Measurement and analysis of clothes dryer air leakage

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Clothes dryer appliances are sold globally in the tens of millions each year. Both vented and ventless types are common and are heated by combustion, electric resistance, or electric heat pumps. In the dyer air path, segments can be defined between components such as the drum, blower, filter, screens or grills, and heat exchangers (where applicable). In this work, a technique was developed to experimentally measure air leakage into and from the segments of a clothes dryer. Detailed leakage measurements were taken on two vented and one ventless residential clothes dryer. The measurements were quantified as a leakage flow coefficient for each segment. For two dryers (one vented and one ventless), these flow coefficients were combined with *in situ* operating pressure measurements to determine leakage flow rates for each segment. For these two units under normal operation with air pressures within 0.5 kPa (50 mm water column) of ambient pressure, volumetric system air leakage was found to be about 20–60% of the blower airflow. Furthermore, a quasi–steady state psychrometric analysis was conducted on vented dryers with negatively pressurized drums. The analysis revealed that leakage quantity, location, and direction are essential to achieving an acceptable energy balance and accurate modeling results for a vented heat pump clothes dryer but are of limited significance for vented electric resistance clothes dryers.

Keywords: Clothes dryer; Psychrometric analysis; Leakage; Heat pump dryer; Drying efficiency

Introduction

Clothes dryers are ubiquitous household appliances found all over the world. Tumble-type clothes dryers generally rely on the interaction between hot, dry air and tumbling, wet fabric to facilitate drying at an accelerated rate. As such, heating air to high temperatures consumes a tremendous amount of energy annually; in the US, clothes dryers consume 0.722 EJ (684 TBtu) of primary energy per year, with associated CO_2 emissions of 32.8 million metric tons [1]. Therefore, it is imperative to improve the efficiency of existing clothes dryers and propose new and efficient technologies through research and development. Ongoing research in clothes dryers involves modeling, experimental development, and performance characterization. One parameter of great interest is air leakage along the airflow path of the dryer, as it can significantly impact overall dryer performance. For example, Stawreberg et al. showed that for certain locations, especially between the heater and drum, air leakage should be minimized to improve the energy efficiency of the dryer [2].

Clothes dryers can be divided into three binary categories based on their leakage properties: (1) closed or open airflow, (2) positive or negative drum pressurization, and (3) the presence of either one or two heat exchangers (HXs). The full factorial of three (k=3) binary options (n=2) would yield $n^k = 2^3 = 8$ combinations, but two combinations are precluded because of the thermodynamic requirement that heat be rejected in a closed cycle. Thus, there are six relevant categories of dryers, as illustrated in Table 1. In this work, the authors focus on one experimental instance each of type 1, type 3, and type 5.

For types 3, 4, 5, and 6 that have two HXs (such as heat pump dryers and condensing dryers), additional distinctions could be made based on the precise location of the blower and the presence or absence of an additional heater. These finer distinctions are not explored in this work.

[Table 1 here]

All three categories studied in this work (type 1, 3, and 5) have negatively pressurized drums. However, the difference between open/closed airflow is more important than the difference between positive/negative drum pressurization. Negative drum pressurization is often preferred because it prevents lint from escaping the drum and building up on the cabinet interior. The leakage measurement methodology presented in this work is applicable to all six types.

Research on the influence of variables on dryer performance has historically focused on type 1 and type 5 dryers. Literature examining open airflow dryers with one HX (type 1) includes the investigation of variables such as heater control algorithms [3]; load size, drum speed, and air leakage through drum seals [4]; airflow rate and heater input [5]; and drum inlet temperature, relative humidity (RH), and airflow rate [6]. Each of these investigations will be discussed in more detail.

Ahn et al. [3] investigated factors that impact the drying performance of commercially available vented dryers. The researchers determined that minimizing heat loss in the inlet duct and drum was critical to improving drying performance. An analysis of the heating algorithm showed that different heater control schedules can optimize dryer performance for dry time and/or energy consumption. In a study by Bassily and Colver [4], experimental results showed that when air leakage through the drum seals was reduced, the flow rate through the heater increased, thereby reducing the air temperature and lengthening the drying time. When the inlet and outlet flow rates were increased by 5% and 100%, the leakage increased. This resulted in a reduction in drying time from 41 min to 37 min and energy consumption from 2.13 kWh to 1.94 kWh, indicating that controlling air leakage had a significant effect on the dryer performance. In research by Hekmat and Fisk [5], four different scenarios were evaluated to improve the energy efficiency of vented dryers: (1) reduced airflow rate, (2) recirculation of exhaust air, (3) heat recovery using an air-to-air HX, and (4) condensing recirculation using a heat pump. All scenarios resulted in an energy savings between 8% and 33%. The last two scenarios had the best energy savings, although scenario 4, which showed 33% energy savings, had a 2 h drying time. Novak et al. [6] examined the impacts of inlet air temperature, RH, and flow rate on vented heat pump dryer performance. They found that to reduce drying time, higher flow rates, higher temperature, and lower RH were preferred. They also found that the moisture evaporation rate during the constant drying rate period was most sensitive to the inlet airflow rate and temperature.

Studies of the performance of closed airflow dryers with two HXs (type 5) include the investigation of variables such as airflow rate and cooling water flow rate on drying performance of alternative condensing HXs [7], load size, drum speed, and airflow rate [8]. Stawreberg et al. used a heat balance approach to estimate leakage along the airflow path in a type 5 dryer [2].

Jian and Zhao [7] studied the impact of airflow rate and water flow rate in a plate-fin HX on a condensing clothes dryer. They found that larger airflow velocities decreased the RH of the air exiting the HX, which resulted in improved drying performance. They also found that the cooling water flow rate did not have a significant effect on drying performance, other than lowering the energy consumption at lower flow rates, which increased the efficiency of the dryer. Gataric et al. [8] investigated how load size, drum speed, and airflow rate impact the performance of a commercial heat pump clothes dryer. They used an experimentally validated model to study the impact of these variables on dryer performance and found that larger mass loads and high drum speeds improved the energy efficiency. They also noted that air leakage plays a role in dryer efficiency and dry time and that at higher airflow rates, the dry time and overall efficiency decrease. Stawreberg et al. [2] extensively studied the impacts of air leakage on the performance of heat pump tumble dryers, finding that any change in the sizes of gaps or holes in the ducting or differential pressure changes between the air in the duct and the surrounding air impacts the amount of leakage from the dryer. The air leakage between the heater and drum is significant because it has a large impact on energy efficiency.

These studies do not include an investigation of dryer performance using direct measurements of air leakage throughout the airflow path. Bansal et al. presented a methodology for measuring air leakage throughout the airflow path of a heat pump clothes dryer but did not use the results to predict or optimize dryer performance [9]. Other studies include air leakage in dryer performance models but do not measure leakage or determine the leakage locations [8, 10]. Shen et al. completed a parametric modeling study of a closed airflow dryer with two HXs (type 5) that showed the predicted impacts on performance by varying the air leakage rate at different locations [11].

Based on the above literature review, further study on the effect of air leakage on clothes dryer performance in this work is necessary for several reasons: (i) the presence of leakage has a significant effect on drying efficiency and drying time and better characterization of leakage can enable optimal performance; (ii) in dryer modeling, a full accounting of leakage and air flow rates at various locations in the dryer air flow path is essential to achieving an accurate energy balance and, therefore, good agreement with experimental data; (iii) few researchers have investigated dryer performance using direct measurements of air leakage which can also easily be integrated with models. To address all the above, we describe an overall methodology of measuring dryer air leakage throughout the air duct path in this paper that can be easily integrated into a simple model. Using a simple model, the leakage measurement methodology, and sample leakage measurements, the impact of leakage on drying performance is presented in this paper.

Leakage measurement methodology

This proposed methodology for characterizing the leakage along a residential clothes dryer air flow path differs from the methodology presented in Bansal et al. in two key aspects [9]. First this methodology attempts to characterize the leakage along the air flow path at any location where the differential pressure between the duct and ambient will change due to a resistive element, such as the air filter. Bansal et al. only characterized leakage locations at the front and rear sliding seal, front grill, air filter, and miscellaneous drum leakage. Second the presented methodology characterizes the leakage at each location as a single fit parameter of a power law function to the measured flow rate versus differential pressure data. Bansal et al. used a two-parameter fit of the power law function and further compute the effective leakage area at 50 Pa for each leakage location. Their approach is more complicated and could include more error due to assumptions when using the effective leakage area equation. Further the effective leakage area is difficult to implement into a model since the differential pressure along the duct will change depending on location and will not be equal to 50 Pa. Using the single fit parameter will allow easier implementation of leakage and its impact on drying performance into a simple a model.

Measuring flow coefficients

A Minneapolis Duct Blaster, designed for measuring the leakage rate of space-conditioning air ducts, was used to characterize air leakage along the airflow path of a clothes dryer. The system operates by pressurizing the air duct of the dryer to a range of pressures in reference to the surrounding ambient air using an external fan. At each state, the pressure difference across a calibrated orifice plate is used with the manufacturer provided flow equation to determine the total volumetric flow rate through the external fan. This flow rate is equal to the air moving out of the cracks and holes in the air duct. Once this process is completed for multiple differential pressures achieved by changing the external fan speed, the measured flow rates versus differential pressures are fit to a power function as shown in Eq. (1), which describes the flow through the leakage points of the duct, where Q is the volumetric flow rate, P is the pressure differential between the duct and the ambient air, C is the flow coefficient, and n is the flow exponent. C and n describe the amount of leakage and can be used to compute a leakage area. The exponent is dimensionless and ranges between 0.5 and 1, where 0.5 describes fully turbulent flow and 1 describes fully laminar flow. For comparing flow coefficients and the relative leakage of sections of dryer ducts and dryer units, fully turbulent flow is assumed and n is set to 0.5. The flow coefficient is then measured in units of $[\mathrm{L} \cdot \mathrm{s}^{-1} \cdot \mathrm{Pa}^{-0.5}]$ and is denoted as C_{vL} in Eq. (2).

$$Q = C\, P^{n} \left[\mathrm{L} \cdot \mathrm{s}^{-1} \right]. \tag{1}$$

$$Q = C_{vL}\, P^{0.5} \left[\mathrm{L} \cdot \mathrm{s}^{-1} \right]. \tag{2}$$

Each dryer model has different leakage locations; therefore, it is important to understand the airflow path and determine a method for measuring leakage at the desired locations. In general, these measurements are taken by first measuring the leakage of the whole airflow path and then sequentially sealing the air leaks while measuring the leakage after each sealing increment. The difference between the measurements is equal to the leakage through the leak that was previously sealed. The objective is to measure the air leakage for each airflow segment. Figure 1 shows example results from this technique: the measured leakage air flow rate versus differential pressure and power law fit for the whole duct system of a prototype heat pump clothes dryer.

[Figure 1 here]

The air leakage was characterized for three different dryers: two open airflow dryers and one closed airflow dryer. The open airflow dryers were a standard commercially available electric resistance dryer (ERD; type 1) and a prototype thermoelectric dryer (TED; type 5) [12]. The closed airflow system was a prototype heat pump clothes dryer (HPCD; type 3) [13]. Figure 2 shows a schematic of the airflow path for each of the three dryers. All the dryers have an electric resistance heater, a filter, a main blower, and front and rear sliding seals (FSS, RSS) on the drum. The electric heater was not activated in any of the testing or analysis presented in this work but is relevant to computation of pressure drop and leakage. The thermoelectric dryer differs from the ERD by the addition of a hot water-to-air HX before the electric resistance heater, a cold water-to-air HX downstream of the blower, and an auxiliary fan at the exhaust downstream of the cold water-to-air HX. The HPCD differs from the ERD by the addition of a condenser before the electric resistance heater, an evaporator after the blower, and ducting connecting the evaporator to the condenser, making this dryer a closed airflow or ventless design. All components are connected with metal ductwork, and all components affect the pressure profile along the airflow path through the dryer.

Each segment between components has an associated leakage flow coefficient, denoted as $C_{vL}\#$ in Figure 2. To the extent possible, the numbering convention was kept consistent among the three dryers for ease of comparison, which resulted in some missing or out-of-order numbers. Typical leakage locations include the FSS and RSS and duct work connections.

[Figure 2 here]

Example procedure using heat pump clothes dyer

The steps for measuring the flow coefficients (C_{vL} values) along the air flow path of the HPCD is used to illustrate the procedure. Since the HPCD had a closed air flow loop design an access port was made in the duct between the electric heater and the rear grill for connection to the Duct Blaster fan. Nine air leakage trials were conducted, each with a different leakage area sealed with tape. Each trial resulted in a C_{vLT} value, denoted with arbitrary trial numbers (C_{vLT1}, C_{vLT2}, etc.) as described in the list below. The relationship between the desired segment C_{vL}s ($C_{vL}1$–$C_{vL}8$ in Figure 2) and the C_{vLT} measurement trials (C_{vLT1}–C_{vLT9}) is not one-to-one because isolating a precise segment as defined in Figure 2 was not always physically practical. Thus, Table 2 shows how these nine trial results were used to determine the eight segment C_{vL} values.

C_{vLT1}: Dryer in as-usual experimental state (whole-dryer leakage)

C_{vLT2}: taped FSS (measure 1)

C_{vLT3}: (1) + taped RSS (measure 2)

C_{vLT4}: (1) + (2) + removed duct between blower and filter (measure 3)

C_{vLT5}: (1) + (2) + (3) + taped front grill (measure 4)

C_{vLT6}: (1) + (2) + (3) + (4) + removed duct from blower to evaporator (measure 5)

C_{vLT7}: (1) + (2) + (3) + (4) + (5) + taped rear grill

C_{vLT8}: Only rear duct from condenser outlet to rear grill (includes C_{vL3} and C_{vL4})

C_{vLT9}: Rear duct plus condenser and evaporator

[Table 2 here]

When measuring the leakage of the sliding seals, the rotational position of the drum influenced the leakage result. Earlier studies showed that the air leakage rate can depend on the position of the dryer drum because the drum is not perfectly circular [9].

The difference in the HPCD drum leakage during ~1 Hz dynamic rotation (without any blower operation) versus at a single static position was measured. Leakage at precise rotational angles of the drum could not be determined during the dynamic rotation test because the differential pressure and flow rate are averaged over many rotations. The rotating drum increased the total dryer leakage by approximately 8%, from 3.13 to 3.40 L/s $Pa^{0.5}$. This represented an 18% increase in the drum C_{vL} value, but because of the minimal effect on the overall leakage, the difference between dynamic and static values was ignored, and the static leakage values for the drum sliding seals were used throughout the analyses in this paper.

Measuring in situ operating pressure and volumetric airflow rate

The volumetric flow rate along the airflow path was determined for the HPCD and TED. This required differential pressure transducers to measure the pressure in each duct segment with respect to the ambient air while the dryers were in operation. For the HPCD, Omega Model PX274 Low Pressure Transducers with a full-scale accuracy of ±1% were used. For the TED dryer, Setra Model 264 Very Low Differential Pressure Transducers with a full-scale accuracy of ±1% were used. During dryer efficiency testing, following the procedure outlined in Appendix D1 of the Uniform Test Method for Measuring the Energy Consumption of Clothes Dryers, the differential pressure transducers were sampled at 1 Hz [14]. The locations of the differential pressure transducers for the HPCD and TED are shown in Figure 2. Note that the transducers are placed between components that alter the state point of the moving air by changing the temperature, humidity, or pressure of the air.

The *in situ* differential pressure can be used with the flow coefficients described previously to determine the volumetric airflow leakage rate into or out of each duct section of the dryer during operation. The measured pressure can also be used to calculate the pressure drop or rise due to resistive elements or blowers in the system. With this information, the volumetric flow rate and air pressure through the entire airflow path can be characterized. The next section describes this procedure as it is applied to a closed airflow dryer, but it can also be applied to an open airflow dryer.

Calculating volumetric leakage along airflow path

To fully characterize the volumetric airflow throughout a ventless dryer, such as the HPCD, the differential pressure with respect to ambient air must be measured at each section of duct between the resistive elements in the airflow path. When combined with the leakage coefficients described above, these two data sets can be used to characterize the airflow along the entire duct system.

Figure 3 illustrates the HPCD with consideration isolated to the pressure profile. Resistive elements in the airflow path are denoted R1, R2, etc. Between each resistive element, pressure losses due to flow in the duct are neglected. Thus, the seven resistive elements plus the blower define eight uniform pressure segments. The leakage flow coefficient for each segment is denoted $C_{vL}1$, $C_{vL}2$, etc., which corresponds to Figure 2. Schematics of airflow paths for three dryers that were measured in this work, showing location of *in situ* pressure measurements ($P_{\#}$), duct leakage measurement locations ($C_{vL}\#$), and components that affect the air state points in the dryer duct. Leakage was measured from an electric resistance dryer, a ventless heat pump dryer, and a vented thermoelectric heat pump dryer, which are examples of dryer types 1, 5, and 3, respectively. 2.

[Figure 3 here]

By solving a set of simultaneous equations, the volumetric air leakage along the airflow path can be calculated. Assuming *N* number of leaks between *N* resistive elements in the airflow path (including the blower), the equation set includes the following.

- *N*-1 equations that describe the relationship between each segment pressure (P_i) and the pressure drop (ΔP_i) across each resistive element in the airflow path (for i = 1 to *N*-1):

$$P_i = \Delta P_i + P_{i+1}\,. \tag{3}$$

- *N* equations for leakage flows, Q_i (for i = 1 to *N*):

$$Q_i = sign(P_i) * C_{vL}i * \sqrt{|P_i|}\,. \tag{4}$$

- One equation for mass balance (constant density assumed). Because the dryer has a closed air loop, leakage out must be balanced with leakage in:

$$\Sigma_{i=1}^{N} Q_i = 0. \tag{5}$$

This system of equations can be solved with simultaneous equation-solving software such as Engineering Equation Solver [15]. The pressure drop between each component is calculated using the measured *in situ* differential pressure between the duct and ambient. These data, along with the calculated C_{vL}s, are used as inputs to the simultaneous equation solver, which will solve for the leakage flow rates.

For the TED, the *in situ* volumetric airflow rate was measured, while the dryer was operating, between the auxiliary fan and the cold water-to-air HX. The airflow was measured with a four-inch Air Monitor Corporation Aluminum LO-flo Pitot Traverse Station connected to an Air Monitor Corporation VELTRON DPT 2500 pressure and flow transmitter with an accuracy of ±0.7 L/s. With these measurements, the airflow rate along the airflow path could be calculated for the TED by subtracting or adding the air leakage rate at each C_{vL} location (depending on the sign of the differential pressure measurement) from the airflow measured with the Air Monitor.

Results and discussion

Measured leakage coefficients

The proposed methodology for measuring leakage flow coefficients for dryer duct sections was completed for three types of clothes dryers: the ERD, TED, and HPCD.

Figure 4 shows the ERD with the Minneapolis Duct Blaster connected to the exhaust of the dryer (left) and certain components of the dryer sealed with tape during the leakage test (right). A similar procedure was followed for the TED and HPCD dryers. Table 3 presents the flow coefficients for the leakage along the airflow path for the three clothes dryers. The uncertainty associated with each C_{vL} is due to the propagation of the C_{vLT} fit uncertainty used to calculate each C_{vL}. A simple check to determine how well the calculated C_{vL}s capture the total leakage was completed for the HPCD. Figure 1 shows that the whole dryer leakage C_{vL} = 3.16 $[\mathrm{L \cdot s^{-1} \cdot Pa^{-0.5}}]$. This can be compared to the calculated C_{vL}tot of 3.24 ± 0.12 $[\mathrm{L \cdot s^{-1} \cdot Pa^{-0.5}}]$ shown in Table 3. The measured total leakage is only 2.5% different than the calculated value and within the uncertainty bounds of the calculated leakage. This shows that almost 98% of the leakage along the HPCD duct path was captured with the measurement methodology presented.

[Figure 4 here]

[Table 3 here]

Calculated leakage flow rates

Type 1 vented with negatively pressurized drum: ERD

The ERD was not outfitted with *in situ* pressure transducers or an apparatus to measure airflow; therefore, leakage flow rates could not be calculated for this clothes dryer.

Type 5 ventless heat pump with negatively pressurized drum: HPCD

For the HPCD, the *in situ* operating pressures were measured with a wet 3,833 g (8.45 lb) test load tumbling in the dryer. The measured *in situ* differential pressure with respect to ambient shown in Table 4 and the C_{vL} values in Table 3 were used with the equations shown in Eqs. (3)–(5) to calculate the air leakage at each duct section. The pressure measurement locations are noted in Figure 2. The flow rate out of the dryer blower was measured using a pitot station as 86.8 L/s. Summing all the leakages shows that the HPCD has a volumetric system air leakage of ~24% of the blower airflow. The duct section air leakages were added or subtracted to the air flow out of the blower to find the air flow rate throughout the dryer duct and is also shown in Table 4. Figure 5 shows the air leakage and pressure along the HPCD air flow path graphically.

[Table 4 here]

[Figure 5 here]

Type 3 vented heat pump with negatively pressurized drum: TED

To calculate the airflow along the TED open airflow path, the simplified procedure described previously was used. The measured volumetric airflow rate after the TED cold water-to-air HX ranged from 64 to 68 L/s using the Air Monitor pitot station. Table 5 shows the *in situ* differential pressure with respect to ambient, the calculated leakage along the airflow path at each location, and the volumetric flow rate along the airflow path. Because the leakage coefficients were measured in more locations than the *in situ* operating pressure, leakages in proximity share *in situ* differential pressure. Eq. (2) is used with the C_{vL} and ΔP to calculate the flow rate into or out of the dryer duct at each leakage location. If all the leakages are summed it is found that the TED has a volumetric system air leakage of ~63% of the blower airflow. Based on the measured flow rate after the cold water-to-air HX, the flow rate into or out of the leakage location can be added to the measured flow rate to find the flow rate along the airflow path. For the TED dryer, the calculated airflow at the air inlet was validated by using a Testo 417 handheld vane anemometer to measure the air inlet flow rate. The calculated air inlet flow rate was within 5% of the handheld measurement. Figure 6 shows the differential pressure and leakage flow rate at each C_{vL} location.

[Table 5 here]

[Figure 6 here]

Impact of leakage on dryer performance

To calculate the impact of leakage on the performance of the whole dryer, a simple model was developed. Performance was quantified in terms of efficiency (electrical energy consumed per unit latent energy removed from the load) and drying rate (g/s of water removed for a fixed dry air mass flow rate and fixed quantity of heat addition).

Electric resistance (ERD) and vented HPCD configurations were studied. The vented ERD was relatively insensitive to leakage. In contrast, the vented HPCD model showed that HPCD performance is affected by the overall quantity of leakage and the locational distribution of leakage. Only the vented HPCD case was analyzed in this study due to higher complexity of modeling the unvented case. The vented case demonstrates a significant impact of leakage, and the unvented case is expected to exhibit similar or more significant impacts. The unvented case is left for future work.

Leakage can be present in the as-built appliance and designed for, or it can develop over time with the wear of components or seals. The emphasis in this section is on the designed-for, built-in leakage.

Type 1 vented electric resistance dryer

First, a model of a vented electric resistance dryer in the conventional configuration (the blower located downstream of the drum) is shown in Figure 7. The drying rate (g/s) is determined by the product of $\dot{m}_{da,3} \times (\omega_{4,premix} - \omega_3)$. This case can be handled by the following analysis, without relying on a computer model.

[Figure 7 here]

Assuming the modeled system has a fixed temperature entering the drum (T_3) and a fixed airflow rate entering the drum ($\dot{m}_{da,3}$), leakage locations can be lumped into two regions.

(1) *Lumped ERD leakage region 1: rear of drum and upstream.* It follows directly from the assumptions that leakages at state points {2} and {3} in Figure 7 do not affect $\dot{m}_{da,3}$. Leakages at {2} and {3} do not affect ω_3 because they involve mixing air streams at an identical humidity ratio. Additionally, since T_3 is assumed to be fixed, the adiabatic humidification process in the drum remains the same, and the humidity ratio leaving the drum $\omega_{4,premix}$ is unchanged. Thus, leakages at the RSS and upstream do not affect efficiency or drying rate.

(2) *Lumped ERD leakage region 2: front of drum and downstream*. Leakages at segments {4} and {5} increase the mass flow through the blower ($\dot{m}_{da,5}$) for a fixed mass flow entering the drum ($\dot{m}_{da,3}$), which does not change the quantity

$\dot{m}_{da,3} \times (\omega_{4,premix} - \omega_3)$ and thus does not impact drying rate. Because of a higher blower flow rate, blower power increases, resulting in lower system efficiency. Quantitatively, typical blowers consume approximately 0.1–0.2 kWh per 3.83 kg load, compared with approximately 2.2 kWh consumed by the heater. Thus, the impact on efficiency of increased blower power due to leakages at {4} and {5} is minor: even if 50% of the blower mass flow is from leakage downstream of the drum, this would lower system drying efficiency by 4–8% compared with the case with no leakage.

In conclusion, for the type 1 dryer, leakage upstream of the drum does not affect efficiency or drying rate, and leakage downstream of the drum can slightly lower efficiency and does not affect drying rate. These conclusions apply when the system has been designed to achieve design targets for drum airflow rate and drum entering temperature, given the leakage rates.

If leakage develops during usage of a type 1 ERD unit (in excess of the designed-for leakage), leakage upstream of the drum still will not affect efficiency or drying rate but could pose a safety risk if airflow to the heater is inadequate. Leakage downstream can lower the drying rate (roughly in proportion to the leakage quantity as a fraction of total airflow), as well as slightly lower the efficiency.

Type 3 vented heat pump dryer

In contrast to the type 1 ERD, the performance of type 3 vented heat pump dryers can be significantly influenced by leakage, and more than two lumped regions of leakage must be considered. To illustrate this, a simplified model of a type 3 dryer was developed.

To define the type 3 dryer model, a schematic with state points {1}–{7} is shown in Figure 8. All pressure drops along the air path are assumed to be discrete across the components. A state point is defined for each unique combination of system pressure and psychrometric state. A “premix” designation is used for the cases of state points {2}, {4}, and {6}, where a psychrometric change occurs upstream of a leakage. No such premix designation is necessary in cases where the only psychrometric change is due to leakage. For example, from {4} to {5} there is a pressure drop and then mixing with leakage air, so no $\{5_{premix}\}$ is needed. This state point naming convention was used because leakage measurements are lumped between a state point and its corresponding premix location. In other words, state points {1} to {7} correspond directly to seven C_{vL} leakage coefficients.

Table 6 and Table 7 provide the equations used to solve the model. For each state point, the determination of two independent state variables is required. The psychrometric state is then fixed, and all additional properties can be determined by psychrometric property routines. All leakages, {2L} through {6L}, flow from the surroundings into the system. Thus, they are assumed to have the same psychrometric state as state point {1}, with mass flows according to the local leakage flow coefficient C_{vL} and local system gauge pressure relative to ambient. The RH_4 (leaving the drum) was set at 90%. This is consistent with the drum leaving humidity reported by Ahn et al. [3] for the first 30 minutes drying a 6.2 kg load, and with the higher values shown in [12]. Exploration of the impact of drum leaving humidity on the analysis is not explored here.

Note that the type 3 system modeled in this section differs slightly from the type 3 HPCD experimentally characterized in a previous section. Specifically, that HPCD had eight segments instead of seven, due to the presence of an electric resistance heater in addition to the condenser. Additionally, the previously discussed HPCD had the blower upstream instead of downstream of the evaporator. Each is a variant of a type 3 system.

[Figure 8 here]

[Table 6 here]

[Table 7 here]

Next, Table 8 shows the baseline assumed values for the system gauge pressure ΔP and the segment leakage coefficient C_{vL}.

[Table 8 here]

The model defined in this section was implemented in Engineering Equation Solver to solve the simultaneous equation set and determine psychrometric properties [15]. The simulation results are presented in Figure 9 through Figure 11.

In Figure 9, all C_{vL} values are varied compared with their baseline values in proportion with the factor f_{CvL} such that when $f_{CvL} = 1$, all C_{vL} values are as shown in Table 8. Lower leakage leads to improved efficiency and faster drying rate. At half the baseline leakage ($f_{CvL} = 0.5$), the efficiency is approximately 2.6, which is 40% higher than the efficiency of 1.8 at the baseline leakage ($f_{CvL} = 1$). The drying rate at half the baseline leakage is 2.45 kg/hr, approximately 20% faster than the rate of 2 kg/hr at the baseline leakage. As leakage approaches zero, the efficiency is about 75% higher and the drying rate is about 35% faster than baseline.

[Figure 9 here]

Whereas Figure 9 varied all C_{vL} values proportionally in unison, Figure 10 and Figure 11 vary the absolute value of each C_{vL} individually, while holding all others at the baseline values.

Regarding efficiency, Figure 10 shows that every leakage segment impacts efficiency in roughly equal measure, with increasing leakage rates lowering efficiency. However, $C_{vL}2$ has a limited effect at low leakage rates, and the effect rapidly accelerates at high leakage rates. This is related to the assumption of fixed heat addition at the hot HX, which mean that the temperature at $\{2_{premix}\}$ will rise (along with the temperature of heat addition at hot HX) as $C_{vL,}2$ increases. Initially the effect is small, but eventually causes significantly higher hot HX temperatures, strongly affecting efficiency.

Regarding drying rate, Figure 11 shows similar trends to Figure 10. Increasing leakage in any region alone decreases efficiency, with lower leakage rates of $C_{vL}2$ having a negligible impact on drying rate. At low leakage levels, the lack of impact can be traced to the assumptions of (a) fixed mass flow at {6} and (b) fixed heat addition at the hot HX. However, as the $C_{vL,}2$ value becomes very high, the downstream pressures are affected, increasing downstream leakage flow rates and reducing the mass flow through the drum to reduce drying rate.

[Figure 10 here]

[Figure 11 here]

The modeling results for a vented heat pump dryer show that leakage has a significant impact on dryer efficiency and drying rate. Different results are expected for other heat pump dryer types, such as a ventless heat pump dryer (type 5), or for positively pressurized drum heat pump dryers (types 4 and 6).

It should be noted that these results were computed at the particular operating conditions defined in Table 6 and Table 7, and the results are expected to vary at different conditions (for example, as drum leaving RH drops towards the end of the cycle). Exploration of these effects is left for future work.

A more detailed evaluation of the impact of leakage is significantly more complex than the impact on vented types. Thus the impact of leakage on these is left for future work.

In addition, experimental data was not available for types 4 and 6. Thus the impact of leakage on these dryer types is also left for future work.

Conclusions

A methodology for measuring air leakage between the dryer duct and the ambient environment using a commercially available system for measuring forced air duct system leakage was presented. After measurements of flow as a function of differential pressure between segments of a duct and ambient environment were taken, the results were converted to leakage coefficients to characterize the volumetric leakage rates in sections of a dryer. The methodology was used to characterize the leakage along the path of three residential clothes dryers. These results were used to investigate model-based design improvements of clothes dryers by investigating both the location and magnitude of leakages for optimum energy efficiency and the cycle time of dryers.

This work demonstrated the impact of leakage on dryer performance for two dryer types. For a vented electric resistance dryer (type 1), an analysis was presented to illustrate that leakage has a negligible impact on performance, under the assumption that drum inlet temperature is fixed. Additionally, a simple model was developed to show that leakage has a significant impact on the performance of a vented heat pump dryer with a negatively pressurized drum (type 3). Leakage from most dryer segments had similar impacts on performance, while performance was relatively unaffected by leakage into the segment between the hot HX and the drum. The results are expected to differ significantly for other dryers (types 4, 5, and 6), and this analysis is left for future work.

Nomenclature

C	flow coefficient ($L \cdot s^{-1} \cdot Pa^n$)
ΔP	differential pressure (Pa)
ERD	electric resistance dryer
FSS	front sliding seal
h	specific enthalpy (kJ/kg)
HPCD	heat pump clothes dryer
HX	heat exchanger
$\dot{m}_{da}$	mass flow rate of dry air circulating through system (kg/s)
$\dot{m}_{evap}$	drying rate (g/s)
n	flow exponent
N	total number of resistive elements in dryer air path
P	pressure (Pa)
Q	volumetric flow rate ($L \cdot s^{-1}$)
R1–R7	resistive element in airflow path
RH	relative humidity
RSS	rear sliding seal
T	temperature (°C)
T1–T9	trial numbers for leakage characterization used for determining leakage of duct section
TED	thermoelectric dryer
W	work required by heat pump
η	drying efficiency
ω	humidity ratio (kg_w/kg_{da})

Subscripts

C	cold
Carnot	efficiency
cooling	cooling side of heat pump
da	dry air
evap	evaporated water from load
H	hot
heating	heating side of heat pump
premix	state points where a psychrometric change occurs upstream of a leakage

vL air leak in dryer duct section with flow exponent = 0.5

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References

[1] EIA, 2019, "Baseline Energy Calculator," https://trynthink.github.io/scout/calculator.html.

[2] Stawreberg, L., Berghel, J., and Renstrom, R., 2012, "Energy losses by air leakage in condensing tumble dryers," Applied Thermal Engineering, 37, pp. 373 - 379.

[3] Ahn, S. P., Kim, S. H., Park, Y. G., and Ha, M. Y., 2019, "Experimental study on drying time and energy consumption of a vented dryer," Journal of Mechanical Science and Technology, 33(5), pp. 2471-2480.

[4] Bassily, A. M., and Colver, G. M., 2003, "Performance Analysis of an Electric Clothes Dryer," Drying Technology, 21(3), pp. 499-524.

[5] Hekmat, D., and Fisk, W. J., 1984, "Improving the Energy Performance of Residential Clothes Dryers," 35th International Appliance Technical ConferenceOhio State University.

[6] Novak, L., Gatarić, P., and Širok, B., 2019, "Influence of drum inlet air conditions on drying process in a domestic tumble dryer," Drying Technology, 37(6), pp. 781-792.

[7] Jian, Q., and Zhao, J., 2017, "Drying performance analysis of a condensing tumbler clothes dryer with a unique water cooled heat exchanger," Applied Thermal Engineering, 113, pp. 601 - 608.

[8] Gatarić, P., Širok, B., Hočevar, M., and Novak, L., 2019, "Modeling of heat pump tumble dryer energy consumption and drying time," Drying Technology, 37(11), pp. 1396-1404.

[9] Bansal, P., Mohabir, A., and Miller, W., 2016, "A novel method to determine air leakage in heat pump clothes dryers," Energy, 96, pp. 1-7.

[10] Huelsz, G., Urbiola-Soto, L., López-Alquicira, F., Rechtman, R., and Hernández-Cruz, G., 2013, "Total Energy Balance Method for Venting Electric Clothes Dryers," Drying Technology, 31(5), pp. 576-586.

[11] Shen, B., Gluesenkamp, K., Boudreaux, P., and Patel, V., 2018, "Model-Based Air Flow Path Optimization of Heat Pump Clothes Dryer," 17th International Refrigeration and Air Conditioning Conference at PurdueLafayette, IN.

[12] Patel, V. K., Gluesenkamp, K. R., Goodman, D., and Gehl, A., 2018, "Experimental evaluation and thermodynamic system modeling of thermoelectric heat pump clothes dryer," Applied Energy, 217, pp. 221-232.

[13] Shen, B., Gluesenkamp, K. R., Bansal, P., and Beers, D., 2016, "Heat Pump Clothes Dryer Model Development," International Refrigeration and Air Conditioning Conference at Purdue UniversityLafayette, IN.

[14] 10 CFR 430, 2013, "Energy Conservation Program for Consumer Products," Subpart B, "Test Procedures"; Appendix D/D1/D2, "Uniform Test Method for Measuring the Energy Consumption of Clothes Dryers."

[15] "F-Chart Software - EES," http://www.fchart.com/ees/pro-comm-versions.php.

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and a vented thermoelectric heat pump dryer, which are examples of dryer types 1, 5, and 3, respectively.

Figure 3. Schematic of flow-resistive elements and leakage locations of the airflow path in the HPCD.

Figure 4. Left: The dryer shown with the Duct Blaster connected to the exhaust duct to allow leakage measurement under controlled pressurization. Right: the blower inlet taped so that the leakage between the blower and the duct outlet can be measured. The red tubing connected to the side of the blower housing measures the duct pressure.

Figure 5: Calculated air leakage rate and gauge pressure at each leakage location in the HPCD closed airflow loop using the simultaneous equation solver. Total flow rate through the blower was 86.8 L/s.

Figure 6. *In situ* measured differential pressure and calculated volumetric air leakage rate along the TED duct. Total flow rate through the blower was 69.1 L/s.

Figure 7. Schematic of model for a vented dryer with one HX and a negatively pressurized drum (type 1 or ERD).

Figure 8. Schematic of model for a vented dryer with two HXs and a negatively pressurized drum (Type 3).

Figure 9. Modeled effect on drying efficiency η and drying rate of scaling all $C_{vL,i}$ leakage flow coefficients equally as fractions (f_{CvL}) of base values for Type 3 dryer.

Figure 10. Modeled effect on drying efficiency η of scaling each individual $C_{vL,i}$ value from zero to double its nominal value for Type 3 dryer.

Figure 11. Modeled effect on drying rate of scaling each individual $C_{vL,i}$ value from zero to double its nominal value for Type 3 dryer.

Table 1. The six categories of dryers with unique leakage characteristics.

		Open airflow (vented)		Closed airflow (ventless)	
		1 HX	2 HX	1 HX	2 HX
Drum pressurization	Negative	1*	3*	(not possible)	5*
	Positive	2	4	(not possible)	6

*explored in this work: type 1 ERD, type 3 TED, and type 5 HPCD

Table 2. Equations for computing C_{vL}s based on the trials listed above.

C_{vL} from Figure 2	Equation
$C_{vL}1$ (blower to evaporator)	$C_{vL}1 = C_{vLT5} - C_{vLT6}$
$C_{vL}2$ (evaporator to condenser)	$C_{vL}2 = C_{vLT9} - C_{vLT8}$
$C_{vL}3$ (condenser to heater)	$C_{vL}3 = C_{vLT7} - C_{vLT9} + 0.5 \times C_{vLT8}$
$C_{vL}4$ (heater to rear grill)	$C_{vL}4 = C_{vLT6} - C_{vLT7} + 0.5 \times C_{vLT8}$
$C_{vL}5$ (RSS)	$C_{vL}5 = C_{vLT2} - C_{vLT3}$
$C_{vL}6$ (FSS)	$C_{vL}6 = C_{vLT1} - C_{vLT2}$
$C_{vL}7$ (front grill to filter)	$C_{vL}7 = C_{vLT4} - C_{vLT5}$
$C_{vL}8$ (filter to blower)	$C_{vL}8 = C_{vLT3} - C_{vLT4}$

Table 3. Leakage flow coefficients (C_{vL} values) for three experimentally measured dryers with absolute propagated uncertainty. Gray boxes indicate leakage areas that are not applicable for the dryer type in that column.

	$C_{vL}\left[\frac{\mathrm{L}}{\mathrm{s \cdot Pa^{0.5}}}\right]$		
	Type 5 HPCD	**Type 3 TED**	**Type 1 ERD**
Total (C_{vL}tot)	3.24±0.12	4.04±0.04	3.92±0.10
Blower to cold HX ($C_{vL}1$)	0.08±0.06	0.48 ±0.002	
Cold HX to aux. blower ($C_{vL}9$)		0.07±0.003	
Cold HX to hot HX ($C_{vL}2$)	0.41±0.01		
Air in to hot HX ($C_{vL}10$)		0.86±0.005	
Hot HX to heater ($C_{vL}3$)	0.47±0.02	0.59±0.004	
Heater to rear grill ($C_{vL}4$)	0.99±0.05	0.37±0.01	2.01±0.09
Blower to exhaust ($C_{vL}11$)			0.18±0.02
RSS ($C_{vL}5$)	0.53±0.10	0.12±0.03	0.60±0.06
FSS ($C_{vL}6$)	0.54±0.15	0.88±0.02	0.79±0.06
Front grill to filter ($C_{vL}7$)	0.12±0.06	0.34±0.01	0.07±0.02
Filter to blower ($C_{vL}8$)	0.18±0.06	0.33±0.01	0.27±0.01

Table 4. Results with propagated uncertainties for the HPCD: measured pressure during operation and calculated flow rates based on Equation (4).

Duct section	ΔP relative to ambient (Pa)	Flow rate out of leakage location (L/s)	Flow rate along main airflow path (L/s)
Blower to evaporator ($C_{vL}1$)	167.1±7.5	1.03±0.78	85.8±1.0
Evaporator to condenser ($C_{vL}2$)	155.3±7.5	5.11±0.18	80.7±1.1
Condenser to heater ($C_{vL}3$)	65.9±7.5	3.82±0.27	76.8±1.1
Heater to rear grill ($C_{vL}4$)	−2.4±7.5	−1.53±3.6	78.4±3.8
RSS ($C_{vL}5$)	−4.7±7.5	−1.15±1.0	79.5±3.9
FSS ($C_{vL}6$)	−30.6±7.5	−2.99±0.91	82.5±4.0
Front grill to filter ($C_{vL}7$)	−171.8±7.5	−1.57±0.79	84.1±4.1
Filter to blower ($C_{vL}8$)	−331.8±7.5	−3.28±1.1	87.4±4.2

Table 5: Results with propagated uncertainties for the TED: measured pressure during operation and calculated flow rates based on Equation (4).

Duct section	ΔP relative to ambient (Pa)	Flow rate out of leakage location (L/s)	Flow rate along main airflow path (L/s)
Air inlet to hot water-to-air HX ($C_{vL}10$)	−37.4±7.5	−5.28±0.53	31.1±1.1
Hot water-to-air HX to heater ($C_{vL}3$)	−37.4±7.5	−3.64±0.37	36.5±0.99
Heater to rear grill ($C_{vL}4$)	−89.7±7.5	−3.55±0.18	40.2±0.92
RSS ($C_{vL}5$)	−89.7±7.5	−1.12±0.32	43.7±0.89
FSS ($C_{vL}6$)	−239.1±7.5	−13.61±0.44	44.8±0.89
Front grill to filter ($C_{vL}7$)	−239.1±7.5	−5.27±0.20	58.4±0.78
Filter to blower ($C_{vL}8$)	−239.1±7.5	−5.03±0.18	63.7±0.77
Blower to cold water-to-air HX ($C_{vL}1$)	47.3±7.5	3.27±0.26	68.7±0.74
Cold water-to-air HX to aux. blower ($C_{vL}9$)	−59.8±7.5	−0.53±0.03	65.5±0.70

Table 6. General thermodynamic and psychrometric modeling assumptions and governing equations for vented heat pump clothes dryer.

Parameter	Value or method of determination	Notes
Heat pump $COP_{cooling}$	40% of Carnot value, where $COP_{Carnot,cooling} = \frac{T_C}{T_H - T_C}$	Simple temperature-dependent model of heat pump efficiency
Heat pump $COP_{heating}$	$COP_{heating} = COP_{cooling} + 1$	Energy balance on a heat pump
T_H (K)	$T_H = T_2 + 10$ K	Assuming a 10 K approach temperature between the heat pump and the hot sink
T_C (K)	$T_C = T_{6,premix} - 10$ K	Assuming a 10 K approach temperature between the heat pump and the cold source
$\dot{m}_{da,6}$	0.085 (kg/s) mass flow rate of dry air into the blower	Corresponds to approximately 254 m^3hr^{-1} or 150 ft^3min^{-1}
$Q_{H\,HX}$	2 kW	Heat pump size is fixed at 2 kW heating capacity
W (kW)	$W = \frac{Q_{H\,HX}}{COP_{heating}}$	Work required by heat pump
$Q_{C\,HX}$ (kW)	$Q_{C\,HX} = Q_{H\,HX}\left(1 - \frac{1}{COP_{heating}}\right)$	Energy balance on a heat pump
Drying rate	$\dot{m}_{evap} = \dot{m}_{da,3}(\omega_{4,premix} - \omega_3)$	Rate of evaporation from load
Drying efficiency	$\eta = \frac{\dot{m}_{evap} h_{fg}}{W}$	Not the same as $COP_{heating}$

Table 7. State point-specific thermodynamic and psychrometric modeling assumptions and governing equations for vented heat pump clothes dryer.

Parameter	**Value or method of determination**	**Notes**
T_1	25°C (298.15 K)	Standard ambient condition
RH_1	50%	Standard ambient condition
$h_{2,premix}$	$h_{2,premix} = h_1 + \frac{Q_{hotHX}}{\dot{m}_{da,1}}$	Energy balance on the hot HX
$\omega_{2,premix}$	$\omega_{2,premix} = \omega_1$	
h_2	$h_2 = \frac{\dot{m}_{da,2,premix}h_{2,premix} + \dot{m}_{da,2L}h_{2L}}{\dot{m}_{da,2}}$	Mixing of {2_{premix}} and {2L} energy balance
ω_2	$\omega_2 = \omega_{2,premix}$	Mixing flows {2_{premix}} and {2L} have the same humidity ratio
h_3	$h_3 = \frac{\dot{m}_{da,2}h_2 + \dot{m}_{da,3L}h_{3L}}{\dot{m}_{da,3}}$	Mixing of {2} and {3L} energy balance
ω_3	$\omega_3 = \omega_2$	Mixing flows {2} and {3L} have the same humidity ratio
$RH_{4,premix}$	90%	Assume adiabatic humidification in drum proceeds to 90% RH
$h_{4,premix}$	$h_{4,premix} = h_3$	Adiabatic process in drum
h_4	$h_4 = \frac{\dot{m}_{da,4,premix}h_{4,premix} + \dot{m}_{da,4L}h_{4L}}{\dot{m}_{da,4}}$	Mixing of {4_{premix}} and {4L} energy balance
ω_4	$\omega_4 = \frac{\dot{m}_{da,4,premix}\omega_{4,premix} + \dot{m}_{da,4L}\omega_{4L}}{\dot{m}_{da,4}}$	Mixing of {4_{premix}} and {4L} water mass balance
h_5	$h_5 = \frac{\dot{m}_{da,4}h_4 + \dot{m}_{da,5L}h_{5L}}{\dot{m}_{da,5}}$	Mixing of {4} and {5L} energy balance
ω_5	$\omega_5 = \frac{\dot{m}_{da,4}\omega_4 + \dot{m}_{da,5L}\omega_{5L}}{\dot{m}_{da,5}}$	Mixing of {4} and {5L} water mass balance
$RH_{6,premix}$	90%	Assume wet coil with 90% leaving RH
$h_{6,premix}$	$h_{6,premix} = h_5 - \frac{Q_{coldHX}}{\dot{m}_{da,6}}$	Energy balance on the cold HX

Table 8. Modeling assumptions for pressure drop and leakage in vented heat pump clothes dryer. Blower head was assumed to be 348 Pa (36 mm or 1.4 in. water column) and absolute pressure of state point {7} was assumed to be 101.325 kPa.

State point	**ΔP (Pa)**	**C_{vL} (L/[s Pa$^{0.5}$])**
{1}	62.2	0
{2}	24.9	1.04
{3}	49.8	0.57
{4}	149.3	0.56
{5}	62.2	0.32
{6}	−348.4	0.32
{7}	0	0