor suction and discharge pressures increase further. It is a critical design consideration to prevent overheating the compressor, i.e., limiting the compressor discharge temperature before the clothes are fully dry.

In addition, the energy exchange with the surrounding air along the closed air circulation path of HPCD (air leakages, in and out, as indicated by blue arrows in Figure 2) cause varied effects on the performance. Understanding the location, magnitude, and direction of air leakage of the heat pump clothes dryer is critical for accurately characterizing the performance and developing a high-performance design.

The standard metric for dryer performance in the United States is the energy factor, as defined in Eq. 1.

$$\textit{Energy Factor} = \frac{\textit{Weight of dry clothes}}{\textit{Energy consumed to dry them}} \tag{1}$$

Shen *et al*. (2016) developed a HPCD system design and simulation model, based on an existing, vapor compression system model, i.e. the ORNL Heat Pump Design Model (HPDM, Shen 2014). HPDM was modified to include HPCD features and components including the drum and air leakages. HPDM's existing segmented fin-and-tube evaporator and condenser models were used, along with a simplified compressor model based on constant volumetric and isentropic efficiencies.

2. MODELING METHODOLOGY

Compressor model: At present, no AHRI 10-coefficient compressor maps are available for compressors used in the HPCD application, because the evaporating and condensing temperatures in the dryer are higher than typical compressor maps provide. To overcome this difficulty, we used basic efficiencies to model the compressor, i.e. volumetric efficiency shown in Equation 2, and isentropic efficiency shown in Equation 3.

$$m_r = Volume_{displacement} \times Speed_{rotation} \times Density_{suction} \times \eta_{vol} \tag{2}$$

$$Power = m_r \times (H_{discharge,s} - H_{suction})/\eta_{isentropic} \tag{3}$$

Where m_r is compressor mass flow rate; $Power$ is compressor power; η_{vol} is compressor volumetric efficiency; $\eta_{isentropic}$ is compressor isentropic efficiency; $H_{suction}$ is compressor suction enthalpy; $H_{discharge,s}$ is an enthalpy obtained at the compressor discharge pressure and suction entropy. We assumed a constant isentropic efficiency of 55% and volumetric efficiency of 90%.

Segment-to-segment fin-and-tube condenser model: A segment-to-segment modeling approach is used. Each tube segment has individual air side and refrigerant side entering states and considers possible phase transition. An ε-NTU approach is used for heat transfer calculations within each segment. The air-side fin is simplified as an equivalent annular fin. Both refrigerant and air-side heat transfer and pressure drop are considered.

Segment-to-segment fin-and-tube evaporator model: In addition to the functionalities of the segment-to-segment fin-tube condenser, the evaporator model is capable of simulating the dehumidification process. The method of Braun *et al*. (1989) is used to simulate cases of water condensing on an evaporating coil, where the driving potential for heat and mass transfer is the difference between enthalpies of the inlet air and saturated air at the refrigerant temperature.

Clothes drum model: The drying process is inherently transient as the clothes dry over time. We simulated the process as a quasi-steady-state process by assuming the vapor compression system reaches steady state at each individual time step. The heat and mass transfer process in the drum is described below [Braun *et al*. (1989)]. This approach for dryer drum modeling was also detailed in Shen *et al*. (2016).

$$\omega_{out,i} = \omega_{s,i} - \left(\omega_{s,i} - \omega_{in,i}\right) \times (1.0 - E_M), \tag{4}$$

$$T_{out,i} = T_{s,i} - \left(T_{s,i} - T_{in,i}\right) \times (1.0 - E_H), \tag{5}$$

$$Q_i = m_{air,circ} \times \left(H_{out,i} - H_{in,i}\right), \text{and} \tag{6}$$

$$WaterFlow_i = m_{air,circ} \times \left(\omega_{out,i} - \omega_{in,i}\right), \tag{7}$$

Where $\omega_{in,i}$ and $\omega_{out,i}$ are the air specific humidity entering and leaving the drum at moment *i* [lbm H_2O/lbm dry air]. $T_{in,i}$ and $T_{out,i}$ are the air temperatures entering and leaving the drum at moment *i* [°F], $T_{s,i}$ is the surface temperature of the clothes load in the drum at moment *i* [°F], and the clothes load is assumed to have a uniform temperature at each moment. $\omega_{s,i}$ is the specific humidity of the saturated air at the surface temperature of $T_{s,i}$ [lbm H_2O/lbm dry air], Q_i is the total heat and mass transfer rate at moment *i* [Btu/hr], and $WaterFlow_i$ is the water rate pick up by the air stream at moment *i*. E_M and E_H are the mass and heat transfer effectiveness, respectively. As a simplification, it is assumed that E_M is approximately equal to E_H. Using this assumption, effectiveness was obtained from laboratory measured data, specific to a particular drum. The heat and mass transfer effectiveness can be curve-fitted as a function of RMC, defined as the remaining water weight per unit dry cloth weight. The RMC was known based on measurements of a high-precision whole-dryer scale, and effectiveness was calculated based on measurements of drum inlet and outlet temperatures, with the assumption that $E_M = E_H$.

To calculate the energy balance between the clothes load and air Stream, Q_i is the energy rate carried away by the air stream in each moment; therefore, the remaining internal energy in the load is

$$\begin{aligned} &M_{clothes} \times U_{clothes,i} + M_{water,i} \times U_{water,i} \\ &\quad = M_{clothes} \times U_{clothes,i+1} + M_{drum} \times U_{drum} + M_{water,i+1} \times U_{water,i+1} + Q_i \times \Delta Time\,, \end{aligned} \tag{8}$$

where $M_{clothes}$ is the thermal inertia mass [lbm] of the clothes bone dry. $M_{drum} \times U_{drum}$ is the enery change due to the thermal mass of the drum metal and other hardware. $M_{water,i}$ and $M_{water,i+1}$ are the water weights in the load, at moment *i*, and the next moment of *i*+1 [lbm], $\Delta Time$ is the time step between the moment *i* and *i*+1 [h], $U_{clothes,i}$ = 0.32*$T_{s,i}$ is the internal energy of the clothes at moment i [Btu/lbm], and 0.32 [Btu/R/lbm] is the specific heat of the clothes load, i.e., assuming it is cotton. $U_{water,i}$ is the water internal energy at moment i [Btu/lbm], which is a function of the load surface temperature $T_{s,i}$, $M_{water,i}$ and $M_{water,i+1}$ are the water weight at moment *i* and moment *i*+1 [lbm], respectively. The relationship between them is given as

$$M_{water,i} = M_{water,i+1} + WaterFlow_i \times \Delta Time. \tag{9}$$

Air leakage model. This paper aims to correlate the effect of air-side leakages in the system modeling and calibration. Overall dryer leakage was quantified experimentally by pressurizing the dryer system and measuring the air flow rate required to maintain that pressurization. Leakages at the various individual locations shown in Figure 2 were quantified experimentally by selectively sealing subsections of the system before pressurization. Then, the measured air leakage rates in and out of the closed HPCD system were inputted to the system model. The surrounding air was set at 75°F temperature and 50% relative humidity.

A time step is determined to reach a temperature increment in the clothes load (e.g., 0.1 K). The transient element (the drum model) updates the boundary condition to the steady-state heat pump system and drives a new vapor compression system balance state at each time step. The quasi-steady-state simulation is conducted as follows (Figure 3). During the quasi-steady-state simulation process, the condenser exit subcooling and compressor suction superheat are approximated as constants.

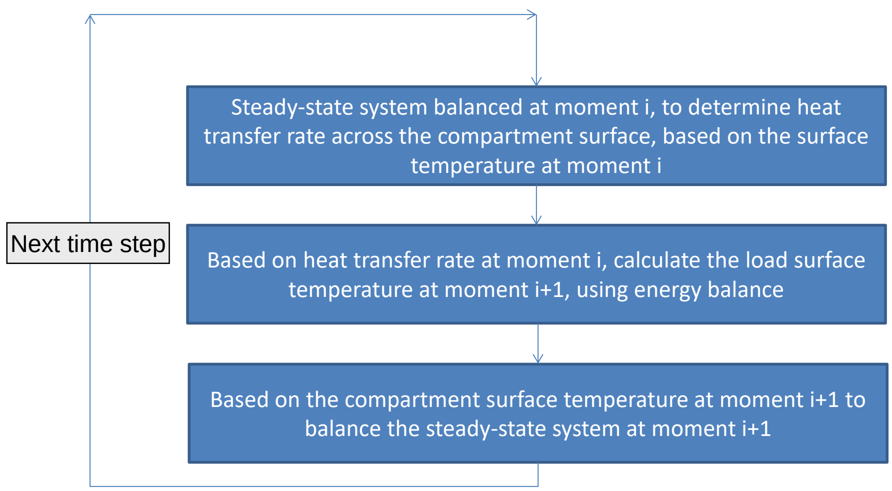

Figure 3. Quasi-steady-state time step.

Air leakages are considered based on mass and energy balances. The total air mass flow rate leaked out equates the total flow rate leaked in. If air leaks out of a location, the system model simply subtracts the amount out of the circulation, while the air temperature and humidity at the point are intact. Where air leaks in, the model adds the amount into the total flow rate, and the mixed air temperature and humidity are calculated based on energy and humidity balances.

3. MODEL VALIDATION

From testing a prototype HPCD, having R-134a refrigerant and a single-speed, rotary compressor, we obtained the component information. We assumed that heat transfer effectiveness is equal to mass transfer effectiveness. Using the experimental data, we first calculated the drum heat and mass transfer effectiveness as a function of the remaining moisture content (RMC) (Figure 4). In the figures plotted with respect to RMC, time proceeds from right (high RMC) to left (low RMC). It should be noted, there is a sharp drop in heat and mass transfer effectiveness, at the end of the drying process, when the RMC is lower than 20%.

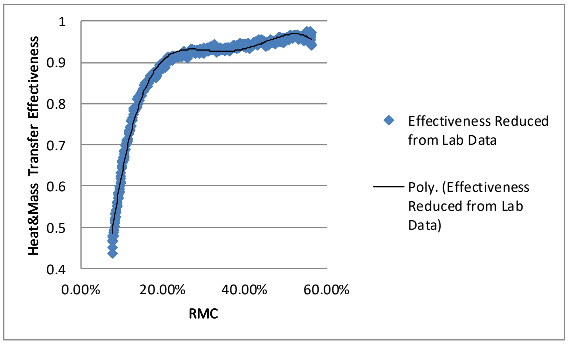

Figure 4. Heat and mass effectiveness of second generation drum.

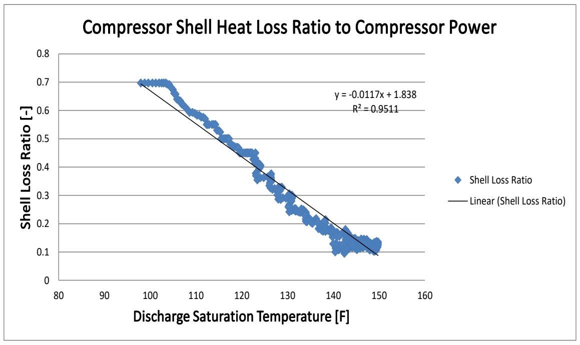

Figure 5. Compressor shell-heat loss ratio to compressor power.

The compressor shell heat loss ratio is defined as the amount of heat lost from, and stored in, the compressor shell divided by the electrical consumption of the compressor. Figure 5 depicts the compressor shell-heat loss ratio relative to the compressor power as a function of the compressor discharge saturation temperature to account for the energy balance across the compressor. The compressor's metal parts tend to reserve more heat at the startup at low-saturation temperature, and thus, the shell-heat loss ratio is larger at the beginning. It reached to 10% shell loss when the system approached to the final state of a drying process.

We set up the HPCD system model and calibrated the air-side heat transfer multipliers of the evaporator and condenser to match near-steady-state suction and discharge pressures. Based on the calibrated system model, we were able to predict the transient clothes drying process. Figures 6 and 7 compare predicted and measured evaporator inlet air temperature and humidity. The results demonstrate that the HPCD design model achieved model validation of 3.6°R or better for the key psychrometric state point temperature and 5% or better for humidity ratio. In the drying process, from high RMC to low RMC, the evaporator inlet relative humidity stays constant with the RMC above 20%, however, it has a drastic drop at the end of the drying process, driven by the degraded heat and mass transfer effectiveness illustrated in Figure 4.

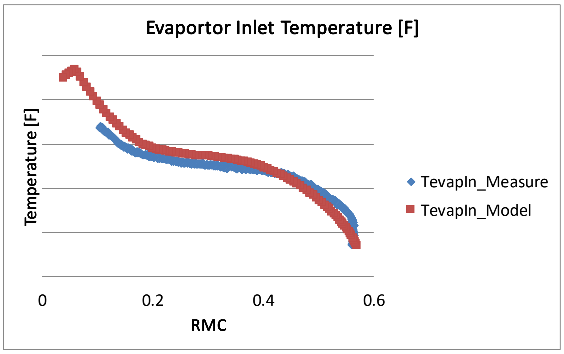

Figure 6. Measured and predicted evaporator inlet air temperatures.

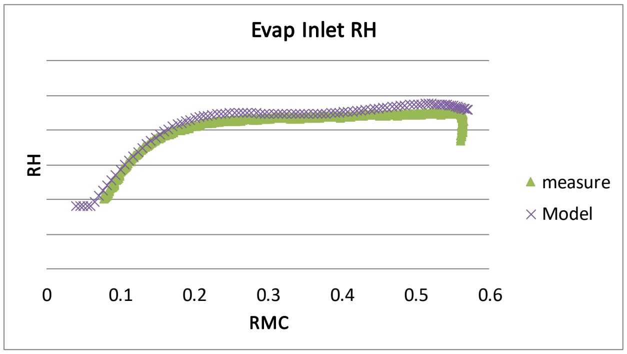

Figure 7. Measured and predicted evaporator inlet air relative humidity.

As shown in Figures 8 and 9, the predicted compressor suction and discharge pressure match the measurements in both the numbers and trends, respectively. It should be mentioned that the system pressures first increase as the system warms up and more moisture joins the air circulation. At the end of the drying process, as the RMC drops below 20%, which decreases the air humidity entering the evaporator coil. As a result, the suction and discharge pressure drop, and the HPCD model is able to capture this trend.

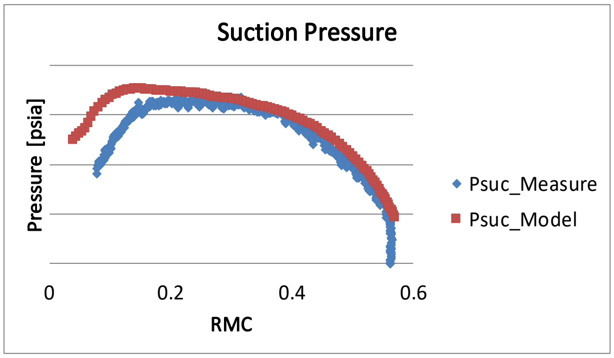

Figure 8. Measured and predicted suction pressures.

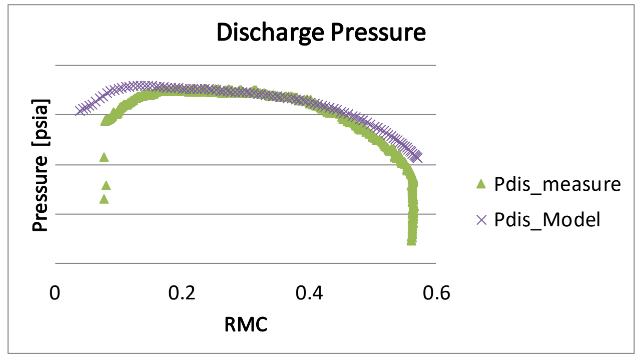

Figure 9. Measured and predicted discharge pressures.

Via the accurate refrigerant-state point predictions, the model predicted the compressor power consumption accurately, as indicated in Figure 10. Figure 11 compares the condenser outlet air temperatures. The compressor power also drops with the RMC below 20%, because of the decreased suction and discharge pressures. The less power input to the closed system decreases the condenser air exit temperature.

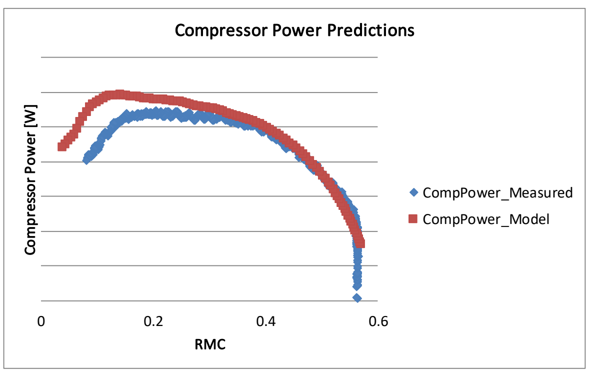

Figure 10. Measured and predicted compressor powers.

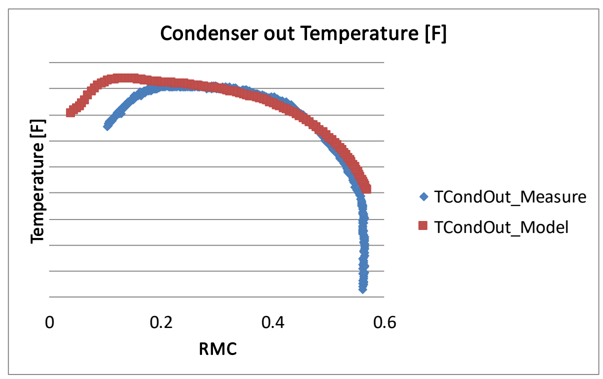

Figure 11. Measured and predicted condenser exit air temperatures.

4. MODEL-BASED OPTIMIZATION

In order to identify the best combination of airflow rate and condenser leaving refrigerant subcooling (indicative of system refrigerant charge mass), we ran a parametric study to simulate the EF as a function of the airflow rate in CFM and subcooling degree, as shown in the contour plot in Figure 12. Additionally, Figure 13 depicts the resultant total drying time.

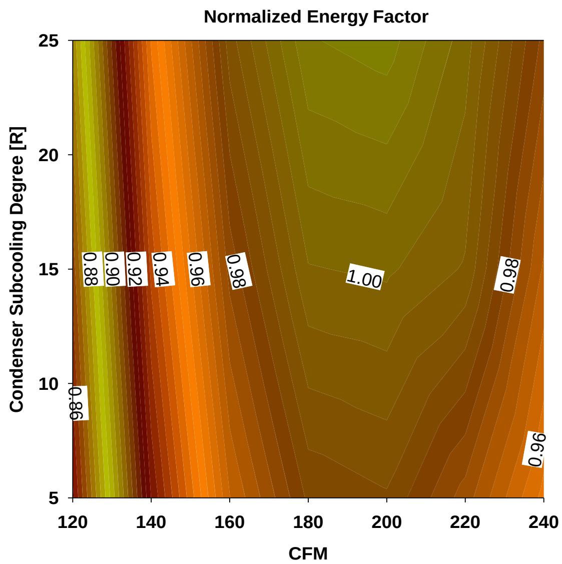

Figure 12. EF changing with airflow rate and condenser subcooling degree.

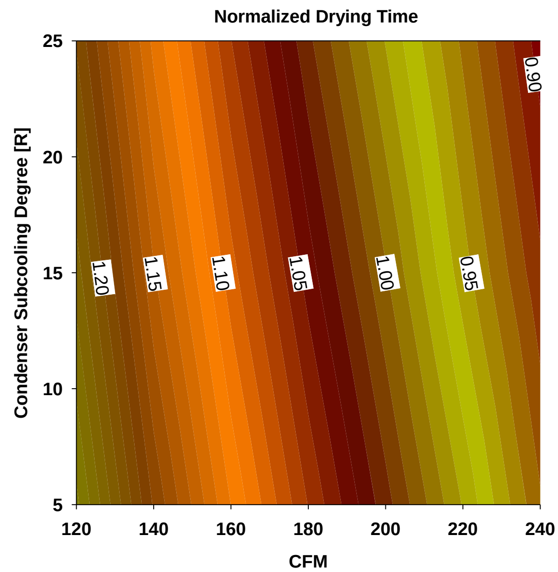

Figure 13. Drying time changing with airflow rate and condenser subcooling degree.

When varying the airflow rate, it was assumed that fan power = fan coefficient * CFM^3, where the fan coefficient was obtained from a baseline single-speed blower. As shown in Figure 12, the airflow rate has a major impact on the EF, with the optimum point from 180 to 200 CFM (note that blower power is neglected in this figure, meaning the actual optimum point is expected to be at slightly lower CFM). The condenser subcooling degree (a proxy for system refrigerant charge mass, with higher charge correlating to greater subcooling) has a secondary impact. Increasing the system charge elevates the condensing temperature and air temperature out of the condenser and boosts the evaporator cooling capacity. This adds more heat to the closed system and accelerates the drying process. However, higher charge leads to larger compressor power consumption. There is a trade-off in adding the system charge. In addition, higher system charge tends to trip the compressor quicker. Figure 13 illustrates that higher air flow rate and condenser subcooling degree both result in shorter drying time.

5. EFFECT OF AIR LEAKAGES AT VARIOUS LOCATIONS

Aiming to identify the effects of the locations and amount for air leaking in and out of the HPCD system, we ran a parametric study to identify optimum leak points, which include

- three leak-outs: LEAKFANOUT (after fan), LEAKEVAPOUT (after evaporator), LEAKCONDOUT (after condenser). These are also referred to as factors A, B and C.
- two leak-ins: LEAKDOWNIN (leak in downstream of the drum), and LEAKUPIN (leak in upstream the drum). These are also referred to as factors E and F.
- the total leak rate, referred to as factor D, adjusts all leakages up and down together in equal proportion

We simulated three total leaked flow rates, 10 CFM total, 20 CFM total, 40 CFM total.

- Three leak-out ratios: 10%, 50%, and 90% of the total leaked rate for each of the three leak-out points
- Three leak-in ratios: 10%, 50% and 90% for the downstream-drum leak-in.

Based on simulation results for a full factorial test matrix involving the six factors A to F, the main effects are plotted in Figures 14 for the predicted energy factor and Figure 15 for the total drying time, using Pareto charts of the standardized effects. Only factors D and E were found to have statistically significant impact.

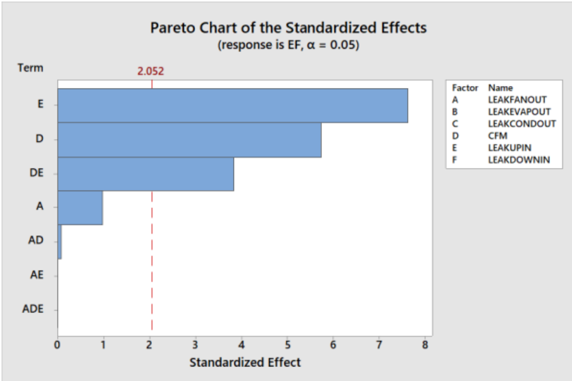

Figure 14: Main effect plot of air leakages on energy factor.

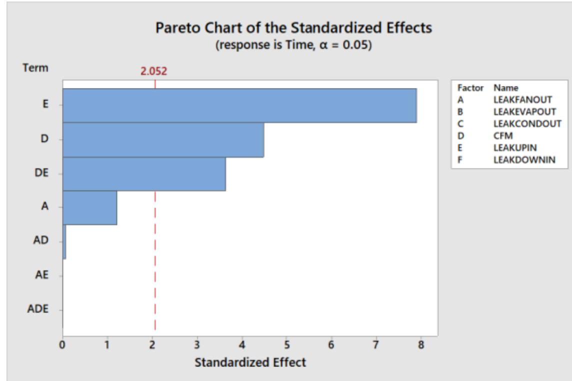

Figure 15: Main effect plot of air leakages on total drying time.

It can be seen that the total leaked CFM (Factor D) and the percentage of air leaking in upstream of the drum (Factor E) have the most significant effects on the energy factor and total drying time. On other hand, where and how much the processed air leaked out of the HPCD system are relatively less important.

Figures 16 and 17 show box plots of EF and total drying time at the three levels of air entering the drum inlet and the total air leakage rates, for the full factorial simulation test matrix. EF and dry time have been normalized against a baseline case. From Figure 16, it is clear that increasing surrounding air entering upstream of the drum from 0.1 to 0.9 leads to higher EFs and shorter drying time. Figure 17 shows that increasing the total leakage rate from 10 to 20 CFM results in higher EF because more relatively dry air is introduced into the system. However, increasing further to 40 CFM complicates the picture, with both the highest and lowest EF cases observed. Increasing total leakage rate overall tends to penalize the total drying time, though the maximum and minimum value ranges both expand.

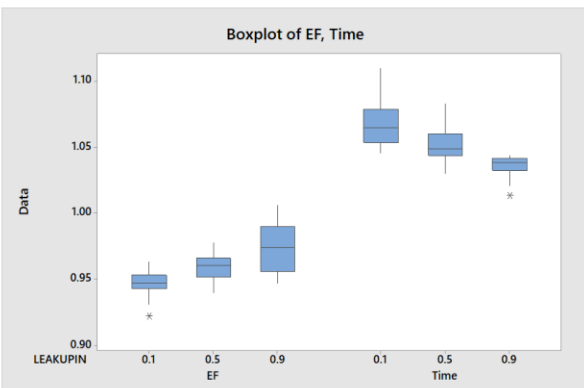

Figure 16: Box plot of EF and drying time impacted by percentage of the air leaking in upstream of the drum.

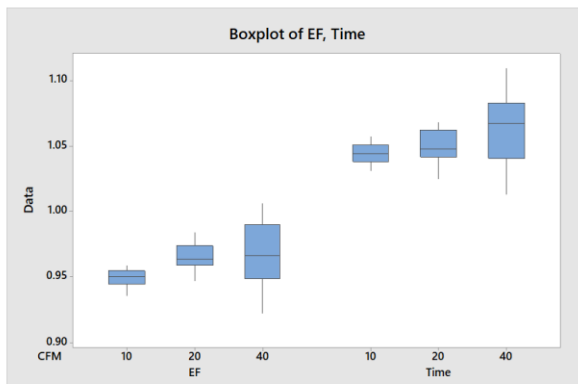

Figure 17: Box plot of EF and drying time impacted by the total air leakage rate.

6. CONCLUSIONS

We developed a quasi-steady-state HPCD system simulation model, which can accurately predict trends of the air side and refrigerant side state points, compressor power consumption, and required clothing drying time. A clothes drum model was developed as the major transient component during the drying process. The drum model is based on a heat and mass transfer effectiveness method, and the effectiveness was obtained as an empirical function depending on a key process variable (RMC).

The system model is able to locate optimum air flow rate and system charge level. Additionally, we investigated the effects of air leakage amount and locations along the closed circulation path. It is found that, for this particular dryer configuration with a closed air loop and negatively pressurized drum, the ratio of the leakage air that enters at the drum inlet has the most significant impact on the performance, and the total leakage rate has a secondary effect. On the other hand, where air leaks out of the system is not important.

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ACKNOWLEDGEMENT

The authors thank Mr. Antonio Bouza, Technology Development Manager for HVAC, WH, and Appliances, Emerging Technologies Program, Buildings Technology Office at the U.S. Department of Energy for supporting this research project.