Efficiency limits of evaporative fabric drying methods

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ABSTRACT
Cloth drying is a major consumer of energy, and most drying is conducted by evaporative methods. The evaporative drying energy efficiency limit is commonly assumed to be 100%: one unit of latent heat removed per unit of energy expended. However, this ignores the “free drying” available from unsaturated ambient air and the possible role of heat pumping. We demonstrate that efficiency limits for evaporative drying are fundamentally related to both drying rate and ambient psychrometric conditions. A relationship among efficiency, drying rate, and ambient conditions is quantified for several evaporative drying technologies, and a comparison among evaporative technologies is provided. In addition, a comparison is provided between efficiency limits and state of the art device performance. This research will help guide future research in the most promising directions.

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1. Introduction
Nearly 80% of households in the United States have clothes dryers, and 30% of these dryers are at least 10 years old.[1] The most common types of clothes dryers use an evaporative thermal process to facilitate moisture removal from fabrics. Electric resistance dryers (ERDs) heat air from the ambient using an electric resistance element, and natural gas dryers (NGDs) heat air from the ambient using natural gas combustion. The heated air is directed into a rotating drum in which the wet fabric is tumbling; the air collects moisture and leaves the drum to be vented. This once-through configuration requires that a vent be installed. Much research effort has been dedicated to understanding the physical processes and characterizing the performance of these types of dryers. Researchers have focused on attempting to improve the energy efficiency of ERDs and NGDs using various techniques such as an air-to-air heat exchanger (HX) in the outgoing air stream to condense moisture,[2] an external hot water supply as an additional heat source,[3,4], or a membrane to enhance dehumidification in a condensing dryer.[5]

Vapor-compression heat pump dryers (VCHPDs) are also based on evaporative thermal drying, but they heat the air entering the drum using the hot side of a heat pump (the condenser in a VC cycle).[6] The moist air leaving the drum is then cooled and the moisture is condensed using the cold side of the heat pump (evaporator in a VC cycle). This type of system can be configured with a vent or with a closed-loop air flow that does not require a vent. Although VCHPDs are based on mature technology and are used extensively in Australia and Europe, they have had poor market penetration in the United States, mainly because of high cost and longer drying times.[7] Recent VCHPD research has been conducted both experimentally[8] and through modeling/analysis.[9,11] Although VCHPDs offer significant energy savings, they typically use hydrofluorocarbon-based working fluids with high global warming potential (GWP) that are being phased out of operation; recent efforts in VCHPD research have therefore considered
other low-GWP working fluids such as CO₂ in transcritical cycles.\textsuperscript{[12–14]}

In addition to VCHPDs, research is ongoing on using solid-state thermoelectric (TE) heat pumps in TEHPDs.\textsuperscript{[15–20]} TE-based heat pumps introduce fewer moving parts into the clothes dryer system and use no refrigerant as a working fluid. When a DC current is applied to the TE element, a temperature difference is created between the two sides and the system can be used as a heat pump. The results of recent efforts show significant increases of up to 40\% in energy efficiency for TEHPDs compared with conventional ERDs.\textsuperscript{[20]} Other early-stage dryer technologies are also being developed, such as microwave drying,\textsuperscript{[21–23]} infrared drying,\textsuperscript{[24,25]} vacuum drying\textsuperscript{[26]} and direct-contact ultrasonic drying.\textsuperscript{[27–34]}

This paper establishes fundamental limits to the theoretical maximum achievable performance of evaporative clothes drying technologies. The work is oriented (but not limited) to residential drying products. To provide dimensional values relevant to common appliances, typical load sizes, typical moisture contents, and typical air flow rates for residential clothes drying technologies\textsuperscript{[35]} are used. The work focuses on efficiency and drying time. Other aspects are not considered, such as appearance, effect of drying on fabric quality and wrinkling, noise level, reliability, cost, ease of use, and other customer expectations. After fundamental limits are established, comparisons are made to state-of-the-art technologies in each category. Particular attention is applied to ERDs and HPDs, which are the most prevalent dryer types around the world, but several other technologies are also mentioned.

### 2. Fabric drying processes and metrics

Figure 1 outlines possible methods of moisture removal. Drying mechanisms can be categorized as either evaporative (thermal) or mechanical; some processes use a combination of the two methods. This work addresses evaporative methods; mechanical methods are outside its scope. Particular emphasis is placed on convective methods with active heat sources, since these are the most widespread and are responsible for the most drying energy consumption.

Most residential and industrial drying processes rely on an accelerated evaporation rate at high temperature with heat input. In residential clothes dryers, an electric resistance heating element or heat from natural gas combustion is used to elevate the temperature of the moist clothes and enhance the evaporation rate. Hot air at a low relative humidity (RH) of 1–10\% blows across the wet clothes, picks up moisture...
and leaves the machine at a RH of ~90%.\[^{4,35}\] A large amount of heat is required for evaporative drying to overcome the large latent heat of evaporation of water. The latent heat of evaporation of water at atmospheric pressure is about 2453, 2358, and 2257 kJ/kg\(_w\) (or \(90 \times 10^6\) Btu/lb\(_w\)) at 20, 60, and 100 °C, respectively.\[^{36}\]

In clothes drying research, various metrics are used to quantify the cloth moisture content, drying rate, and energy efficiency of the drying process. Some commonly used metrics are defined in this section and summarized in Table 1 (which contains Eqs. [1–13]). A measure of the water content in cloth at any given time, \(t\), is defined by a mass ratio, \(y(t)\) (Eq. [1]), where \(m_{w,t}\) is the instantaneous mass of water contained in the cloth at time \(t\) and \(m_c\) is the mass of the cloth when it is completely bone-dry. The mass ratio \(y(t)\) is also referred to as the remaining moisture content, RMC\((t)\).

A brief note about the definition of “dry” is warranted. In common use, as well as indicated by regulatory and legal statutes in test procedures, “bone dry” is defined as the driest a fabric can be made by a conventional drying appliance; i.e., the steady state level of dryness achieved through indefinitely long operation of a conventional evaporative tumble dryer. Although the moisture is imperceptible to human senses, bone dry cloth still contains some molecules of water in the microscopic pores of the cloth.

The initial mass ratio, \(y_i\), (also referred to as starting moisture content, or SMC) is defined in Eq. (2), where \(m_{w,i}\) is the initial mass of water contained in the cloth before drying. Similarly, the final mass ratio, \(y_f\), (also referred to as final moisture content, or FMC) is defined in Eq. (3), where \(m_{w,f}\) is the final mass of water contained in the cloth after drying. Since the initial moisture content of fabric can vary, \(y_i\), must be overcome. We can introduce a more consistent definition that relates the dryness of the clothes to the energy used by the dryer, the standard definition used for rating appliances in the United States is the energy factor (EF).\[^{37}\] The EF is defined based on the dry mass of the cloth (also known as the bone-dry weight or BDW, and measured in lb\(_{BDW}\)) divided by the energy required to dry the load from an initial moisture content (\(y_i\) or SMC) of 57.5% to a final moisture content (\(y_f\) or FMC) of less than 4%. It is given by Eq. (4), where \(E\) is the total energy (in kWh) consumed during the drying process.

Another metric used in clothes drying research to quantify the water removal per unit of energy consumed is known as the specific moisture extraction rate, SMER, defined in Eq. (5), for which all quantities are as defined above.

For any evaporative drying process, the latent heat of vaporization for water, \(h_{fg}\), must be overcome. We can use this information and the definition of SMER to develop an expression for the overall drying efficiency \(\tilde{\eta}\), given by Eq. (6), which is a measure of the drying process efficiency relative to a reference process in which all energy input to the dryer is used to overcome the water latent heat of vaporization. To determine the instantaneous drying efficiency, \(\eta(t)\), at any given time, we can use rates of water removal and energy input in Eq. (7), where \(m_{wr}\) is the rate of water removal (in kg\(_w\)/s) and \(E\) is the rate of energy input (in kWh/s).

The EF is commonly used in US clothes dryer research (e.g.,\[^{7,11}\]) and industry to designate energy

| Table 1. Summary of definitions used in clothes drying research. | Definition | Equation | Eq. # |
| --- | --- | --- | --- |
| Drying metrics | Relative moisture content | \(y(t) = \frac{m_{w,t}}{m_c} = \text{RMC}\(t)\) | 1 |
|  | Initial moisture content | \(y_i = \frac{m_{w,i}}{m_c} = \text{SMC}\) | 2 |
|  | Final moisture content | \(y_f = \frac{m_{w,f}}{m_c} = \text{FMC}\) | 3 |
| Efficiency metrics\(^a\) | Energy factor | \(\text{EF} = \frac{y_i}{y_i - y_f} \left(\frac{h_{fg}}{E}\right)\) for \(y_i - y_f < 0.04\) | 4 |
|  | Specific moisture extraction rate | \(\text{SMER} = \frac{m_{w,i} - m_{w,f}}{m_{w,f} \cdot h_{fg}}\) | 5 |
|  | Overall drying efficiency | \(\tilde{\eta} = \frac{m_{w,i} - m_{w,f}}{m_{w,f}}\) | 6 |
|  | Instantaneous drying efficiency | \(\eta(t) = \frac{m_{w,i}(t)}{E(t)}\) | 7 |
| Efficiency conversion\(^b\) | EF to SMER | \(\text{SMER} = 0.454 \cdot \text{EF}(\frac{y_i}{y_f})\left(\frac{h_{fg}}{E}\right)\) | 8 |
|  | SMER to EF | \(\text{EF} = \frac{\text{SMER}}{(y_i - y_f) \cdot h_{fg}}\) for \(y_i - y_f < 0.04\) | 9 |
|  | \(i\) to SMER | \(\text{SMER} = \frac{y_i}{(y_f - y_i) \cdot h_{fg}}\) | 10 |
|  | \(i\) to EF | \(i = \frac{\text{SMER} \cdot h_{fg}}{y_i}\) | 11 |
|  | \(\eta\) to EF | \(\eta = 0.454 \cdot \text{EF}(y_i - y_f) \cdot h_{fg}\) | 12 |
|  | \(\tilde{\eta}\) to EF | \(\tilde{\eta} = \frac{\text{SMER} \cdot h_{fg}}{y_i}\) for \(y_i - y_f < 0.04\) | 13 |

\(^a\)Note that \(E\) and \(h_{fg}\) must use consistent energy units in the equations. For example, if \(E\) is in units of kWh, then \(h_{fg} = 0.6782\) kWh/kg\(_w\) for 25 °C.

\(^b\)Note that \(h_{fg}\) must use consistent energy units in the equations. For example, if \(E\) is in units of kWh, then \(h_{fg} = 0.6782\) kWh/kg\(_w\) for 25 °C.
efficiency, whereas SMER is commonly used in the literature (e.g., [9,36]). It is useful to define a conversion equation from one metric to another. For example, Equations (4) and (5) for EF and SMER can be related to each other using Eq. (8), where the value 0.454 is used for unit conversion. Equation (8) can be used to convert from EF (in lbBDW/kWh) to SMER (in kgw/kWh) for particular initial and final moisture contents. For converting the reverse, from SMER to EF, the initial and final moisture contents should match the EF definition, as shown in Eq. (9); otherwise a clear translation is not available, since knowledge of the moisture-content-dependent efficiency would need to be considered. This translation asymmetry is a consequence of defining EF on the basis of dry fabric mass, whereas SMER is defined on a water mass basis. If SMER is measured under conditions different from EF measurements, no clear conversion is available.

The remaining metrics for drying rate and efficiency can be related to one another, as shown in Eqs. (10–13) (note that the units of \( h_{fg} \) must be specified in kWh/kgw for correct conversion using these equations).

An ideal evaporative drying process can be defined as the reference case in which all energy input is used to overcome the water latent heat of vaporization (Eq. [14]).

\[
\eta_{ideal} = 1 \quad [-] \tag{14}
\]

Applying the conversion in Eqs. (10–13) results in the ideal expressions shown in Table 2. Note that EF depends on the change in water mass ratio, while SMER and \( \eta \) are independent of cloth water content. SMER is the inverse of \( h_{fg} \) assumed to be evaluated at 25°C. In the table, EF has been computed for \( y_i \) of 0.575 and \( y_f \) of 0.04. With values of 0.575 and 0.00, \( EF_{ideal} \) changes from 6.08 to 5.65 lbBDW/kWh.

The value of each ideal efficiency metric varies with temperature, since the latent heat of vaporization of water varies with temperature. Figure 2 shows this temperature dependence. It shows an approximately 1% efficiency gain for every 10°C increase in the drying temperature due simply to reduction of the latent heat of evaporation.

In the case of \( \eta_{ideal} \), the reference point is \( h_{fg} \) itself. Thus, \( \eta_{ideal} \) has no temperature dependence. For instructive purposes, Figure 2 also plots \( \eta_{ideal} \) relative to a reference of \( \eta_{ideal,25°C} \). This is equivalent to a plot of the temperature dependence of the latent heat relative to the reference of 25°C.

Evaporative drying at ambient temperature would typically be impractically slow, and evaporation at higher temperatures requires sensible heating of the load and its moisture. For an ideal process at elevated temperature, in which all energy use results only in sensible heating of the moist load followed by evaporation of the moisture at the final temperature, the energy required is shown in Eq. (15). This calculation assumes that the final water mass \( (m_{w,f}) \) is zero, a reasonable approximation for most drying processes that allows the following expressions to be straightforward while still capturing the phenomena of interest. Note also that the energy and latent heat are written as functions of the final temperature, \( T_f \).

\[
E_{L+S}(T_f) = (T_f - T_i) (\Delta m_w \cdot C_{p,w} + m_c \cdot C_{p,c}) + \Delta m_w h_{fg}(T_f) \quad [\text{kWh}] \tag{15}
\]

To express Eq. (15) as \( \eta \), we can divide both sides by \( \Delta y h_{fg} \) then use the definitions in Eqs. (2) and (3) (where \( \Delta y \equiv y_f - y_i \)) to substitute \( \Delta m_w = \Delta y m_c \). After algebraic manipulation, this yields the ideal efficiency as a function of temperature shown in Eq. (16). Note that, unlike when sensible effects are ignored, the expression involves the moisture content in the term \( \Delta y \).

\[
\eta_{ideal,L+S}(T_f) = \frac{h_{fg}(T_f)}{h_{fg}(T_f) + (T_f - T_i) (C_{p,w} + C_{p,c})} \quad [-] \tag{16}
\]

The same process can be carried out with EF instead of \( \eta \) to obtain Eq. (17) and with SMER to obtain Eq. (18).

\[
EF_{ideal,L+S}(T_f) = \frac{1}{(T_f - T_i) (\Delta y C_{p,w} + C_{p,c} + \Delta y h_{fg}(T_f))} \quad \frac{\text{lbBDW}}{\text{kWh}} \tag{17}
\]

\[
SMER_{ideal,L+S}(T_f) = \frac{1}{h_{fg}(T_f) + (T_f - T_i) (C_{p,w} + C_{p,c})} \quad \frac{\text{lbBDW}}{\text{kWh}} \tag{18}
\]

### Table 2. Expressions for ideal evaporative drying efficiency.

| Efficiency metric | Expression for ideal evaporative drying efficiency | Value of ideal efficiency at 25°C |
|-------------------|--------------------------------------------------|----------------------------------|
| SMER              | \( SMER_{ideal} = \frac{1}{h_{fg}} \)            | 1.474                            |
|                   | \( \eta_{ideal} = 1 \)                           | 1 [-]                           |
| EF                | \( EF_{ideal} = \frac{1}{(y_f - y_i) h_{fg}} \)   | 6.08                            |

| Note to Table 2: | |
|------------------|.--.|
These expressions are plotted as a function of temperature in Figure 3. \( C_{p,c} \) was set to \( 0.389 \, \text{kWh/kg} \cdot \text{K} \) and \( C_{p,w} \) was evaluated at the average temperature of \( T_f \) and \( T_i \).

As temperature rises, the increased sensible heating tends to decrease ideal efficiency, but the decreased latent heat of vaporization partially counteracts this effect. Real processes will typically have thermal masses in addition to the thermal mass of the load itself, leading to efficiencies lower than this ideal.

### 3. Evaporative drying methods

In this section, process-specific efficiency limits are developed for various drying processes. Each process-specific efficiency limit applies for assumptions that define a specific process. In contrast, the generic definition of efficiency from the prior section (\( \eta_{\text{ideal}} = 1 \)) is simply defined based on a reference process in which all energy input to the dryer is used only to overcome water’s latent heat of vaporization. Emphasis is placed in this section on ERD and HPD.

#### 3.1. Natural air drying

Line drying, or drying clothes by hanging them, is a traditional method. In this configuration, the moisture content in the fabric evaporates and forms water vapor. The rate of the drying generally depends on the RH and the ambient temperature of the air, temperature of the fabric, type and thickness of the fabrics, airflow around the fabrics, and the surface area for evaporation. This form of drying does not use any primary energy and thus has an infinite value of \( \eta_{\text{ideal}} \): However, line drying generally requires more time than active drying methods, and space limits and visual considerations prohibit it in many locations. Computation of theoretical drying time is complicated, because it depends on so many uncontrolled factors, so it is not attempted here.

#### 3.2. Unheated forced air drying

When the airflow rate around wet clothes increases, the mass transfer boundary layer becomes thinner, providing an enhanced water concentration gradient.

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**Figure 2.** Ignoring sensible heating, ideal efficiencies rise with temperature (since the latent heat of vaporization declines with temperature). Values for \( EF \) computed for \( y_i = 0.575 \) and \( y_f = 0.04 \).

**Figure 3.** Accounting for the sensible heating of the cloth and moisture, ideal efficiencies are lower at higher temperatures because of the additional sensible heating required. Values for \( EF \) were evaluated at \( y_i = 0.575 \) and \( y_f = 0.00 \).
and hence a faster drying rate compared with natural air drying.

Modern clothes dryers in widespread residential and commercial use are almost exclusively of a heated forced-air type, with various heat sources. This section on forced-air drying looks at unheated forced air drying (UFAD), with heated methods covered in later sections.

Most conventional clothes dryers already have an “air fluff” or “no heat” or similar setting in which the heating element is turned off and a blower fan motor works to blow indoor air through the drum. In this mode, drying can still occur since indoor air is less than saturated (RH less than 100%). Because no heating is involved, the drying rate is typically slow, but the drying efficiency can be high. The main power input to such systems is to drive the drum and blower. When indoor conditioned air is used and discharged to the ambient air, this method discharges more conditioned air than other methods because of the long drying time, imposing a larger heating, ventilation, and air-conditioning (HVAC) burden on the building. Such building HVAC impacts were not considered in this study.

A relatively straightforward test was conducted with a conventional dryer. An

\[ \eta = 1.18 \] (EF of 7.2 lb\textsubscript{BDW}/kWh) at a drying time of 5 hours and 54 minutes was achieved when the intake ambient air temperature and humidity were at 21.3 °C (70.4 °F) and 68%, respectively, using a 3.83 kg\textsubscript{BDW} (8.446 lb\textsubscript{BDW}) load of US Department of Energy standard cloth. Figure 4 shows the weight of the load and the cumulative energy used by the system during this test. The figure uses an

![Figure 4](image_url)

**Figure 4.** Experimentally measured load wet weight and cumulative energy consumed by a conventional tumble-type clothes dryer in non-heated mode.

### 3.3. Electric resistance drying

Two models for ERD are described in this section. Most current household dryers fall into this category. In ERD, the heat comes from the dissipation of electrical energy across a resistive element. A summary of residential clothes dryer performance results can be found in Gluesenkamp,\footnote{35} in which two standard-size electric dryers, two compact-size electric dryers, and one standard-size gas dryer were studied. All the evaluated dryers were vented types, i.e., dryers that use indoor air as a source and exhaust warm, humid air through a duct to the outdoors.

As of January 1, 2015, the minimum EF efficiency requirements (from 10 CFR 430, Subpart C, §430.32\footnote{37}) for ERD in the United States are 3.73, 3.27, and 3.30 for electric standard, electric compact, and gas dryers, respectively. These requirements correspond to \( \eta = 0.54–0.66 \).

First, a simple model for analyzing the behavior of a conventional dryer is discussed. It is used to explain two phenomena of particular interest to the fundamentals of a vented ERD: in some conditions, \( \eta \) is limited to less than 1 (because of losses of sensible heat in the exhaust); and under other conditions, \( \eta \) can exceed 1 (because of the inherent, passive moisture removal capacity of unsaturated air, which a vented dryer uses).

The principle behind heating air before blowing it through moist cloth is to increase the moisture removal capacity of the air. But if the ambient air is less than saturated, then it starts with some inherent moisture removal capacity. The fact that non-saturated air has moisture removal capacity means that it is possible to exceed the ideal ERD efficiency of \( \eta = 1 \), if the heat input is small enough.

The model in this section assumes that ambient air is heated by the electric resistance heating element, undergoes an adiabatic (isenthalpic) humidification process in the drum, and leaves the drum in a saturated condition. The model has three independent variables. The one controllable variable in this model is the temperature to which air is heated before entering the drum; the model also depends on the (uncontrolled) ambient temperature and humidity.

Results for this model were computed using the psychrometric properties and simultaneous equation-solving capabilities of Engineering Equation Solver (EES)\footnote{39} with five ambient conditions (0 °C/50% RH, 25 °C/0% RH, 25 °C/50% RH, 25 °C/100% RH, and 50 °C/50% RH) and drum entering temperatures starting from 150 °C and decreasing to just over ambient temperature, as shown in Figure 5. The figure uses an
inverse y-axis to more clearly show that efficiency approaches infinity as the drum entering temperature approaches ambient temperature for non-saturated ambient conditions. If the ambient air is saturated, then the efficiency predicted by this model is relatively constant regardless of the drum inlet temperature, and drying time is reduced by a higher drum inlet temperature. If the ambient air is less than saturated, then there is a tradeoff between drying time and efficiency, with lower temperatures leading to higher efficiency but longer drying time.

For the case with saturated air, efficiency for this idealized model is about \( g_{\text{max}} = 0.74 \) (i.e., \( g^{-1} = 1.35 \)), significantly less than \( g = 1 \). This is because the saturated air leaving the drum has a dry bulb temperature above ambient, representing unusable sensible energy. For a modified process with an additional HX, this sensible energy could be used to pre-heat incoming air and bring \( g_{\text{max}} \) closer to 1. For the cases with non-saturated ambient air, \( g = 1 \) when the air leaving the drum has a dry bulb temperature equal to the ambient. With enough drying time, unsaturated air can be used with zero heat input, resulting in \( g_{\text{max}} = \infty \) (i.e., \( g^{-1} = 0 \)).

Thus, it is apparent that ERD and UFAD exist on a continuum. One extreme is defined by a lack of added heat, whereas the other is defined by a process in which the moisture removal capacity of the source air is negligible compared with heat addition (owing to nearly saturated ambient conditions, and/or large heat additions). Actual ERD devices operate somewhere on this continuum, with units operated in hot, dry surroundings taking advantage of the favorable ambient psychrometric conditions, and having a performance limit that can be greater than \( g = 1 \).

Stated another way, the performance limit for any vented dryer depends on ambient conditions. The simplest case to analyze is when the ambient air is saturated—then the performance limit is a function of ambient temperature and drum entering temperature and is always \( g < 1 \) (0.74 at \( T_{\text{ambient}} = 25 \degree \text{C} \), lower for colder and higher for higher \( T_{\text{ambient}} \)). However, for unvented dryers in unsaturated ambient air, the performance limit defies a concise definition, since it depends on (1) ambient temperature, (2) ambient humidity, and (3) the dry time. A simple analytical solution is not available, so a map of the relationship was established graphically for a few selected cases in Figure 5, which relies on specific assumptions about the load and the air flow. To compute a map for a more general case (for different load size, air flow rate, starting moisture content, and/or final moisture content) would require re-solving the set of simultaneous equations (including psychrometric properties) used to generate the figure.

For each curve in Figure 5, the left-most point is at 150 °C drum entering temperature, and each

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**Figure 5.** The inverse of \( g \) vs. drying time for various ambient conditions and drum entering temperatures, based on a simple ERD process model in which ambient air is heated by ER, enters the drum, then follows an adiabatic humidification process to saturation.
subsequent point represents an incremental decrease of 25°C. Each non-saturated curve ends at $\eta_{\text{max}} = \infty$ (i.e., $\eta^{-1} = 0$) for a drum entering temperature equal to the ambient (UFAD). The saturated curve does not reach zero, since no drying can occur without heating in saturated air. Drying times were computed assuming the drying time parameters (load size, air flow, starting and final moisture contents) in Table 3.

For the standard case of ERD at $T_{\text{ambient}} = 25^\circ\text{C}$ and $RH_{\text{ambient}} = 50\%$, $\eta_{\text{max}} = 0.79$ for $T_{\text{drum, entering}} = 100^\circ\text{C}$ (corresponding to a drying time of 20.3 minutes), $\eta_{\text{max}} = 0.83$ for $T_{\text{drum, entering}} = 75^\circ\text{C}$ (corresponding to drying time of 29.6 minutes), and $\eta_{\text{max}} = 1.13$ for $T_{\text{drum, entering}} = 40^\circ\text{C}$ (corresponding to drying time of 74.3 minutes). Note that all cases with unsaturated ambient air will have a higher efficiency limit than cases with saturated air at the same dry bulb temperature (recall $\eta_{\text{max}} = 0.74$ for vented ERD with saturated 25°C ambient air).

### 3.4. Natural gas drying

Although the focus of this paper is on electrical energy efficiency, the large market presence of gas-powered dryers merits mention. In NGDs, the heat comes from direct combustion of natural gas. NGDs have the same maximum efficiency $\eta$ on a site energy (or final energy) basis as ERDs, although the calculation differs on a source energy (or primary energy) basis. Figure 5 generally applies to NGDs as well as ERDs.

### 3.5. Heat pump drying

HPD differs from ERD because air is heated by the hot HX of a heat pump, and the exhaust from the drum is used as a heat source for a cold HX. Ambient heat can also be used in addition to or instead of heat from the exhaust. Since an HPD can condense moisture from the exhaust, it can be configured as unvented as well as vented. Sections 3.5.1 and 3.5.2 analyze the performance of a generic HPD process for unvented and vented operation. Sections 3.5.3 through 3.5.5 analyze more specifically Carnot HPDs, which require more in-depth model development. First, general considerations are discussed that apply to both unvented and vented dryers (3.5.3), then unvented and vented units are explored individually (Sections 3.5.4 and 3.5.5, respectively). Finally, VCHPDs and TEHPDs are discussed in Sections 3.5.6 and 3.5.7.

#### 3.5.1. HPD—unvented

In the case of unvented operation without air leakage, moisture removed from the clothes can be removed from the dryer system only when it condenses at the cold evaporator (neglecting the small mass of water vapor present in the drum and in any ductwork). Furthermore, any sensible heat removed from the closed air stream at the evaporator must be added again at the condenser, so sensible cooling does not contribute to moisture extraction. In other words, only the latent heat fraction (LHF) of the evaporator cooling capacity contributes to moisture extraction, where the LHF is defined as the air-side water condensation capacity at the evaporator divided by the total evaporator cooling capacity. In addition, the cooling coefficient of performance (COP$_c$) is defined as the total (latent plus sensible) cooling capacity divided by electrical consumption. Thus, the minimum energy for a generic unvented HPD can be defined as shown in Eq. (19). Substitution of the minimum energy term for the “E” term in Eq. (6) yields the ideal (or maximum) efficiency shown in Eq. (20).
Emin, HPD, unvented \[= \frac{(m_{w,i} - m_{w,f}) \cdot h_g \cdot \text{LHF}}{\text{LHF} \cdot \text{COP}_c} \] \[= \eta_{\text{ideal, HPD, unvented}} = \text{LHF} \cdot \text{COP}_c \] \[= \text{COP}_c \]

COPc is a function of various heat pump operating and design variables and can be evaluated using any relevant heat pump of interest.

Evaluating the LHF of the cold HX requires consideration of air psychrometric properties. The LHF is a function of four parameters: the temperature and humidity entering and leaving the cold HX. Figure 6 shows the LHF as a function of temperature leaving the drum, with contours shown for a typical range of RH leaving the drum (which relates to drum heat and mass transfer effectiveness) and RH leaving the cold HX (which relates to the HX condensation effectiveness). Typical heat and mass transfer effectiveness values for the drum are provided in Gluesenkamp et al. Figure 6 shows that LHF is fairly sensitive to the RH leaving the drum and thus to drum effectiveness, with high drum effectiveness resulting in a higher LHF and therefore a higher drying efficiency \( \eta \).
The evaporator condensation capability also helps efficiency, especially for low air temperatures leaving the drum. Another observation is that a higher evaporator-leaving air temperature (i.e., Figure 6b vs. a) leads to a higher LHF.

3.5.2. HPD—vented

In the case of vented operation, the drying capacity is not limited by condensation at the cold HX. Thus, the expression for maximum drying efficiency is closer to the ERD case, with a modification for the heating $\text{COP}_h$. However, as shown in Section 3.3 for vented ERDs, vented efficiency is a function of ambient temperature, ambient humidity, and drying time. Saturated ambient conditions ($\text{RH}_{\text{ambient}} = 100\%$) are the simplest to analyze. In the ERD case, vented efficiency with saturated ambient air was always $< 1$. Similarly, for a generic vented HPD, we can use Eqs. (21) and (22), which apply for saturated ambient conditions (indicated by the superscript “saturated.”) When the ambient conditions are not saturated (i.e., $\text{RH}_{\text{ambient}} < 100\%$), then the actual performance limit will be higher and will depend on the drying time as well as the ambient conditions. Graphical and/or analytical mapping of these more detailed performance limits for non-saturated ambient conditions for generic vented HPDs is left for future work.

3.5.3. Carnot HPD—general considerations

A Carnot cycle is defined as operating between a high-temperature reservoir and a low-temperature reservoir. In the following discussion, the high-temperature reservoir temperature is referred to as $T_H$ and the low-temperature reservoir temperature as $T_C$. The difference between them (called the heat pump temperature lift, or simply “lift”) is referred to as $\Delta T_{\text{lift}} = T_H - T_C$.

As shown in Figure 7, $T_H$ corresponds to the air temperature leaving the hot HX (entering the drum), and $T_C$ corresponds to air temperature leaving the cold HX. Figure 7 also shows the psychrometric chart for one representative condition of each Carnot dryer. For the present discussion, it is assumed that all heat is transferred at $T_H$ and $T_C$. Since the air side of the cycle is not at fixed $T_H$ and $T_C$ but rather undergoes temperature glide, a slightly more ideal system could be defined in which heat is the transfer heat at the isentropic average temperatures $T_{H,s}$ and $T_{C,s}$ that result from the air-side temperature glide. The simpler, fixed-temperature approach is taken in this work. Energy consumption required to implement...
forced air flow (blower power) is neglected in the present Carnot analysis.

With a zero-order analysis, the theoretical maximum efficiency will approach infinity for zero $\Delta T_{lift}$ (the same as for natural air drying). However, going to higher temperature lifts is desirable to reduce drying time.

A quantitative ideal, Carnot-based unvented HPD and a vented one can be defined based on the general assumptions indicated in Table 3 and the state-point–by-state-point assumptions enumerated in Table 4. Table 4 indicates free variables in bold, as discussed in the sections on unvented and vented Carnot HPD.

### 3.5.4. Carnot HPD—unvented

For the unvented Carnot HPD, the governing equation is Eq. (20), where COP$_c$ can be obtained by solving for Carnot COP$_c$. To obtain the Carnot COP$_c$, the $T_H$ and $T_C$ must be solved based on psychrometric properties and the assumptions outlined in Table 3 and Table 4. Results were computed using the psychrometric properties and simultaneous equation-solving capabilities of EES.$^{[39]}$ The result for a single case is shown in Table 5, where $T_H = 60^\circ\text{C}$ and $T[5] = 25^\circ\text{C}$.

Note that the unvented Carnot HPD has two degrees of freedom. $T_H$ and $T_C$ may seem natural choices for the two free variables; but the authors have chosen $T_H$ and $T[5]$ as free variables in solving the system of equations, where $T[5]$ is bounded between ambient temperature and $T_H$. This choice was made because $T[5]$ has a clear lower bound (i.e., the ambient temperature), whereas the bound on $T_C$ is implicit, making $T[5]$ a better choice for a free input variable.

The computed results for a range of values for $T_H$ and $T[5]$ are shown in Figure 8 (a, b, and c). These can be viewed as “Pareto plots,” showing the performance (efficiency and drying time) for various Carnot system design choices. The most desirable system would reside in the upper left of the diagram, with high efficiency and fast dry time.

Figure 8a shows a fixed $T[5]$. Importantly, Figure 8a shows a fundamental tradeoff between efficiency and drying time. For a given $T[5]$, raising $T_H$ makes drying time faster but decreases efficiency. Furthermore, the tradeoff is almost directly proportional for this unvented Carnot HPD system: a two-fold increase in drying time corresponds to an approximately two-fold increase in efficiency.

Figure 8b assumes a fixed $T_H$. For a given $T_H$, lowering $T[5]$ also makes drying time faster at the expense of decreased efficiency. In this case, the tradeoff is fairly linear, but with a different proportionality. A two-fold increase in drying time corresponds to an approximately three-fold increase in efficiency.

### Table 4. Assumptions by state point used to define Carnot heat pump dryers.

| State point | Unvented | Vented |
|-------------|----------|--------|
| Ambient condition: state point [1] | $T[1] = 25^\circ\text{C}$ unless otherwise indicated | $RH[1] = 50\%$ unless otherwise indicated |
| Leaving the hot HX, entering the drum (state point [2]) | (ambient RH is not applicable: no mass exchange with ambient) | $T[2]$ (free variable) |
| Drum psychrometric process (state point [3]) | $w$ | $w[2] = w[1]$ |
| Air leaving cold heat exchanger (state point [4]) | $h$ | Adiabatic (isenthalpic) process $h[3] = h[2]$ |
| Additional heat rejection (state point [5]) | $RH$ | None |

*Bounds for $T[5]$ are ambient temperature and $T_H$, i.e., $T_{ambient} = T[1] < T[5] < T[2] = T_H$.

### Table 5. State points and performance metrics for one representative case of unvented Carnot heat pump dryer, with $T_H = 60^\circ\text{C}$ and $T[5] = 25^\circ\text{C}$.

| State points | T [°C] | RH [%] | $h$ [kJ/kg] | $w$ [g w/gdA] |
|--------------|--------|--------|-------------|---------------|
| [1]          | 25.0   | NA     | NA          | NA            |
| [2]          | 60.0   | 15.9   | 112.7       | 0.0201        |
| [3]          | 32.4   | 100    | 112.7       | 0.0313        |
| [4]          | 25.9   | 100    | 80.0        | 0.0212        |
| [5]          | 25.0   | 100    | 76.3        | 0.0201        |

*Assumes load parameters and air mass flow in Table 3.
Figure 8. Ideal efficiency of an unvented Carnot heat pump dryer. In (a), T[S] is fixed at 25°C while $T_H$ is varied in 5 K increments from 35 to 120°C. In (b), $T_H$ is fixed at 60°C while T[S] is varied in 5 K increments from 10 to 45°C. In (c), the “temperature lift” ($T_H - T_C$) is fixed at 35°C while $T_H$ is varied in 10 K increments from 30 to 120°C. Drying times are based on the load parameters and air mass flow in Table 3.
In contrast to the inherent tradeoffs in Figure 8a and b, in Figure 8c, we observe that both faster drying times and higher efficiency are possible by simultaneously elevating $T_C$ and $T_H$. As we move from low to high temperature (right to left in the figure), large improvements occur in efficiency as results of improved LHF (according to governing Eq. [20] for unvented systems). In addition, improvements in drying time occur as result of the enhanced moisture holding capacity of warmer air. The continued rise in efficiency even as LHF levels out (around $T_H = 100^\circ C$) is due to the nature of the Carnot heat pump: since $\text{COP}_{C,\text{Carnot}} = \frac{T_C}{(T_H - T_C)}$, when the denominator (the lift) is fixed, the COP will be directly proportional to the absolute temperature $T_C$. Of course, it is important to remember that real systems will pay a penalty for heating dead thermal masses (of the load and other components) when elevating all cycle temperatures, an effect assumed to be zero in this Carnot analysis.

Finally, although Figure 8 shows only a limited range of values, extended computations with this model confirm the prediction of the zero-order model in the previous section. At one limit, as $T_H$ approaches $T_C$, drying time and efficiency approach infinity (results are shown for a drying time of up to 2.5 hours, approaching the natural air drying duration). At the other limit, as $T_H - T_C$ approaches infinity, the drying time and efficiency approach zero (results are shown up to $T_H = 120^\circ C$). Indeed, at $T_H = 314^\circ C$ and $T[5] = 25^\circ C$, the ideal unvented Carnot HPD has $\text{COP}_{\text{cooling, Carnot}} = 1.00$ (approaching parity with an ideal ERD), with drying time $= 4.5$ minutes, $T_C = 45.3^\circ C$, $\text{COP}_{\text{heating, Carnot}} = 1.19$, and LHF $= 0.84$. This theoretical calculation ignores practical impacts of such a high $T_H$, such as unavailability of suitable compressor technologies and melting of polymer fabrics.

### 3.5.5. Carnot HPD—vented

There are significant differences between unvented and vented Carnot dryers. In vented operation, the cold HX does not have to condense the water to achieve drying; that HX serves only to recover sensible and latent heat from the exhaust. In addition, the ambient psychrometric condition must be considered (as was the case with a vented ERD), adding complexity to the analysis. The reader can contrast the assumptions used to construct the Carnot vented HPD with unvented assumptions by inspecting Table 3 and Table 4. One difference that merits mention is that, in the unvented case, moisture removal occurred via condensation at $T_C$, and $\text{COP}_C$ was the relevant heat pump efficiency metric. In contrast, in the unvented case, moisture removal does not depend on condensation, and $\text{COP}_h$ is the relevant heat pump efficiency metric. This is reflected in the assumptions for state point [4] in Table 4.

For the vented HPD, a special case exists when ambient air is saturated, for which the governing equation is Eq. (22), in which the $\text{COP}_h$ can be obtained by solving for Carnot $\text{COP}_h$.

To solve the vented Carnot HPD with unsaturated ambient air, a model must be used such as the one developed for the ERD, in which psychrometric properties are considered. This does not lend itself to a concise expression, and is addressed in this work by a system model, defined by the simultaneous equation set shown in Table 3 and Table 4. Ambient air is heated by the Carnot heat pump hot HX and then undergoes an adiabatic humidification process in the drum ending in saturation. The Carnot $\text{COP}_h$ is calculated based on the system temperatures, and the energy consumption is the heat added to the air divided by the $\text{COP}_h$. Note that the heat added to the air ($Q_H$) can be less or more than the latent heat removed from the load, depending on ambient humidity and the drying time. Drier ambient air can yield efficiency limits higher than indicated in Eq. (22), especially for long drying times.

Previously, in the unvented case, a choice had to be made about how much heat to reject from the system. If more were rejected, $T_C$ would be lower for a given $T_H$, and vice versa. However, there is no analogous choice in the vented system, since we have assumed that all heat at $T_C$ comes from the process air. Thus, whereas the unvented Carnot HPD has two degrees of freedom related to the system design ($T_H$ and $T[5]$), the vented system has only one ($T_H$).

On the other hand, whereas the unvented system is impacted only indirectly by $T_{\text{ambient}}$ (which imposes a lower bound on $T[5]$), the vented system is directly affected by both $T_{\text{ambient}}$ and $RH_{\text{ambient}}$. From the modeling perspective, these two ambient conditions can be considered two additional degrees of freedom, although they are not free variables for system design nor controls.

In summary, the unvented Carnot dryer has two degrees of freedom ($T_H$, $T[5]$), and both are controllable. The vented Carnot dryer has three degrees of freedom ($T_H$, $T_{\text{ambient}}$, $RH_{\text{ambient}}$), and only one is controllable ($T_H$). To obtain the Carnot $\text{COP}_h$, for a vented Carnot dryer, $T_H$ and $T_C$ were solved based on psychrometric properties and the assumptions outlined in Table 3 and Table 4. Results were computed...
using the psychrometric properties and simultaneous equation-solving capabilities of EES \(^{[39]}\). The result for a single case is shown in Table 6, where \(T_H = 60^\circ C\), \(T_{ambient} = 25^\circ C\), and \(RH_{ambient} = 50\%\). This is comparable to the representative case in Table 5 for unvented operation, but the vented system has higher efficiency with faster drying time, despite the lower cycle COP. The cycle COP is lower because of a lower \(T_C\), but the drying efficiency is higher because of the use of unsaturated air which provides some “free” cooling.

Figure 9 shows the performance of the vented Carnot HPD for \(T_H\) varying from 30 to 80 °C. Notably, the \(T_C\) is lower than the assumed ambient temperature of 25°C, indicating that higher performance could be obtained by using the ambient as a heat source, instead of the air leaving the drum. In the limit of \(T_H\) approaching \(T_{ambient}\), \(T_C\) will approach the ambient wet bulb temperature (17.9 °C in the present case of 25 °C/50% RH). Exploration of a Carnot HPD sourcing heat from the ambient is left for future work.

Next, the impact of ambient conditions for the vented Carnot HPCD is explored in Figure 10. The performance of the Carnot vented heat pump shows counterintuitive trends as ambient conditions vary. The impacts of ambient humidity and of dry bulb temperature are discussed separately.

First, higher ambient humidity (at a given dry bulb temperature) increases cycle COP, has negligible effect on drying efficiency (\(\eta\)), and lengthens drying time. This behavior can be traced to the fact that higher drum entering humidity causes the drum’s adiabatic humidification process to stop at a higher temperature. The resulting higher temperature leaving the drum raises the cold HX temperature and improves the cycle COP. This increase in COP is compensated for by the “free” drying capacity of the unsaturated ambient air (i.e., the COP rises approximately proportionally to \(Q_H\)), leading to a similar drying efficiency. Finally, a longer drying time results from the decreased difference in the humidity ratio across the drum.

Second, a higher ambient dry bulb temperature (at a given humidity ratio) increases efficiency and has no

| State points | T [°C] | RH [%] | h [kJ/kg] | w [g/gda] |
|--------------|--------|--------|-----------|-----------|
| [1]          | 25.0   | 50     | 50.3      | 0.0099    |
| [2]          | 60.0   | 7.9    | 86.1      | 0.0099    |
| [3]          | 27.2   | 100    | 86.1      | 0.0230    |
| [4]          | 19.2   | 100    | 54.7      | 0.0139    |

Performance metrics

| COPheating [-]          | 8.16 |
| \(\eta\) [-]           | 7.31 |
| EF \(\eta_W/\eta\) [\%] | 44.4 |
| SMER \(\eta_\text{SMER}\) [\%] | 10.8 |
| Dry time [minutes]      | 40.2 |
| LHF [-]                 | NA   |

*Assumes load parameters and air mass flow in Table 3.
effect on drying time. The increased efficiency can be attributed to the lower capacity required of the hot HX for a given drum inlet temperature. Meanwhile, the drying time is unaffected because the ambient dry bulb does not influence the change in humidity ratio across the drum. Note that this conclusion is based on assuming a constant mass flow rate of dry air through the system (0.0646 kg_da/s). An alternate assumption of constant volume flow can result in small changes in drying time with changes in the ambient dry bulb temperature because of changes in air density at some state points.

3.5.6. Vapor compression HPD
The VCHPD has seen significant market penetration in many countries. In a VCHPD, air is heated by a VC cycle condenser, and the exhaust from the drum is used as a heat source for the VC system evaporator. It can be configured as either vented or unvented, and

Figure 10. Heat pump cycle COP (a) and efficiency (b) for a vented Carnot heat pump clothes dryer. Higher ambient absolute humidity lengthens drying time and has a negligible impact on drying efficiency (despite higher cycle COP). Higher ambient dry bulb temperature increases drying efficiency, with no impact on drying time. Contours are shown for three ambient dry bulb temperatures. The right-most point of each contour line is at 100% RH; the left-most point is at 0% RH. Starting from 0 humidity, absolute humidity increments of 0.00762 g_w/g_da are shown (the final step to reach 100% RH is of a different size).
both vented and unvented product types are currently offered on the market.

The performance limits of a VCHPD will correspond to Eqs. (19)–(20) for unvented operation and Eqs. (21)–(22) for saturated vented operation, where the COPc or COPh is computed for the VC heat pump under the relevant conditions. Graphical and/or analytical mapping of the HPD performance limits is restricted in this work to the Carnot HPD cases.

Thermoelectric HPD. It is possible to use any heat pump technology to heat the air entering the drum and extract heat from the drum exhaust. Besides the VC cycle, the TE heat pump has been applied to fabric drying. Like the VCHPD, the TEHPD can be implemented in a vented \[16,20\] or unvented \[19\] configuration. The equations for maximum efficiency are thus very similar to those for a VC system, but “LHF” in the case of a TE-based HPD refers to the LHF at the cold HX (not the evaporator), and “COP” refers to the TE COP.

3.6. Vacuum drying

Some research has been conducted on vacuum drying, e.g., by the Environmental Protection Agency \[26\]. A fundamental challenge of vacuum drying is that, although resistance to evaporation and condensation is dramatically lower in a vacuum, air is no longer available as a heat transfer medium. In addition, heat needs to be added from a source, which will likely make this technique similar to atmospheric pressure drying methods. Thus, vacuum drying can generally be thought of as a drying speed enhancement, rather than an efficiency improvement tactic, and its efficiency limits are described in other sections of this work, according to the heat source used in vacuum drying.

3.7. Microwave drying

Microwave drying is based on the volumetric heating of water by electromagnetic radiation at 915 or 2450 MHz \[41\]. This technology has been used for potential applications in clothes drying \[21,22\], textile drying \[23\], and drying of lossy food products \[41,42\]. For clothes drying, microwave drying might not be an appealing option mainly because of the thickness of the garments in a full load of a laundry and the interaction between microwaves and metal components of garments. Besides these challenges, considering the 80–90% coupling efficiency between microwaves and moisture, the 60–85% efficiency of conversion of electricity to microwaves, and the 80–90% efficiency of AC to DC conversion, microwave fabric drying is expected to have an efficiency range between 38 and 68%. This is no higher than the efficiency of ER-based conventional residential clothes dryers. But if energy conversion losses are recovered and applied to the drying process, the ER efficiency limit can be approached.

3.8. Infrared drying

Similar to microwave energy, if infrared (IR) energy is produced electrically, it will have a maximum efficiency approaching that of electric resistance because the conversion of electricity to radiation has a maximum efficiency of 100%.

One case of IR heating–based drying technology for residential clothes drying has been reported \[25\]. In this work, an IR heater with a wavelength tuned to 2.5–10 micron (to correspond with the 3 micron water absorption peak) was combined with an electrostatic precipitator in a ventless design. Other work has involved a pilot-scale IR oven with tubular radiant heaters used for pre-drying of textiles \[24\]. The experimental study consisted of passing sheets of fabric through a 20 kW electric IR dryer and measuring the moisture content and surface temperature of the fabric as a function of its speed. Several different fabric types were tested of different widths (including cotton, polyester, nylon, and blends). View factor calculations were performed based on the dryer geometry, and the effects of distance (between IR sources and fabric) and fabric width were studied for medium- and short-wave IR sources. The results showed that a maximum drying efficiency of 73% (i.e., \(\eta = 0.73\)) was possible for this specific dryer design. Although these studies show the potential of IR drying, they are limited in scope and application. In this paper, IR drying is not explored further and is effectively lumped together with ERD.

4. State of the art performance

In this section, the state of the art is established for ERD and HPD dryer types through a brief review of drying types selected from the available literature. They are summarized in Figure 11, and each is described below.

4.1. Electric resistance drying

Commercially available performance for standard efficiency vented ERDs is reported in \[43\]. Multiple cloth
types were investigated, but in this work we report only results for the standard DOE cloth. Across four commercially available units, depending on the cycle termination criteria, drying time ranged from 22 to 40 minutes, and energy factor ranged from 2.54 to 3.99 ($\eta = 0.418$ to 0.657).

Gluesenkamp\[35\] also evaluated four standard efficiency commercially available ERD units with multiple cloth types. With standard DOE cloth, drying duration ranging from 18.62 to 48.99 minutes (again primarily depending on the cycle termination criteria, and also user-selectable dryer settings), and energy factor ranged from 2.98 to 3.92 ($\eta = 0.491$ to 0.646).

Commercially available performance for higher-efficiency vented ERDs is available in the ENERGY STAR qualified product list\[44\]. Two key criteria for inclusion in this list are combined energy factor of at least 3.93 ($\eta = 0.647$), and a dry time faster than 80 minutes. Of the hundreds of entries in the list for ERDs, the fastest dry time reported is 50 minutes, and the longest is 78 minutes.

Next, a prototype (non-commercially available) unit was selected for inclusion in the state of the art to represent improved efficiency possible with techniques under research to improve ERD performance. Jian and Luo \[45\] modified an ERD to improve efficiency by including a heat pipe heat recovery heat exchanger. The reported SMER was up to 1.328 ($\eta = 0.901$), and the drying rate was 0.418 g/s, which would correspond to a drying time of 81.8 minutes for a 3.83 kg load with 57.5% SMC and 4% FMC.

Regarding unvented ERD units, drying time is generally significantly longer than vented ERD. Jian et al. \[46\] reported on the application of a heat recovery heat exchanger to an unvented ERD, with SMER up to 1.448 ($\eta = 0.982$). The drying rate was reported as 0.203 g/s which would correspond to a 168 minutes drying time for the 3.83 kg load with 57.5% SMC and 4% FMC.

### 4.2. Heat pump drying

Commercially available performance for HPDs is reported in \[44\]. A vented product is reported at CEF of 4.5 ($\eta = 0.741$) with a dry time of 75 minutes. An unvented product is reported with CEF of 5.2 ($\eta = 0.857$) and a dry time of 70 minutes. Performance is also characterized for another unvented commercially available HPD in \[8\], with CEF of 7.75 ($\eta = 1.28$), with a dry time of 120 minutes. The same researchers modified the off the shelf prototype to incorporate a vapor-injected two-stage vapor compression cycle, and achieved a CEF of 10.295 ($\eta = 1.70$) with a dry time of 120 minutes.

Other prototype HPDs reported in the literature include an unvented unit reported by Bengtsson et al. \[48\] with SMER of 2.167 ($\eta = 1.47$). The drying rate was reported as 0.511 g/s which would correspond to a 67 minute drying time for a 3.83 kg load with 57.5% SMC and 4% FMC.

Finally, in a model-based study, TeGrotenhuis et al. \[11\] project that the addition of a R134a-based vapor compression cycle and passive recuperative heat exchanger to a vented dryer should deliver CEF between 5.4 and 6.0 ($\eta = 0.890$ to 0.988) with dry time of 57-71 minutes. Mean values ($\eta = 0.939$ and 64 minutes) are shown in Figure 11.

### 5. Discussion

Based on the analysis in this paper, Figure 12 summarizes the feasible performance range of four drying technologies (ERD, HPD-vented, HPD-unvented, and UFAD) and the corresponding state-of-the-art performance for ERD, HPD, UFAD, and natural air drying. Since the ideal technology would be in the upper left of the figure (high efficiency and short drying time), the feasible region for a technology includes everything below and to the right of the performance limit curve. Drying time was calculated assuming a cloth load mass of 3.83 kg, starting water mass ratio $y_i = 0.575$, ending water mass ratio $y_f = 0.04$, and mass flow of air $= 0.0646$ kg$_{ds}$/s. For different $y_i$, $y_f$, or
efficiency metrics, the reader is referred to Table 1 for conversions. Performance limit curves were computed for ERD and HPD by solving the governing efficiency and drying time equations developed in Section 3 across a range of drum air entering temperatures. For UFAD, since blower power was neglected in the analysis, the performance limit is a point at infinite efficiency and 182 minutes drying time.

Comparing performance limits with actual state-of-the-art performance reveals that ERD is achieving a high fraction of its fundamental potential—approximately 65–75% of ideal efficiency at a given drying time achieved in real machines. In contrast, VCHPDs are very far from their ideal—approximately 2–20% of their ideal efficiency at a given drying time.

The analysis presented also reveals the critical importance of reporting both efficiency and drying time for any drying technology. Efficiency cannot be evaluated in isolation, because it fundamentally depends on drying time. Furthermore, for drying time to be meaningful, at least y_i, y_f, and m_c are needed.

It is instructive to analyze the temperatures shown for each technology in Figure 12. For each technology, a higher drum entering temperature corresponds to lower efficiency and faster drying time. The minimum and maximum drum air entering temperatures for each of the fundamental limit curves in Figure 12 are shown in Table 7, along with the corresponding efficiency and drying time.

The performance of all vented processes (ERD, vented Carnot HPD, and UFAD) converge as the amount of heat addition approaches zero. As this limit is approached, the efficiency approaches infinity and the drying time approaches approximately 3 hours for the load and ambient conditions specified in Figure 12. With very low heat addition, the performance is

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Table 7. Minimum and maximum drum air entering temperatures in Figure 12.

| Drying Technology          | Lowest drum air entering temperature shown in Figure 12 | Highest drum air entering temperature shown in Figure 12 |
|----------------------------|---------------------------------------------------------|---------------------------------------------------------|
|                            | Temp. [°C] | η [-] | Drying time [min] | Temp. [°C] | η [-] | Drying time [min] |
| ERD                        | 25.07      | 100   | 181             | 150        | 0.76  | 12.2             |
| Carnot HPD-vented           | 33.7       | 100   | 88              | 100        | 4.0   | 20.3             |
| Carnot HPD-unvented         | 27.1       | 100   | 818             | 100        | 3.3   | 21.6             |
| UFAD                       | 25.0       | infinite | 182         | 25.0       | infinite | 182             |

Figure 12. Fundamental performance limits and state of the art of evaporative fabric drying technologies, at a standard ambient air condition.
strongly dependent on ambient humidity. At high heat addition (higher drum air entering temperature), the performance is only weakly dependent on ambient humidity. In contrast, unvented processes (assuming zero leakage to/from the ambient) do not approach infinite efficiency at a finite drying time. In other words, as heat addition and removal approach zero for an unvented drying process, the drying time approaches infinity.

As the TH continues to rise for the Carnot HPDs, their theoretical performance limit eventually approaches that of ERDs. With \( T_H = 100 \, ^\circ C \), Carnot dryers have approximately 25 times higher efficiency than an ideal ERD. Such a dramatic performance differential persists even up to temperatures that would involve melting or combustion of fabrics, and \( T_H \) up to only \( 100 \, ^\circ C \) is shown in the diagram.

Additional observations from Figure 12 include these:

- At the same drum air entering temperature, the ERD and unvented HPD have the same minimum drying time.
- There is a cross-over in performance of ERD and unvented Carnot HPD (at approx. 2.5 hours and \( \eta = 20 \)). This is because the unvented system suffers from the inability to take advantage of the “free” evaporative capacity of unsaturated ambient air.
- For the case of saturated ambient air (not shown), the vented and unvented Carnot HPD performance curves converge.

6. Conclusions

Fundamental performance limits were derived for ERD and Carnot HPD and compared with state-of-the-art fabric drying performance. Efficiency and drying time were found to be directly dependent on each other, a finding that requires that future discussions of efficiency address drying time. State-of-the-art ERD was found to operate at 65–75% of its maximum theoretical efficiency for a given drying time, while state-of-the-art HPD operates at only 2–20% of its performance limit.

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Disclosure statement

The authors report no conflicts of interest. The authors alone are responsible for the content and writing of the paper.

Nomenclature

| Symbol | Definition |
|--------|------------|
| BDW    | bone-dry weight, lb |
| \( C_{P,w} \) | specific heat of water, kWh/kg°C |
| \( C_{P,c} \) | specific heat of dry cloth, kWh/kg°C |
| COPc   | cooling coefficient of performance |
| COPe   | heating coefficient of performance |
| \( E = \) | energy, kWh |
| \( EF = \) | energy factor, lbBDW/kWh |
| FMC    | final moisture content |
| \( h_{lg} \) | latent heat of evaporation of water, kWh/kg, or kJ/kg |
| \( H = \) | enthalpy, J/mol |
| HPD    | heat pump dryer/drying |
| HX     | heat exchanger |
| LHF    | latent heat factor |
| \( m = \) | mass, kg |
| \( \dot{m} = \) | mass flow rate, kg/s |
| NA     | not applicable |
| \( P = \) | pressure, kPa |
| \( Q_C = \) | heat transfer at the cold reservoir of a Carnot heat pump, kW |
| \( Q_H = \) | heat transfer at the hot reservoir of a Carnot heat pump, kW |
| \( R = \) | universal gas constant, J/mol·K |
| RMC    | relative moisture content, % |
| SMC    | starting moisture content, % |
| SMER   | specific moisture extraction rate, kgw/kWh |
| \( T = \) | temperature, °C |
| TE     | thermoelectric |
| \( V = \) | volume, m³ |
| VCS    | vapor compression heat pump system |
| \( y = \) | mass ratio, kgw/kg |

Subscripts/superscripts

| Symbol | Definition |
|--------|------------|
| BDW    | bone dry weight |
| c      | cloth |
| Ct     | Carnot |
| da     | dry air |
| \( i = \) | initial |
| \( f = \) | final |
| fab    | fabric |
| l      | liquid |
| L+S    | both latent and sensible heat |
| \( sat = \) | saturation |
\[ v = \text{vapor} \]
\[ w = \text{water} \]
\[ ww = \text{wet weight (combination of cloth and liquid water)} \]

**Greek symbols**

\[ \rho = \text{density, g/cm}^3 \]
\[ \eta = \text{instantaneous drying efficiency} \]
\[ \bar{\eta} = \text{average drying efficiency} \]

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