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An Experiential Learning Exercise: Optimization of Evaporators and Condensers in a Vapor Compression Cycle

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Introduction

One way to engage students in the learning process is to provide them with a practical, but significant challenge. Such a challenge for first semester thermodynamic students was to optimize a refrigeration cycle by minimizing total life-cycle cost as contained in [1] and [2]. Another effective way to improve student learning was to add an experiential element to the process. A vapor compression apparatus, see Figure 1, was used to provide this experiential element for the class through a demonstration.

Data from the apparatus were collected for two purposes. First, a heat balance between the refrigerant and air sides of the heat exchangers was performed. Students were led deductively from the heat transfer rate in the conservation of energy to the measurements required to calculate the rate. Poor agreement between the refrigerant-side and air-side measurements showed the instruments needed to be calibrated or replaced.

Second, the overall heat transfer coefficient (U) for the selected operating condition was determined. The data from the experiment was fed into a computer analysis program developed by an upperclassman as part of an independent study project. The program calculated the fin efficiency to adjust the area of the fins to an equivalent area. The overall heat transfer coefficient was a key input into the system model which the thermodynamic students then used to perform the optimization.

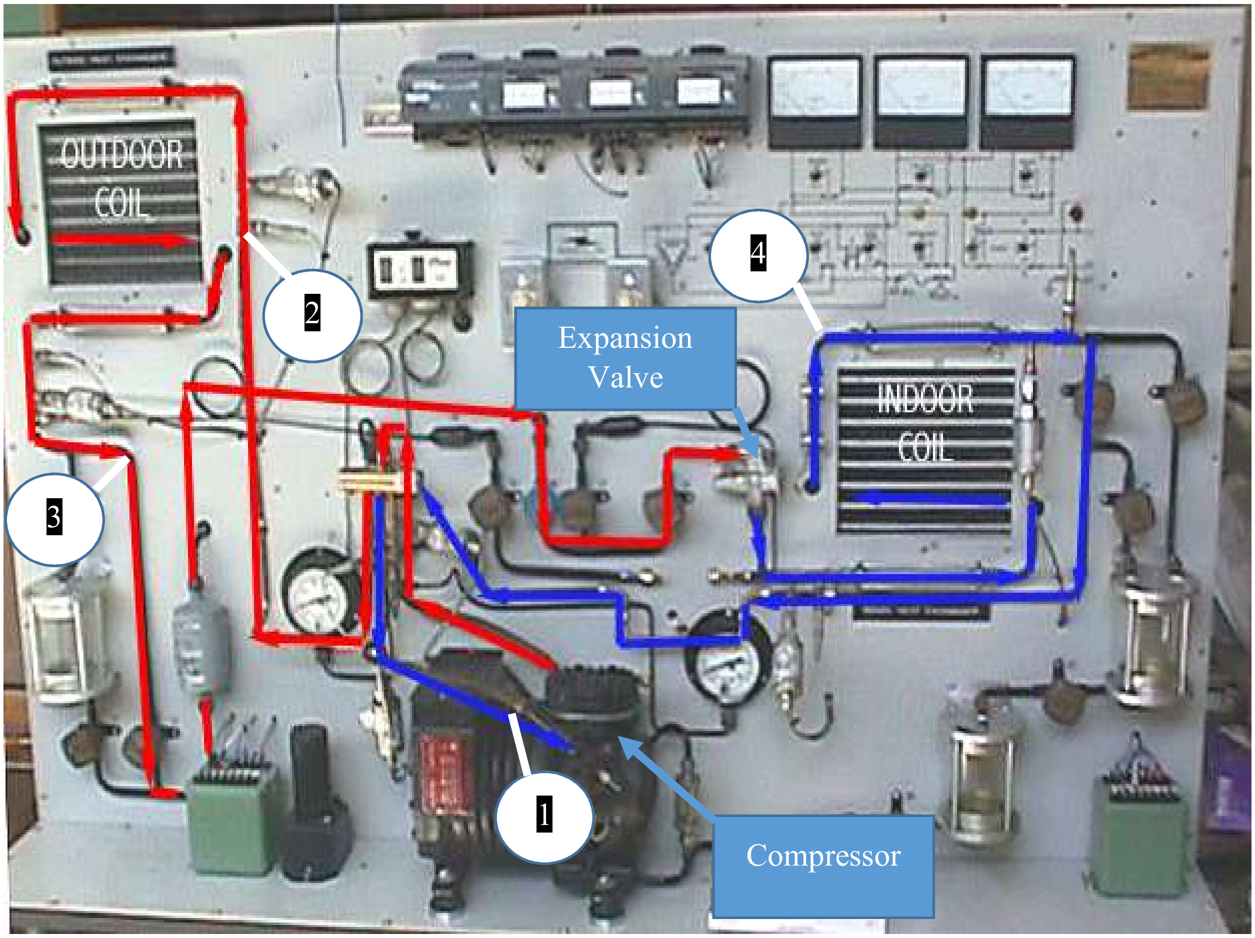

Figure 1. Vapor Compression Apparatus (Air Conditioning Mode)

Theory

The goal of the system model, contained in [1] and [2], was to relate total cost to design variables. In a series of two video lectures, totaling 100 minutes, viewed before class, the students learn how to develop the system model. As they work their way through the model, the students learn the connections between thermodynamics and heat transfer and the costs of an air conditioning system.

To set up the system model some key equations will be presented. In addition to traditional thermodynamics, equations from engineering economics, and heat transfer are essential to optimize the system. The equations provided are only a small subset of all the equations listed in [1] and [2] which are necessary to complete the system model.

$$IC_{Compressor} = \frac{ccc*\dot{W}}{(1-\eta)^e} \tag{1}$$

In Equation 1, the initial cost of the compressor is determined by regression analysis from supplier data. It is a function of the size of the compressor and its efficiency. The multiplier (*ccc*) and exponent (e) are empirical parameters determined from regression analysis. The next equation from engineering economics will convert and annual distribution of future dollars to present dollars.

$$PA = \frac{(1+i)^{n-1}}{(i(1+i)^n)} \tag{2}$$

The PA factor is a function of the life of the system (n) in years and the minimum acceptable rate of return (i). The next equation uses the conservation of energy around the evaporator to calculate the mass flow rate of the refrigerant.

$$\dot{m}_r * h_4 + \dot{Q}_{er} = \dot{m}_r * h_1 \tag{3}$$

In Equation 3, $\dot{m}_r$ is the mass flow rate of the refrigerant through the system. Next, h_4 is the enthalpy of refrigerant at the entrance of the evaporator. Lastly, the enthalpy at the evaporator outlet is h_1. All of these state points can be seen in the picture of the apparatus, and the P-h diagram located in Figures 1 and 2, respectively.

The last key equation and the most important comes from heat transfer. Equation 4 calculates the heat transfer rate through the evaporator using the conductance form the heat transfer equation. This equation is vital to connecting design variable, area, to the system performance.

$$\dot{Q}_e = U_e * A_e * (T_{aei} - T_{sate}) \tag{4}$$

U_e represents the overall heat transfer coefficient of the evaporator, and A_e is the area of the heat exchanger. T_{aei} is the temperature of the air at the evaporator inlet, and T_{sate} is the saturation temperature of the refrigerant in the evaporator. Although the effectivess-NTU relationship will be more accurate this simple form allows the student to see the connection between area and system performance.

The heat transfer rate on the air-side was crucial for validation of instrument accuracy. Having a heat balance between the air-side and refrigerant-side outside 10% would demonstrate that the sensors on the apparatus were inaccurate and need to be calibrated or replaced. The equations were used to solve for the heat balance:

$$\dot{m}_{ac} * h_6 + \dot{Q}_{ca} = \dot{m}_{ac} * h_7 \tag{5}$$

Above in equation 5, the heat transfer rate through the air-side of the condenser was calculated using the conservation of energy equation. From Equation 5 additional equations were needed for mass flow rate of the air through the condenser [$\dot{m}_{ac}$] and the enthalpy at state points 6 (condenser inlet) and 7 (condenser outlet). These equations can be seen below.

$$\dot{m}_{ac} = \rho * \dot{V}_c \tag{6}$$

From Equation 6, which comes from the definition of density, the density and the volumetric flow rate of air through the condenser needed additional equations as shown in equations 7 and 8.

$$\rho = \mathbf{density}(\boldsymbol{Air_{ha}}, \boldsymbol{T} = \mathrm{T}_{aci}, \boldsymbol{P} = \mathrm{P}_{aci}) \tag{7}$$

$$\dot{V}_c = V_c * A_{inlet} \tag{8}$$

After measuring the temperature and pressure at the condenser inlet, density was calculated using a property relationship. The inside (fan motor) and outside diameter (inside of fan casing) of the fan inlet were measured to calculate the flow area of the inlet shown below.

$$A_{inlet} = A_o - A_i \tag{9}$$

$$A_o = \pi * (\frac{D_{ci_o}}{2})^2 \tag{10}$$

$$A_i = \pi * (\frac{D_{ci_i}}{2})^2 \tag{11}$$

$$h_6 = \mathbf{Enthalpy}(\boldsymbol{Air_{ha}}, \boldsymbol{T} = \mathrm{T}_{aci}, \boldsymbol{P} = \mathrm{P}_{aci}) \tag{12}$$

$$h_7 = \mathbf{Enthalpy}(\boldsymbol{Air_{ha}}, \boldsymbol{T} = \mathrm{T}_{aco}, \boldsymbol{P} = \mathrm{P}_{aco}) \tag{13}$$

The equations shown above were necessary to solve for the heat transfer rate through the air-side measurements of the condenser. Similar equations were used for the refrigerant-side calculations. In these equations the properties were for R134a rather than air and the volumetric flow rate was measured rather than velocity.

This example of equation development, which follows the deductive problem solving strategy contained in [1], helped the student discover what measurements were needed. The deductive problem solving strategy also kept the equations in their simplest form so they were easy to comprehend. Once the system of equations was solved, the value of heat transfer rate on the air-side was compared to that of the refrigerant-side as shown in the Equation 14 below.

$$HB_{e,percent_{diff}} = \frac{(\dot{Q}_{ea} - \dot{Q}_{er}) * 100}{\dot{Q}_{er}} \tag{14}$$

Procedure

Before the apparatus was operated for the class, data needed to be collected to calculate the overall heat transfer coefficients for both heat exchangers. Advanced knowledge of heat transfer was required to determine these coefficients. This includes convective heat transfer coefficient correlations and fin efficiency. This calculation was beyond the scope of first semester thermodynamic students. Therefore, an upperclassman performed this analysis for the students.

The apparatus contained sensors to record refrigerant-side data including the temperature and pressure at each state point of the energy cycle, see Figures 1 and 2, along with the volumetric flow rate of refrigerant. During the experiment, the pressures and temperatures were observed in real time on a graph to insure they were at steady-state conditions. Reaching steady state took approximately 8-10 minutes. After steady-state was achieved, the data were recorded and then exported into a spreadsheet for the students to perform the heat balance analysis.

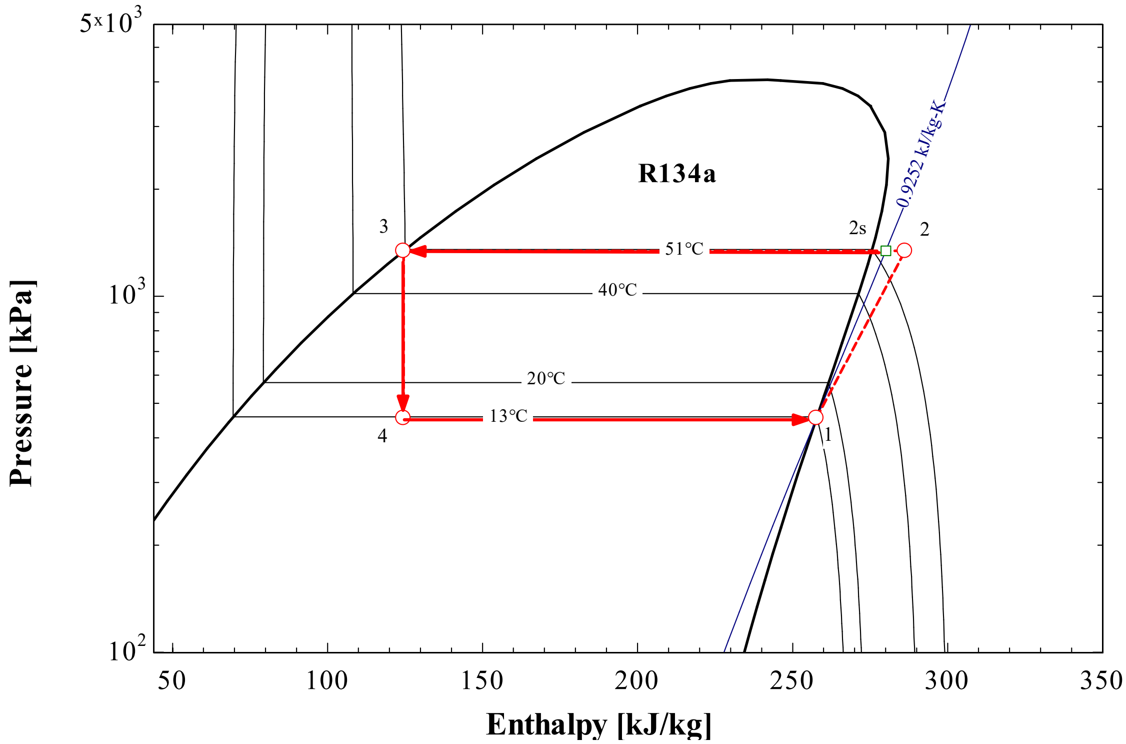

Figure 2: Property Plot of Pressure vs Enthalpy-Optimum Refrigeration Cycle.

During the demonstration of the refrigeration apparatus the students needed to fill out the assignment sheet located in Appendix A. In the assignment sheet, there is a list of all the variables to be measured. The air velocity at the condenser and evaporator inlet and outlet needed to be measured along with the temperatures using portable sensors. To properly measure these variables, there needed to be multiple data points across the circular entrance to the fan and the rectangular exit of the heat exchanger. Once numerous (at least 15) data points were taken, the average was determined to obtain a more representative value. Additionally, the relative humidity of the air at the heat exchanger inlet and outlets was recorded throughout the duration of steady-state operation, and an average was used.

Results

The apparatus was operated until steady-state conditions were reached as shown in Figure 3.

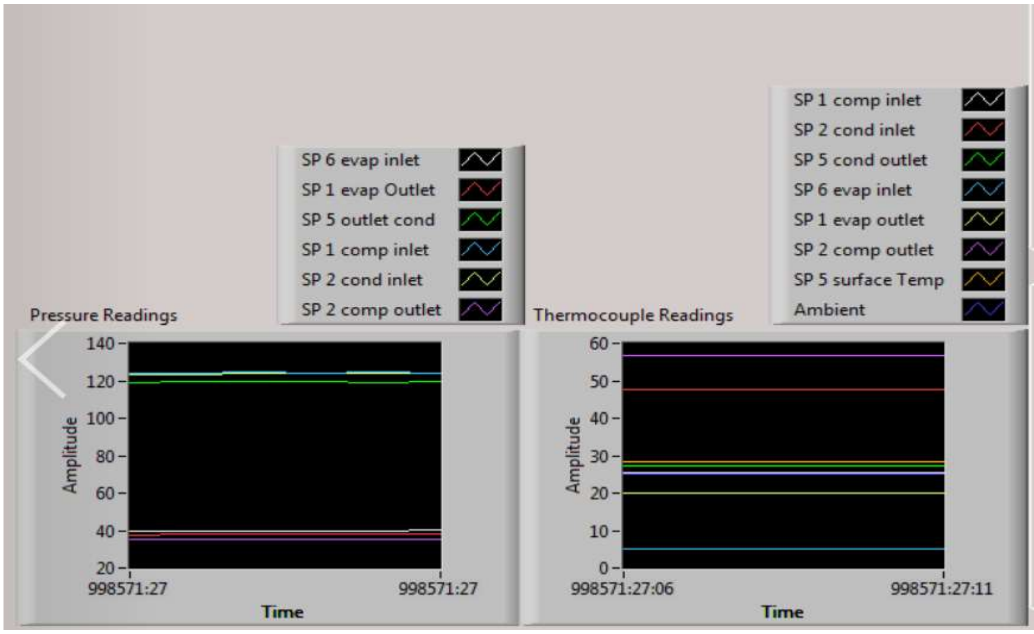

Figure 3: Temperatures and Pressures as a Function of Time

Using the Equations 5 through 14 and the data collected during the fall of 2017, the heat balance for the condenser was 17.4 %. Since the heat balance was off by more than 10% the instrumentation needs calibration or more temperature data was needed at the outlet. Possible sources of error could result from the propagation of uncertainty in the instrumentation. The uncertainty of the thermocouples being +/- 1 [C]. Also, the relative humidity sensor being +/- 3[%]. It was noticed during the experiment, that the measurement of the temperature out of the condenser and the evaporator varied at different locations. To find a representative value multiple data points were needed.

The students were given the values for the overall heat transfer coefficient to perform the optimization. These values are contained in Table 1. To complete the optimization, the students will vary the size of the evaporator and condenser until the total life-cycle cost of the system is at a minimum as shown in Figure 4. The overall heat transfer coefficient was 0.020 [kW/m^2-C] for the condenser and 0.0309 [kW/m^2-C] for the evaporator. These values depend on the fin efficiencies of the condenser or the evaporator. The values calculated from the analysis are summarized below in Table 1.

Table 1. Calculated and measured values that were collected for analysis

Calculated Variables	**Results**	**Calculated Variables**	**Results**
Heat transfer rate through condenser refrigerant side (**Q_dot_cr**)	0.5217 [KW]	Heat transfer rate through condenser air side (**Q_dot_ca**)	.4308 [KW]
Heat transfer rate through evaporator refrigerant side (**Q_dot_er**)	0.4909 [KW]	Heat transfer rate through evaporator air side (**Q_dot_ea**)	0.3503 [KW]
Heat Transfer Coefficient (**U_e**)	0.0309 [kW/m^2-C]	Heat Transfer Coefficient (U_c)	0.020 [kW/m^2-C]
Fin Efficiency Evaporator	0.895 [-]	Fin Efficiency Condenser	.878 [-]
Measured Variables	**Results**	**Measured Variables**	**Results**
Fan Speed →	Low	Temperature of Air at the Evaporator and Condenser inlet	23.3[C]
Temperature of Air at Evaporator Outlet	14.9 [C]	Temperature of Air at Condenser Outlet	37.0 [C]
Relative Humidity of Air at Evaporator Inlet	25 [%]	Atmospheric pressure	102.6 [kPa]
Air Velocity at evaporator inlet	2.65[m/s]	Air velocity at condenser inlet	3.3[m/s]
Pressure of refrigerant at evaporator exit	303[kPa]	Pressure of refrigerant at condenser inlet	1067 [kPa]
Temperature of refrigerant at evaporator exit	23.9 [C]	Pressure of refrigerant at condenser outlet	1025 [kPa]
Volumetric flow rate of refrigerant at condenser outlet	2.4×10^{-6} [m^3/s]	Temperature of refrigerant at condenser inlet	51 [C]
Relative Humidity at Evaporator Outlet	34.9 [%]	Temperature of refrigerant at condenser outlet	33 [C]
Diameter of fan motor (located within the opening of the housing)	0.09525 [m]	Diameter of opening in fan housing	0.143 [m]

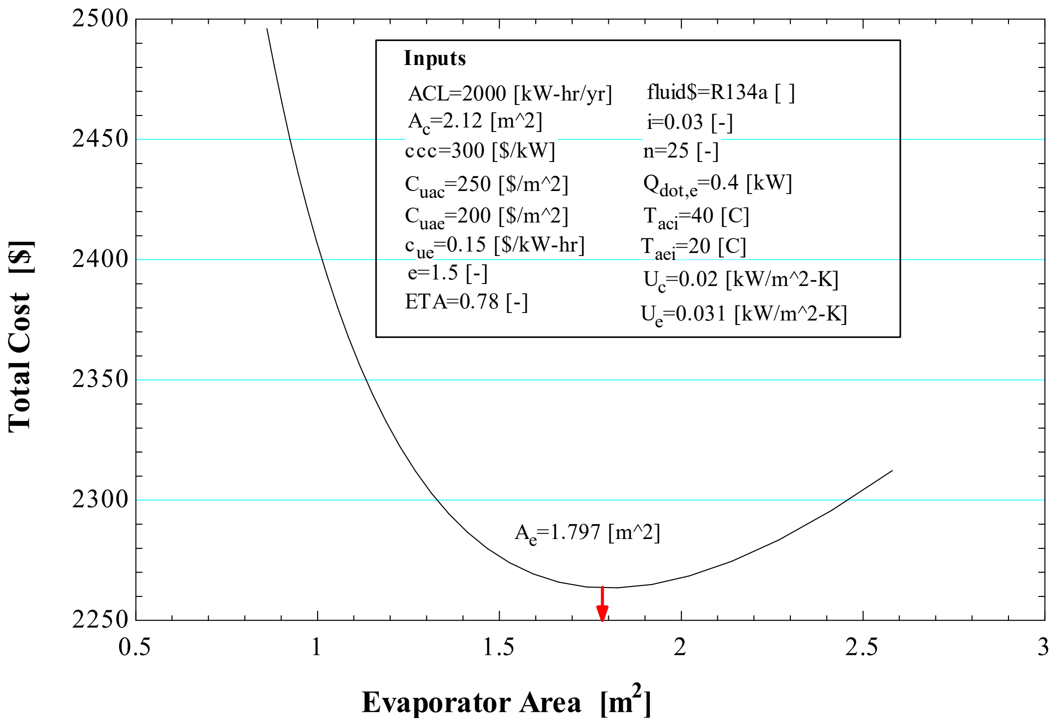

Figure 4: Total Cost of the System Compared to the Size of the Evaporator.

The optimum size of the evaporator was 1.797 square meters (See Figure 4). This area was where the total life cycle costs are at a minimum assuming the conditions listed in the box labeled “Inputs”. To understand why the costs increase on either side of this optimum the students plotted the initial cost of the system and the present worth of the operating costs as a function of the area as shown in Figure 5. To understand why the operating costs decreased as the area increased requires further exploration as shown in Figure 6. Although the relationship between the initial cost and the area of the evaporator was straight forward, the non-linearity in the curve needs to be understood.

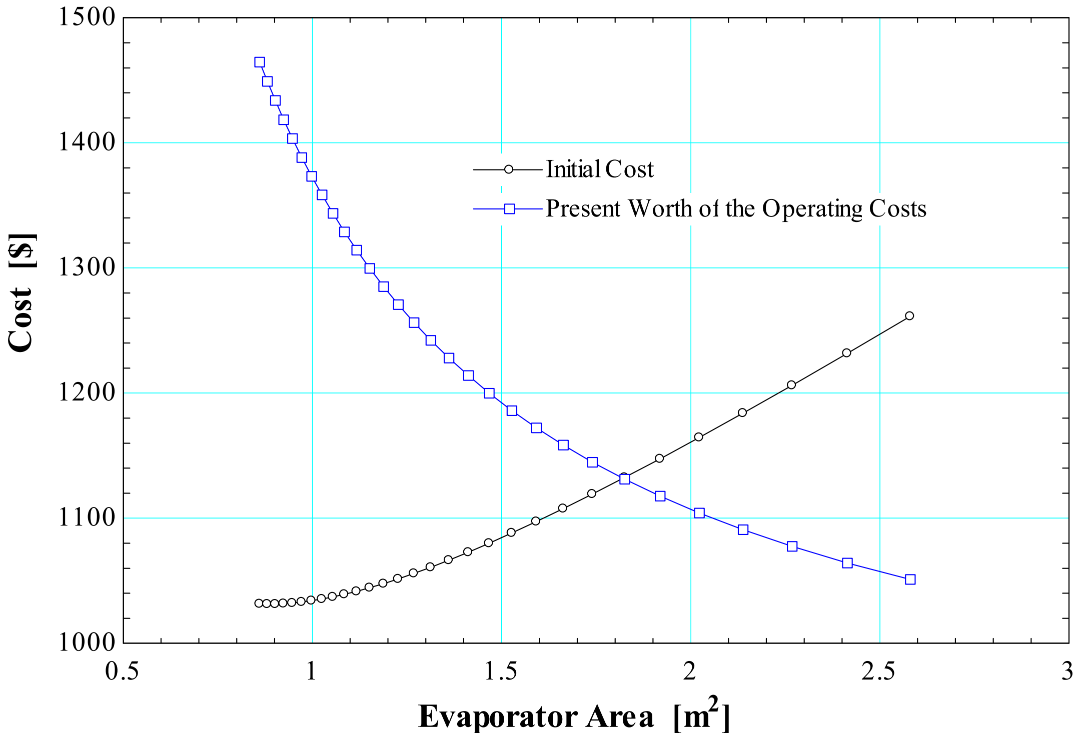

Figure 5: Initial and operating cost relationship to the area of the evaporator.

In Figure 6, as the area increased the temperature difference between the air and the refrigerant decreased. This relationship was provided in the conductance form of the heat exchanger as given in Equation 4. For a given application, the rate of heat transfer through the evaporator was fixed along with the temperature of the air entering the evaporator. Fixing the overall heat transfer coefficient provided insight into how the area of the evaporator and the saturation temperature vary. As the area increased, the saturation temperature approaches the inlet air temperature. The reduced temperature difference lowered the irrevesibilities of the heat transfer which in turn increaseed the COP.

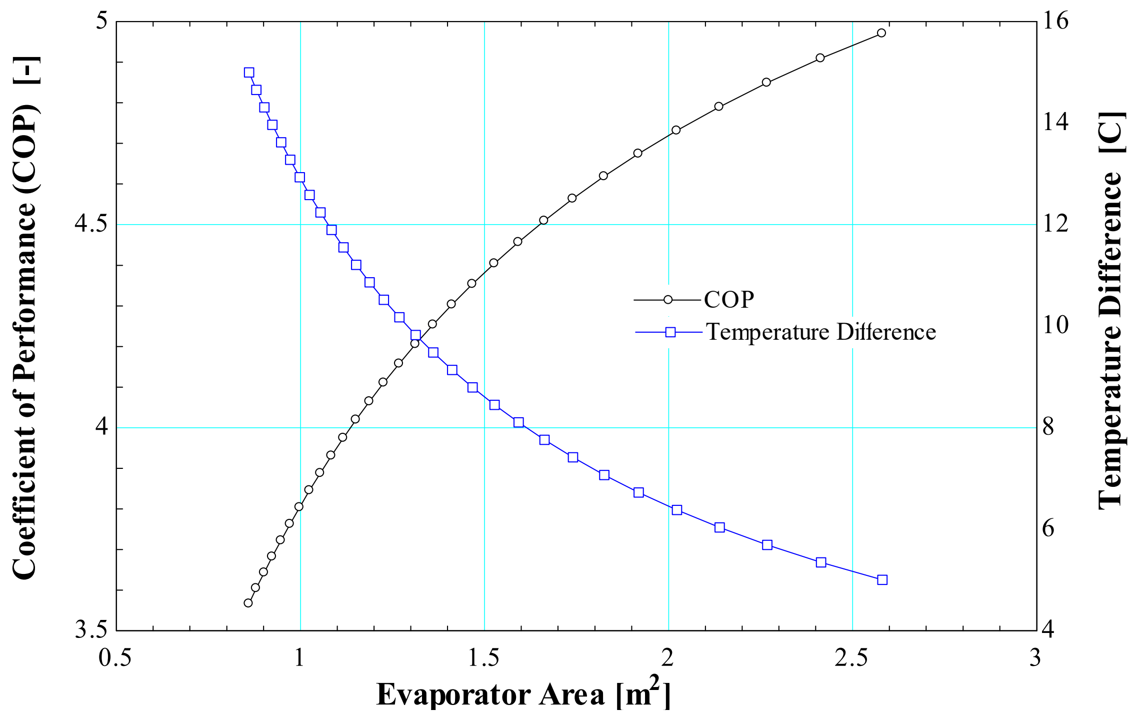

Figure 6: Relationship of coefficient of performance along with temperature difference against the size of the evaporator.

Although the initial cost of the evaporator was modeled in the system model as a linear relationship, it was clearly non-linear in Figure 5 especially at smaller areas. As the area decreased so did the saturation temperature in the evaporator. This lower temperature also lowered the pressure. As the pressure dropped, the work of the compressor increased therefore increasing its initial cost. The condenser pressure was held constant so the area of the condenser needed to increase to remove the extra work from the compressor. Therefore, the initial cost of the condenser was increasing as well. These relationships are shown in Figure 7.

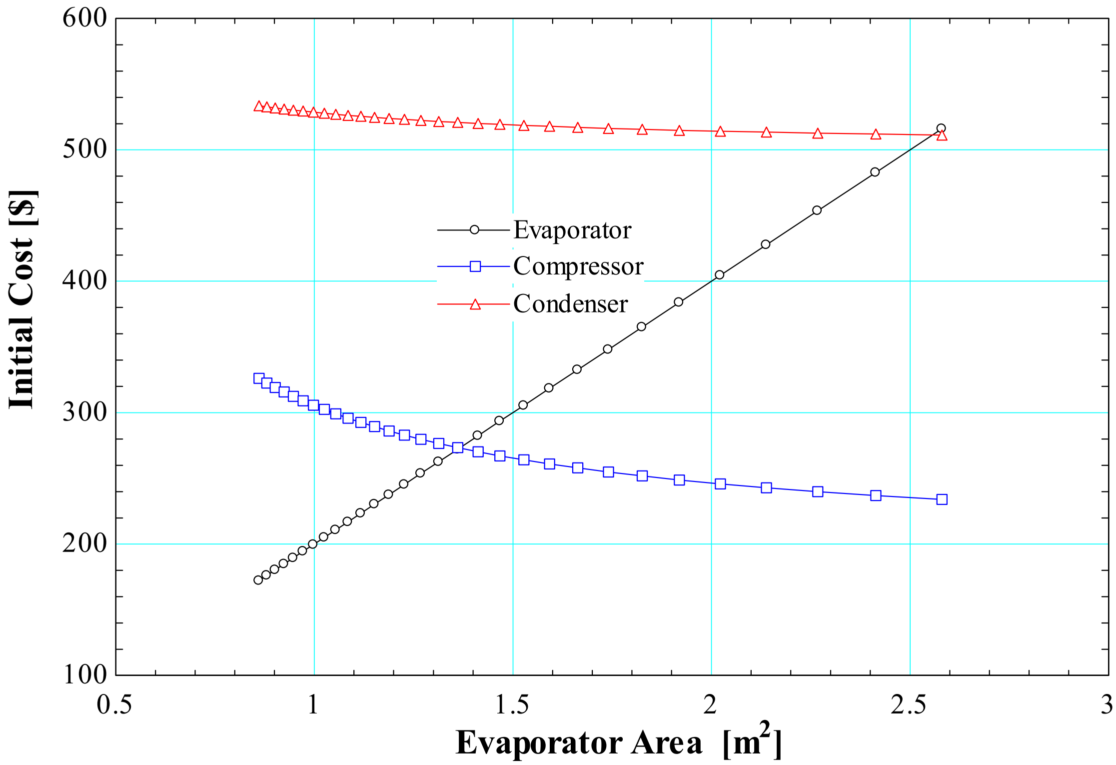

Figure 7: Initial Cost as a Function of the Size of the Evaporator.

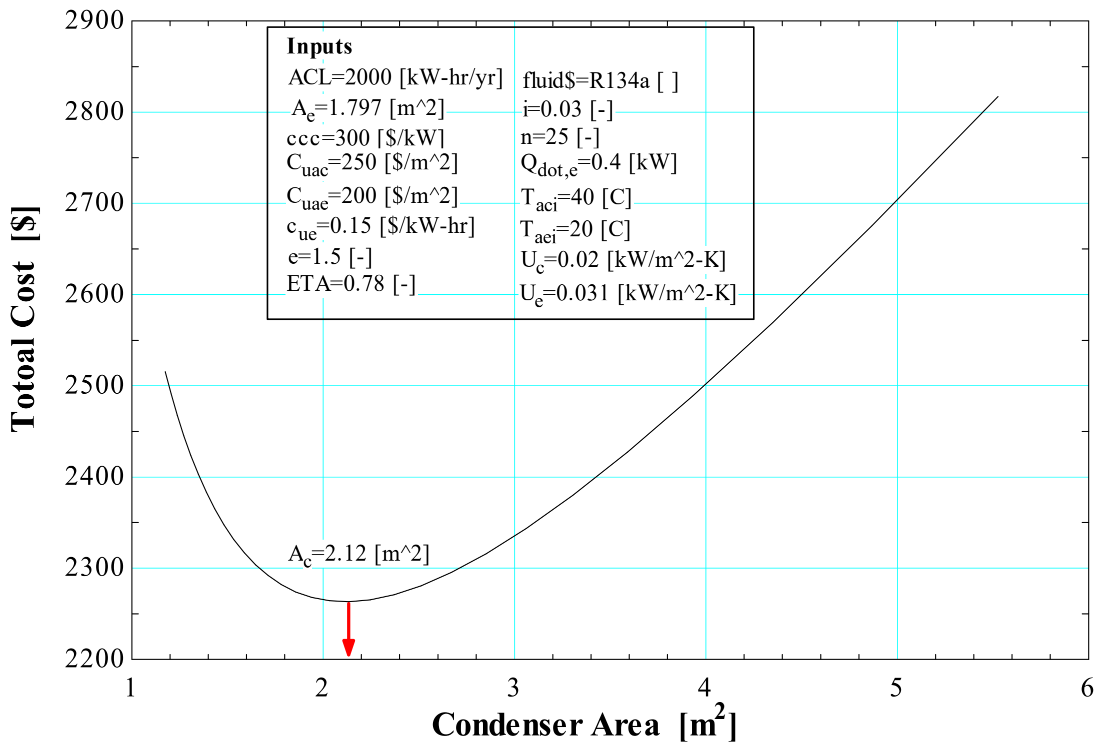

Figure 8: Total Cost as a Function of the Size of the Condenser.

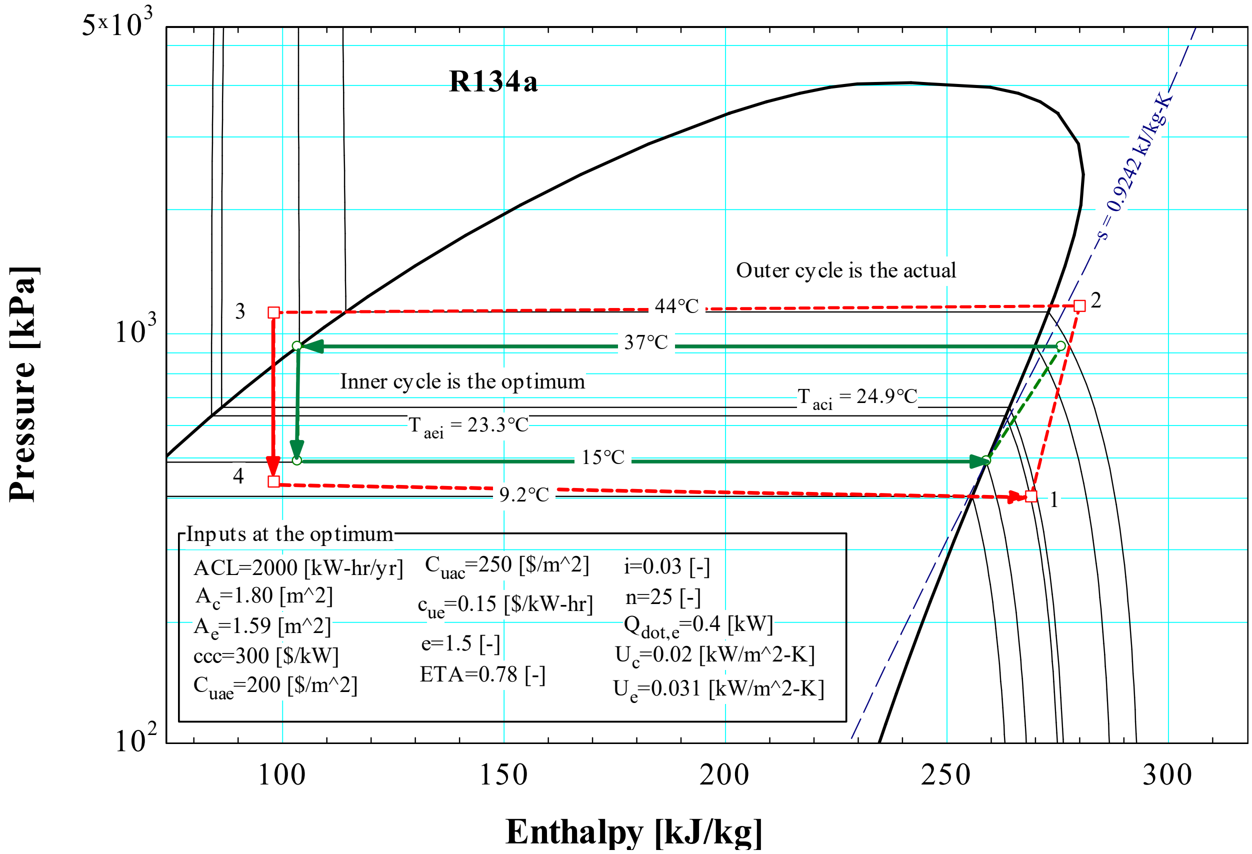

Figure 9. Comparison of Optimum System with Actual System.

Data from the apparatus were plotted on a property plot along with the optimum under the same operating conditions. The only difference in operating conditions between this plot and the previous plot are the inlet air temperatures to the evaporator and condenser. The effective area of both the evaporator and condenser on the apparatus is 1.1 [m^2]. The optimum areas were 45% and 63% higher than the areas for the evaporator and condenser, respectively which were installed on the apparatus. The larger evaporator reduced the temperature difference from 14.1 [˚C] down to 8.3 [˚C]. The larger condenser lowered the temperature difference between the refrigerant and air from 19.1 [˚C] to 12.1 [˚C]. Two other differences between the apparatus and the system model were the pressure losses on the refrigerant-side of the heat exchangers and the sub-cooling at the condenser outlet and the superheat at the evaporator outlet.

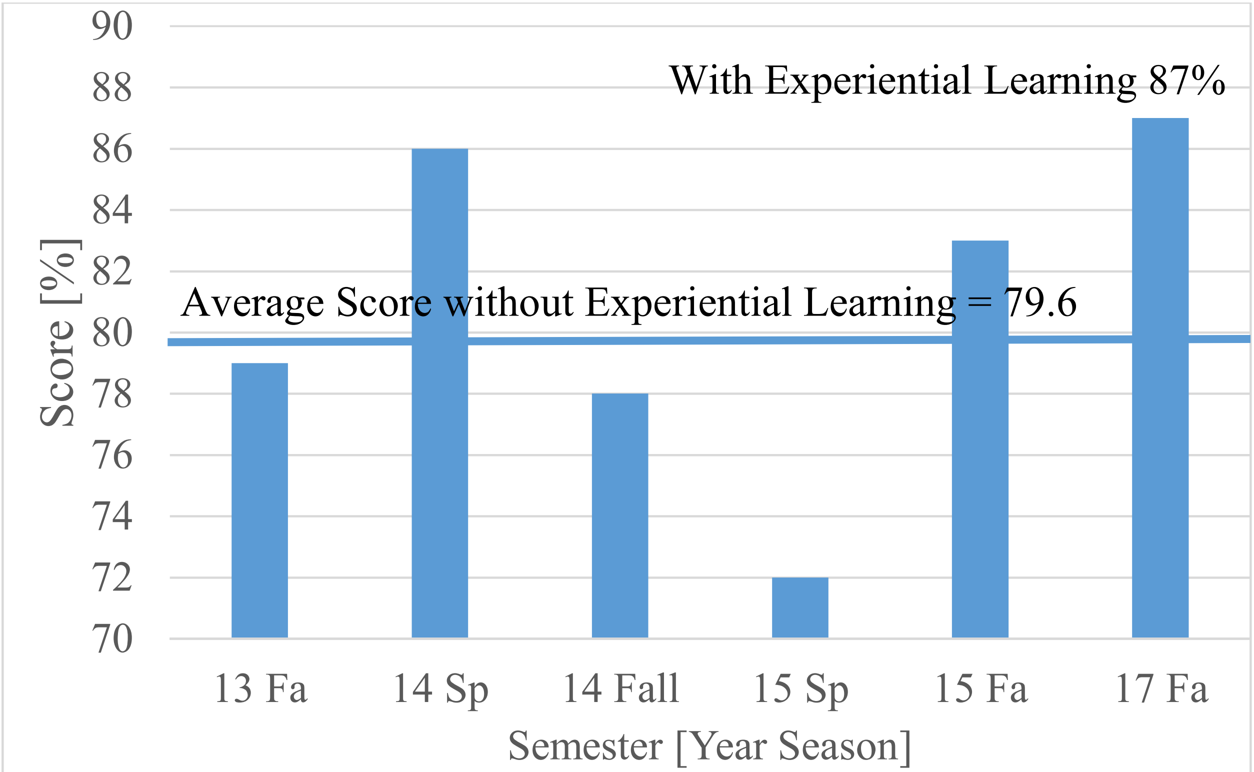

Figure 10. Comparison of Assignment 12 Scores (%) by Semester

Figure 10 shows the effect of including the experiential learning module in the class. Prior to the fall of 2017, the students were given the same optimization assignment but without the benefit of the experiential learning module. The average score for all the students that were enrolled in the class was 79.6%. Once the experiential learning module was added in the fall of 2017 the score increased to 87%. There were many factors which may have influenced the improvement besides the experiential learning. Some of those factors may have been class size, and the difference in grader. Table 2 quantifies these factors with the number of students and the average score, assigned by the grader, for all the assignments for the semester. Qualitatively the other factors influence the scores as you would expect. When the grader provided a lower score for assignments overall (spring 15) the score for assignment 12 in Figure 10 was lower as well. In the spring of 2015 the number of students were higher as well. How much these additional factors quantitatively affect the outcome was beyond the scope of this study. Quantifying these effects would be a valuable contribution to the educational community.

Table 2. Other factors which may affect the score for assignment 12

Semester	# of Students Sec 01	# of Students Sec 02	Average Score
13 Fall	15	27	90
14 Spring	32	28	96
14 Fall	12	26	93
15 Spring	29	24	83
15 Fall	18	20	84
17 Fa	9	30	88

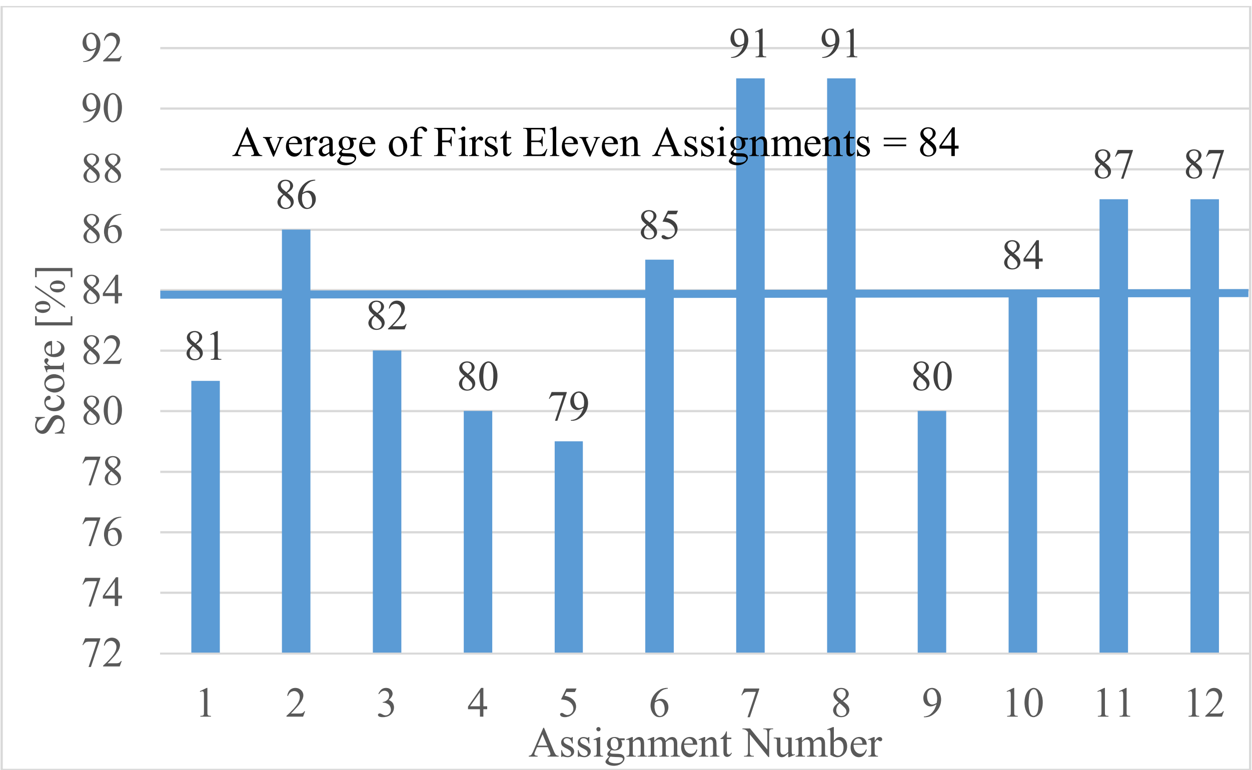

Figure 11. Scores from All Assignments During the Fall of 2017

Over the semester students were given twelve assignments. The twelfth assignment was the optimization of an air conditioner with both the experiential learning exercise and video lecture. The previous eleven assignments had varying degrees of experiential learning and video lecture. From Figure 11, the score for assignment 12 was three percentage points above the average for the previous eleven assignments.

One of the other experiential learning exercises was a demonstration of Joule's experiment. The students understanding of Joule's experiment was tested in assignment 3 and a problem on the final exam. It is difficult to determine the effect of the experiential learning because most of assignment 3 included material for which there was no experiential learning or video. A question on the final exam however showed that Joule's experiment was well understood with a score of 92% [n=9]. For assignment 7 the experiential learning included a demonstration of the components that were evaluated in the problems. For example a cut-away of a compressor using a working piston and cylinder was presented and circulated among the students.

Finally an assessment of student perceptions about the value of the demonstration of the air conditioning trainer was conducted in the Spring of 2018. Out of the 29 students who responded to the assessment:

- 93% agreed or strongly agreed that the trainer helped them connect the schematic to the hardware.
- 83% agreed or strongly agreed the trainer helped them connect the processes to the pressure-enthalpy diagram.
- 86% of the students thought the trainer gave them a deeper understanding of the instrumentation needed to measure the performance of air conditioners.

One student wrote "*The air conditioner was really interesting. I enjoyed being able to compare the schematic to the real life application of it. I felt that by seeing the energy flow (how the components connect and work together) I gained a deeper understanding of how this system operates. It helped me understand thermodynamics in general because you could see and further understand why each component is so important and special when working in a system. Having the trainer in class is not something every professor would go out of their way to show; however, I felt that for students learning the topic it was very important and helpful*."

Conclusions

Adding the experiential learning module to the class improved the scores on the assignment where the students optimize an air conditioning system. Based on data from five semesters where the experiential learning was not incorporated the module improved the average score on the same assignment by over 7%. During the same semester the score improved by 3% over the previous eleven assignments. The smaller improvement is likely due to the fact that other experiential learning modules were implemented for some the other assignments as well.

By the end of first semester, thermodynamic students can develop a system of equations for a refrigeration system and optimize the condenser and evaporator based on minimum total cost. This was done by adding three simple tools to traditional thermodynamics. Which were: 1) the conductance form of the heat exchanger equation from heat transfer, 2) the equation to take a uniform series of payments and bring them to present worth from engineering economics, and 3) regression analysis to relate the design variables to the initial cost.

The system model relating the design variables to cost provided the students with a practical and significant challenge. The students could see how the area of the heat exchangers affect the temperature difference, irreversibilities, and COP of the system. They were also, able to see how the change in one component impacts the other component of the system.

A heat balance was performed on the condenser showing a difference of 17.4%. Thus, showing the instrumentation needs to be calibrated or replaced. The outlet condenser temperature and inlet velocity or refrigerant flow rate were the most probable sources of error.

The optimized system showed significant improvement in performance over the actual air conditioning cycle. Increasing the size of the heat exchangers reduced the operating cost. The added investment in the larger heat exchangers did produce a return on that investment through the energy cost savings. Along with the reduced energy consumption came a reduced impact on the environment. So investing in energy efficiency was good for both business and the environment.

Future Work

The same apparatus could be used for second semester thermodynamic students where the compressor performance can be correlated to the pressure ratio. Then the system model could be validated with experimental data taken at different air flow rates. Also, the evaporator heat balance can be performed since the students will have psychrometrics.

In an air conditioning design class the pressure drops on both the air and refrigerant sides can be modeled and validated with an apparatus like the one presented here. This would allow for optimization of fans and fin and tube spacing along with tube diameter.

Develop a model to determine how each independent factor (e.g. class size, demonstration, and grader) influence the learning of students.

Nomenclature

Variable	Units	Description
$\dot{m}_{ac}$	[kg/s]	Mass flow rate of air through the condenser
$\dot{m}_{ae}$	[kg/s]	Mass flow rate of air through the evaporator
h_6	[kJ/kg]	Enthalpy (ex: state point 6)
$\dot{Q}_{ca}$	[KW]	Heat transfer rate of air through the condenser
ρ	[kg/m^3]	Density of air
$\dot{V}_c$	[m^3/s]	Volumetric flow rate of air through the condenser
V_c	[m/s]	Velocity of air through the condenser
A_{inlet}	[m^2]	Area of the fan inlet
A_o	[m^2]	Area of the outer diameter of fan
A_i	[m^2]	Area of the inner diameter of fan
D_{ci_o}	[m]	Outer diameter of condenser fan inlet
D_{ci_i}	[m]	Inside diameter of condenser fan inlet
T_{aci}	[C]	Temperature at the condenser inlet
P_{aci}	[kPa]	Air pressure at the condenser inlet
T_{aco}	[C]	Temperature of air at the condenser outlet
P_{aco}	[kPa]	Pressure of air at the condenser outlet
$HB_{epercent_{diff}}$	[%]	Heat balance
$\dot{Q}_{ea}$	[KW]	Heat transfer rate of air through the evaporator
$\dot{Q}_{er}$	[KW]	Heat transfer rate of refrigerant through evaporator
$IC_{Compressor}$	[$]	Initial cost of the compressor
A_C	[m^2]	Area of the heat exchanger in the condenser
CU_{AC}	[$/m^2]	Cost per unit area of the heat exchanger in the condenser
PA	[-]	PA factor
i	[-]	Minimum acceptable rate of return
n	[yr]	Number of years for the total lifecycle of system
$\dot{m}_r$	[kg/s]	Mass flow rate of the refrigerant
$\dot{Q}_e$	[KW]	Heat transfer rate through the evaporator
U_e	[kW/m^2-C]	Heat transfer coefficient of the evaporator
T_{aei}	[C]	Temperature of air at the evaporator inlet
T_{sate}	[C]	Saturation temperature of the refrigerant
$\dot{W}$	[KW]	Work through the compressor
η	[-]	Efficiency of the compressor
η_f	[-]	Fin efficiency
P_{atm}	[kPa]	Atmospheric pressure
CU_{AE}	[$/m^2]	Cost per unit area of the heat exchanger in the evaporator
C_{uE}	[$/KW-hr]	Cost per unit energy
ccc	[$/KW]	Cost coefficient of the compressor
A_e	[m^2]	Area of the heat exchanger in the evaporator
ACL	[KW/hr-yr]	Annual cooling load
e	[-]	Empirical exponent for IC of compressor

Nomenclature (continued)		
IC	[$]	Initial cost of the entire system
PWOC	[$]	Present worth of the operating cost for the system
TC	[$]	Total cost of the system

Works Cited

[1] Zietlow, David C. (2016) *Optimization of Cooling Systems.* New York, NY: Momentum Press

[2] Zietlow, David C. (2014) *Optimization of Vapor Compression Cycles*. "121st ASEE Annual Conference & Exposition. Indianapolis, IN"

Appendix A: Student Handout

Objective: Enter the data values collected during class into the table below for use in determining the heat balance.

Measured Variables	**Results**	**Measured Variables**	**Results**
Fan Speed →		Temperature of Air at the Evaporator and Condenser inlet	
Temperature of Air at Evaporator Outlet		Temperature of Air at Condenser Outlet	
Relative Humidity of Air at Evaporator Inlet		Atmospheric pressure	
Air Velocity at evaporator inlet		Air velocity at condenser inlet	
Pressure of refrigerant at evaporator exit		Pressure of refrigerant at condenser inlet	
Temperature of refrigerant at evaporator exit		Pressure of refrigerant at condenser outlet	
Volumetric flow rate of refrigerant at condenser outlet		Temperature of refrigerant at condenser inlet	
Relative Humidity at Evaporator Outlet		Temperature of refrigerant at condenser outlet	
Diameter of fan motor		Diameter of opening in fan housing	