Waste heat recovery from a vented electric clothes dryer utilizing a finned-tube heat exchanger

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Abstract

Conventional residential clothes dryers continuously vent moist, hot air during the drying process. The vented air leaves the home but still has useful temperature and humidity that could be recovered to offset other heating demands in the home. A study is carried out to quantify the amount of heat extracted from the waste heat stream of a conventional, vented clothes dryer. To extract the heat, a water cooled, fin-and-tube heat exchanger is located within the exhaust duct. A steady state thermodynamic dry coil and wet coil model was built in Engineering Equation Solver (EES). The model accounts for the heat exchangers geometry and applies a dimensionless heat and mass transfer analogy (Colburn-j-factor) determined empirically to calculate an overall heat transfer coefficient for both dry and wet areas of the coil. Assuming water and moist air inlet temperatures and air and water side flow rates, a rate of heat transfer and outlet temperatures of both streams are predicted. Comparing the model prediction to experimental results identifies the accuracy of the model. Using energy balance, the potential heat available and the heat recovered are calculated and the effectiveness of the finned-and-tube heat exchanger are determined. It was observed that approximately 0.1 kWh of energy was recovered leading to a heat exchanges effectiveness of 55%.

Keywords: Waste Heat Recovery, Heat Exchangers, Thermodynamic Modeling, Clothes Dryer, Finned-and-tube Heat Exchangers

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1. Introduction

As of the year 2017, 1347 billion kWh of electricity was utilized across all sectors in the United States. A sector-by-sector breakdown indicates that 56 billion kWh (4%) of the electricity was consumed just by clothes dryers[1]. Within the United States, more than 80% of homes contain clothes dryers [2]. There are multiple types of clothes dryers including a heat-pump clothes dryer, a condensing clothes dryer, and a vented clothes dryer. This study focuses on vented clothes dryers, mainly obtaining energy dissipated away in the hot, moist air stream exiting the dryer.

Many authors investigated the impact of modifying the electric heating element within the unit. Bansal et al. [3] replaced the conventional electric heating element of the dryer with a tube-to-fin heat exchanger utilizing water and air. Their research indicated that the novel heat exchanger caused the temperature of air entering the drum to be higher than it was with the conventional heating element. This increased inlet temperature generated a lower moisture extraction rate (less kWh of electricity use per kg of water removed) and dried the clothes in roughly 15 to 18 minutes less time than the conventional dryer. Zhao et al. [4] did a similar study where they replaced the conventional heating element of a dryer with modified electric resistance wire. The resistance wire was modified to have aluminum splints in attempts to increase the effective area of contact the heating element has with the air. As a result of this modification, the inlet and outlet temperatures of the drum increased by 13 C and 8 C (w.r.t standard heating element). The moisture extraction rate decreased indicating a 4.5 percent improvement in energy efficiency and the final moisture content decreased from 7.9 percent to 5.8 percent. The drying time decreased from 170 minutes to 98 minutes. Bassily [5] took a different approach of optimizing costs of the dryer. A conical heating element was used parameters like cone angle, coil wire length, diameter, and resistance, along with insulation thickness as well as coil and surface emissivity. This study indicated that the minimum achievable values of cone angle, coil wire diameter, coil emissivity along with the reduction of insulation thickness while air outlet temperature decreases, achieved the lowest cost.

Huelsz et al. [6] used energy and exergy analysis to determine the need for improvement within a dryer. They plugged some of the leakages within the dryer and provided lower electricity to the heating element. The results indicate 11 percent less consumption of energy with only a 0.8-minute increase in drying time.

Another open loop process used was a discussed in [7]. TeGrotenhuis et al. designed a hybrid heat pump system where a vapor compression heat pump along with a heat recovery heat exchanger was utilized and R-134a was used as a refrigerant. This system used a standard vapor compression cycle but added another heat exchanger after drum outlet where heat was recovered. At each iteration of the cycle, ambient air as being input into the system. The heat recovery heat exchanger preheats the air inlet into the drum. With a constant condensing temperature of 70 C, approximately 30 percent energy savings along with a drying time 1/3 shorter than the conventional dryer is attainable. The energy factor, a term indicating the mass of water evaporated per kWh nearly doubled that of the conventional dryer.

Related to the field of study focused in this paper, authors discussed the utilization of waste heat. Han et al. [8] completely change the standard drying process to a method that captures wasted heat from a residential air conditioning unit (RAC) and sends it into a chamber containing the moist clothes. The results conveyed that the RAC waste heat dryer utilizes approximately 1.2 percent of the energy a conventional dryer utilizes. Jian et al. [89 modified the dryer to use a heat pipe heat recovery system to capture waste heat expelled from the dryer and to preheat the drum inlet air. This modification resulted in energy efficiency rising from 47.2 percent to 55.8 percent, a reduction of energy consumption from 4.2 kWh to 3.5kWh, and a reduction in the specific moisture extraction rate (electricity consumed per kg of moisture removed) from 1.33 kWh/kg to 1.1 kWh/kg. The final moisture content of the dryer load was less than the base condition. Tomlinson et al. [10] designed a home that captured the waste heat expelled from multiple appliances and stored it in the form of hot water. Focusing on the dryer specifically, n air to water heat exchanger was made that consisted of 2 copper tubes wrapped in parallel around the exhaust. Heat was recovered from the exhaust with an effectiveness (actual heat transfer as compared to the maximum possible heat transfer) of 10 percent.

A not so common method for drying clothes in the United States is utilizing a heat pump or a closed loop process in the dryer. In Braun et al.s study [11], a reverse Brayton cycle with air as the flow substance is implemented with a heat exchanger recovering some of the heat and preheating the air. In this closed loop system, the energy efficiency was found to have increased by 40 percent as compared to a conventional dryer with the benefit of not requiring venting. Comparing this method to a vapor compression cycle, the reverse Brayton cycle heat pump dryer costs less and low pressures within this system allow for easy access of the heat exchanger to remove fouled lint. Jian et al. [12] studied the impact of changing the condenser of a condensing dryer to a water-cooled heat exchanger. Increasing the air flow rate increases the rate of heat transfer of the heat exchanger providing a lower relative humidity at the condenser outlet. This results in an increase in moisture evaporation rate of water within the drum and MER to decrease. However, the energy consumption increases. Increasing the water flow rate has similar outcomes while maintaining the energy consumption.

Of the aforementioned papers, only 1 [10] applies directly to the research we are conducting. In this paper, it is clear that the heat recovery heat exchanger was haphazardly designed as the paper judged its heat exchanger to only attain 10 percent effectiveness. The effective area of contact between the exhaust air and the water side is not enough for there to be substantial heat recovery. Our study will attempt to increase the effectiveness by using a tube-to-fin heat exchanger so that more heat from the exhaust air can be recovered in the form of hot water.

2. Methods

2.1. Experimental Setup

The test stand consisted of a commercial residential vented clothes dryer with components a heat exchanger, secondary lint trap, sensors, a flow straightener, and an auxiliary fan attached to it. Initially, the dryer duct was 10.2 cm in diameter. Due to the purchased heat exchanger being 6 inches in diameter, the dryer duct was expanded from 10.2 cm to 15.25 cm and protruded 2.54 meters outside the dryer to accommodate the additional components. The specific section lengths break down is provided in Table 2. In Figure 1, the attachments onto the dryer duct are illustrated. To reduce heat losses from the duct, the dryer duct was insulated with 5.1 cm of VersaMat insulation [13] and foil tape was used to seal any gaps between duct connections.

In Figure 1, circles with numbers inside represent thermocouples, diamonds represent the relative humidity sensor, vertical lines represent the heat exchanger, horizontal lines represent the flow straightener, and the fan shaped symbol represents the auxiliary fan. A secondary lint trap is attached to the

Table 1: Uncertainties

Sensor	
Type T Thermocouple	±0.75%
Relative Humidity	±2%
Ebtron (Flow Meter)	±3%

Table 2: Dryer Duct Section Breakdown (Duct Work Sections indicate sections of hollow piping)

Dryer Sections	Length (cm)
Secondary lint trap	15.2
Duct Work Section 1	48.1
1st Section of T and RH sensors	9.3
Heat Exchanger	27.4
2nd Section of T and RH sensors	9.3
Flow Straightener	16.2
Duct Work Section 2	57.1
Flow Meter	9.3
Duct Work Section 3	44.5
Auxiliary Fan	18.2

existing dryer duct to ensure lint traveling through the duct is reduced. To the outlet of this, the first section of thermocouples and relative humidity sensors is attached. The finned-and-tube heat exchanger (Table 3) is connected in between the previous sensor section and another section of thermocouples and relative humidity sensor. Details on how these attachments were made will be provided later. After the second section of sensors, a long pipe section is attached consisting of a flow straightener at the beginning and a flow meter (Ebtron) at the end to measure flow rate. The pipe section is long to warrant a fully straightened flow. Lastly, an auxiliary fan is attached with speed controlled by a slider to control the flow rate of the moist air. All these components provide an obstruction to the path of the moist air reducing flow rates. The auxiliary fans goal is to account for this loss in flow rate.

To place the thermocouples and relative humidity sensor into the dryer duct, holes were drilled around the duct. High gauge (thinner) wires were tied around the duct to create and x-shaped bracket inside of the duct. Using the centroids of the 4 quarters created by the x-shaped bracket, the locations

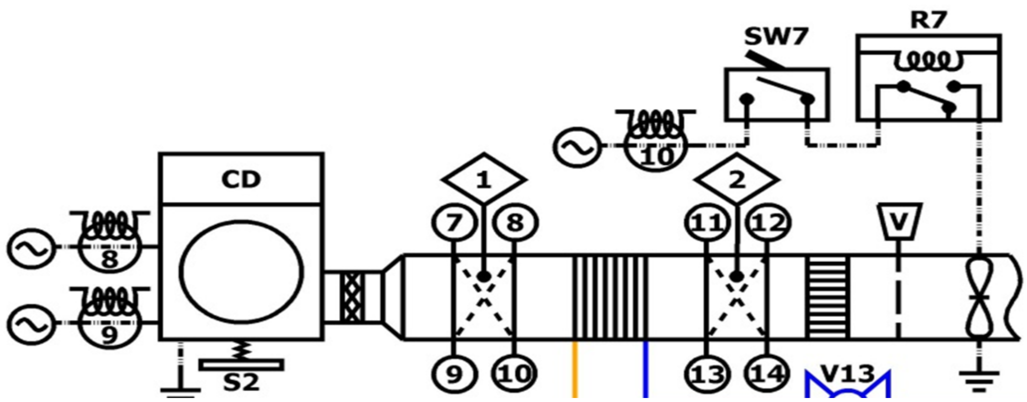

Figure 1: Dryer duct layout

Table 3: Fin Parameters

Fin	
Thickness	0.5 mm
Width	15 cm
Depth	6.5 cm
Number of fins	102 fins
Fin Spacing	0.5 cm

for thermocouple beads were determined. T-type thermocouple wires were used to create the 4 thermocouple beads and these beads were tied to the x-bracket to provide support. A hole was drilled for the relative humidity sensor and its support braces were drilled into the duct work. Type T thermocouple beads were secured to the water side of the heat exchanger using zip-ties. All the mentioned sensors were wired into an Agilent 34972a which was connected to a DAQ system and a LabView [16] code was developed to read the sensor values.

The Ebtron (flow meter) was attached to the dryer duct using the same method as the relative humidity sensor and was read into the mentioned LabView VI [16].

2.2. Modeling Effort

To best understand the data that would be retrieved by the experimental setup a model is necessary for the heat exchanger. This model was spread into

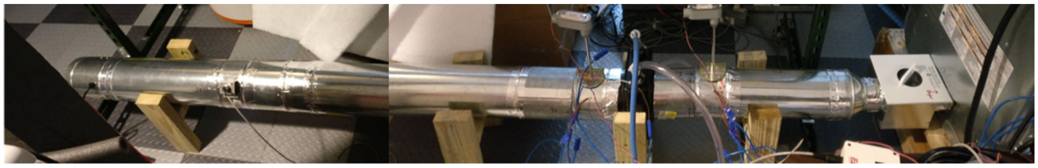

Figure 2: Physical Layout of Dryer Duct

a wet and dry coil analysis. A wet coil analysis indicates that condensation occurs through the cycle at an extreme level as the inlet temperature of the water side is lower than the dew point temperature of the incoming waste moist air stream. A dry coil analysis occurs when the air streams dew point temperature is higher than the water side inlet temperature of the heat exchanger, thus not leading to condensation. Though the actual system is a transient system, the modeling system was assumed to be at steady state to simplify calculations. To create a model, Ian Bells (former Herrick Laboratories student) open source method on calculating dehumidification in a heat exchanger was utilized [14] A key parameter is the overall heat transfer coefficient α. To determine this parameter, the Engineering Equation Solver software [15] was utilized. A geometric analysis of the air side of the finned and tubed heat exchanger was conducted as mentioned in [14]. Using the a empirical heat and mass transfer analogy called Colburn-j-factor in equation 1 and assuming fixed water flow rate throughout the cycle, the model is developed. It takes experimental data inputs like inlet relative humidity of air side, mass flow rates for both sides, and inlet temperatures for both air and water sides to determine the thermo-physical properties and calculate α_a in equation 2. For the water side, the Churchill equations represented in represented in equations 3 through 7 was used to determine α_w. The overall heat transfer coefficients of both fluids are combined in eq. 5 to determine the overall heat transfer coefficient of the system. These equations are found from empirical data from other systems and have been assumed to apply to our experimental system. The wet and dry coil heat transfer analysis is done using enthalpies and specific heats to determine a prediction for the outlet temperatures of both water and air side given air and water side inlet conditions like the air flow rate, water flow rate, air input relative humidity, air inlet temperature, and water inlet temperature for the heat exchanger. The rest of the equations used have been provided in the appendix below.

Nomenclature	
j	colburn-j-factor
α_a	overall heat transfer coefficient (air)
Re_D	Reynolds Number
$A_{a,total}$	Air side total area
A_{tube}	Tube Area
N_{bank}	Number of Banks
p_f	twice amplitude of fin corrugation
D	Diameter of duct
$alpha_w$	overall heat transfer coefficient (water)
ρ_{ha}	density humid air
u_{max}	max air velocity
$c_{p,a}$	specific heat air
ϵ	surface roughness
k	conductivity of water
Pr	Prandtl Number

$$j = 16.06 * Re_D^{-1.02*(p_f/D)-0.256}(A_{a,total}/A_{tube})^{-0.601}N_{bank}^{-0.069} * (p_f/D)^{0.84} \quad (1)$$

$$\alpha_a = j * \rho_{ha} * u_{max} * c_{p,a}/Pr^{2/3} \quad (2)$$

$$A = (-2.457 * log[(7/Re_D]^{0.9} + 0.27(\epsilon/D)])^{16} \quad (3)$$

$$B = [37530.0/Re_D]^{16} \quad (4)$$

$$f = 8[(8/Re_D)^{12} + 1/(A+B)^{1.5}]^{1/12} \quad (5)$$

$$\alpha_w = (k/D) * (f/8)(Re_D - 1000)Pr/(1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)) \quad (6)$$

This temperature predictions from this modeling effort is analyzed in the results sections and compared with the experimental data to assess the accuracy of the predictions.

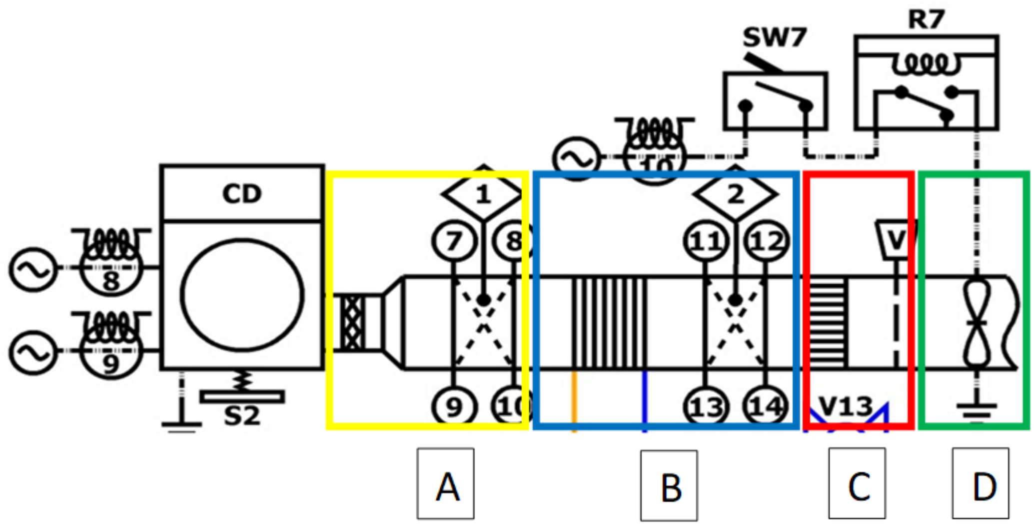

Figure 3: Dryer Duct Section Breakdown

2.3. Experimental Procedure

Before each run, the lint trap inside the dryer, the secondary lit trap, the heat exchanger, and the flow straightener were checked for lint. It was assumed that if no lint was present in the secondary lint trap, there would be no lint in the heat exchanger. Clothes were washed and dried at consistent settings between runs.The clothes washer was run at the quick wash setting with cold water and a high rinse speed. The dryer was operated for a fixed time of 60 minutes per cycle at the highest temperature setting and a mixed setting for the clothes type setting. The load sized were maintained at approximately 3.4 kilograms.

To begin collection of heat recovery data, three different types of test runs were conducted to begin obtaining data regarding heat exchanger performance. As indicated by Figure 3, the dryer duct was split into 4 different sections. Section A consists of the secondary lint trap and the first dryer duct section consisting of thermocouples and the relative humidity probe. Section B consists of the heat exchanger and the second duct section of thermocouples and relative humidity probe. Section C consists of the flow straightener and the Ebtron (flow meter). Lastly, section D consists of some duct work and the auxiliary fan. The first run was a baseline run. In this run, only sections A and C were attached to the dryer. This test run determines a

Table 4: Dryer Results from Test Runs

		Baseline		With HX no HR		With HX with HR	
Dryer Run Time	min	61	61	61	61	61	61
Weight of wet clothes	lbs	11.605	11.65	11.79	12.175	11.84	11.89
	kg	5.264	5.284	5.348	5.522	5.371	5.393
Total Energy per cycle	kWh	3.955	3.935	3.888	3.995	3.920	3.864
Potential Heat in Energy Stream	kWh	-	-	-	-	0.194	0.194
Energy Recoved in Hx	kWh	-	-	-	-	0.109	0.106
Effectiveness	-	-	-	-	-	0.560	0.550
MER	kWh/kg	0.7513	0.7447	0.7270	0.7234	0.7299	0.7165
Weight of dry clothes	lbs	7.595	7.65	7.575	7.54	7.66	7.54
	kg	3.445	3.470	3.436	3.420	3.475	3.420

desired flow rate for moist air that will be matched with the completed set up. The second run was a preliminary run for the complete system which included sections A, B, C, and D. However, no water side connections had been made. This test run varies flow rate of the moist air to be within 10% of the flow rate measured by the baseline run. The third set up test run is the same as the second test run except water side connections for the heat exchanger have been attached. The heat exchanger receives water from a hose and expels it to a drain. The flow rate for the water side was measured before the test run by filling a 4-gallon bucket and measuring the time it took to fill the bucket. This value was assumed constant throughout the dryer cycle.

3. Results and Discussion

The experimental results are presented in table 4. In this table, the weight of the clothes before washing, after washing, and after drying are provided. Over the 6 runs, these values were consistent. This table also provides the energy the dryer takes in to dry the clothes including the energy to operate the auxiliary fan when necessary. Over all runs the dryer required an input of approximately 4 kWh. The MER or the moisture extraction rate for the clothes, using equation 8, was found to be approximately 0.75 kWh/kg for the baseline runs and 0.7 kWh/kg for the other runs meaning less energy was required to cool the clothes when the heat exchanger was implemented into the dryer.

$$MER = EnergyPerCycle/MoistureRemoved \tag{7}$$

For the heat exchanger system with heat recovery, approximately 0.2 kWh of energy was potentially available in the waste stream. This was determined by calculating the energy required to cool the heat exchanger air side inlet state to the ambient state. The ambient state was assumed to be 25.56 C and have a relative humidity of 67 %. The finned-and-tube heat exchange was able to extract around 0.11 kWh of the energy from the waste stream resulting in a heat exchanger effectiveness of 0.55. This effectiveness was determined by dividing the measured recovered heat by the heat exchanger with the potential heat wasted (0.2 kWh). These calculated values were determined using equations 9 and 10.

$$Q = m_{dot,air} * (h - h_0) \tag{8}$$

$$Effectiveness = Q_{exhaustinlet}/Q_{Hx} \tag{9}$$

The flow rate results have been plotted in Figure 4. As previously mentioned, the objective of the auxiliary fan was to account for the loss in air stream flow rate observed as compared to the baseline run. The plots in Figure 3 indicate that the flow rate for all runs consisting of the heat exchanger were less than the baseline. The flow rate gradually decreases overtime due to lint fouling within the dryer system blocking air passage. The largest difference in flow rate is found in the baseline runs where flow rate decreases by approximately 25 meters cubed per hour. Key points to notice are that the runs with the heat exchanger attached operate at lower flow rates than that of the baseline run. The auxiliary fan was operating at its maximum speed yet the flow rates were smaller in magnitude than the baseline flow rates.

Plots of the inlet and outlet temperatures of the heat exchanger for both heat recovery runs are presented in Figure 5. From this plot, the drying process can be broken down into 4 different regions. The first region, indicated by the inlet temperature rising from the ambient temperature to 50 C, is referred to as the warm-up region. The next region is the drying region where the clothes are being dried. The reason for this naming is that the moisture in the clothes cools down the air stream coming into the drum by absorbing heat to dry the clothes. The temperatures in this region are much lower than peak temperatures due to this drying process occurring. A rise in the dryer exhaust temperature or exhaust inlet temperature signals the clothes drying

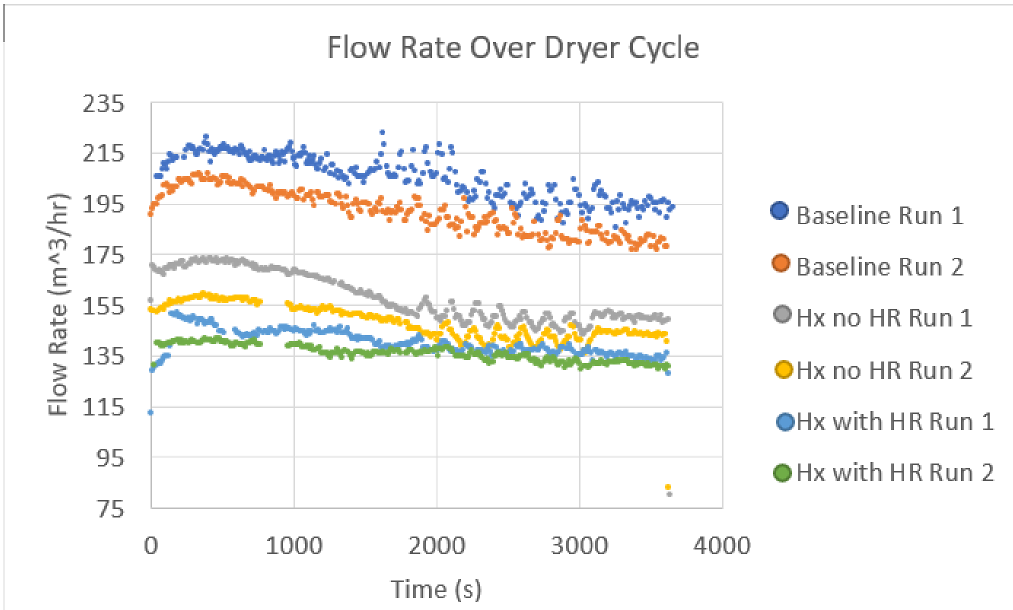

Figure 4: Flow rates of moist air stream in dryer duct

process reaching completion. However, the selected dryer settings force the dryer to run for 1 hour. Therefore, the heating element within the dryer periodically switches between being on and off indicated by the region of the plot where the inlet and outlet temperatures are oscillating. The final region is where the temperature of the dryer exhaust begins to cool down as the cycle approaches its cooling region. In the drying region, the temperature difference between air side inlet and outlet is 20 C whereas it is approximately 30 C in the oscillating region.
The water side inlet and outlet temperatures are presented in Figure 6 for both runs conducting heat recovery. The water side inlet temperature for both runs is approximately 20 C throughout the cycle. The outlet temperature for the water follows the 4 regions mentioned in the air side temperature analysis. However, the outlet temperature reaches only about 23 C leading to a change in temperature of only 3 C or to 3.5 C.

The plots in Figure 7 plot the air side and water side rates of hate transfer measured for heat recovery run 1. Regions within this plot can be seen where the water side rates of heat transfer measured are larger than the rate of heat

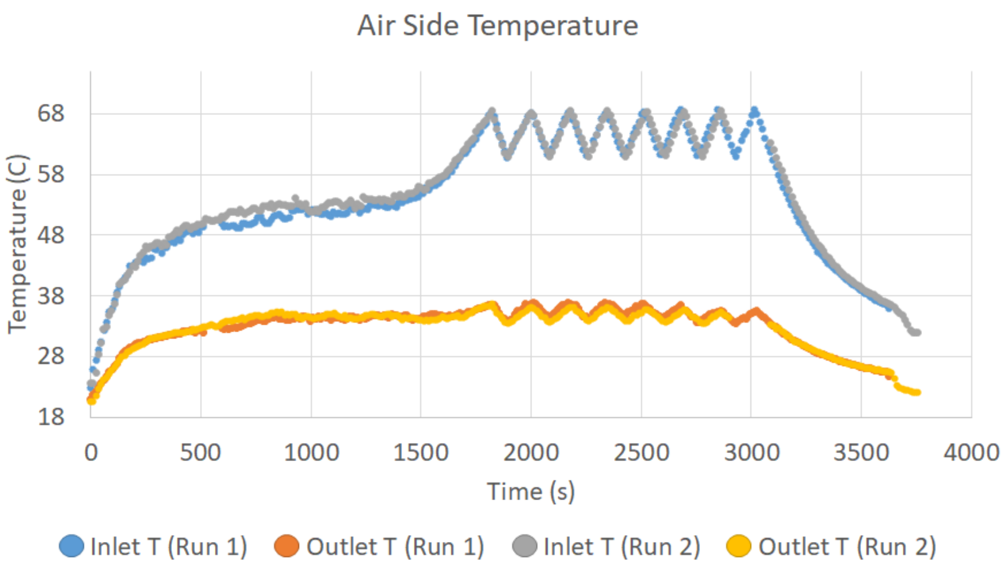

Figure 5: Air Side Inlet and Outlet Temperatures of Heat Exchanger

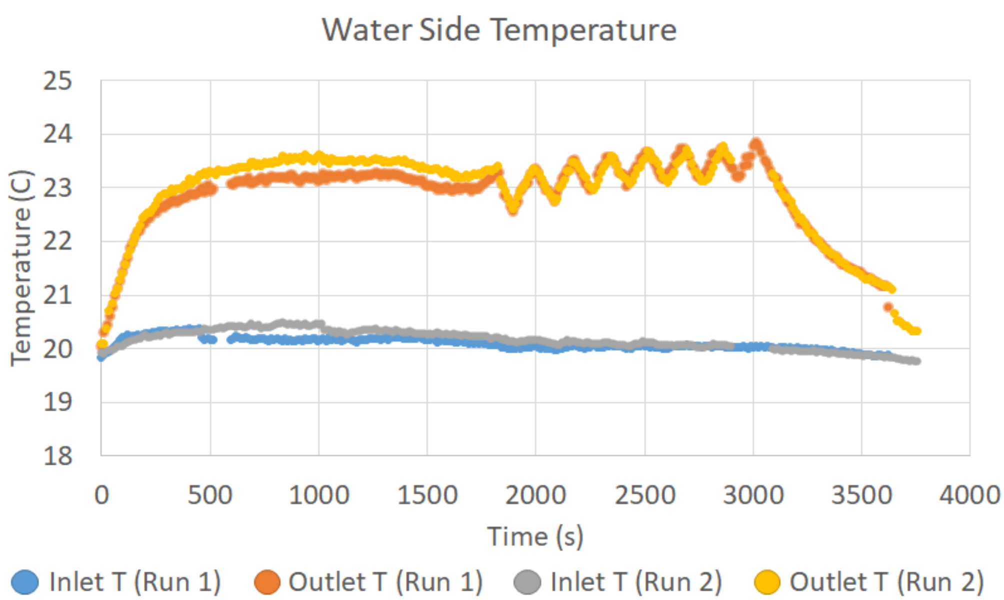

Figure 6: Water Side Inlet and Outlet Temperatures of Heat Exchanger

Table 5: Comparison Between Experimental and Model Values

	time (s)	Air out T (degrees C)	Wet Coil Air out T (degrees C)	Dry Coil Air out T (degrees C)	Water out T (degrees C)	Wet Coil Water Out T (degrees C)	Dry Coil Water out T (degrees C)	Q wet coil (W)	Q dry coil (W)
	322	31.15475	21.03	20.92	22.763	26.53	22.64	3039	1133
	476	31.9975	20.9	20.8	22.911	26.58	22.75	3146	1271
Run 1	1247	34.32575	20.99	20.89	23.23	27.04	23.09	3351	1417
	2489	35.849	20.74	20.81	23.569	24.29	23.78	2068	1818
	2800	34.69525	20.7	20.78	23.44	23.73	23.64	1799	1756
	1524	33.98	20.98	20.92	23.332	26.29	23.13	1460	3074
	1894	33.4245	20.75	20.83	22.605	23.34	23.36	1646	1634
Run 2	2025	35.54025	20.84	20.9	23.161	24.33	23.73	1824	2129
	2121	34.478	20.74	20.8	23.044	23.91	23.43	1695	1937
	2336	35.74725	20.8	20.85	23.489	24.6	23.71	1834	2287

transfer measured on the air side. A possible explanation for this is that the fins have or the water tubes have absorbed the heat from the air stream and this heat is not immediately absorbed by the water. This absorbed energy may be recovered by water when less heat was recovered from the air stream leading to a larger rate of heat transfer on the water side.

The generated model is compared to the experimental data at select input temperatures. These select points are chosen when the air side and water side rates of heat transfer have small magnitudes of differences (within 5 % of air side heat transfer rate). This was done to compare the steady state model with steady state experimental points. The values chosen for comparison are indicated in Table 6. The dry coil model and wet coil model predicted air and water side outlet temperatures are then compared to the experimental data in Figure 8. A 45-degree linear line is plotted to that indicates points that would have perfect correlation between the model and the experimental data. The wet coil and dry coil model significantly under predicted the air side outlet temperatures as seen on Figure 8. The wet coil model predicted water outlet temperatures close to experimentally found values for both runs with some points being over predicted . However, the water side outlet temperatures predicted by the dry coil model for both runs are almost all on the or close to 45-degree line indicating that the dry coil model predicted water side outlet temperatures within a small percentage of error of experimental values.

Figure 8 overlays the dry and wet coil model heat transfer rates (Q) at the

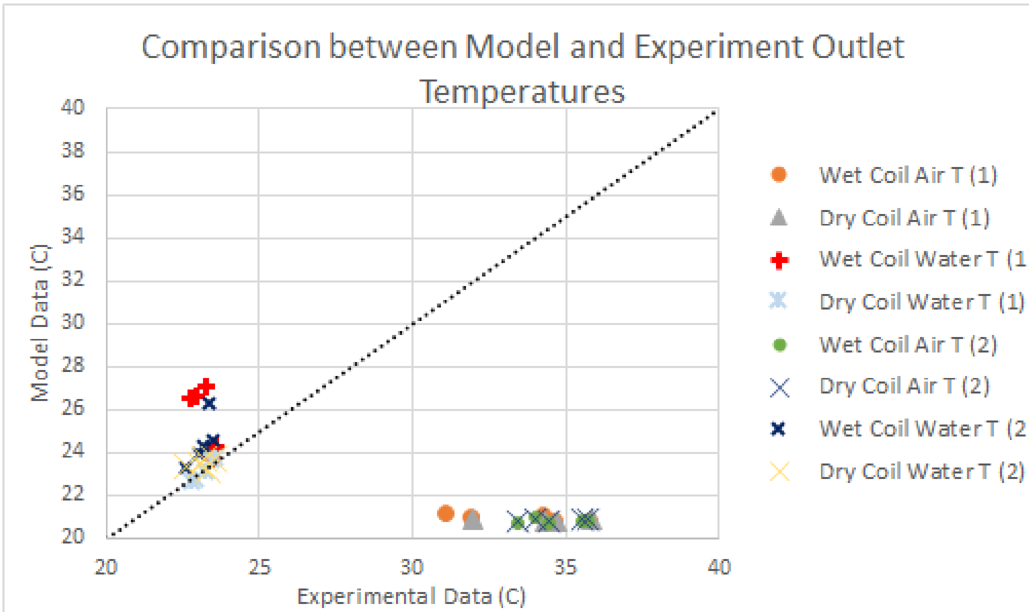

Figure 7: Comparison Between Specific Experimental Values and Corresponding Model Outputs

specific points compared earlier on a plot with heat transfer rates found from the inlet and outlet temperatures of the air side of the heat exchanger for both runs. The dry coil heat transfer predictions for run 1 are approximately equal to the experimental value of that point as seen by these points being amidst the experimentally found heat transfer rate data points. For run 2, the dry and wet coil Qs are almost twice that of the experimental value, sometimes even 3 times as large as well as seen by the first triangle on Figure 8. The wet coil model is able to predict heat transfer rates near experiential values in the later portion of the dryer cycle. However, in the earlier portion of the cycle, the wet coil model over predicts the experimental Q by almost three times.

4. Conclusion

This study considered a conventional residential clothes dryer as a heat source and aimed to extract heat from the heat moist air stream exiting the dryer. The approach utilized a fin and tube heat exchanger with hot moist air on the air side and cool water on the water side. The changes in temperatures

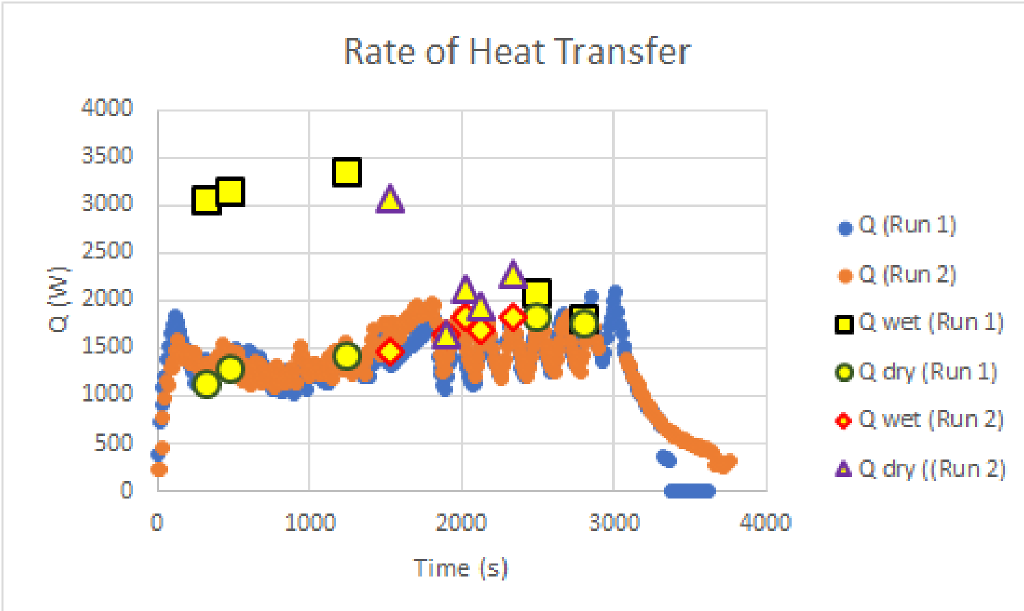

Figure 8: Model Heat Transfer Rate Compared with Experimental Data Heat Transfer Rates

on the air and water side indicated that about 0.1 kWh of heat was recovered. Comparing this to the incoming air stream having potentially 0.2 kWh, it was found that the heat exchanger was 55% percent effective in extracting heat from the hot moist air released by the dryer (waste stream). These numbers were consistent over two test runs. The consistency in the data is also seen in the temperatures represented in Figures 5 and 6, and the rates of heat transfer represented in Figure 8. The two runs have similar inputs reported in Table 3 and yield similar outputs.

A model was generated representing the heat exchanger part of the dryer system to be able to predict outlet temperatures. The model was compared with the experimental data at specific steady state points were air side and water side heat transfer rates for the heat exchanger were almost equal. It was found that both dry coil and wet coil temperature predictions for the air side were significantly larger than experimental values while water side predictions were almost equal to experimental values.

4.1. Future Work

It is evident from Figure 4 that the auxiliary fan is unable to produce air flow rates close to the baseline tests. A fan with a larger capacity is needed to replace it. As previously mentioned, the flowrate of the water side is measured before the run. A flow meter is necessary on the water side to measure fluctuations in water side flow rate. These fluctuations can then be implemented into the model and calculations.

The model currently is a steady state model while the drying cycle is a transient process. The code should be modified to take consider this along with a partially wet and partially dry coil for the heat exchanger. Empirical relations like the Colburn-j-factor and the Churchill correlations were assumed to fit the heat exchanger system presented even though they are specific to certain scenarios and based off experimental data. It would be beneficial to determine empirically how these relations would fit into the presented waste heat recovery set-up and modify the model.

A method needs to be determined to measure how much of the energy that is coming in to the heat exchanger from the air side is being stored within heat exchanger components and how to account for this in the model.

5. Acknowledgments

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6. References

[1] U.S. Energy Information Administration. (2018). U.S. Energy Information Administration - EIA - Independent Statistics and Analysis. Retrieved from https://www.eia.gov/tools/faqs/faq.php?id=96&t=3

[2] U.S. households with clothes dryer, by energy source — Statistics. (n.d.). Retrieved from https://www.statista.com/statistics/220466/household-penetration-of-clothes-dryers-by-energy-source-in-the-us/

[3] Bansal, P., Islam, S., & Sharma, K. (2010). A novel design of a household clothes tumbler dryer. Applied Thermal Engineering, 30(4), 277-285. doi:10.1016/j.applthermaleng.2009.09.002

[4] Zhao, J., Jian, Q., Zhang, N., Luo, L., Huang, B., & Cao, S. (2018). The improvement on drying performance and energy efficiency of a tumbler clothes dryer with a novel electric heating element. Applied Thermal Engineering, 128, 531-538.

[5] Bassily, Ashraf Maurice, "Modeling and optimization of heating and drying processes in a clothes dryer " (2000). Retrospective Theses and Dissertations. Paper 470.

[6] Huelsz, G., Urbiola-Soto, L., Lpez-Alquicira, F., Rechtman, R., & Hernndez-Cruz, G. (2013). Total Energy Balance Method for Venting Electric Clothes Dryers. Drying Technology, 31(5), 576-586. doi:10.1080/07373937.2012.746977

[7] Tegrotenhuis, W., Butterfield, A., Caldwell, D., Crook, A., & Winkelman, A. (2017). Modeling and design of a high efficiency hybrid heat pump clothes dryer. Applied Thermal Engineering, 124, 170-177. doi:10.1016/j.applthermaleng.2017.05.048

[8] Han, H., & Deng, S. M. (2003). A Study on Residential Clothes Drying Using Waste Heat Rejected from a Split-Type Room Air Conditioner (RAC). Drying Technology, 21(8), 1471-1490. doi:10.1081/drt-120024488

[9] Jian, Q., & Luo, L. (2018). The improvement on efficiency and drying performance of a domestic venting tumble clothes dryer by using a heat pipe heat recovery heat exchanger. Applied Thermal Engineering, 136, 560-567. doi:10.1016/j.applthermaleng.2018.03.029

[10] Tomlinson, J. J., Christian, J., & Gehl, A. C. (2012). Evaluation of Waste Heat Recovery and Utilization from Residential Appliances and Fixtures. Oak Ridge National Laboratory. doi:10.2172/1055065

[11] Braun, J., Bansal, P., & Groll, E. (2002). Energy efficiency analysis of air cycle heat pump dryers. International Journal of Refrigeration, 25(7), 954-965. doi:10.1016/s0140-7007(01)00097-4

[12] Jian, Q., & Zhao, J. (2017). Drying performance analysis of a condensing tumbler clothes dryer with a unique water cooled heat exchanger. Applied Thermal Engineering, 113, 601-608. doi:10.1016/j.applthermaleng.2016.11.086

[13] VersaMat. (n.d.). Retrieved from http://www.indratech.com/versamat

[14] Heat Exchangers with and without Dehumidification. (n.d.). Retrieved from http://achp.sourceforge.net/ACHPComponents/ WetDryHeatExchanger.html

[15] F-Chart Software, E. (2018). F-Chart Software : Engineering Software. Retrieved from http://www.fchart.com/ees/

[16] National Instruments. (2018). What is LabVIEW? - National Instruments. Retrieved from http://www.ni.com/en-us/shop/labview.html