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ABSTRACT

A heat pump clothes dryer (HPCD) is an innovative appliance that uses a vapor compression system to dry clothes. Air circulates in a closed loop through the drum, so no vent is required. The condenser heats air 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. We developed a physics-based, quasi-steady-state HPCD system model with detailed heat exchanger and compressor models. In a novel approach, we applied a heat and mass transfer effectiveness model to simulate the drying process of the clothes load in the drum. The system model is able to 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 on a prototype HPCD. In the paper, the modeling method is introduced, and the model predictions are compared with experimental data measured on a prototype HPCD.

1. INTRODUCTION

Approximately 85% of US households have a clothes dryer and 6-7 million new dryers are sold every year. However, the majority of clothes dryers (about 80%) in the US use electric resistance heaters of approximately 5 kW heating capacity that provide heat. A typical electric dryer is estimated to use about 718 kWh per year. These dryers consume about 71 TWh/year in the US or approximately 4% of the total annual residential electricity use [EIA 2010].

Heat pump clothes dryers innovatively apply vapor compression systems in the clothes drying application. 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 heat pump clothes dryer 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 (7), a compressor (4), a condenser (5) and an expansion valve (9). The processed air is re-circulated in the cycle continuously until drying is complete. Following Figure 2, the heated and dried air enters the clothes drum at state 1. It extracts the moisture from the wet clothes in the drum at state 2, where its temperature decreases and its relative humidity increases. At the drum exit at state 3, 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 at state 7. Due to the low temperature refrigerant flowing inside the evaporator coil, a significant amount of moisture from the air is condensed out at state 8 and is collected in a tray, while the drier and cooler air is blown by the fan at state 6 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 at state 1. 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.

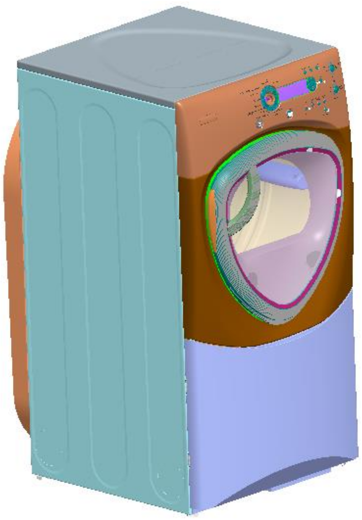

Figure 1: View of a heat pump dryer

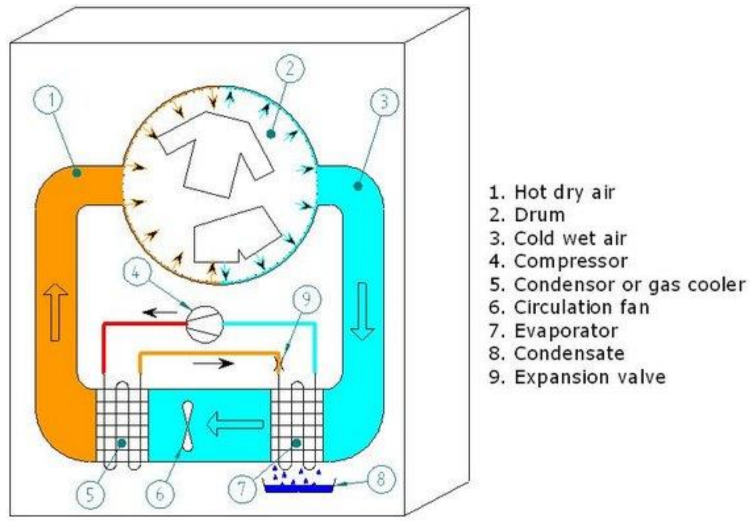

Figure 2: Schematics of a heat pump clothes dryer

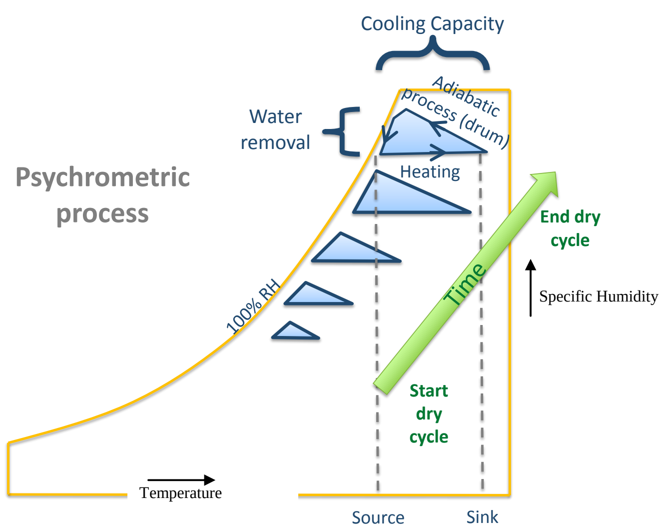

Figure 3: Transient clothes drying process in a psychrometric chart

Clothes drying in a HPCD is a transient process. In one complete air flow path, the air stream circulates through the evaporator, condenser, circulation fan, drum and ducts. In the condenser, the air temperature is increased and the specific humidity is unchanged; after that the air passes the drum to pick up moisture. Due to the evaporative cooling effect, the air temperature decreases through the drum. 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. As shown in Figure 3, air circulation starts at a low temperature and humidity, and ends at a high temperature and humidity. It is a critical design consideration to prevent overheating the compressor, i.e. limiting the compressor discharge temperature before the clothes are fully dry.

The standard metric for dryer performance in the US is the energy factor, as defined in equation (1).

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

The minimum energy factor (EF) of an electric dryer required by the 2015 DOE minimum efficiency standard is 3.73 lb/kWh. Regarding an efficient HPCD, its EF is expected to be higher than 6.0. During a standard EF test, the initial moisture content (MC, defined as the ratio of water weight divided by dry clothes weight) is 57.5%, and the final MC is 4%. The total dry timing is also an important design target.

It can be seen from above, that designing a HPCD involves complicated physics, i.e. sizing heat exchangers, air flow rate and compressor for better efficiency and lower cost, predicting EF and total drying time, and estimating the maximum compressor discharge temperature in the transient drying process. Ling (2013) used CoilDesigner to design heat exchangers for a two-stage HPCD, which is a good example for modeling the HPCD at the component level. However, a complete HPCD system model, able to integrate all the components and simulate the transient process, is still absent. The paper introduces the development of a first-in-the-kind, quasi-steady-state HPCD system model based on detailed, hardware-based component information and first-principle.

2. MODEL DESCRIPTION

Development of the HPCD system model was implemented in an existing, steady-state vapor compression system model, i.e. ORNL Heat Pump Design Model (HPDM), with the addition of new HPCD features and components of drum, duct heat loss. HPDM was improved to simulate the quasi-steady-state process, by assuming the vapor compression system reaches steady state at each individual time step (time step is determined to reach a temperature increment in the clothes load, for example, 0.1K). The transient element (the drum model) updates the boundary condition to the steady-state heat pump system, and drives a new system balance state at each time step.

2.1 ORNL Heat Pump Design Model (HPDM)

ORNL Heat Pump Design Model (HPDM, Rice 1997 and Shen 2014) is a well-recognized, public-domain HVAC equipment modelling and design tool. It has a free web interface to support public use, which has been accessed over 300,000 times by US and worldwide engineers. Some features of the HPDM, related to this study, are introduced below:

Compressor model: At present, no AHRI 10-coefficient compressor maps are available for compressors used in the HPCD application, because they work at very high evaporating and condensing temperatures, which go beyond typical compressor map conditions. 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.

Heat Exchanger models:

Segment-to-segment fin-&-tube condenser: It uses a segment-to-segment modeling approach, which divides a single tube to numerous mini segments; 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. Air-side fin is simplified as an equivalent annular fin. Both refrigerant and air-side heat transfer and pressure drop are considered; the coil model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant side entering and exit states, misdistribution, and accept two-dimensional air side temperature, humidity and velocity local inputs; the tube circuitry and 2-D boundary conditions are provided by an input file. The flow-pattern-dependent heat transfer correlation published by Thome (2003a, b) is used to calculate the condenser two-phase transfer coefficient. The pressure drop correlation published by Kedzierski (1999) is used to model the two-phase pressure drop.

Segment-to-segment fin-&-tube evaporator: In addition to the functionalities of the segment-to-segment fin-tube condenser, the evaporator model is capable of simulating 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. The flow-pattern-dependent heat transfer correlation published by Thome (2002) is used to calculate the evaporator two-phase transfer coefficient. The pressure drop correlation published by Kedzierski (1999) is used to model the two-phase pressure drop

Expansion Devices: The compressor suction superheat degree and condenser subcooling degree are explicitly specified. As such, the expansion device is not solved here – a simple assumption of constant enthalpy process is assumed.

Fans and Blowers: Single-speed fan: the air flow rate and power consumption were direct inputs from the laboratory measurements.

Refrigerant Lines: Heat transfer in a refrigerant line is ignored and the pressure drop is calculated using a turbulent flow model, as a function of the refrigerant mass flux.

Refrigerant Properties: Interface to REFPROP 9.1: (Lemmon et al., 2010) We programmed interface functions to call REFPROP 9.1 directly; our models accept all the refrigerant types in the REFPROP 9.1 database, and also we can simulate a new refrigerant by making the refrigerant definition file according to the REFPROP 9.1 format. REFPROP 9.1 runs fairly slow. To speed up the calculation, we have an option to generate property look-up tables, based on REFPROP 9.1; our program uses 1-D and 2-D cubic spline interpolation algorithms to calculate refrigerant properties via reading the look-up tables, this would greatly boost the calculation speed, given the same accuracy;

2.2 Transient Heat and Mass Transfer Process in the Drum

Heat and mass transfer in the drum is the major transient process that has been modeled. The heat and mass transfer process is described below (reference: Braun et al. (1989))

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

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

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

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

Where:

$\omega_{in,i}$ and $\omega_{out,i}$ are the air specific humidity entering and leaving the drum at moment i, [lbm H2O/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 H2O/lbm dry air].

Q_i is the total heat and mass transfer rate at moment i, [Btu/hr].

$WaterFlow_i$ 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, circulation air flow rate, and the standard clothes load. The figure below shows the heat and mass transfer effectiveness as a function of remaining moisture content (RMC), defined as 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 $E_M = E_H$. It can be seen in Figure 4 that the effectiveness increases almost linearly with the RMC. The strong dependence of the effectiveness on the RMC indicates that the equations 4 to 7 capture the major physics.

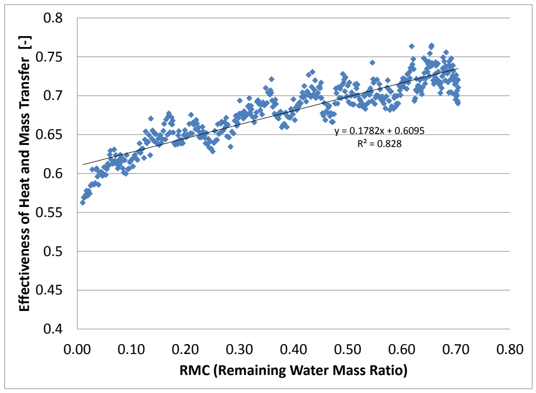

Figure 4: Effectiveness of heat & mass transfer as a function of RMC

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,

$$M_{clothes} \times U_{clothes,i} + M_{water,i} \times U_{water,i} = M_{clothes} \times U_{clothes,i+1} + M_{water,i+1} \times U_{water,i+1} + Q_i \times \Delta Time \quad (8)$$

Where:

$M_{clothes}$ is the thermal inertia mass, [lbm], including clothes, drum metal, and other parts, etc.

$M_{water,i}$ 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 as 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], the relationship between them is given as below:

$$M_{water,i} = M_{water,i+1} + WaterFlow_i \times \Delta Time \quad (9)$$

2.3 Heat Loss

The heat losses upstream and downstream of the drum are modeled using simple effectiveness method, i.e.

$$Heat\ Loss = E_{loss} \times m_{air,circ} \times Cp_{air} \times (T_{air,in,circ} - T_{air,enviro,circ}) \quad (10)$$

Where:

$m_{air,circ}$ is the dry air circulation mass flow rate [lbm Dry Air/hr]

Cp_{air} is the air specific heat [Btu/R/lbm Dry Air]

$T_{air,enviro,circ}$ is the surrounding air temperature, which is assumed the same as the indoor temperature, i.e. 70°F.

$T_{air,in,circ}$ is the air temperature entering a heat loss section, i.e. upstream or downstream of the drum.

E_{loss} is the heat transfer effectiveness.

2.4 Quasi-Steady-State Simulation Steps

The quasi-steady-state simulation is conducted following the steps below:

Step 1: at the beginning of one time step, i.e. moment i, the vapor compression system model calculates the steady-state refrigerant side state points based on the evaporator and condenser air flow rate, entering air temperatures and humidities. In addition, it calculates the water condensation rate and the air state exiting the condenser.

Step 2: Before entering the drum, the air stream picks up additional heat from the circulation fan and the drum rotator, while losing some heat to the environment. The energy transfers are treated as steady-state processes.

Step 3: During the heat and mass transfer process in the drum, the warm air flows over the clothes load with the surface temperature at moment i. Using equations 4 to 7, the heat transfer and water evaporation rates, and the air outlet state are calculated. By multiplying the heat transfer rate and water evaporation rate by the time step duration, the moisture loss and internal energy change in the clothes load are determined. The internal energy and thermal mass changes lead to a new load surface temperature at moment i+1, as shown in the equations 8 to 9.

Step 4: the air stream flows out of the drum, and the air temperature decreases due to the heat loss before entering the evaporator. The updated air temperature, humidity and air flow rate entering the evaporator are the new air side boundary conditions used for moment i+1, and the simulation goes back to step 1 with incrementing one moment.

3. MODEL VALIDATION

For validating the compressor component model, described in the equations 2 to 3, the compressor isentropic and volumetric efficiencies were reduced from a set of experimental data as a function of the HPCD running time in Figure 5. It can be seen that the isentropic efficiency is approximately equal to 60%, and the volumetric efficiency decreases from 90% at the beginning to 83% at the end. Those variations are not significant, and thus, it indicates the assumption of constant compressor efficiencies is acceptable. Figure 5 also plots the compressor shell loss ratio, (compressor power – energy added to the refrigerant flow in and out of the compressor)/compressor power, as a function of the running time. The compressor shell loss ratio starts at 70%, and decreases towards 30% at the end. This means, at the beginning, there were two parts of heat losses. One part was absorbed by the compressor's metal parts, and the other part was lost to the air surrounding the compressor. When approaching to the end, the compressor suction and discharge states changed less, and the major heat loss was due to the heat transfer from the compressor shell to the surrounding air. To improve the prediction accuracy compared with assuming constant compressor efficiencies and shell loss ratio, an empirical correlation was used, i.e. curve-fitting the efficiencies as a function of the key process parameter of RMC, and determining a key moment for the compressor reaching a constant shell loss ratio.

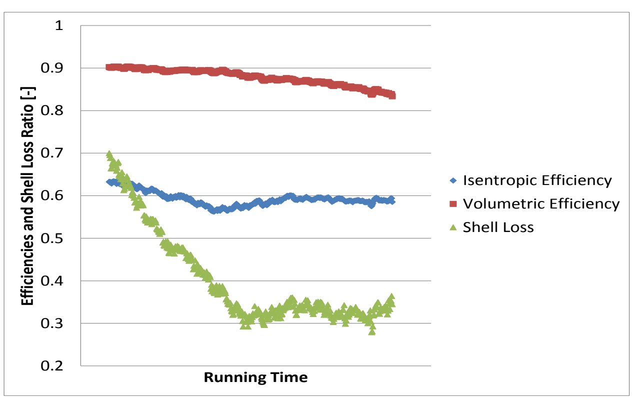

Figure 5: Measured compressor efficiencies and shell loss ratio

Approaching to the end of a drying process, the heat pump system almost reaches steady-state, when the measured air side and refrigerant side data are useful for the model calibrations. The following calibration steps were used:

1. Adjust evaporator and condenser heat transfer multipliers to match the suction and discharge saturation temperatures at "steady-state".

2. Adjust effectiveness of heat loss upstream of the drum to match the drum inlet temperature;
3. Adjust effectiveness of heat loss downstream of the drum to match evaporator inlet air temperature.

Figures 6-10 below show comparisons between the model predictions and a monitored HPCD drying process in the laboratory. Note that the numbers of the X-axis and Y-axis are intentionally hidden to protect proprietary data. Figure 6 and 7 compare the compressor suction and discharge saturation temperatures as a function of the running time. Figure 8 shows the comparison between the predicted and measured compressor powers. Figure 9 validates the air side predictions regarding the air temperature leaving the drum. It can be seen that all the predictions follow the measured trends consistently, and the prediction errors are small. In general, the calibrated HPCD model accurately predicted the total power consumption and the required time to dry the clothes.

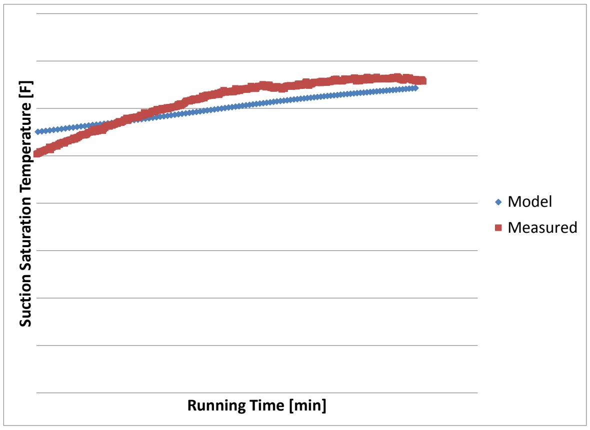

Figure 6: Comparing predicted compressor suction saturation temperatures to measured values

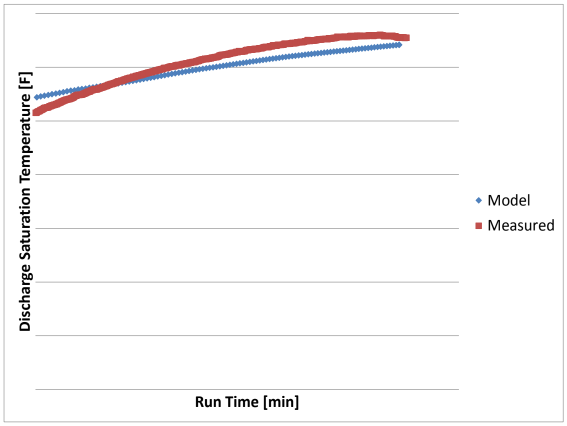

Figure 7: Comparing predicted compressor discharge saturation temperatures to measured values

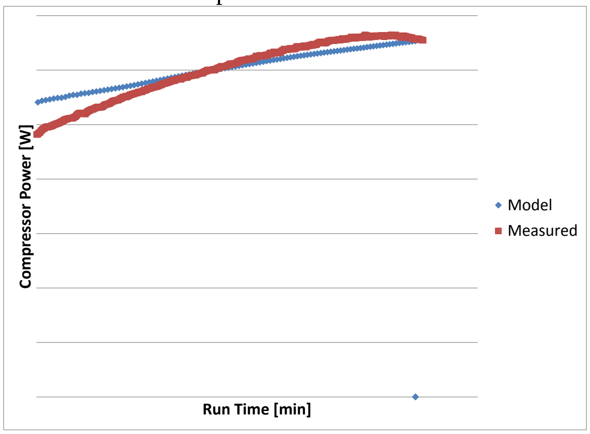

Figure 8: Comparing predicted compressor power consumptions to measured values

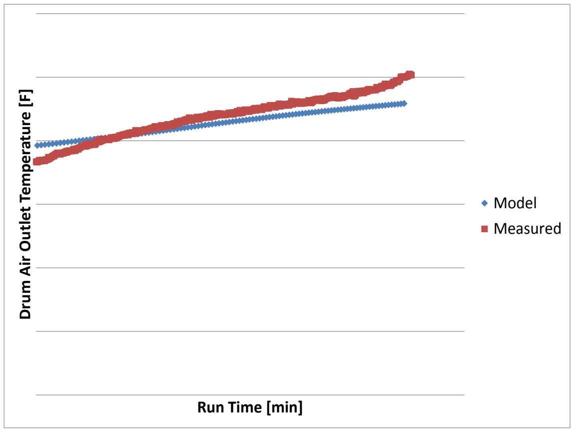

Figure 9: Comparing predicted air states leaving the drum to measured values

Figure 10 compares the predicted compressor discharge temperatures to the measured values in the whole drying process. The model accurately captures the maximum discharge temperature at the end of the process. It will be an effective design tool to predict the operation limits, and size the components and circulation air flow rates to maximize the energy factor and avoid overheating the compressor.

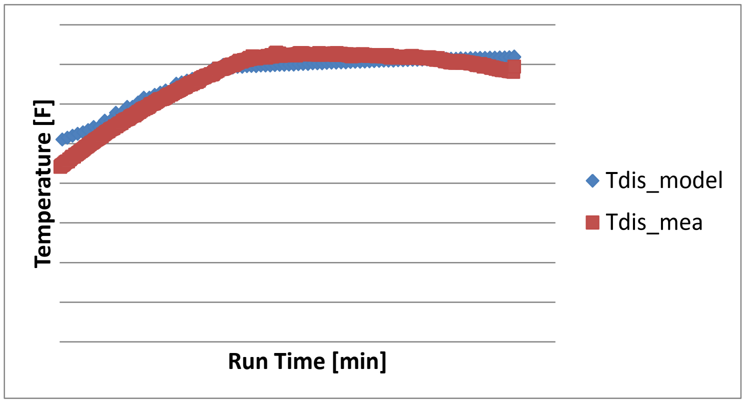

Figure 10: Comparing predicted compressor discharge temperatures to measured values

4. SUMMARY

HPCD is an innovative and very efficient clothes drying technology. This paper introduces the development of a quasi-steady-state system model to simulate the transient clothes drying process. The vapor compression system was modelled as a steady-state cycle, using ORNL HPDM with segment-to-segment heat exchanger models and an efficiency-based compressor model. 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). To improve the modeling accuracy, the compressor efficiencies, shell loss ratio, heat transfer multipliers, heat loss factors were curve-fitted to experimental data. The comparisons between the model predictions and the lab measured data demonstrate that the model follows the trends of the key refrigerant and air state points consistently, and it is able to predict the total power consumption, required drying time, and capture the operation limits accurately. It will be a useful tool to guide the HPCD design practice.

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