Design of an Efficient Clothes Dryer

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Abstract

A dryer was designed with the objective of improving its efficiency through the addition of a heat exchanger. The dryer is to operate in an apartment with surrounding conditions of 21.1 ˚C and 50% relative humidity according to [1]. The dryer is required to run for 45 minutes and remove 2.268 kg (5 lbs) of water from the clothes that enter the dryer at the ambient conditions. The maximum temperature in the dryer, 100˚C, occurs immediately after the heating element [1]. The final design of the dryer specifies the relative humidity at the exit of the dryer as 90%. This sets the temperature of the air exiting the dryer drum at 35.96 ˚C. The air mass flow rate through the dryer is 0.03167 kg/s, and the volumetric flow rate is 1.584 m^3/min (55.94 cubic feet per minute). The dryer draws an instantaneous power load of 2.554 kW and requires 1.916 kWh of energy for a 45-minute drying cycle. This translates to a cost of $0.25 per load based on current electricity costs in Northeast Ohio [2]. The pressure loss through the ducting and the lint filter of the dryer is 1117 Pa which requires a fan driven by 38 W to pull the air through the entire system. While a heat exchanger was designed to accompany the dryer and extract heat from the hot exhaust of the dryer, based on economic considerations, the inclusion of the heat exchanger cannot be recommended. The designed heat exchanger is a concentric counterflow double pipe heat exchanger with air and water vapor gas mixture as the fluid in both the annulus and inner pipe. The final heat exchanger design calls for copper pipes with a 5" (0.127 m) nominal inner diameter outer pipe, a 3" (0.07062 m) nominal outer diameter inner pipe, and a wall thickness of 1/16" (0.001587 m). The size and the length of the heat exchanger were limited by the physical dimensions of the dryer because the heat exchanger was designed to fit entirely within the dryer. The heat exchanger could have a maximum length of 2.5' (0.75 m) which resulted in an effectiveness of 0.0512 and a paltry 49.3 W extracted from the air and water vapor gas mixture exhausted from the dryer. This resulted in a savings of only $0.00554 per load. The heat exchanger would cost $304.45 to be mass produced and would have a payback time of nearly 140 years with 400 loads of laundry washed per year. Based on the economic analysis of the heat exchanger, it cannot be recommended for inclusion in the dryer. A more compact heat exchanger, potentially of the shell-and-tube, or plate and fin variety, would be more appropriate and could be feasible given the size constraints and the efficiency required to make the heat exchanger a sensible inclusion to the drying system.

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Section 1: Introduction

A typical electric dryer uses around 800 kWh of energy per year which equates to $100 at current energy prices. Nationwide, Americans spend $9 billion a year drying clothes clothes at a cumulative energy usage of 60 billion kWh [3]. Although new washing machines consume 70% less energy than models two decades older, no substantial progress has been made on the dryer efficiency front in decades, and therefore the energy required to run dryers nationwide generates 40 million metric tons of carbon dioxide per year [3]. Clearly, there exists a need for a more efficient clothes dryer, both to reduce the environmental impact of drying clothes, and to reduce the financial burden to consumers of running this energy-intensive appliance. For that reason, Dryer Inc. has commissioned this report to examine the feasibility of adding a heat exchanger to a dryer and to design such a system. The dryer is to be manufactured for use in an apartment. The air and water vapor gas mixture that will be drawn into the dryer is at the ambient conditions of the apartment defined in [1] as 21.1 ˚C and 50 % relative humidity (ω). Furthermore, according to [1], the dryer should remove 2.286 kg (5 lbs) of water from the clothes entering the dryer at the ambient conditions and should have a drying time of 45 minutes. The maximum temperature in the dryer, 100 ˚C, occurs at the entrance to the drum, directly after the heating element. The heat exchanger is designed to extract power from the exhausting air and water vapor gas mixture from the dryer. According to [4], the heat exchanger should be of the concentric double pipe style with air and water vapor gas mixture as the fluid in both the inner and outer pipes. The heat exchanger is intended to improve the efficiency of the dryer by delivering power back to the heating element, thereby reducing the energy needed to run the dryer. Numerous theoretical methods exist to reduce the energy load of dryers (see [3], [5], and [6]), but a heat exchanger is an attractive option because it takes advantage of the exiting stream of air and water vapor from the dryer that normally goes unutilized.

Section 2: Nomenclature

- Subscript numbers (1,2,3) indicate the stage of drying at which the variable is evaluated.
- Subscript letter v indicates water vapor, subscript letter a indicates air, and subscript letter l indicates liquid water.
- Subscripts c and C refer to the stream of cold air in the heat exchanger and subscripts h and H refer to the stream of hot air in the heat exchanger.
- Subscript letters i and p indicate the inner pipe in the heat exchanger and subscript letters o and a indicate the outer pipe (annulus) in the heat exchanger.

Symbol	Description	Units
$\dot{m}$	Mass flow rate	kg/s
ω	Absolute (specific) humidity	-
M	Mass	kg
P	Pressure	Pa
P_{sat}	Saturation pressure	Pa
$\emptyset$	Relative humidity	-
h	Enthalpy	kJ/kg
$\dot{Q}$	Rate of heat transfer	Watts (or J/s)
T	Temperature	K or ˚C
t	Time	s
E	Energy	kWh
$\dot{V}$	Volumetric flow rate	m^3/s
$h_{conv,i}$	Inner convective heat transfer coefficient	W/m^2K
$h_{conv,o}$	Outer convective heat transfer coefficient	W/m^2K
A	Area	m^2
D_h	Hydraulic diameter	m
D_e	Equivalent heat transfer diameter	m

ID	Inner diameter	m
OD	Outer Diameter	m
L_c	Characteristic length	m
$U_{OA,i}$	Overall inner heat transfer coefficient	W/m^2K
c_p	Specific heat	J/kg K
f_d	Darcy friction factor	-
r_w	Wall radius	m
Δ	Change in quantity	-
ΔT	Temperature difference	°C
ε	Effectiveness factor	-
C	Heat capacity rate	J/k s
C_{min}	Minimum Heat-Capacity Rate	J/k s
CR	Heat capacity ratio	-
NTU	Number of transfer units	-
V	Velocity	m/s
ρ	Density	Kg/m^3
ΔP	Pressure drop	Pa
Po	Power	W
h_L	Head loss	m
k	Thermal conductivity	W/m k
Nu	Nusselt number	-
Re	Reynolds number	-
Pr	Prandtl number	-
K	Loss coefficient for pressure drop	-

Section 3: Methods

3.1 Assumptions

- A steady drying state is assumed with the dryer operating at constant conditions throughout the drying cycle [1]. If the rate at which water enters the drum from the clothes between stage 2 and 3 is assumed to be constant, and the dryer does not cycle, the thermal analysis is greatly streamlined and can be completed with simple mass and energy conservation analyses at several control volumes. A thermodynamic analysis with a varying mass flow rate of water would either be difficult without empirically measuring the mass flow rate.
- The mass flow of air is constant throughout the entire drying process. According to [6], mass airflow slightly decreases as lint accumulates in the lint filter. The assumption is valid because, although lint accumulates at the lint filter and creates a blockage for the passage of air, lint filters generally do not cause significant blockage to deter airflow until approximately 50% of the lint filter is covered according to page 20 of [7].
- There is no heat loss to the environment from either the heat exchanger or the dryer process. In other words, both processes are adiabatic. This is not completely true because the temperature of the outer pipe of the heat exchanger is greater than the ambient conditions and there would be convection and conduction from the pipe to the surroundings. However, this is a negligible amount when compared with the heat exchanger occurring in the heat exchanger between the annulus and inner pipe.
- There is no pressure loss from the pipes to the environment. This is valid because there is no porosity in the pipes to allow pressure to escape.
- The pipes in the heat exchanger are hydraulically smooth, meaning that the roughness of the pipe walls is less than the thickness of the laminar sub-layer of the turbulent flow as described in [8]. This assumption is valid because the design calls purchasing standard heat exchanger piping which can be considered to have undergone processing for a smooth surface finish.
- The flow through the heat exchanger pipes is fully developed, i.e. the flow profile is identical in the direction of the flow along the length of the pipe. Additionally, the boundary layers on the pipe walls have completely merged. This assumption is valid because the Reynolds number for the flow in the heater exchanger is much greater than 4000, listed in [8] as the minimum Reynolds number for fully developed, turbulent internal pipe flow.
- The properties of the air and water vapor gas mixture are constant throughout the heat exchanger process. Properties slightly change as either temperature or pressure are slightly changed, but it is safe to assume a fluid property value at an average temperature and state because the flows are considered to be well-mixed with uniform properties. Accordingly, the density for each state is assumed constant.

- The thermal entry length for both streams of air and water vapor gas mixture is insignificant because both flows are turbulent and fully developed.
- The pipes are horizontal and there is no considerable potential energy difference from the inlet to the outlet. The pressure drop in the ducting and the pipes is caused only by the friction in the pipes and minor losses from bends.
- According to the material safety data sheet for copper, neither air nor water vapor reacts with the copper tubing over the course of its use [9].
- The contribution of the mass flow rate of liquid water into the dryer drum is negligible in the design of the heat exchanger. In other words, the mass flow rates of the streams of gas mixture in the heat exchanger are equivalent. The mass flow rate of the liquid water is roughly 2% of the mass flow rate of the streams of air and water vapor in the heat exchanger.

3.2 Design Procedure for Clothes Dryer

The design of the overall dryer system was split into two phases. The electric clothes dryer by itself was designed first and then, based on the selected parameters and thermodynamic outputs of the dryer, the heat exchanger was designed. The diagram of the dryer itself is shown in Figure 1. The drying process was split into four stages to simplify the thermodynamic analysis.

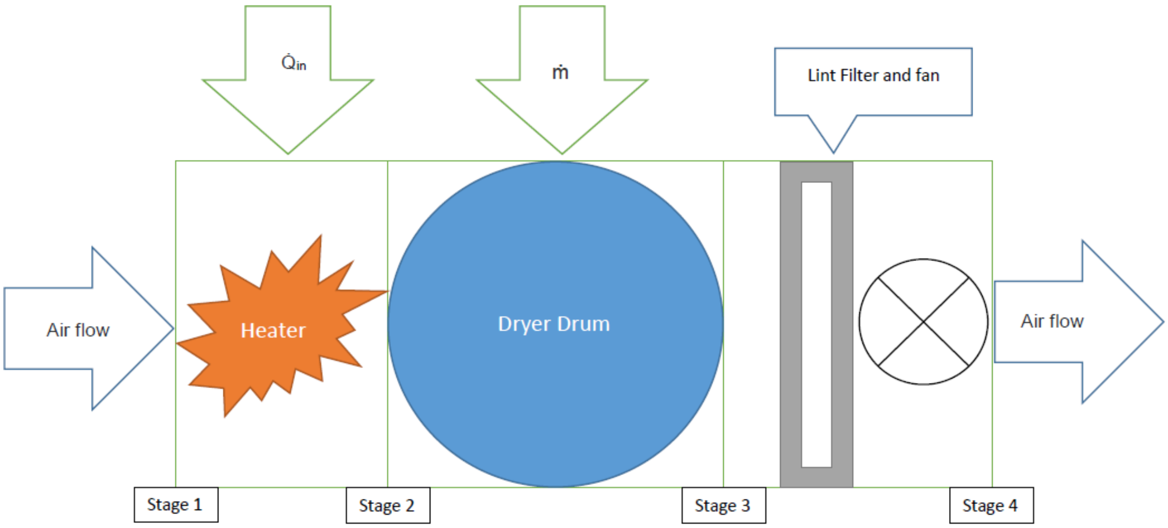

Figure 1: Dryer Schematic

Stage 1 is the inlet into the dryer where the dry air and water vapor mixture enters the system at a temperature and relative humidity of the ambient environment defined in [1] as 21.1 ˚C and 50 % respectively. In between stages 1 and 2 is the heating element which adds heat to the incoming gas mixture before it enters the dryer drum. Stage 2 is the inlet into the dryer drum, where the gas mixture reaches its highest temperature which was established as 100˚C as specified in [1]. In between stages 2 and 3 is the dryer drum, where the wet clothes enter. As required in [1], the clothes enter at the ambient conditions of 21.1 ˚C and 50% relative humidity and contain a total of 2.268 kg of water. Stage 3 is the exit of the dryer drum. In between stages 3 and 4 is the lint screen which causes a pressure drop but does not affect the mass or energy balance of the control volume. The final stage of the dryer is 4 which is the fan driven by a small motor that is required to pull the air through the dryer and sized based on the overall pressure drop in the system from the ductwork and the lint screen.

The design of the electric clothes dryer was simplified by considering mass and energy balances. The mass balance was written for entire drying process while for the energy conservation equations, the drying process was broken down into three phases, a control volume was drawn around each phase, and the appropriate energy conservation equation was then developed for that stage.

The governing equation for overall mass conservation can be written:

$$\dot{m}_a + \dot{m}_{v,1} + \dot{m}_l = \dot{m}_a + \dot{m}_{v,3} \qquad \textbf{(1)}$$

Where $\dot{m}_a$ is the mass flow rate (in kg/s) of the inlet air, $\dot{m}_{v,1}$ is the inlet mass flow rate of the water vapor, $\dot{m}_l$ is the liquid water mass flow rate that enters the dryer drum in the wet clothes, and $\dot{m}_{v,3}$ is the mass flow rate of the water vapor exiting the drum. Absolute humidity can be used to simplify the mass conservation equation as shown in [7] where absolute humidity can be written:

$$\omega \equiv \frac{M_{H20,v}}{M_a} = \frac{\dot{m}_{H20,v}}{\dot{m}_a} \qquad \textbf{(2)}$$

Where ω, the absolute humidity (also referred to as the specific humidity), is defined as the ratio of the mass of the water vapor in the air, $M_{H20,v}$, to the mass of the dry air, M_a, which can also be expressed in terms of the respective mass flow rates. Using the absolute humidity, Equation (1) can now be written as:

$$\dot{m}_a(1 + \omega_1) + \ \dot{m}_l = \ \dot{m}_a(1 + \omega_3) \qquad \textbf{(3)}$$

With ω_1 the absolute humidity of the air and water vapor mixture entering the dryer before the heating element and $\omega_3$the absolute humidity of the air and water vapor mixture exiting the dryer drum. The relative humidity was specified in the design requirements, which meant that an expression needed to be developed to relate the relative humidity to the absolute humidity. [7] lays out the procedure for deriving this equation. The first step is to express the absolute humidity in terms of the ambient pressure, P, and the partial pressure, $P_{H2O,v}$, of the water vapor:

$$\omega = 0.622 * \left(\frac{P_{H2O,v}}{P - \ P_{H2O,v}}\right) \qquad \textbf{(4)}$$

The definition of relative humidity is found on page 1025 of [8] and it can be expressed as the fractional amount of actual water vapor in the air at a given temperature to the theoretical maximum that the air can hold at the temperature. This is expressed in equation form as:

$$\emptyset = \frac{P_{H2O,v}(T)}{P_{sat}\ (T)} \qquad \textbf{(5)}$$

Where $\emptyset$ is the relative humidity, $P_{H2O,v}(T)$is the pressure of the water vapor at the given temperature, and $P_{sat}\ (T)$ is the saturation pressure for water vapor at the temperature under consideration. Equations (4) and (5) can be combined to yield an expression for the absolute humidity as a function of the relative humidity:

$$\omega = 0.622 * \left(\frac{P_{sat}(T)}{P - P_{H2O,v}(T)}\right) * \emptyset \quad \textbf{(6)}$$

The next step in the design of the dryer is to develop the energy conservation equations for each control volume. The first control volume to consider is that encompassing stages 1 and 2. A gas mixture of dry air and water vapor enters the drying system at stage 1, is heated by the heating element, and then exits into the dryer drum at stage 2. The energy conservation equation for this control volume comes from [7]:

$$\dot{m}_a\left(h_{a,1} + \omega_1 h_{v,1}\right) + \dot{Q_{in}} = \dot{m}_a(h_{a,2} + \omega_2 h_{v,2}) \quad \textbf{(7)}$$

Where $h_{a,1}$ is the enthalpy of the entering air (in kJ/kg), $h_{v,1}$is the enthalpy of the entering water vapor, $\dot{Q_{in}}$ is the heat input from the heating element (in Watts), $h_{a,2}$is the enthalpy of the air entering the dryer drum, and $h_{v,2}$is the enthalpy of the water vapor entering the dryer drum. The mass flow rate of the air is a constant for the process as is the absolute humidity. This is because there is no change in the mass of the water vapor contained in the dry air and therefore, $\omega_1 = \omega_2$. The relative humidity of the air does decrease because the ability of the air to hold moisture increases as the air is heated. After being heated to the maximum temperature of 100 ˚C by the heating element, the air and water vapor mixture enters the dryer drum at stage 2. Moving from stage 2 to stage 3, the energy conservation equation for the control volume can be written:

$$\dot{m}_a\left(h_{a,2} + \omega_2 h_{v,2}\right) + \dot{m}_l h_l = \dot{m}_a(h_{a,3} + \omega_3 h_{v,3}) \quad \textbf{(8)}$$

Where h_l is the enthalpy of the water entering the dryer drum in the clothes (in kJ/kg), $h_{a,3}$ is the enthalpy of the air leaving the dryer drum, $\omega_3$is the absolute humidity at stage 3 of the dryer, and $h_{v,3}$ is the enthalpy of the exiting water vapor. Moving from stage 2 to 3, the temperature drops as do the enthalpies of the water vapor and the air. The water from the clothes is added to the air, which raises the absolute humidity of the gas mixture leaving the tumbler. In other words, the absolute humidity at stage 3 is greater than at stage 2 as shown by:

$$\omega_3 = \omega_2 + \frac{\dot{m}_l}{\dot{m}_a} \quad \textbf{(9)}$$

Finally, an expression for the incoming mass flow rate of air into the drying process can be derived from the previous two equations as shown in [7]:

$$\dot{m}_a = \frac{\dot{m}_l\left(h_{v,3} - h_l\right)}{h_{a,2} - h_{a,3} + \omega_1(h_{v,2} - h_{v,3})} \quad \textbf{(10)}$$

This allows for the overall constant mass flow rate of the dry air to be calculated. The final control volume, moving from stage 3 to stage 4 is of little interest for energy and mass conservation because there is no change in the stage of the gas mixture. There is a pressure drop across the lint screen as is addressed in the design. However, Equation (10) still has an unknown term in $h_{v,3}$, the enthalpy of the water vapor exiting the dryer tumbler. This term is a function of the temperature at the dryer exit, T_3, and the saturation pressure, $P_{sat,3}$, at this temperature. Once the temperature at the exit of the drum is determined, the saturation pressure and the enthalpy of the water vapor can be found. Because the properties of the gas mixture (relative humidity and temperature) are known at stage 2, it is possible to determine the needed temperature at stage 3 by relating the energy balances at the two stages. Using the previous equations (3-8) and the relation between absolute and relative humidity, this governing equation as shown in [7] becomes:

$$h_{a,2} + \omega_2 h_{v2} + \left[\left(\frac{0.622 * \emptyset_3 * P_{sat,3}}{P_3 - \emptyset_3 P_{sat,3}}\right) - \omega_2\right] h_l = h_{a,3} + \left(\frac{0.622 * \emptyset_3 * P_{sat,3}}{P_3 - \emptyset_3 P_{sat,3}}\right) h_{v,3} \quad \textbf{(11)}$$

The unknowns in this equation are $h_{a,3}$, $h_{v,3}$, $P_{sat,3}$. However, all of these unknowns are functions of the temperature at stage 3. Therefore, for a given relative humidity at stage 3, $\emptyset_3$, the process to solve the equation is as follows: select a reasonable temperature based on the literature [11], evaluate the unknown gas mixture properties at that temperature, calculate the right hand side (RHS) and left hand side (LHS) of the equation, compare the two sides, and adjust the temperature until the RHS and LHS converge. This will allow for the determination of the temperature of the gas mixture exiting the drum for a given relative humidity that also should be selected based on a study of the literature. Then, once the temperature at stage 3 has been established, the mass flow rate of the dry air can be determined. Furthermore, the mass flow rate of the air will allow equation 7 to be solved for the heat transfer rate from the heating element:

$$\dot{Q}_{in} = \dot{m}_a[(h_{a,2} + \omega_2 h_{v2}) - (h_{a,1} + \omega_1 h_{v1})] \quad \textbf{(12)}$$

Again, $\omega_1 = \omega_2$ because no water vapor is added to the air in this stage of the dryer. This procedure establishes the critical parameters of mass flow rate of dry air, the heat rate required by the heating element, and the exiting temperature of the air from the dryer drum. As the lint screen does not alter the temperature or the mass flow rate of the air, the exit parameters of the gas mixture from the dryer drum are the same as the parameters of the gas mixture leaving the dryer through the fan. These specifications can then be fed into the heat exchanger equations to determine the amount of heat it is possible to extract from the exiting air and return to the dryer to improve efficiency.

The last aspect of the dryer design procedure is the pressure loss through the ducting. These were completed by examining the literature to determine appropriate ducting for a dryer. Once the ducting had been selected, the pressure drop was calculated from the following equation on page 796 of [8]:

$$K \equiv \frac{h_L}{\frac{\left(V_{avg}\right)^2}{2 * g}} \tag{13}$$

Where the loss coefficient, K, is defined for different types of valves and elbows in Table 10.8 on page 797 of [8], and the head loss, h_L, is in meters. Once the total head loss has been calculated, the pressure drop can be determined from the definition of the heat loss as in [8]:

$$\Delta P = h_L * \rho * g \tag{14}$$

Where ΔP is the pressure drop (in Pa) and ρ is the density of the fluid (in kg/m^3). The last aspect of the dryer to design is the fan required to pull the air through the dryer. The power required by the fan can be calculated from the pressure drop as in [8]:

$$Po = \Delta P * \dot{V} \tag{15}$$

Where Po is the power (in W) and $\dot{V}$ is the volumetric flow rate (in m^3/s). Once the fan power had been specified, the design of the dryer was complete. The next step was to take the exhaust products of the dryer and feed those into the heat exchanger equations to calculate how much more efficient the dryer can be made with the addition of a simple thermal device.

3.3 Selection of Dryer Parameters

The procedure outlined in Section 3.2 was implemented in Microsoft Excel to allow for a range of different variables to be evaluated. In particular, the effective of changing the relative humidity at stage 3, the exit of the dryer drum, could be studied as this determined the temperature at the exit of the dryer drum, the mass flow rate of dry air, the heat transfer required from the heating element, and ultimately the cost to the consumer to run the dryer for a cycle. The fixed variables in the design as given in [1] are shown in Table 1:

Parameter	**Value**
Inlet temperature of gas mixture, T_1	21.1 ˚C
Inlet relative humidity of gas mixture, $\emptyset_1$	50 %
Inlet absolute humidity of gas mixture, ω_1	0.00785
Drying Time	45 minutes
Temperature of gas mixture at drum inlet, T_2	100 ˚C
Inlet temperature of wet clothes	21.1 ˚C
Water contained in clothes	2.268 kg
Mass flow rate of water from clothes, $\dot{m}_l$	8.4×10^{-4} kg/s

Table 1: Specified System Parameters

The variables for the dryer to be determined were: the relative humidity at the exit of the drum, the temperature at the exit of the drum, the mass flow rate of dry air in the system, and the heat transfer required for the heating element. Suitable starting ranges for the relative humidity and the temperature at the exit of the dryer drum could be found in the literature. Excellent figures for establishing initial estimates to begin the analysis were found in [11] and are shown in Table 2. Using these ranges, the actual values for the efficient clothes dryer could be calculated using the design procedure.

Measurement	Range
Surface temperatures of heater duct	45–120°C (depending on location and voltage)
Coil temperature	350–800°C (depending on location and voltage)
Voltage input (ac rms)	110–220 VAC
Volumetric air flow rate	50–70 m^3/h (inlet); 70–150 m^3/h (exit)
Energy input: watts (instantaneous)	2000–4200 W
Energy input: kWh (to dry)	1.7–2.3
Relative humidity exiting drum	65–80% (start of test); 30–35% (end of test)
Air temperature drum inlet	90–160°C
Air temperature drum outlet	20–35°C (start of test); 40–50°C (end of test)
Air temperature in wet bulb drum outlet	20–35°C
Fan rpm	1000–2000
Drum rpm	35–50
Weight of clothes and (% initial and final moisture content)	2.2–3.2 kg (53–76% start of test and 2–5% end of test)
Ambient temperature	22–25°C
Ambient relative humidity	32–65%
Time to dry	30–55 min (depends on applied voltage, weight/moisture content of clothes)

Table 2: Experimental Dryer Parameters from [11]

The first step was selecting an appropriate relative humidity to use at the outlet of the tumbler. Table 2, from [11] provided an estimated relative humidity range of 65-80% based on numerous experimental test runs, which was correlated by other industry papers. However, according to Professor J. Kadambi in the Department of Mechanical and Aerospace Engineering (EMAE) at Case Western Reserve University (CWRU) [4], the relative humidity at the exit of the drum can reach as high as 90%. Therefore, it was decided that calculations from 65-90% relative humidity in increments of 5% would be completed. Once a given relative humidity had been selected, a temperature for the exit of the dryer also needed to be assumed. Again, Table 1 from [11] proved invaluable as it listed typical dryer drum outlet temperatures to be from 40-50 ˚C. This served as a starting point for the calculations in Excel. Data for the enthalpies of water vapor and dry air were evaluated using [12] across a range of temperatures from 35-45 ˚C. A selection of this data is shown in Table 3 (full data sets can be viewed in the Appendix). The pressure at stage 3 was set to the ambient pressure, outlined in the design requirements as 1 atm (101.325 kPa).

T_3 (˚C)	T_3 (k)	P_sat,3 (kPa)	P_3 (kPa)	h_a,3 (kPa)	h_v,3 (kPa)
35	308.15	5.609	101.325	434.6	2564.5
35.25	308.4	5.687	101.325	434.86	2565
35.5	308.65	5.766	101.325	435.11	2565.4
35.75	308.9	5.846	101.325	435.36	2565.9
36	309.15	5.927	101.325	435.61	2566.3
36.25	309.4	6.008	101.325	435.86	2566.8
36.5	309.65	6.091	101.325	436.11	2567.2
36.75	309.9	6.175	101.325	436.37	2567.7
37	310.15	6.260	101.325	436.62	2568.1
37.25	310.4	6.346	101.325	436.87	2568.6

Table 3: Gas Mixture Properties at Dryer Exit

The saturation pressure of water at a given temperature was calculated using the Antoine Equation as shown in [13] and given by:

$$P_{sat}(T) = 10^{8.07131 - \frac{1730.63}{233.426+T}} \quad \textbf{(16)}$$

Where P_{sat}, the saturation pressure of the water vapor, is given in Torr and T is in Celsius. According to [13], this equation is an empirical representation based on experimental data and is valid for temperatures from 1-100 ˚C, within the range under consideration in the design of the dryer. The saturation pressure at a given temperature was converted to kPa in keeping with the International System of units (SI system) convention adhered to in the design. Once a relative humidity at stage 3 had been selected, the LHS and the RHS of Equation 11 were evaluated across the range of temperatures as shown in Table 4. (Selected values are shown for a relative humidity of 90%. Additional tables can be found in the Appendix).

T_3 (˚C)	LHS (kJ/kg)	RHS (kJ/kg)
35	523.3841821	518.2357978
35.25	523.426536	519.7375593
35.5	523.4694616	521.2429862
35.75	523.5129665	522.7689707
36	523.5570581	524.3089773
36.25	523.6017443	525.8701013
36.5	523.6470328	527.4456148
36.75	523.6929315	529.0528218
37	523.7394486	530.6647968
37.25	523.7865921	532.2990579

Table 4: LHS and RHS of Equation 11 for Relative Humidity of 90%

From these tables, it was possible to determine the exit temperature of the drum at which the two sides converged. However, these tables were only accurate to the nearest quarter degree. In order to achieve a higher degree of specificity, graphs of both the LHS and the RHS as a function of temperature were made and then fitted with polynomials in Excel. Therefore, the exact temperature at which the two sides converged could then be determined. This is shown in Figure 2 for a relative humidity of 90 %.

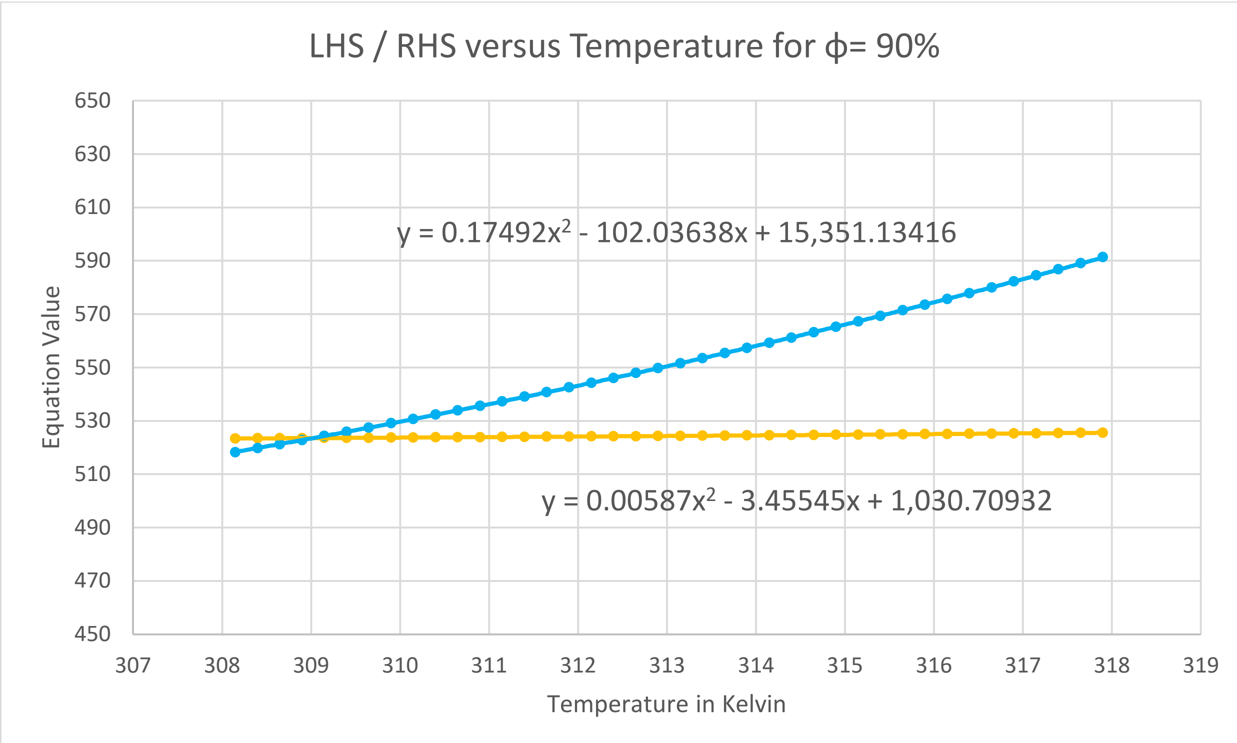

Figure 2: LHS and RHS versus Temperature

Once the temperature at the dryer tumbler exit had been established, the necessary mass flow rate of dry air could be calculated. The mass flow rate of liquid water used in this calculation was determined from the amount of water to be removed from the clothes. As stated in the design requirements, the dryer was required to remove 5 lbs (2.268 kg) of water from the clothes. It was assumed that all the water was to be removed from the clothes because the dryer operated at a steady state. Therefore, the mass flow of the liquid water into the control volume could be determined by:

$$\dot{m}_l = \frac{m_l}{t} = \frac{m_l}{(45\,\mathrm{min} * 60\,sec/\mathrm{min})} \tag{17}$$

Where the mass flow rate of the water (in kg/s) was calculated by taking the mass of the water removed spread out over the entire drying cycle in seconds. This assumed constant operation of the dryer, that is, the dryer did not run through heating cycles as in the operation of most commercial dryers as shown in [6]. Once the mass flow rate of the dry air had been established, the heat rate transfer required by the heating element could be determined from Equation 12. Additionally, the flow rate of air can be expressed as a volumetric flow rate by:

$$\dot{V}_a = \frac{\dot{m}_a}{\rho_a} \quad \textbf{(18)}$$

Where $\dot{V}_a$ is the volumetric flow rate of the air in m^3/s and ρ_a is the density of the air in kg/m^3. The results from this procedure are shown in Table 5. Calculations were carried out at a range of relative humidities at the dryer drum exit. From this information, it was decided that the dryer should be run with a relative humidity of 90% at the dryer exit as this minimized the mass flow rate of the dry air and hence the needed heat transfer from the heating element. Ultimately, this leads to a more efficient clothes dryer that consumes less energy.

Relative humidity	Drum Output(˚C)	(kJ/kg)	(kJ/kg)	(kg /s)	(m^3 / s)	(m^3 / min)	(kWatts)
ϕ_3	T_3	h_a,3	h_v,3	mdot_a	vdot_a	vdot_a	Qdot
0.65	40.75	440.39	2574.9	0.034467352	0.02873	1.72379857	2.779084
0.7	39.75	439.39	2573.1	0.033875319	0.028236	1.69418948	2.731349
0.75	38.75	438.38	2571.3	0.033297146	0.027755	1.66527363	2.684731
0.8	38.48	437.88	2570.4	0.033017709	0.027522	1.65129829	2.6622
0.85	37.26	436.87	2568.6	0.032467029	0.027063	1.62375737	2.617799
0.9	35.96	435.36	2565.9	0.031675615	0.026403	1.58417681	2.553988

Table 5: Dryer Parameters across Range of Dryer Exit Relative Humidities

The final temperature at the exit of the dryer drum was then set based on the intersection point of the RHS and LHS of Equation 11. For a relative humidity of 90%, this translated into a $T_3$of 35.96 ˚C. The mass flow rate of dry air was calculated to be 0.03167 kg/s and the required heat transfer rate for the dryer was 2.554 kW. The dryer parameters selected are summarized in Table 10 in the results section of the report.

To determine the energy use of the dryer without a heat exchanger, the instantaneous power needed to be converted into an energy requirement for a drying cycle. This is done by:

$$E = \dot{Q} * 45\,mins * \frac{1\,hr}{60\,mins} \quad \textbf{(19)}$$

Where E, the energy required by the dryer is measured in kWh, and the instantaneous energy usage, $\dot{Q}$, which is also the rate of heat transfer, is multiplied by the cycle time in hours. Again, this assumes that the dryer is continuously operated at a steady state (at a constant temperature) and thus the same heat transfer rate is constant across the entire drying cycle. To find the average cost of drying a load of clothes for the consumer, the energy can be multiplied by the current average cost of energy in Cleveland, Ohio as found in [2] and currently at $0.127/ kWh. The results for the cost of the dryer at a range of relative humidities can be seen in in Table 6.

Relative humidity	Drum Output(	(kWatts)	(kWh)	
ϕ_3	T_3	Qdot	Energy	Cost for one cycle
0.65	40.75	2.779	2.084	$ 0.26
0.7	39.75	2.731	2.049	$ 0.26
0.75	38.75	2.685	2.014	$ 0.26
0.8	38.48	2.662	1.997	$ 0.25
0.85	37.26	2.618	1.963	$ 0.25
0.9	35.96	2.596	1.947	$ 0.25

Table 6: Cost of Running Dryer Based on Relative Humidity

The ducting of the dryer was designed based on the relevant literature, mainly [15] and [16]. The ducting schematic can be seen in the results section. The pressure drop through the complete ducting of the dryer and the lint screen was also calculated. The results from the calculations described in the design procedure are shown in Table 7:

Description:	Quantity:	Area (m^2):	Velocity (m/s):	Loss Coefficient (unitless):	Pressure Drop (Pa)
5" Duct Bend	6	0.013	2.687	0.019	68.580
Dryer Drum	1	0.292	0.117	0.1	0.680
Lint Filter 75% blocked	1	0.021	1.649	0.5	679.477
Inner Heat Exchanger	1	0.005	7.463	n/a	18.487
Outer Heat Exchanger	1	0.013	2.687	n/a	7.822
Total calculated pressure drop (Pa):					
1117.944					
Power required by fan assuming 100% electrical efficiency (W):					
38.050					

Table 7: Pressure Loss through Dryer

The pressure loss through the dryer determined the power required by the fan to pull air through the system. As the pressure loss and the associated power requirement was minimal compared to the heating element requirements, no steps were taken to minimize the pressure loss through innovative ducting. Instead, the literature was closely adhered to in the design of the ducting. As can be seen, the greatest contribution to the pressure drop is the lint screen, which ultimately is controlled by the consumer. In order to decrease the pressure drop, the most effective step would be to educate consumers about the crucial step of cleaning the lint filter. That phase of the product development is beyond the reach of the design team.

3.4 Design of Heat Exchanger

Specification of Fluid and Pipe Properties

After the parameters for the dryer alone had been specified, the next step was to design a heat exchanger based on the outlet thermodynamic numbers from the dryer. The heat exchanger could then be analyzed based on the cost to determine if adding a heat exchanger to the system was a sound economic design decision.

The first step in the design of a heat exchanger is to specify the type of heat exchanger. According to the project manager, a concentric double pipe heat exchanger would be an adequate choice for the system. Based on the authors' previous experience with heat exchanger design [14], it was determined that the heat exchanger should be counterflow to optimize the amount of heat transfer and therefore energy efficiency of the heat exchanger for a given area. Therefore, the heat exchanger designed was a concentric double pipe heat exchanger with the air and water vapor gas mixture as the fluid in both the inner and outer pipes.

Once the style of heat exchanger had been specified, the next part of the design was to specify the fluid properties of both the hot and cold fluids. Based on preliminary calculations and the conclusions reached in [14] and [4], it was determined that a high level of accuracy could be attained if the properties for the cold fluid were evaluated at the cold fluid inlet temperature, and the properties of the heating fluid were evaluated at the heating fluid inlet temperature. This is because the preliminary calculations determined that the temperatures of the two fluids (both air and water vapor gas mixtures) do not change significantly from the inlet to the exit of the heat exchanger. Moreover, the fluid properties of the gas mixture do not exhibit dramatic changes across the range of operating temperatures considered. Therefore, the fluid properties of the cold air and water vapor mixture were evaluated at a temperature of 21.1 ˚C, the ambient temperature, and at a relative humidity of 50%. The fluid properties of the heating air and water vapor were evaluated at the exit conditions from the dryer, 35.96 ˚C and a relative humidity of 90%. The mass flow rates were set equal, because the cold gas mixture drawn into the heat exchanger ultimately becomes the warm fluid after passing through the dryer, meaning that the mass flow rates must be equal according to mass conservation.

Fluid	T	ω	ṁ	ρ	cp	k	ν = μ/ρ	Pr	α = k / (ρ * cp)	μ
Cold	21.1	50	0.0317	1.1998	1006.4	0.025956	1.52E-05	0.70798	2.21496E-05	1.83E-05
Hot	35.96	90	0.0317	1.0878	1007.7	0.028177	1.81E-05	0.70442	0.000025703	1.97E-05

Table 8: Fluid Properties

The pipe properties for the heat exchanger also needed to be specified before the calculations could begin. This required studying the relevant literature and looking at several commercially available dryers to determine the pipe diameter that could realistically fit into a dryer and also the length of a pipe that could be placed inside dryers currently on the market. According to [15] and [16], typical dryer ductwork has a maximum nominal inner diameter (ID) of 5" (0.127 m).

Moreover, the maximum possible length of pipe that could fit in the dryer was determined to be 2.5' (around 0.75 m). These dimensions were likely oversized and represented the maximum size heat exchanger that could be fit inside a commercial dryer. As can be seen from equations on page 902 of [8], the overall heat transfer rate is directly related to the area of the heat exchanger. Therefore, the maximum possible diameter and length of pipe were desired for the heat exchanger. The initial dimensions of the heat exchanger were set at 5" (0.127 m) ID for the outer pipe, 3" (0.0762 m) nominal outer diameter (OD) for the inner pipe, 1/16" walls of the inner pipe, an ID for the inner pipe of 2 7/8" (0.073025 m), and a length of 0.75 m. These could be adjusted as needed if the dimensions proved unrealistic for incorporating into the dryer.

	ID_a	ID_p	OD_p	A_p	A_a	D_h	D_e
Nominal	ID of outer pipe	ID of inner pipe	OD of inner pipe	Area of inner pipe	Area of annulus	Hydraulic diameter	Heat transfer diameter
	m	m	m	m^2	m^2	m	m
5" x 3"	0.127	0.073025	0.0762	0.004188254	0.00810732	0.0508	0.135466667

Table 9: Heat Exchanger Pipe Properties

NTU-Effectiveness Method

As only the inlet temperatures of the heating and cooling fluid streams were known before starting the calculations, the NTU-effectiveness method had to be employed in the design of the heat exchanger. The method can be completed with only the two inlet temperatures and depends on expressing the actual heat transfer versus the theoretical maximum heat transfer for the given inlet streams of hot and cold fluid as described on pages 917-919 of [8]. The first step to take is to determine the heat capacity rates of both the hot and cold fluid:

$$C_C \equiv \left(\dot{m}c_p\right)_C \qquad C_H \equiv \left(\dot{m}c_p\right)_H \qquad \textbf{(20)}$$

Where C is the heat-capacity rate (in J/ k s) and c_p is the specific heat capacity of the fluid in (J / kg k). The lower of the two heat-capacity rates is then called C_{min} and the fluid is called the minimum fluid, which determines the maximum possible heat transfer for the two given fluids. The maximum possible heat transfer occurs when the minimum fluid is heated or cooled from its inlet temperature to the inlet temperature of the other fluid. This is described on page 918 of [8]:

$$\dot{Q}_{\max possible} = C_{min}(T_{H,in} - T_{C,in}) \qquad \textbf{(21)}$$

The maximum temperature difference is that between the inlet temperature of the hot fluid, $T_{H,in}$, and that of the cold fluid, $T_{C,in}$. The effectiveness of the heat exchanger is defined as the actual rate of heat transfer over the maximum possible heat transfer or as on page 918 of [8]:

$$\varepsilon = \frac{\dot{Q}_{actual}}{\dot{Q}_{\max possible}} = \frac{\Delta T_{\min fluid}}{T_{H,in} - T_{C,in}} \quad \textbf{(22)}$$

The effectiveness depends only on two dimensionless parameters, the number of transfer units, NTU, and the heat capacity ratio, CR, for the two fluids. The NTU can be expressed as:

$$NTU = \frac{U_{OA,i} * A_i}{C_{min}} \quad \textbf{(23)}$$

Where $U_{OA,i}$ is the overall inner heat transfer coefficient (in W/ m^2 k), A_i is the area of the inner pipe (in m^2) and C_{min} is the heat capacity rate of the minimum fluid (in J / k s). The heat capacity ratio is:

$$CR = \frac{C_{min}}{C_{max}} \quad \textbf{(24)}$$

Once these two parameters, NTU, and CR, have been determined for the given specifications of the heat exchanger, the final temperature of the minimum fluid can be calculated as given in [8]:

$$T_{min,out} = T_{min,in} \pm \varepsilon(T_{H,in} - T_{C,in}) \quad \textbf{(25)}$$

Where the sign is positive if the minimum fluid is the cold stream, and negative if the minimum fluid is the heating fluid stream. Before the heat transfer can be calculated, a value for the effectiveness must be determined. This can be found in a chart, as given in Figure 11.54 on page 919 of [8] or it can be calculated from an equation:

$$\varepsilon = \frac{NTU}{1 + NTU} \quad \textbf{(26)}$$

This equation is valid for a counterflow concentric tube heat exchanger if the specific heat ratio is equal to unity (CR = 1). For this case, CR = 0.999 which was assumed to be near enough to one that the equation is valid for the fluids. Finally, the actual heat transfer of the heat exchanger can be determined as shown in [8] (assuming the minimum fluid is the fluid being heated) :

$$\begin{aligned} \dot{Q}_{actual} &= \varepsilon * \dot{Q}_{\max possible} \\ &= C_{min}(T_{min,out} - T_{min,in}) \end{aligned} \quad \textbf{(27)}$$

All of the variables needed to carry out the NTU-effectiveness method could be determined from the fluid and pipe characteristics except for the overall heat transfer coefficient, $U_{OA,i}$.

Overall Heat Transfer Coefficient

The first calculation to be carried out was that velocity of the air and water vapor gas mixture in the pipes. The velocity can be found from:

$$V = \frac{\dot{m}}{\rho * A} \quad \textbf{(28)}$$

Where V, the velocity (in m/s) is found from the mass flow rate $\dot{m}$ (in kg/s), the density ρ (in kg/m^3), and the area of the pipe, A (in m). The next step was to calculate the Reynolds number of both the flow in the inner pipe and in the outer pipe (also referred to as the annulus) as in [8]:

$$Re = \frac{V * D}{\nu} \quad \textbf{(29)}$$

Where Re is the unitless Reynolds number, D is the diameter of the pipe, and ν is the kinematic viscosity of the fluid (in m^2 /s). Care must be taken to use the correct heat transfer equivalent diameter in the equation. For the inner pipe this is simply the inner diameter of the pipe. The equivalent heat transfer diameter for the outer pipe is expressed as given in [17]:

$$D_e = (ID_a^2 - OD_p^2)/\ OD_p^2 \quad \textbf{(30)}$$

Where De is the equivalent heat transfer diameter, ID_a is the inner diameter of the annulus, and OD_p is the outer diameter of the inner pipe (all in meters). According to page 746 of [8], internal flow begins to transition from laminar to turbulent at a Reynolds number of 2300 and is fully turbulent at a Reynolds number of 4000. For both the inner and the outer pipe, based on the calculations, the flow was fully developed and turbulent. The next parameter to determine was the dimensionless Nusselt number for heat transfer. The definition of the Nusselt number is provided on pages 602-604 of [8]:

$$Nu \equiv \frac{h_{conv} * L_C}{k} \quad \textbf{(31)}$$

Where Nu is the dimensionless Nusselt number, h_{conv} is the convective heat transfer coefficient of the fluid (in W/m^2 k), L_C is the characteristic length (in this case the equivalent heat transfer diameter in m as shown in [17]), and k is the thermal conductivity of the fluid (in W/ m k). As both flows were fully developed and turbulent, the Dittus-Boelter equation could be employed to calculate the Nusselt number. The equation is given in Table 10.5 on page 774 of [8]:

$$Nu = 0.023 * Re^{0.8} * \mathrm{Pr}^{\mathrm{n}} \quad \textbf{(32)}$$

Where Pr is the dimensionless Prandtl number for the fluid and n is 0.4 for a liquid being heated (the colder gas mixture) and 0.3 for a liquid being cooled (the heating fluid). Several assumptions were employed in order to use this expression. First, the Dittus-Boelter equation requires a Reynolds number > 10,000 and a Prandtl number between 0.7 and 160. Both of these assumptions are correct for the selected design and the fluid used. The other two assumptions are that for internal flow, the walls of the tube are hydraulically smooth (meaning that the laminar sub-layer thickness is greater than the roughness of the tube) and that the equation is evaluated for fully developed flow (meaning the boundary layers have merged). Both of these assumptions are valid in the design because the relative roughness of the selected tubing (copper) is small enough to classify the pipes as hydraulically smooth and the flow will be fully developed based on the Reynolds number. Combining equations (20) and (21), the convective heat transfer coefficient, h_{conv}, could be found for both streams of air and water vapor mixture.

The next step was to calculate the overall heat transfer coefficient for the inner pipe, $U_{OA,i}$ (in W/m^2 k). The overall heat transfer coefficient takes into account the sum of the individual thermal resistances in a system, $R_{thermal,j}$, and is defined on page 902 of [8]:

$$U_{OA} \equiv \frac{1}{A \sum_j R_{thermal,j}} \qquad \textbf{(33)}$$

For a double pipe heat exchanger, the thermal resistances are associated with convection at the tube inner wall, conduction through the tube wall, and convection at the tube outer wall. The equation for the overall heat transfer coefficient is provided on page 903 of [8]:

$$U_{OA,i} = \frac{1}{\frac{1}{h_{conv,i}} + \frac{r_{w,i}}{k_w} \ln\left(\frac{r_{w,o}}{r_{w,i}}\right) + \frac{r_{w,i}}{r_{w,o}} * \frac{1}{h_{conv,o}}} \qquad \textbf{(34)}$$

Where $h_{conv,i}$ is the heat transfer coefficient of the inner fluid (in W/m^2 k), $r_{w,i}$ is the radius of the inner pipe (in m), k_w is the thermal conductivity (in W/m k) of the pipe material, $r_{w,o}$ is the equivalent heat transfer radius of the outer pipe (in m), and $h_{conv,o}$ heat transfer coefficient of the outer fluid (in W/m^2 k). The material selected for the heat exchanger was copper because of its high thermal conductivity and low cost. The properties of copper were found on page 770 of [8]. After the overall inner heat transfer coefficient had been found, the NTU-effectiveness method could be carried out to determine the amount of heat transfer possible given the fluid properties and the dimensions selected for the heat exchanger. This procedure was implemented in Microsoft Excel which allowed for the effect of changing the dimensions of the heat exchanger to be studied and optimized based on the space available within the dryer design.

Pressure Loss

The next step was to determine the friction factor within the pipe and the associated pressure loss. A Moody Chart, located on page 770 of [8], was used to obtain the friction factor rather than an explicit calculation. This chart plots the friction factor versus Reynolds number for a range of relative roughness parameters. The relative roughness of a pipe is the absolute roughness divided by the hydraulic diameter. The absolute roughness values for stainless steel was provided by [18]. The Reynolds number used in the calculation of the friction factor must be found using the hydraulic diameter. After the friction factor had been determined, it was possible to calculate the pressure loss due to friction. This was done using the Darcy-Weisbach equation provided on page 736 of [8]:

$$\Delta P = f_D * \frac{L}{D} * \frac{1}{2} * \rho * V^2 \qquad \textbf{(35)}$$

Where ΔP is the pressure drop across the heat exchanger (in Pa), f_D is the unitless friction factor, ρ is the density of the fluid within the pipes (in kg / m^3), and V is the velocity of the fluid (in m/s). Equation (32) uses the hydraulic diameter for the annulus because both the inner and outer walls cause friction losses within the annulus. There are also minor pressure losses associated with flow in the system. According to pg 796 of [8], minor losses can be calculated from:

$$K \equiv \frac{h_{L,minor}}{\frac{\left(V_{avg}\right)^2}{2 * g}} \qquad \textbf{(36)}$$

Where the loss coefficient, K, is defined for different types of valves and elbows in Table 10.8 on page 797 of [8], and the minor head loss, $h_{L,minor}$, is in meters. The head loss can be converted to a pressure drop from the following equation [8]:

$$\Delta P = \rho * g * h_L \qquad \textbf{(37)}$$

Where the pressure drop is ΔP (in Pa), ρ is the density of the fluid (in kg/m^3), and g is gravitational acceleration. Minor losses were added to the pressure loss calculated from Equation (32) to arrive at the total pressure loss within the heat exchanger. These pressure losses can then be used to calculate the pump needed for the system. The equation for power required to overcome a pressure loss is from [19]:

$$Po = \Delta P * \dot{V} \qquad \textbf{(38)}$$

Where Po is the power (in Watts) and $\dot{V}$ is the volumetric flow rate of the fluid (in m^3/s). The pump size required for the heat exchanger can be calculated from these equations. The hot stream of air and water vapor can be driven through the heat exchanger by the dryer exhaust fan. In order to size the fan therefore, the pressure loss of the heat exchanger should be added to the pressure loss of the dryer to arrive at the total pressure the fan must overcome to drive air through the system.

This design process was implemented in Microsoft Excel so that the effect of changing one variable could be studied. The main variables of concern in this design were the material of the heat exchanger and the area of the heat exchanger which was limited by the size constraints imposed by the dryer.

3.5 Selection of Heat Exchanger Parameters

The incoming hot air and water vapor gas mixture properties were set by the outputs of the dryer. The gas mixture exited the final stage of the dryer at a mass flow rate of 0.03167 kg/s and a temperature of 35.96 ˚C (324.5 K). The mass flow rate of the incoming cold air and water vapor mixture had to be equal to that of the hot mixture based on mass conservation as the cold stream would eventually pass through the dryer and become the hot stream after completing the cycle. The mass flow rate of the water entering the dryer drum from the clothes was neglected as it was trivial compared to the mass flow rate of the gas mixture exiting the dryer. The temperature of the cold stream was assumed to be that of the ambient environment, or 21.1 ˚C (294.3 K). The thermodynamic properties of neither of the streams could be changed, and therefore, the only way to affect the amount of heat transfer was to alter the size of the heat exchanger pipes. However, this parameter was also limited by the physical considerations of the dryer. Based on a study of the relevant commercial dryers available, primarily [15] and [16], it was determined that the maximum nominal inner diameter of the annulus could be 5" (0.127 m) and the maximum length of the heat exchanger could be 2.5' (0.75 m). Even these dimensions were likely to be oversized, but, in order to maximize heat transfer, these were the parameters selected for the design. Even if there were no physical limits to the size of the heat exchanger, there still exists a thermodynamically determined maximum for the amount of heat transfer that can take place between the hot and the cold streams which is shown by Equation (21). Copper was chosen as the material because of its high thermal conductivity and inexpensive cost. However, as the heat exchanger is expected to operate in a damp environment, copper is not the optimal selection for a long-term application. A better material for corrosion resistance would be galvanized or stainless steel, both of which would have the two drawbacks of lower thermal conductivity and higher per unit cost than copper. The thermal conductivity of both copper and stainless steel were found from [20]. Cost estimates for both copper and stainless steel were obtained from [21]. If the heat exchanger were to be included in the dryer, the corrosion would have to be taken into account for any long term use of the heat exchanger. Table 11 shows the final specifications for the heat exchanger.

Section 4: Results

4.1 Dryer Specifications

The final parameters of the selected dryer design are shown in Table 10. These figures do not account for the inclusion of the heat exchanger.

Parameter	Value
Relative Humidity at Stage 3	90%
Temperature at Stage 3	35.96 ˚C (309.11 K)
Mass Flow Rate of Dry Air	0.03167 kg/s
Volumetric Flow Rate of Dry Air	1.584 m^3/s (55.94 cfm)
Instantaneous Required Heat Transfer for heating element	2.554 kW
Energy used in a drying cycle	1.947 kWh
Cost for a Drying Cycle	$0.25
Pressure Drop (total)	1118 Pa
Power Required for Fan	38 W
Diameter of ducting	0.127 m (5”)
Power of Purchased Heating Element	3500 W
Basic Parts cost estimate	$271.97

Table 10: Dryer Specifications

4.2 Heat Exchanger Specifications

The final specifications for the heat exchanger are shown in
Table 11. The heat exchanger will be the concentric double pipe style with counterflow. The cold stream in the inner pipe will be air and water vapor gas mixture pumped in at the ambient temperature (21.1 ˚C) and relative humidity (50%) of the surroundings. The hot stream in the annulus of the heat exchanger will be the air and water vapor mixture exhausted from the dryer and entering at the temperature (51.3 ˚C) and relative humidity (90%) at the exit of the dryer drum, which corresponds to the properties at the exhaust fan of the dryer.

Parameter	Value
Material	Copper
Inner diameter of outer pipe (ID_a)	5” (0.127 m)
Outer diameter of inner pipe (OD_p)	3” (0.07062 m)

Inner diameter of inner pipe (ID_p)	2.875” (0.073025)
Wall Thickness of Pipes	0.0625” (0.0015875 m)
Length	2.5’ (0.75 m)
C_C	31.88 J / k s
C_C	31.92 J/ k s
$T_{H,in} - T_{C,in}$	30.20 ˚C (or K)
$\dot{Q}_{\max possible}$	962 W
$U_{OA,i}$	9.98 (W/ m^2 K)
NTU	0.0538
CR	0.999
ε	0.0512
$\dot{Q}_{actual}$	49.26
Pressure Drop Inner Pipe Total	20.94 Pa
Pressure Drop Outer Pipe (Annulus) Total	8.62 Pa
Power Required for driving flow in inner pipe	0.68 W
Power Required for driving flow in outer pipe	0.31 W

Table 11: Heat Exchanger Design Specifications

4.3 Engineering Drawings

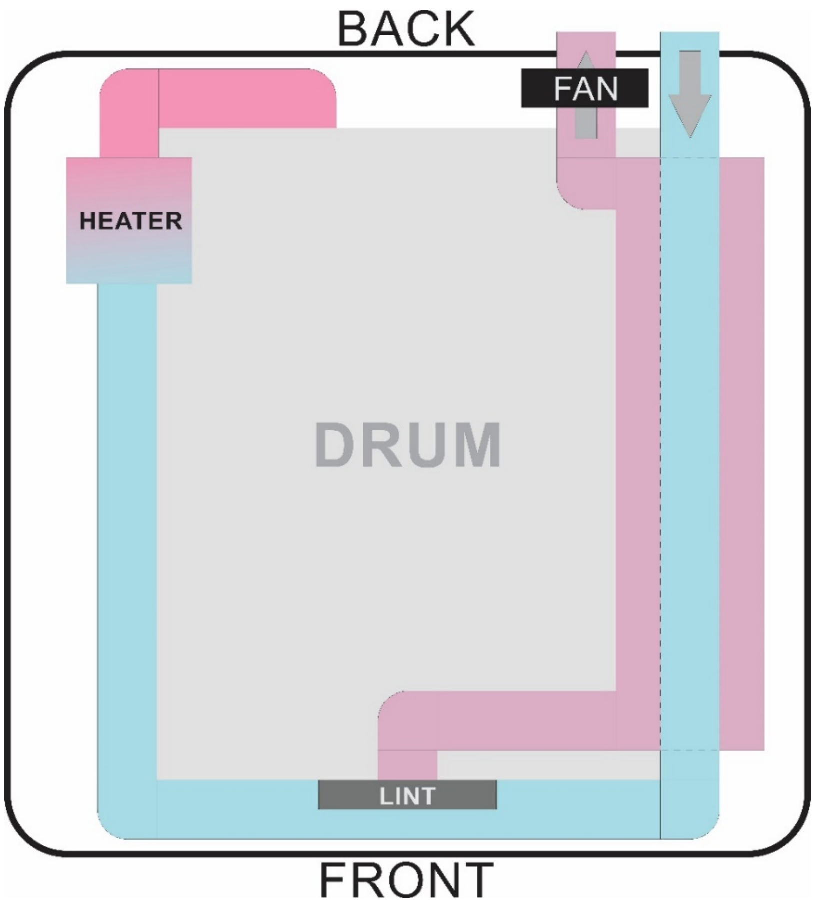

Figure 3: Dryer Ducting Schematic

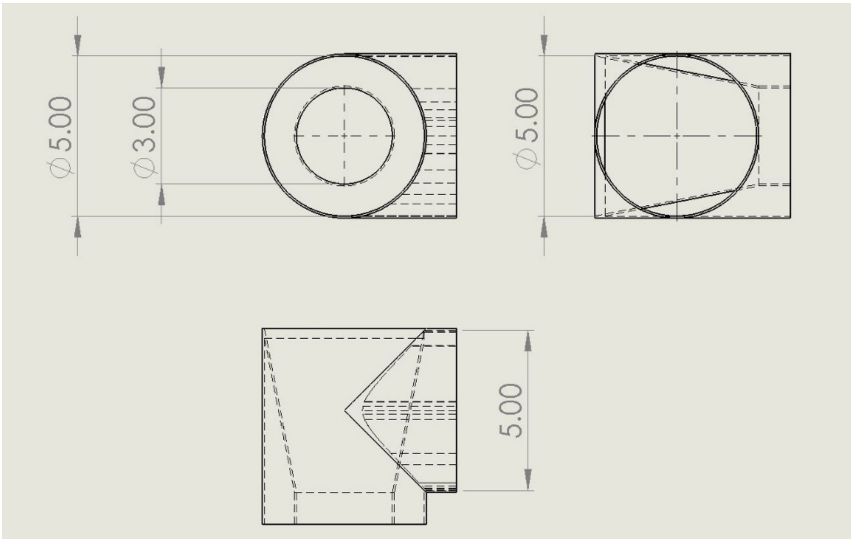

Figure 4: Detailed View Heat Exchanger End Caps

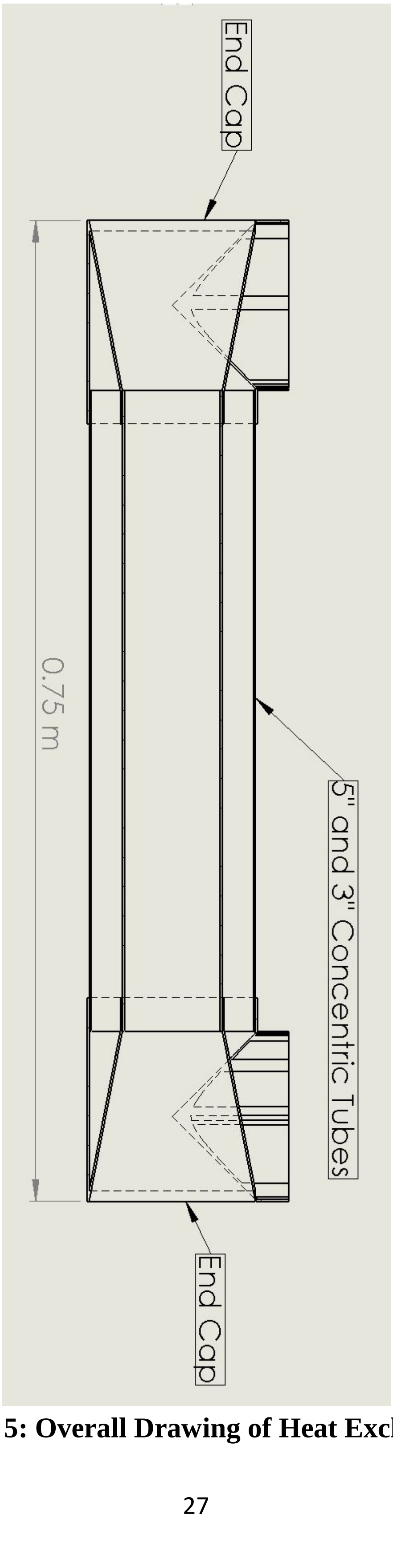

Figure 5: Overall Drawing of Heat Exchanger

4.4 Bill of Materials and Cost Analysis

Part Number	Description	Quantity	Price	Subtotal
01 [31]	3500W Heating Element	1	18.99	18.99
02 [32]	Lint Filter Assembly	1	45.75	45.75
03 [33]	5” 90˚ Elbow Duct	6	4.81	28.86
04 [34]	5” Straight Duct	1.4 m	3.94 / m	5.57
05 [35]	Blower Motor Assembly	1	172.80	172.80
Total				**$ 271.97**

Table 12: Bill of Materials for Dryer

Part Number	Description	Quantity	Price	Subtotal
01 [36]	5” Copper Tube	0.75 m	87.11 / m	65.33
02 [36]	3” Copper Tube	0.75 m	52.17 / m	39.13
03 [36]	Flared and Notched End Caps	2	100.00	200.00
Total				**$304.46**

Table 13: Bill of Materials for Heat Exchanger

Cost Analysis

It is important to note that the prices for bought products are based on a Sub-A profit margin per a standard mass-production contract. Prices do not reflect the cost to create a single prototype, but rather the steady-state cost of production on a large scale.

A 3500W heating element was chosen due to its availability and capacity to provide adequate heating per the design. An entire lint filter assembly was also chosen to provide a self-contained apparatus with low maintenance and replacement costs. Copper tubing was selected for the heat exchanger pipes due to copper's exceptional thermal conductivity and lower relative cost than stainless steel. Instead of custom machining the heat exchanger's end caps (which would be appropriate in a prototype scenario), the end caps are made of flat copper stock that is rolled, shaped, and brazed in place to create an adapter that not only fits the large 5" internal ducting but also allows for continuous heat transfer. Standard zinc-plated steel ducting was chosen for the internal network pipes within the dryer to reduce cost and a blower motor assembly was selected to handle the desired mass flow rate given the target blockage of the lint filter.

Section 5: Discussion

5.1 Heat Exchanger Efficiency Analysis

For the heat exchanger designed, the effectiveness is slightly greater than 5%, as calculated by the equation

$$\varepsilon = \frac{NTU}{1 + NTU} \quad \textbf{(39)}$$

With a heat capacity ratio, CR ≈ 1. This effectiveness is very low, as the heat exchanger considered for addition to a clothes dryer in the literature [6] had an effectiveness of 90%. It follows that the designed heat exchanger will not reduce the power usage of the dryer by a significant effect. As calculated, the heat exchanger saved a total of 50 W from the power requirements of the heating element. Assuming a dryer cycle time of 45 minutes with constant energy usage and an electric rate of $0.127 [2] per kWh, each load of clothing dried in the dryer with the addition of a heat exchanger costs a minuscule $0.00554 less than a load in a normal dryer. Considering each load of laundry can cost 32 to 41 cents in a normal electric dryer [16], this equates to a cost savings of 1.2% to 1.5%. The limiting factor for the efficiency of the heat exchanger is the area that the heat exchanger can occupy within the dryer. As the area of a concentric double pipe heat exchanger increases, its efficiency increases as well, as shown by:

$$NTU = \frac{U_{OA,i} * A_i}{C_{min}} \quad \textbf{(40)}$$

In order to fully explore the potential savings of a heat exchanger addition to the clothes dryer, two theoretical design were tested in Excel and evaluated. The results are shown in Table 14. The average number of loads washed per year is for an American [3] family and could be even less considered the intended placement in small apartments [1]. The theoretical heat exchangers are shown along with the what energy/cost savings that would be achieved. Two different effectiveness parameters were evaluated, 50% and 90%. 50% was chosen as a halfway point between the maximum possible for the heat exchanger and 90% was chosen as the literature [6] stated that is close to the maximum effectiveness practically achievable.

	Actual Design	Theoretical Model 1	Theoretical Model 1
Effectiveness	0.0512	0.5	0.9
Cost savings per load	$0.00554	$ 0.05635	$0.10142
Area Required (m^2)	0.17	3.33	29.97
Length Required (m)	0.75	14.51	130.62
Cost	$304.45	$2,221.32	$18,391.90
Payback loads	54912	39423	181339
Payback time (400 loads per year)	137	99	453

Table 14: Comparison of Heat Exchangers with Varying Effectiveness

With a dryer with a theoretical heat exchanger operating at 50% effectiveness, the system would save $0.05 per load. However, using 5” diameter outer piping and 3” diameter inner piping, the heat exchanger alone would be nearly 15 meters long and would occupy over 3 cubic meters, far too big to fit in a standard size dryer. In addition, the heat exchanger alone would cost more than even the most expensive dryers available on the market today [27]. For a dryer with 90% effectiveness, the heat exchanger would save 10 cents per load but it would need to be 130 m long, take up 30 cubic meters, and cost nearly $20,000. Both of these options are clearly not possible to put in a consumer dryer. Unfortunately, the actual heat exchanger design is also not suitable for commercial production. To improve the effectiveness, possibly to the point where a dryer and heat exchanger design could be viable, the design must be significantly reworked. A major design change could be to use a different style of heat exchanger, one that has much more surface area to enable the transfer of more heat as air passes through it. Two possible designs are the shell and tube design and the fin and plate design as illustrated in Chapter 11 of [8]. Both styles of heat exchanger achieve much higher effectiveness levels in a much smaller area than a concentric double pipe heat exchanger. However, according to [4], both styles of heat exchanger were beyond the necessary scope for evaluation and design.

5.2 Recommendations Regarding Heat Exchanger

In order for this dryer to be a sound financial choice for a potential owner, the cost savings the heat exchanger provides must be greater than the additional cost of design and manufacture added to the price of the dryer. The estimated cost of the heat exchanger as seen in

Table 12, is $304.45. This amount includes both materials and machining and is representative of mass production. For an average American family that washes 400 loads [4] of laundry a year, this equates to a cost savings of $2.22 every year which would mean that the family would need to use the dryer for about 137 years and 3 months to break even. Moreover, this dryer is designed for a small apartment, which would mean that the potential user would use the dryer even fewer times per year, increasing the payback time substantially. In addition, an added cost to the dryer with the heat exchanger both upfront and per hour would be the 2nd fan that would be required to push air through the heat exchanger initially. The fan would increase the upfront cost as it is an additional part which needs to be purchased and the operation of the fan would increase the price per hour that the dryer costs to run. For a customer looking for a dryer with cheaper operating costs and higher purchase costs, a potential option is a gas dryer. While gas dryers can cost $50 to $150 more than their electric counterparts [3] their lower cost per load $0.15-$0.33 results in a much greater cost savings over the lifetime of the dryer. Even if the customer has to pay for gas lines to be run to the laundry room, the economics of a gas dryer far outweigh those of an electric dryer with a heat exchanger. For these reasons, it is not recommended that the company not produce the heat exchanger equipped dryer. The dryer by itself could be economical, although it is slightly more inefficient than comparable products, [11] but the addition of the concentric double pipe heat exchanger to the clothes dryer cannot be recommended.

While the dryer does not make sense to a consumer looking for the cheapest possible cost over the lifetime of the product, it may appeal to an environmentally conscious buyer willing to pay more for a product which uses less energy. According to Nielsen, a large group of consumers is now purchasing items with particular reflection on how green the product, packaging, and company are [28] and [29]. From a survey of people residing in and around a large North American city, 13.1% of subjects were willing to pay significantly more for green products with a larger group of undecided consumers who could possibly be convinced to purchase products from that category [30]. These early adopters of green technology are willing to spend more if they believe that the product is beneficial to the environment. Moreover, considering the relative paucity of dryer efficiency progress in recent decades, [3], this report of a potential improvement to dryer efficiency could set off a desperately needed competition amongst manufacturers to produce the most efficient clothes dryer. If the market research team determines that there are enough consumers willing to pay more to reduce their energy consumption that selling the dryer and heat exchanger combo will be profitable for Dryer Inc., then the company could pursue the manufacturing of “the efficient clothes dryer.”

5.3 Alternative Designs

Dryer

The main parameter that could be varied with the design of the dryer was the relative humidity at the exit of the dryer drum, $\emptyset_3$. As can be seen in Figure 6, as the relative humidity is increased, the temperature at stage 3 of the dryer decreases. This, in turn, decreases the mass flow rate of dry air through the dryer and the heat transfer required from the heating element, lowering the overall cost to run the dryer. The relationship is demonstrated in Figure 7 and Figure 8. For this reason, it was decided that the dryer should be run at the maximum feasible relative humidity at the exit of the dryer drum. According to Professor J. Kadambi [4], this figure can be as high as 90%. The relative humidity determined the temperature at stage 3 based upon the conservation of energy as determined by Equation 11. The selection of relative humidity also determined the mass flow rate of dry air and the heat transfer required to dry the clothes in the allotted time. Therefore, the design space consisted of the relative humidity range of 65-90% at the exit of the dryer drum. Properties of the dryer were evaluated at increments of 5% across this range. The relative humidity at stage 3 of the dryer was selected to be 90% based on the calculations and the resulting graphs demonstrating the benefits of increasing the relative humidity.

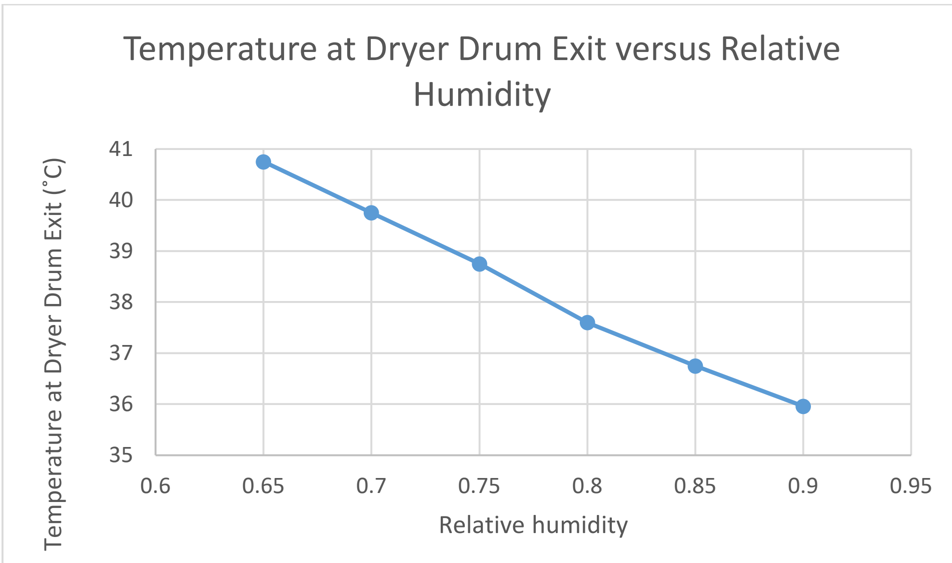

Figure 6: Temperature at Dryer Drum Exit versus Relative Humidity

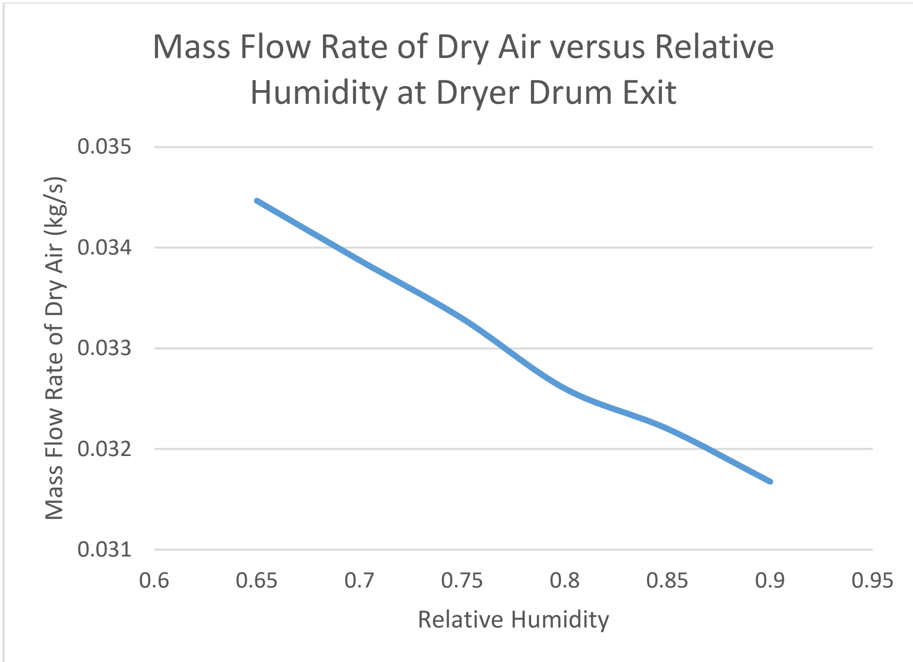

Figure 7: Mass Flow Rate of Dry Air versus Relative Humidity at Dryer Drum Exit

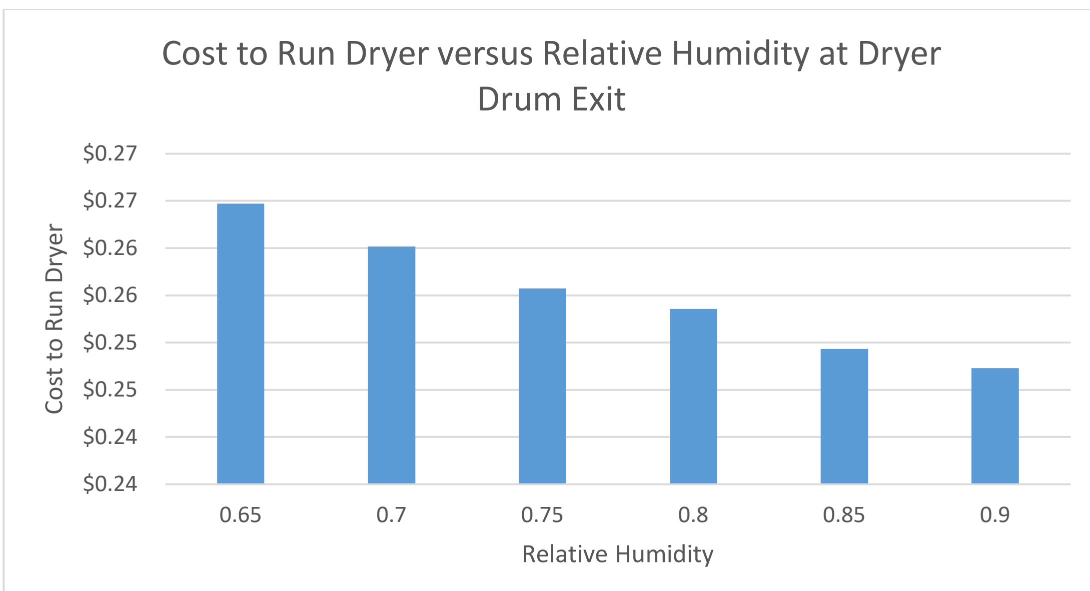

Figure 8: Cost to Run Dryer versus Relative Humidity at Dryer Drum Exit

Relative humidity	Drum Output(˚C)	(kg /s)	(m^3 / s)	(kWatts)	(kWh)	
ϕ_3	T_3	mdot_a	vdot_a	Qdot	Energy	Cost for one cycle
0.65	40.75	0.034	0.029	2.779	2.084	$ 0.26
0.7	39.75	0.034	0.028	2.731	2.049	$ 0.26
0.75	38.75	0.033	0.028	2.685	2.014	$ 0.26
0.8	38.48	0.033	0.028	2.662	1.997	$ 0.25
0.85	37.26	0.032	0.027	2.618	1.963	$ 0.25
0.9	35.96	0.032	0.027	2.596	1.947	$ 0.25

Table 15: Summary of Alternative Designs for Dryer

Heat Exchanger

The design space for the heat exchanger was fairly limited. The thermodynamic properties of both entering fluids were fixed based on the design of the dryer, and the physical dimensions were limited based on the dimensions of commercially available dryers. One parameter that could be varied was the material. Copper and stainless steel were the two materials considered as shown in Table 16.

	k (W/ m k)	U_OA,i (W / m^2 k)	NTU	CR	ε	Qdot (W)
Copper	401	9.977102007	0.053850709	0.99871	0.051099	49.19432
Stainless Steel	16	9.967828573	0.053800656	0.99871	0.051054	49.15093

Table 16: Comparison of Materials for Heat Exchanger

There is a small benefit to the heat transfer rate using copper as shown in the table. However, the real benefit of copper is in the price. Copper is significantly cheaper than stainless steel as represented in Table 17. The dual benefits of copper make it the clear choice for the design.

Prices Listed in Dollars per Foot							
Size	Copper Inner	Copper Outer	SS Inner	SS Outer	Copper Total	SS Total	SS / Copper
5 x 3	16.48	31.80	25.66	31.51	48.28	57.17	118.41%

Table 17: Cost Analysis of Copper versus Stainless Steel for Heat Exchanger Pipes (from [21])

The other design space that was considered for the heat exchanger was the effectiveness, ε. However, this was a theoretical exercise as any effectiveness above the actual effectiveness of 0.0491 would require a larger area heat exchanger which was not physically possible given the constraints of the dryer. This was done merely to show that the economics for a concentric double pipe heat exchanger would not work out even if the heat exchanger efficiency was increased by more than an order of magnitude. See Section 5.1 for a more thorough analysis of the difference theoretical effectiveness designs that were examined.

5.4 Implications of Assumptions

- If the drying process is assumed to be steady, then dryer operates at a constant drying temperature, and the heating element draws the same wattage the entire drying cycle. According to [6], typical dryers have a bulk drying stage and a high-heat stage. Most of the water is evaporated during the bulk drying stage, then the heater cycles on and off during the high-heat stage, so potentially damaging temperatures are avoided. Hence, the steady-state assumption is not true. Because of this variance in water evaporation, the mass balance equation becomes a differential equation and the energy required calculation changes. Furthermore, according to [6], the cycling nature of the high heat stage reduces total energy consumption. During the high heat stage, almost all of the water has been removed from the clothes which typically are at a remaining moisture content (RMC) of 5% during this process. Therefore, running the dryer at the same constant temperature as during the bulk drying stage is inefficient because the clothes have already essentially been completely dried. No attempt was made to account for this as it was beyond the scope of the project, but according to [11], a typical clothes dryer in experiments with a water weight removed of 1.67- 2.43 kg consumed between 1.7-2.3 kWh. This suggests that the designed dryer is slightly more inefficient than a typical commercial clothes dryer.
- As the drying process was assumed to be constant, the mass flow rate of the water and air, and the rate at which the water evaporated from the clothes was held to be constant in this analysis. As it is not constant in an actual application, the thermal analysis becomes a differential equation and the energy spent calculation would change although it would not be expected to have as great of an effect as the inefficiencies introduced by running the drying at a constant operating temperature.
- Because there is no heat loss or pressure loss, all energy is maintained in the system. Thus, the claimed efficiency will be higher than the actual efficiency and energy usage will be lower than if the design was implemented.
- Uniform developed flow in the heat exchanger allows for the use of average values of the flow profile in order to make a simple thermal analysis. Fully developed turbulent flow also conducts heat much more effectively than laminar flow as described in [8].
- Assuming that the air and water properties are nearly the same imply that the properties change very little and that the average is a good estimate of the true value.
- Neglecting the thermal entry length implies step-wise immediate heat transfer within the heat exchanger. Entrance effects in the heat exchanger are not accounted for and would reduce the heat exchanged that occurs [8].
- Neither the air nor water vapor will react with copper, and thus the heat extracted from the heat exchanged was assumed to be constant over the operation of the exchanger. No fouling was accounted for, which would tend to reduce the heat transfer over a considerable enough amount of time (on the order of 1 year).

Section 6: Conclusion

This report documents the design of a clothes dryer along with a concentric double pipe heat exchanger to accompany the dryer and intended to improve its efficiency. However, although the heat exchanger does improve the efficiency of the dryer, based upon the economic analysis, most crucially the payback time for the additional cost of the heat exchanger, the design team must recommend that Dryer Inc. not include the heat exchanger in the clothes dryer. The final specifications for the dryer are as follows. The incoming air and water vapor gas mixture is at the ambient conditions of the surroundings which are 50% relative humidity, ω, and a temperature of 21.1 ˚C. The dryer is designed to run for 45 minutes and remove 2.268 kg of water from the clothes operating in a steady state. The maximum temperature in the dryer, 100˚C, occurs immediately after the heating element [1]. The water is evaporated from the clothes at a constant rate over the course of the drying cycle and the dryer operates in a steady state without heat cycling. The conditions at the exit of the dryer drum are a relative humidity, ω, of 90% and a temperature of 35.96 ˚C. The air mass flow rate through the dryer is 0.03167 kg/s, and the volumetric flow rate is 1.584 m^3/min (55.94 cubic feet per minute). The dryer draws an instantaneous power load of 2.554 kW and requires 1.947 kWh of energy for a 45-minute drying cycle. This translates to a cost of $0.25 per load based on current electricity costs in Northeast Ohio [2]. The pressure loss through the ducting and the lint filter of the dryer is 1117 Pa which requires a fan driven by a 38 W motor to pull the air through the entire system.

The heat exchanger was entirely designed, and although its inclusion in the dryer cannot be recommended, the following design specifications were made. The heat exchanger is the concentric double pipe style, with the hot air and water vapor mixture exhausting from the dryer in the outer pipe (annulus) and a stream of cold air and water vapor mixture entering at the surrounding conditions in the inner pipe. The final heat exchanger design calls for copper pipes with a 5” (0.127 m) nominal inner diameter outer pipe, a 3” (0.07062 m) nominal outer diameter inner pipe, and a wall thickness of 1/16” (0.001587 m). The size and the length of the heat exchanger were limited by the physical dimensions of the dryer which severely restricted the heat exchanger effectiveness. The heat exchanger could have a maximum length of 2.5’ (0.75 m) which resulted in an effectiveness of 0.0512 and a paltry 49.3 W extracted from the air and water vapor gas mixture exhausted from the dryer. The payback time for the additional $304.45 cost of the heat exchanger is easily over 100 years even for a family washing 400 loads of laundry per year. Based on an economic study, the heat exchanger cannot be recommended for inclusion in the clothes dryer. A heat exchanger that increases heat transfer area in a relatively small volume compared to a concentric double pipe, such as a shell and tube or a plate and fin heat exchanger, would be a much better choice and could even be practical given the efficiency possible with these designs. Although there clearly exists a need for a reduction in the extreme energy usage of commercial clothes dryers, the concentric double pipe heat exchanger is not the ideal way to create the efficient clothes dryer.

Section 7: Acknowledgements

Our design team would like to thank the following individuals for their assistance with the design process. The efficient clothes dryer would not have been completed without their generous counsel.

Professor Kadambi: for providing a suggested method for the design process, general all-around advice, and for being willing to answer our numerous questions related to every aspect of a clothes dryer and heat exchanger.

Teaching Assistant Scott Rubeo: for general assistance and for being willing to read and suggest improvements to an initial draft of this report.

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Section 9: Academic Integrity Statement

This report was created by the group members listed below in compliance with Case Western Reserve University's Academic Integrity policy available at:
https://students.case.edu/community/conduct/aiboard/policy.html

By signing below, each group member is giving their affirmation to the above statement.

Name:	Signature:
William Koehrsen	William
Tyler Eston	
Arturo Garcia	Arturo Garcia
Theodore Bastian	

Section 10: Appendices

I: Data Tables

State 1	
T_1 (k)	294.26
ϕ_1	0.5
ω_1	0.00785
P_sat,1 (kPa)	2.495
P_1 (kPa)	101.325
h_a,1 (kJ/kg)	420.62
h_v,1 (kJ/kg)	2539.4
P_v,1 (kPa)	2.5051
ρ_a,1 (kPa)	1.1997

Table 18: Conditions at Stage 1 of Dryer

State 2	
T_2 (k)	373
ω_2	0.00785
P_2 (kPa)	101.325
h_a2 (kJ/kg)	500.03
h_v2 (kJ/kg)	2675.3
h_l,2 (kJ/kg)	88.651
m_dot,l (kg/s)	0.000839842
P_sat,2 (kPa)	101.325
P_v,2 (kPa)	100.88

Table 19: Conditions at Stage 2 of Dryer

	Velocity (m/s)	Reynolds Number	Nusselt Number	Convective Heat Transfer Coefficient
Inner (cold)	6.30	30246.05	76.96	27.36
Outer (hot)	3.59	26872.54	72.36	15.05

Table 20: Heat Exchanger Fluid Calculations

Number of Transfer Units		
NTU		0.053851
Heat Capacity Ratio		
CR = Cmin / Cmax		0.99871
Effectiveness factor		
ε		0.051099
delta_T,max		30.2
T_c, out		22.64
Q_dot		49.26
T_h, out		49.75

Table 21: Heat Exchanger NTU-Effectiveness Calculations

II: Sample Calculations for Final Design

Included are the calculations for the final design. Some of the numbers may be in slight disagreement with the data tables provided. Rounding errors in the calculations presented below are to blame for any discrepancy. For the final design, numbers calculated in Excel were assumed to be correct and were the figures used in the final design specifications.

Dryer Calculations

The absolute humidity can be a found from equation 6:

$$\omega = 0.622 * \left(\frac{P_{sat}(T)}{P - P_{H2O,v}(T)}\right) * \emptyset$$

$$\omega_1 = 0.622 * \left(\frac{2.495}{101.325 - 2.505}\right) * 0.5 = 0.00785$$

The absolute humidity in stage 2 is the same because only heat was added:

$$\omega_1 = \omega_2 = 0.00785$$

The mass flow rate of water is:

$$\dot{m}_l = \frac{\frac{5\ lbs}{2.205\frac{lb}{kg}}}{45\ s * 60\frac{s}{m}} = 0.000839842\ kg/s$$

A relative humidity at stage 3 is selected, and Excel is used to evaluate the LHS and RHS at a range of temperatures to find the temperature at which the two sides are equal:

$$h_{a,2} + \omega_2 h_{v2} + \left[\left(\frac{0.622 * \emptyset_3 * P_{sat,3}}{P_3 - \emptyset_3 P_{sat,3}}\right) - \omega_2\right] h_l = h_{a,3} + \left(\frac{0.622 * \emptyset_3 * P_{sat,3}}{P_3 - \emptyset_3 P_{sat,3}}\right) h_{v,3}$$

For chosen $\emptyset_3 = 0.9$

At $T_3 = 36.00\ ℃$

$$500.2 + 0.0078 * 2566 + \left[\left(\frac{0.622 * 0.9 * 13.089}{P_3 - 0.9 * 13.089}\right) - 0.0078\right] * 88.651$$

$$= 435.6 + \left(\frac{0.622 * 0.9 * 13.089}{101.325 - 0.9 * 13.089}\right) * 2566.3$$

$$523.9 = 523.8$$

At this point, the graph of the LHS versus the RHS can be used to obtain the precise temperature at which the energy conservation equation is satisfied. This occurs at

$$T_3 = 35.96\,^{\circ}C$$

Now that T_3 is known the mass flow rate of air can then be solved by:

$$\dot{m}_a = \frac{\dot{m}_l\left(h_{v,3} - h_l\right)}{h_{a,2} - h_{a,3} + \omega_1\left(h_{v,2} - h_{v,3}\right)} = \frac{0.000839842 * (2566.3 - 88.651)}{500.18 - 435.6 + 0.0078(2675.6 - 2566.3)}$$

$$= 0.03167\ kg/s$$

The absolute humidity at stage 3 can be calculated:

$$\omega_3 = \omega_2 + \frac{\dot{m}_l}{\dot{m}_a} = 0.0078 + \frac{0.000839842}{0.0389} = 0.0344$$

Now the heat rate of the element can be solved:

$$\dot{Q}_{in} = \dot{m}_a[(h_{a,2} + \omega_2 h_{v2}) - (h_{a,1} + \omega_1 h_{v1})]$$

$$= 0.03167 * [(500.18 + 0.00785 * 2675.6) - (420.62 + 0.00785 * 2539.4)$$

$$= 2.554\ kW$$

Then the head loss is calculated. The total head loss is found from the individual head loss of each pipe fitting in the dryer ductwork. Values for the head loss coefficient are from [8]:

$$K \equiv \frac{h_L}{\frac{\left(V_{avg}\right)^2}{2 * g}}$$

The average velocity can be determined from the volumetric flow rate of air divided by the area of the duct through which the air flows:

$$V_{avg} = \frac{0.027\,\frac{m^3}{s}}{0.0127\ m^2} = 2.68\,\frac{m}{s}$$

For 5” duct bend K = 0.019

$$h_L = K * \frac{\left(V_{avg}\right)^2}{2 * g} = 0.019 * \frac{2.687^2}{2 * 9.81} = 0.00699\ m$$

From the head loss, the pressure drop can be calculated for the 6 bends:

$$\Delta P = h_L * \rho * g = .00699 * 998.82 * 9.81 = 68.508\ Pa$$

For the dryer drum K = 0.1

$$h_L = K * \frac{(V_{avg})^2}{2 * g} = 0.01 * \frac{0.117^2}{2 * 9.81} = 6.977 * 10^{-6}\ m$$

Now pressure drop for the drum:

$$\Delta P = h_L * \rho * g = 6.977 * 10^{-6} * 998.82 * 9.81 = .0683\ Pa$$

For the lint filter 75% blocked K = 0.5:

$$h_L = K * \frac{(V_{avg})^2}{2 * g} = 0.5 * \frac{1.649^2}{2 * 9.81} = 0.0693\ m$$

Now the pressure drop for the filter:

$$\Delta P = h_L * \rho * g = 0.0693 * 998.82 * 9.81 = 678.99\ Pa$$

The total pressure drop is the sum of the component pressure drops:

$$\Delta P = 678.99 + 0.0683 + 68.508 * 6 = 1118\ Pa$$

Now the power output needed to drive the fan is calculated:

$$Po = \Delta P * \dot{V} = 1118\ Pa * 0.032 \frac{m^3}{s} = 38 \frac{J}{s} (W)$$

Heat Exchanger Calculations

The heat exchanger is of the concentric double pipe style utilizing counterflow with the hot gas mixture exhausted from the dryer in the outer pipe (annulus) and the cold gas mixture drawn in from the ambient surroundings in the inner pipe. The heat exchanger was designed with the NTU-Effectiveness method as outlined in the Methods and based on the description from [8].

First the heat capacity rates of the fluids are obtained:

$$C_C \equiv (\dot{m}c_p)_C = 0.03167 * 1006.4 = 31.878 \frac{J}{s * K}$$

$$C_H \equiv (\dot{m}c_p)_H = 0.03167 * 1007.7 = 31.920\ \frac{J}{s * K}$$

Then the maximum possible heat transfer is obtained:

$$\dot{Q}_{\max possible} = C_{min}(T_{H,in} - T_{C,in}) = 31.878 * (309.11 - 294.25) = 963 \frac{J}{s}$$

The heat capacity ratio is:

$$CR = \frac{31.878}{31.920} = 0.9987$$

In order to find the overall heat transfer coefficient, first the fluid velocities are calculated:

$$V_i = \frac{\dot{m}}{\rho * A} = \frac{0.03167}{1.1998 * 0.004188254} = 6.30\ m/s$$

$$V_o = \frac{\dot{m}}{\rho * A} = \frac{0.03167}{1.0878 * 0.00810732} = 3.59\ m/s$$

Then the Reynolds number of the fluids:

$$Re_i = \frac{V * D_e}{\nu} = \frac{7.7465 * 0.073025}{0.000015219} = 30246$$

$$Re_o = \frac{V * D_e}{\nu} = \frac{4.4139 * 0.135466667}{0.000018106} = 26872$$

As can be seen, the flow in both pipes is fully developed and turbulent. The Nusselt numbers are then calculated

$$Nu_i = 0.023 * Re^{0.8} * \mathrm{Pr}^{\mathrm{n}} = 0.023 * 30246^{0.8} * 0.70798^{0.4} = 76.96$$

$$Nu_o = 0.023 * Re^{0.8} * \mathrm{Pr}^{\mathrm{n}} = 0.023 * 26872^{0.8} * 0.70442^{0.3} = 72.36$$

From the Nusselt number the convection heat transfer coefficient can be determined:

$$h_{conv,i} = \frac{Nu * k}{D_e} = \frac{76.96 * 0.025956}{0.073025} = 27.36\ \frac{W}{m^2 k}$$

$$h_{conv,o} = \frac{Nu * k}{D_e} = \frac{72.36 * 0.028177}{0.135466667} = 15.05 \frac{W}{m^2 k}$$

Next the overall inner heat transfer is obtained:

$$U_{OA,i} = \frac{1}{\frac{1}{h_{conv,i}} + \frac{r_{w,i}}{k_w} \ln\left(\frac{r_{w,o}}{r_{w,i}}\right) + \frac{r_{w,i}}{r_{w,o}} * \frac{1}{h_{conv,o}}}$$

$$= \frac{1}{\frac{1}{27.36} + \frac{0.0365}{401}\ln\left(\frac{0.0381}{0.0365}\right) + \frac{0.0365}{0.0381} * \frac{1}{15.05}} = 9.98\ \frac{W}{m^2 k}$$

The inner area of the heat exchanger is calculated:

$$A_i = \pi * 0.073025\ m * 0.75\ m = 0.172\ m^2$$

The number of transfer units (assuming CR ≈ 1):

$$NTU = \frac{9.98\ \frac{W}{m^2 k}\ * 0.172061\ m^2}{31.9\ \frac{J}{s * k}} = 0.05385$$

The effectiveness of the heat exchanger:

$$\varepsilon = \frac{0.05385}{1 + 0.05385} = 0.0512$$

The actual heat transfer is found from:

$$\dot{Q}_{actual} = \varepsilon * \dot{Q}_{\max possible} = 0.0512 * 962 = 49.3\ \frac{J}{s}$$

Outlet temperatures are found:

$$T_{c,out} = T_{c,in} \pm \varepsilon\left(T_{H,in} - T_{C,in}\right) = T_{min,out} = 21.1 + 0.049136(51.3 - 21.1) = 22.64^{\circ}C$$

$$T_{h,out} = T_{h,in} \pm \varepsilon\left(T_{H,in} - T_{C,in}\right) = T_{min,out} = 51.3 - 0.049136(51.3 - 21.1) = 49.75^{\circ}C$$

The pressure drop through the heat exchanger is from friction and losses in the end caps:

$$f_D = 0.05\ from\ Moody\ Chart\ (function\ of\ Re\ and\ \frac{\varepsilon}{D})$$

Pressure drop in the inner tube:

$$\Delta P = \frac{1}{2} * 0.05 * \frac{0.75m}{0.073025m} * 1.1998\frac{kg}{m^3} * \left(6.30\frac{m}{s}\right)^2 = 18.49\ Pa$$

Pressure drop in the outer tube:

$$\Delta P = \frac{1}{2} * 0.05 * \frac{0.75m}{0.127m} * 1.1998\frac{kg}{m^3} * \left(3.59\frac{m}{s}\right)^2 = 7.822\ Pa$$

Loss coefficient from three-way end cap:

$$k = 0.8\ from\ page\ 767\ of\ [8]$$

Head loss from three-way end cap in inner tube:

$$h_L = \frac{0.8 * \left(6.30\,\frac{m}{s}\right)^2}{2 * 9.801\,\frac{m}{s^2}} = 2.45\ Pa$$

Head loss from three-way end cap in outer tube:

$$h_L = \frac{0.8 * \left(3.59\,\frac{m}{s}\right)^2}{2 * 9.801\,\frac{m}{s^2}} = 0.794\ Pa$$

Total Pressure Loss inner tube:

$$\Delta P = 18.49\ + 2.45 = 20.94\ Pa$$

Total Pressure Loss outer tube:

$$\Delta P = 7.822 + 0.794 = 8.62\ Pa$$

Pump Power Required for inner pipe:

$$Po = 20.94\ Pa * 0.0324\,\frac{m^3}{s} = 0.68\ W$$

Pump Power Required for outer pipe:

$$Po = 8.62\ Pa * 0.0358\,\frac{m^3}{s} = 0.31\ W$$

Always Moving Forward.®

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SAFETY DATA SHEET

SECTION 1. IDENTIFICATION

Copper Tubing (all sizes and wall thicknesses)

Cerro Flow Products LLC
PO Box 66800, St Louis, MO 63166-6800 Telephone number 618-337-6000

Recommended use: Plumbing and industrial copper tubing. Restricted use: None known

SECTION 2. HAZARD IDENTIFICATION

CAUTION

Inhalation Hazard Fumes are created by heating copper past its melting point. Proper soldering or sweating copper tubes will not produce fumes. Brazing of copper tube may produce fumes. Consult the Copper Development Association Inc. (CDA) “The Copper Tube Handbook” for proper joining methods, and recommended solders, fluxes and filler metals (see CDA link on www.cerroflow.com to obtain handbook).

Ingestion Hazard Ingestion of metallic copper is not a primary route of exposure. Metallic copper may be moderately irritating to the gastrointestinal tract.

SECTION 3. COMPOSITION/INFORMATION ON INGREDIENTS

MATERIAL OR COMPONENT	C.A.S. No.	WT. %
Copper	**7440-50-8**	**99.9+**

SECTION 4. FIRST AID MEASURES

Inhalation: Remove from exposure; place individual under care of a physician.
Ingestion: Induce vomiting in conscious individual and call a physician.
Skin or Eyes; Flush with plenty of water. If symptoms develop, consult a physician.

SECTION 5. FIRE FIGHTING MEASURES

FIRE AND EXPLOSION HAZARDS	FIRE EXTINGUISHING AGENTS RECOMMENDED	FIRE EXTINGUISHING AGENTS TO AVOID .
Not Applicable	**No specific agents recommended**	**No specific agents recommended**

SPECIAL FIRE FIGHTING PRECAUTIONS .

Copper tube will not burn or give off toxic gases in normal fires Use fire fighting methods compatible with surrounding materials.

SECTION 6. RELEASE MEASURES

SPILLS OR LEAKS .

Proper installation of copper tubing will not produce dust. Consult Copper Development Association, Inc (CDA)”The Copper Tube Handbook” for proper joining methods (See CDA link on http://www.cerroflow.com to obtain handbook) Vacuuming is preferred for dust. Do not use compressed air for cleaning. Recycle unused or scrap copper tube at a local scrap metal dealer.

SECTION 7. HANDLING AND STORAGE

NORMAL HANDLING .

Avoid conditions which create fumes or fine dust. Use of approved respirators is required where adequate ventilation cannot be provided. Do not use copper tubing where incompatible materials may be present, (see section X).

STORAGE .

Avoid storage near incompatible materials, see Section 10.

SECTION 8. EXPOSURE CONTROLS/PERSONAL PROTECTION

Permissible Air Conc. (mg/m3)			
	OSHA		ACGIH
Dust	1.0		1.0
Fume	0.1		0.2

ENGINEERING CONTROLS . .

Local exhaust is recommended for dust and/or fume generating operations where airborne exposure may exceed permissible air concentrations.

PERSONAL HYGIENE .

Avoid inhalation or ingestion. Practice good housekeeping and personal hygiene procedures. Showering is recommended if significant dust exposure occurs.

SPECIAL: PRECAUTIONS/PROCEDURES/LABEL INSTRUCTIONS LABEL SIGNAL WORD: .

No special precautions. NOT APPLICABLE

RESPIRATORY PROTECTION .

Where airborne exposures may exceed OSHA/ACGIH permissible air concentrations, the minimum respiratory protection recommended is a negative pressure air purifying respirator with cartridges that are NIOSH/MSHA approved against dust, fumes, and mists having a TWA not less than 0.05 mg/m3

EYES AND FACE .

Safety glasses recommended when dust or shavings may exist.

OTHER CLOTHING AND EQUIPMENT .

Protective clothing is recommended to prevent burns during installation of tube or splattering of fluxes, solder or filler metals.

SECTION 9. PHYSICAL/CHEMICAL PROPERTIES

MATERIAL IS (AT NORMAL CONDITIONS)	APPEARANCE AND ODOR
Solid	Yellow-red metal, various shapes and sizes.

MELTING POINT (DEGREES C)	BOILING POINT (DEGREES C)	SPECIFIC GRAVITY (H2O = 1)	VAPOR DENSITY (AIR = 1)
1083	2595	8.96	Not applicable

SOLUBILITY IN WATER (% BY WT.)	pH	VAPOR PRESSURE (mm Hg)	EVAPORATION RATE
Insoluble	Not Applicable	Not Applicable	Not Applicable

SECTION 10. STABILITY AND REACTIVITY

STABILITY	CONDITIONS TO AVOID
Stable	Not Applicable

INCOMPATIBILITY (MATERIALS TO AVOID) .

Reacts violently with acetylene, hydrogen peroxides, gaseous chlorine, ammonia nitrate, bromates, chlorates, hydrogen sulfide, lead azide, and hydrazine.

HAZARDOUS DECOMPOSITION PRODUCTS	HAZARDOUS POLYMERIZATION	CONDITIONS TO AVOID .
Copper does not decompose	Will not occur	Not Applicable

SECTION 11. TOXICOLOGICAL INFORMATION

PRIMARY ROUTES OF ENTRY	INHALATION	CARCINOGENICITY
	X	Not listed as a carcinogen by NTP, IARC, or OSHA

ACUTE OVER EXPOSURE (SYMPTOMS AND EFFECTS)

A. Fumes are created by heating metallic copper past its melting point. Proper soldering or sweating copper tubes will not produce fumes. Brazing of copper tube may produce fumes. Consult Copper Development Association, Inc. (CDA) "The Copper Tube Handbook" for proper joining methods, and recommended solders, fluxes and filler metals (see CDA link on Cerro Flow Products, LLC website to obtain handbook). Use approved ventilation or respiratory protection if the possibility of fumes exists. Inhalation of fume may cause irritation of the respiratory tract or metal fume fever (chills, fever, aching muscles, dry mouth and throat, headache, nausea, vomiting, and diarrhea). Onset may be delayed several hours.

B. Ingestion of metallic copper is not a primary route of exposure. Metallic copper may be moderately irritating to the gastrointestinal tract.

CHRONIC OVEREXPOSURE (SYMPTOMS AND EFFECTS)

No long term effects. Skin irritation or discoloration of the skin and hair are short term.

MEDICAL CONDITIONS POSSIBLY AGGRAVATED

Wilson's Disease (an abnormal genetic condition) could be aggravated.

LD50 (SPECIES, ROUTE)	LC50 (SPECIES)	MUTAGENICITY
Copper: 3.5 mg/kg (mouse, intraperitoneal)	Not Available	Not positive in Ames test

Health Hazard Ratings				
	Health	Flammability	Reactivity	Other
NFPA	1	0	0	
HMIS	1	0	0	E

SECTION 12. ECOLOGICAL

ECOTOXICITY	ENVIRONMENTAL FATE
The LC50 for copper in the fathead minnow is 12 mg/L.	Acid solutions promote mobility and solubility of copper.

SECTION 13. DISPOSAL CONSIDERATIONS

WASTE DISPOSAL METHODS (DISPOSER MUST COMPLY WITH FEDERAL, STATE, AND LOCAL DISPOSAL OR DISCHARGE LAWS) .

Recycling or disposal must be in accordance with the appropriate federal, state, or local statutes or regulations.

SECTION 14. TRANSPORT

DOT REGULATION AND ID (OR PIN) NUMBER .

Not a DOT controlled material (United States).
Special Provisions for Transport: Marine Pollutant

SECTION 15. REGULATORY INFORMATION

WHMIS CLASSIFICATION, SARA REGULATION AND OTHER INFORMATION .

WHMIS does not classify this material
TSCA Status --------------------------- On TSCA Inventory
Regulated under SARA Title III:
Sect. 302 ----------------------------- None
Sect. 311/312 ----------------------- Immediate and Delayed
Sect. 313 Chemicals ------------- Copper
CERCLA Reportable Quantity ------ 5,000 pounds for Copper Powder

Federal and State Regulations:
Pennsylvania RTK: Copper Massachusetts RTK: Copper TSCA 8(b) inventory: Copper CERCLA: Hazardous substances. Copper

WHMIS (Canada): CLASS D-2A: Material causing other toxic effects (VERY TOXIC).

SECTION 16. OTHER INFORMATION

ISSUED DATE	**SUPERSEDES .**
August 15, 2013	**October 1, 2011**

PERMISSIBLE CONCENTRATION REFERENCE .
OSHA regulations for airborne contaminants 29 CFR 1910.1000 and 1018; ACGIH Threshold Limit Values for Chemical Substances

HAZARD INFORMATION REFERENCES .
Documentation Up to date, curated data provided by Mathematica's ElementData function from Wolfram Research, Inc

GENERAL .
Copper Development Association, The Copper Tube Handbook, 2010

Notes

No additional information.