Performance of an R-410A Room Air Conditioner Modified for Use with R-1234ze

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ABSTRACT

This paper presents the results of a senior design project that challenged a team of undergraduate students to reduce the environmental impact of a room air conditioner (RAC) by reducing its energy consumption and/or use of high-global warming potential (GWP) refrigerants. Over the course of an academic year, the team was able to investigate, design, model, evaluate, and build a prototype improved RAC.

The team began by reviewing literature on approaches that have been proposed to meet or exceed existing energy efficiency and refrigerant selection regulations. Based on these findings and the specified needs of the project sponsor, the team evaluated the appropriateness of different concepts for improving the existing R-410A RAC design and decided to pursue modifications to adapt the unit for R-1234yf.

The first step in the redesign process was to develop a thermodynamic model of the existing system. Because very little information was known about the performance of the individual components in the existing RAC, some rough performance estimates were obtained through measurements. The model of the existing system was then modified to provide the same cooling capacity as the original unit using R-1234yf and a replacement compressor was selected based on the model results.

After the replacement compressor and resized capillary tubes were installed in the RAC, the team was asked to test the prototype unit using R-1234ze instead of R-1234yf. Therefore, the model was modified to predict the cooling capacity of the unit using R-1234ze as the working fluid. The unit was tested using an environmental chamber to simulate the outdoor air conditions and a large room as the indoor environment. Although this setup could not ensure steady-state operation, air temperature measurements indicated that the room temperature did not vary more than 0.83 °C over 12 minutes of RAC operation. The cooling capacity calculated based on experimental measurements agreed within 16% of the model predictions.

While the team was able to modify the RAC to operate with R-1234ze and was able to predict the unit's performance with reasonable accuracy, the modifications required a significantly larger compressor and capillary tubes; therefore, the project clearly illustrated that fitting within the space and weight constraints of window units presents a significant challenge to implementing R-1234ze in RACs.

1. INTRODUCTION

The Energy Information Agency (EIA) conducted the Residential Energy Consumption Survey in 2009. The survey showed that approximately 23% of homes in the U.S. possess window or wall air conditioners, and 50% of those residences have two or more units; therefore, small improvements in the performance of room air conditioners (RACs) can result in significant reductions in national energy usage ("Residential Energy," 2009). The benefits of increasing energy efficiency would be twofold: the RACs would cost less to operate and the carbon footprint would also be reduced.

Although most consumers are aware of the benefits of reducing an appliance's energy consumption, not many are aware of the danger that current refrigerants like R-410A, with a high GWP of 2,088, pose to the environment ("Transitioning to," 2010). The need to adapt current RACs for use with more environmentally-friendly refrigerants is one of the primary concerns in the continuing development of refrigeration and air conditioning systems today. The definition of GWP, as referred to in this document, is the ratio of heat trapped by one unit mass of the greenhouse gas to that of one unit mass of gaseous carbon dioxide (CO_2) over a specified time period. In mathematical terms,

$$GWP_i = \frac{\int_0^{TH} a_i x_i(t)dt}{\int_0^{TH} a_{CO_2} x_{CO_2}(t)dt} \tag{1}$$

where GWP_i is the global warming potential of chemical i and a_i is the irradiative forcing of chemical i, which is the energy absorbed by the molecule for a 1 part per billion by volume (ppbv) concentration as infrared energy attempts to leave the earth. The variable x_i represents the time dependent concentration of species i after an initial pulse of release;it will be determined by the removal of chemical i due to degradation in the atmosphere, removal by dissolution into water droplets and subsequent rain-out, deposition onto dry particles, and any other mechanisms for reducing the chemical load in the atmosphere. Finally, *TH* is an arbitrary time horizon chosen for evaluation of the global warming potential of a species (Blowers). The specified time horizon is commonly defined to be 100 years by the Environmental Protection Agency ("High GWP," 2011). The numerical GWP values utilized by the project sponsor can be found in Table 1 below and the corresponding designations will be used throughout the rest of this document.

Table 1: Sponsor's classification of GWP values

GWP Value (100 yrs.)	Designation
1-30	Ultra Low
31-100	Very Low
101-300	Low
301-1000	Moderate
1001-3000	High
3001-10000	Very High
> 10000	Ultra High

The implementation of ultra-low GWP refrigerants in RACs offers an opportunity to reduce or eliminate the usage of refrigerants with a high GWP. This is advantageous because leaks in the system and improper disposal at the end of a unit's life can allow the average refrigerant charge of 0.7 kg to leak out of the system and evaporate into the atmosphere ("Transitioning to," 2010). If the entire charge of R-410A should leak out of the system, the definition of GWP explained above predicts that the equivalent mass of CO_2 emitted into the atmosphere is around 1500 kg. Also, evaporated R-410A in the atmosphere remains there, contributing to the warming of the atmosphere, for up to seventeen years (Matsunaga, 2002). Finally, R-410A belongs to a group of refrigerants called hydrofluorocarbons (HFCs) which. due to their high GWP, are expected to be completely phased out by the year 2020 through global treaties such as the Kyoto protocol ("HFCs in," 2009). Therefore, it is imperative that companies which currently use R-410A in their systems understand how to adapt their current systems to use refrigerants other than HFCs so that they may have adequate time to satisfy the requirements of any future mandates.

2. OBJECTIVE

The primary objective of this project is to analyze the operating performance of a 2.93-kW room air conditioner using two different refrigerants: R-410A, the current refrigerant, and R-1234yf, the proposed replacement refrigerant which has an ultra-low GWP of 4 ("Transitioning to," 2010). Since adapting the current unit to use R-1234yf will require the redesign and resizing of several components, the secondary objective, if it is feasible and practical, will be to improve the current energy efficiency rating of the unit during the modification and resizing of components.

More specifically, a mathematical model of the RAC operating with R-410A is to be developed based on the first and second laws of thermodynamics using data provided by the manufacturer. After the model's ability to predict the performance of the system with R-410A is verified, the model will be used to determine the modifications necessary to achieve a cooling capacity of 2.93 kW with R-1234yf as the working refrigerant. Finally, the feasibility of applying the model to the system with refrigerants other than R-410A will be evaluated based on the experimental performance of the modified RAC utilizing R-1234yf.

3. MATHEMATICAL MODEL

The mathematical model of the 2.93-kW RAC was developed in Engineering Equation Solver (EES) software, which contains a library of callable thermodynamic property data for both R-410A and R-1234yf. The following assumptions were used in the development of the model:

- System is operating at steady-state
- Pressure drop across each heat exchanger (evaporator and condenser) is negligible
- Refrigerant leaves the condenser as a saturated liquid (quality of 0)
- Volumetric efficiency of the compressor is 100% for R-410A
- Heat exchanger performance does not vary with working fluid
- Changes in kinetic and potential energies of working fluid are negligible
- Pinch temperatures (obtained from manufacturer test data) are constant
    - Evaporator: ΔT_{evap}=22°C below indoor ambient air temperature
    - Condenser: ΔT_{cond}=15°C above outdoor ambient air temperature

Using these assumptions, test data from the manufacturer, and lab measurements of the power into the compressor and heat loss from the compressor, the thermodynamic states of the cycle were predicted as shown in Figure 1. Once these state points were established, the mass flow rate of refrigerant could be estimated based on three different mass and energy balances: (1) an energy balance on the compressor assuming adiabatic operation and an isentropic efficiency of 68%, (2) an energy balance on the condensation process assuming saturated liquid at the exit, and (3) an energy balance on the evaporator assuming isenthalpic expansion occurs from a saturated liquid state at the condenser exit to the evaporation pressure. The standard deviation of these mass flow rates was found to be 7.4%, which was considered reasonable based on the stated assumptions and the uncertainty associated with the direct measurements taken by the team in the absence of manufacturer-supplied data. The most accurate mass flow rate was assumed to be the rate calculated using the evaporator energy balance because the manufacturer supplied data on both cooling capacity, $\dot{Q}_{evap}$, and the compressor suction conditions. Therefore, the enthalpy at the evaporator exit, h_1, can be determined from property relations and the enthalpy at the evaporator inlet, h_4, can be estimated by assuming saturated liquid at the condenser exit:

$$\dot{m} = \frac{\dot{Q}_{evap}}{h_1 - h_4} \qquad (2)$$

where $\dot{m}$ represents the refrigerant mass flow rate.

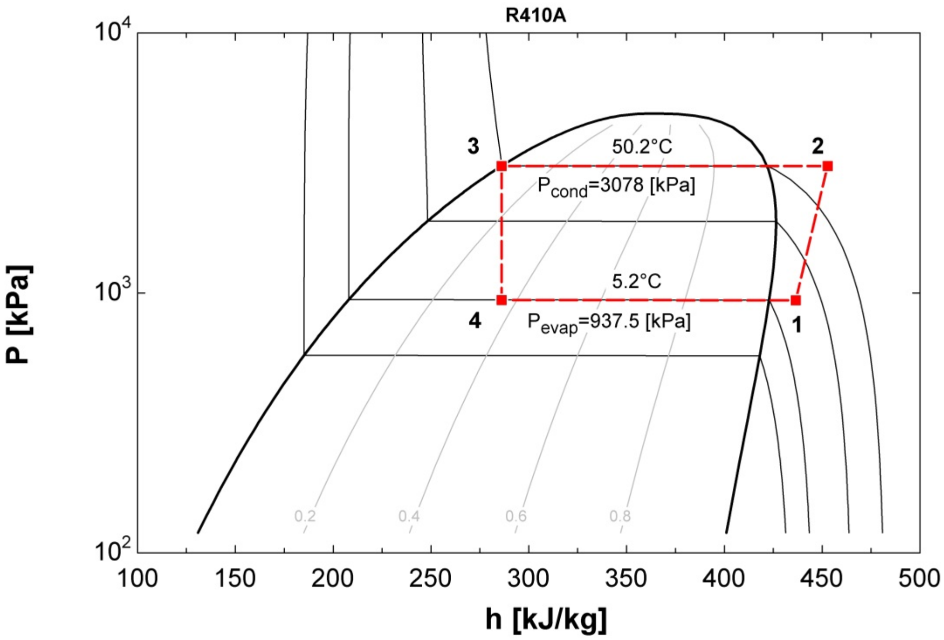

Figure 1: P-h Diagram with System States for RAC with R-410A

4. SYSTEM MODIFICATIONS

4.1 Predicted System Performance with R-1234yf

One objective of the senior design project was to modify the RAC such that it provides the same cooling capacity as the original unit but using R-1234yf as the working fluid. Therefore, adapting the verified model to use R-1234yf as the refrigerant enabled the team to resize the main components of the RAC to achieve this objective. As can be seen by comparing Figures 1 and 2, the main difference between the R-410A and R-1234yf cycles are the working pressures in the condenser and the evaporator. Assuming that the condensing and evaporating temperatures do not change, the working pressures with R-1234yf are much lower than in the R-410A system. The densities of the refrigerants also differ, which will impact the mass flow rate through each component.

The single most important component of the system which had to be modified was the compressor. Assuming that the conditions of the refrigerant entering and leaving the evaporator are primarily a function of the heat source and sink temperatures, which do not change, the primary method of adjusting the cooling capacity of the system is through adjusting the mass flow rate of refrigerant. Figure 2 confirms that the refrigerant experiences a smaller change in specific enthalpy across the evaporator in the R-1234yf system and thus the system requires a significantly larger mass flow rate to ensure that the cooling capacity remains above the goal of 10,000 Btu/hr.

In order to resize the compressor, the required mass flow rate through the evaporator was calculated using Equation (2) with the target cooling capacity. The resulting volumetric flow rate entering the compressor can be calculated based on the density of the refrigerant in the suction line, ρ_s:

$$\dot{V}_s = \frac{\dot{m}_s}{\rho_s} \tag{3}$$

Therefore, the minimum displacement of the new compressor can be estimated by assuming that the compressor operates with a volumetric efficiency of 100%,

$$V = \frac{\dot{V}_s}{\dot{N}} \tag{3}$$

where $\dot{N}$ represents the rotational speed of the compressor crankshaft in rotations per second. In addition, the ideal swept displacement of the original R-410A compressor was calculated for comparison. It was found that the compressor's swept displacement must increase from 10.1 cm^3 to 23.61 cm^3 to satisfy the cooling capacity requirement with R-1234yf.

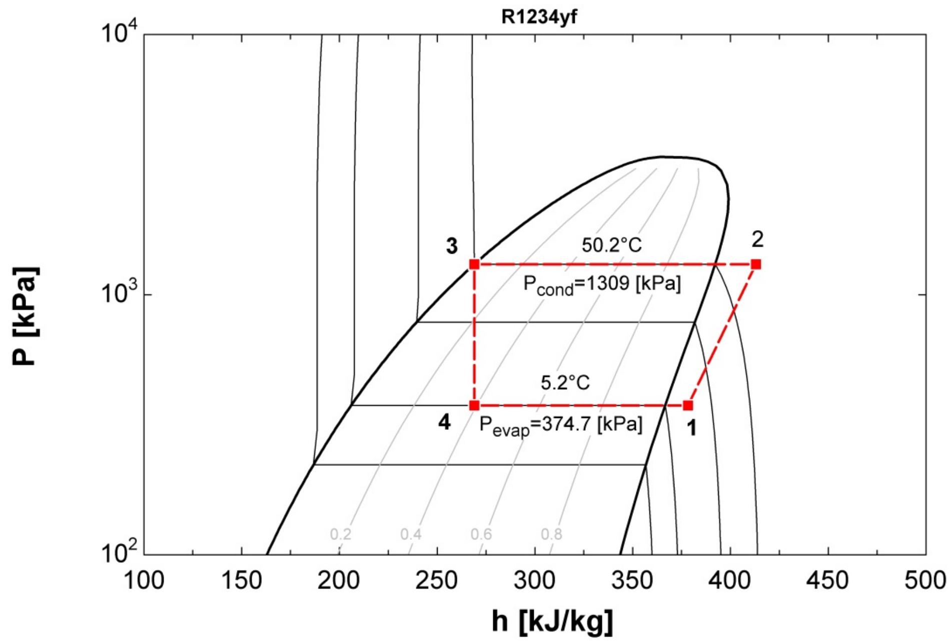

Figure 2: P-h Diagram with System States for RAC with R-1234yf

Due to the current developmental nature of R-1234yf and the similarities between the properties of R-1234yf and R-134a, a compressor designed for use with R-134a was chosen for use in the RAC ("Product Bulletin"). After examining multiple R-134a compressors, an Embraco NT6220Z was chosen based upon the swept displacement and electrical requirements. With a swept displacement of 26.11 cm^3, the compressor was oversized for the application and the model predicted it would achieve a cooling capacity of 3.37 kW under the design conditions, which is approximately 15% above the required cooling capacity. However, the additional capacity offered an allowance for deviations between the mathematical model, which was based on simplifying assumptions, and the real system behavior.

In addition to resizing the compressor, the capillary tubes used in the system were replaced in order to obtain the required isenthalpic pressure drop between the condenser (State 3) and evaporator (State 4). They were sized based on the desired evaporating temperature and the R-134a compatibility recommendations provided in the compressor literature. Compared to the original R-410A unit, the new capillary tubes are longer and have a larger internal diameter.

It is important to note that the changes to these components required a structural modification to the RAC housing because the replacement compressor was significantly larger and heavier than the original R-410A compressor. Figure 3 shows the extension added to the RAC housing, painted in black, which makes the unit much bulkier and less practical for household use.

Figure 3: RAC housing expansion to accommodate compressor for R-1234yf system

4.2 Predicted System Performance with R-1234ze

After sizing, purchasing and installing the new components required for the RAC to utilize R-1234yf, the project sponsor requested that the team test the unit using R-1234ze instead of R-1234yf. This change was proposed because R-1234ze was less expensive and did not have the flammability concerns of R-1234yf. However, the model estimated that the RAC would achieve a cooling capacity of 2.68 kW with R-1234ze as opposed to the capacity of 3.37 kW that was predicted for R-1234yf. Figure 4 shows that the pressure-enthalpy plot of R-1234ze varies noticeably from the same diagram for R-1234yf, which is the reason for this change in capacity. Because the cooling capacity with R-1234ze was below the marginal value of the original target specification (2.93 kW), the ultimate objective of the project had to be modified accordingly. The primary project goal became to verify the ability of the model to accurately predict system performance with an alternative refrigerant.

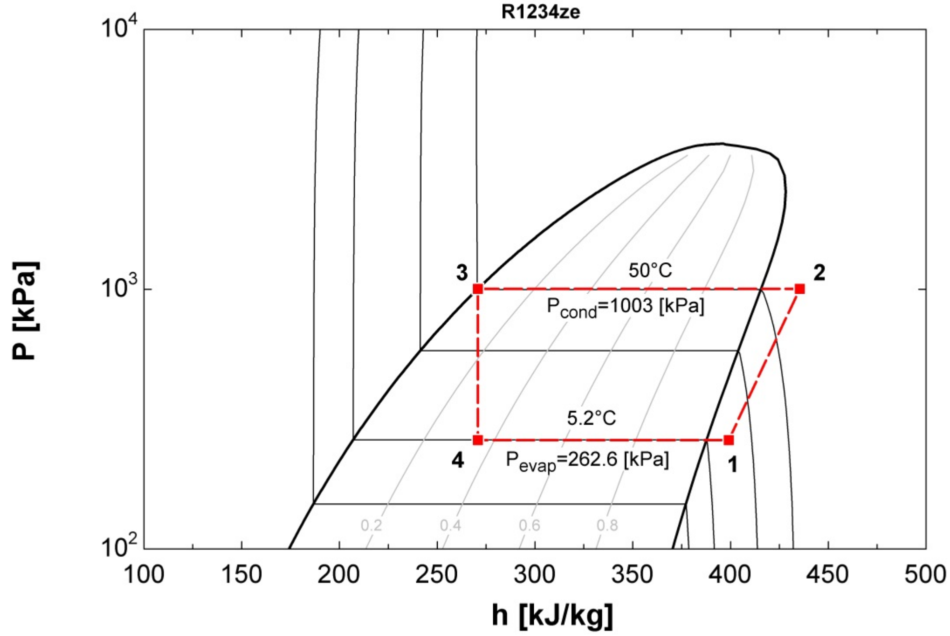

Figure 4: P-h Diagram with System States for RAC with R-1234ze

5. EXPERIMENTAL RESULTS

In order to verify that the model accurately predicts the prototype RAC performance with R-1234ze, a single performance test was designed and performed twice in a pseudo-environmental chamber. One test was performed with the heat sink maintained 11 °C above the internal environment's temperature to simulate normal operation, and another test was performed the heat sink at 28 °C above the source temperature, to simulate operation under extreme conditions. Due to limitations on the environmental chamber control, the tests could not reproduce the conditions used in the RAC manufacturer's testing standard, which involved an indoor temperature of 27 °C and an outdoor temperature of 35 °C.

With only one environmental chamber without humidity control available for testing, the extent of testing was limited to "pseudo-steady state" operation. It was assumed that the laboratory was a large enough space to serve as the conditioned reservoir, and that variations in the humidity over the period of the test would be negligible. A temporary barrier was built around the RAC to form a seal between the environmental chamber and the laboratory, and the temperature in each space was monitored continuously.

The laboratory environment was measured to be at an average temperature of 19.1 °C and a relative humidity of 31%. For the first test, the temperature of the chamber was set to 30.2 °C, resulting of a temperature difference of 11 °C across the chambers, and the uncontrolled relative humidity was measured to be 23.3%. For the second test, the temperature of the chamber was set at 46.1 °C in order to test the unit at an extreme operating condition, which raised the relative humidity to 68.5% and resulted in a temperature difference of 28 °C between the reservoirs. The collected data is summarized in Table 2 and includes air temperatures and relative humidities of the unconditioned room air, conditioned room air and the environmental chamber ("outdoor") air once the RAC had reached pseudo-steady state operation. The test was considered pseudo-steady because the appropriate temperature profiles did not vary by more than 0.83 °C over 12 minutes of RAC operation.

Table 2: RAC test data

Conditions	Unconditioned Air Temperature (°C)	Unconditioned Relative Humidity (%)	Conditioned Air Temperature (°C)	Conditioned Relative Humidity (%)	Outdoor Temperature (°C)	Outdoor Relative Humidity (%)
ΔT=11 °C	19.1 ± 0.56	31 ± 2.5	9.06 ± 0.56	73.7 ± 2.5	30 ± 0.56	23.3 ± 2.5
ΔT=28 °C	19.1 ± 0.56	31 ± 2.5	10.4 ± 0.56	68.5 ± 2.5	47 ± 0.56	15.4 ± 2.5

Because the environmental chamber is not instrumented to measure the thermal load, an energy balance was performed on the air stream flowing over the evaporator to estimate the unit's cooling capacity. Using the recorded temperature data and estimating the air flow rate as 15.2 m^3/min based on velocity measurements, the cooling capacity of the unit was calculated at the two test conditions. Table 3 shows that the cooling capacity for the 11°C temperature lift case was predicted to be 2.13 kW by the model and was calculated as 2.48 kW based on experiments. Similarly, the model predicted a cooling capacity of 1.74 kW for the 28°C temperature lift case which was less that the capacity of 2.13 kW calculated based on experiments. Although the predicted and measured values for the two test cases differ by 16.35% and 11.62%, respectively, the results are considered satisfactory given the simplifying assumptions of the model and the limited availability of testing equipment. The results indicate a promising correlation of the mathematical model with the performance of the modified unit and suggest that further testing of the unit is warranted in an environment that can be accurately and precisely controlled to higher standards.

Table 3: Predicted and measured cooling capacities

Conditions	Cooling Capacity (kW)		Percent Difference Between Measured and Predicted (%)
	Predicted	Measured	
ΔT=11 °C	2.13	2.48	16.35%
ΔT=28 °C	1.74	1.94	11.62%

6. CONCLUSIONS

The refrigeration cycle of a 2.93-kW RAC that utilizes R-410A (GWP of 2,088) was successfully modeled using a set of simplifying assumptions, test data and EES. The model was then used to determine the system changes required for the unit to utilize R-1234yf, an alternative refrigerant with an ultra-low GWP of 4, while maintaining the current cooling capacity. Based upon the model results, both the compressor and capillary tubes were resized in order to maintain the cooling capacity with R-1234yf as the working fluid. After modifying the RAC with the resized components, the project goal was altered to evaluate the performance of the alternative refrigerant R-1234ze in the system. The model was updated accordingly to predict the cooling capacity of the modified RAC and the unit was then experimentally tested. Although the unit was unable to be tested in strict accordance with industry standards, it was determined that the model predicted the experimental cooling capacity to within 16% under the given testing conditions, warranting further testing in a facility with dual psychrometric chambers.

Even though the modified RAC was significantly larger and heavier than the original unit, making it less practical for implementation in the typical household, the project demonstrated the feasibility of utilizing an alternative, ultra-low GWP refrigerant in the place of an HFC refrigerant with high GWP. In addition, the project provided valuable design experience and exposed the team of senior engineering students to the HVAC industry. Therefore, the project was considered a success for both the project sponsor and the supporting university.

NOMENCLATURE

h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
$\dot{N}$	crankshaft speed	(rev/s)
$\dot{Q}$	heat transfer rate	(kW)
ρ	density	(kg/m^3)
$\dot{V}$	volume flow rate	(m^3/s)

Subscript

s	suction

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