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Creation of an Exergetic Based Leak Detection and Diagnosis Methodology for Automotive Carbon Dioxide Air Conditioning Systems*

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Abstract

Carbon dioxide automotive air conditioning (AC) systems have been under development for over a decade. Although the AC system construction is important, a reliable refrigerant leakage detection system is also vital. A detailed thermodynamic simulation model and fault detection and diagnosis (FDD) system, with proposed validation plans, has been developed at Rochester Institute of Technology. A discussion of simulation models that have been developed for various compressors and heat exchangers is presented; they are compared to actual AC systems to develop a realistic experimental setup. Assumptions from previous work are examined and improved.

Keywords: *Carbon dioxide, air conditioning systems, exergy, leak detection.*

1. Introduction

With concerns regarding global warming and ozone depletion, refrigerant emissions have become an area of interest. The emissions released through automotive applications, such as leaks from air conditioning (AC) systems, are of great concern.

Bailey et al. (2000) and Braun (2003) discuss the benefits of fault detection and diagnosis (FDD) of any system as falling into two categories: improving safety or reducing cost to operate. Currently, commonly used FDD technology detects and diagnoses significant faults, and problems are only detected and diagnosed after occupants have complained. Refrigerant leakage is not only a problem with regards to comfort and service calls, but also with regards to the environmental. It is possible that early leakage detection could eventually be mandated for unitary equipment. The loss of refrigerant tends to lower certain pressures within a cycle, leading to reductions in both evaporating pressure and gas cooler temperatures. The lower evaporating pressure results in both higher evaporator superheat and a higher refrigerant discharge temperature from the compressor.

For over a decade, there has been exploration into using naturally occurring refrigerants as the working fluid in all sectors of refrigeration, including automotive applications. Carbon dioxide (CO_2) is receiving attention as a possible replacement for synthetic refrigerants due to its abundance in nature and benign characteristics. However, the abundance of CO_2 in the atmosphere makes it a very difficult to detect should anything but a catastrophic leak occur. Despite its environmental neutrality, fault detection and diagnosis is still necessary to make CO_2 a viable alternative refrigerant.

Currently, most research conducted concerning the new CO_2 AC system is focused on developing new system components that can withstand the pressures and temperatures associated with a carbon dioxide cycle. Changes to previous thermodynamic models for a vapor compression refrigeration cycle must be made in order to accommodate the thermophysical properties of CO_2. Carbon dioxide has a low critical temperature (31.1°C), which means that all CO_2 systems are subcritical, or move between sub and supercritical pressures during the cycle. The system shown in Figure 1 was developed by Lorentzen and Pettersen (1993). Figure 2 shows the associated Temperature vs. Entropy diagram. With the current design in hand, researchers have been developing new system components, such as compressors and heat exchangers.

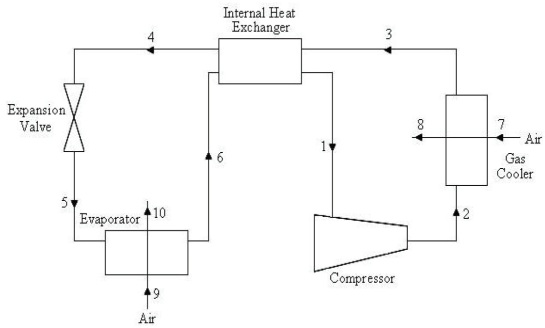

Figure 1. CO_2 Refrigeration System Schematic.

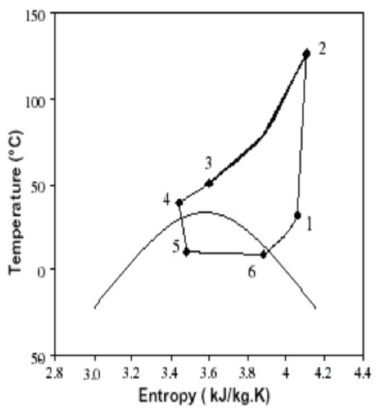

Figure 2. Temperature vs. Entropy Diagram for CO_2 Refrigeration Cycle.

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2. Research Objective

The purpose of this research is twofold: to continue the development of an automotive carbon dioxide air conditioning leak detection and diagnosis system, and second, to enhance simulations with more in-depth analysis of system components. A previously developed simulation model is used and improved upon by refining certain assumptions. The compressor and the air-to-refrigerant heat exchangers (evaporator and gas cooler) are of the most concern. With the use of past models and through collaboration with industry, a more realistic model is developed. Leak locations are the same as those of the previous model, which were based on results from a fleet of prototype vehicles in Europe that currently use CO_2 AC systems.

3. Previous Research

Canfield and Bailey (2005) developed an exergetic based fault detection and diagnosis methodology for the carbon dioxide AC system. In this development, the authors utilized the Engineering Equation Solver software (2008). This model simulated the AC system using gross assumptions based upon previous work. Using the system simulation, the authors applied leaks to what were considered leak prone areas. As seen in Figure 1, the leaks were placed at the outlet of the compressor (State 2), the inlet of the evaporator (State 5) and the outlet of the evaporator (State 6). These same locations are used for the new simulation model.

The leak locations were chosen for the following reasons. The compressor used for this system is prone to leakage because it is non-hermetically sealed, thus consisting of a sealed rotating shaft in an open configuration. The evaporator is an area of concern because of its proximity to joints. It is important that manufactures dampen the evaporator vibrations because it is the closest refrigerant system component to the passengers. Because the refrigerant flowing into and out of the evaporator is subcritical, meaning a liquid-vapor mixture, manufacturers can reduce the vibrations using hoses between metal lines and components; however, the use of hoses increases the likelihood of refrigerant leaks.

Differences between exergy losses and exergetic efficiencies were found once the leaks were applied to the aforementioned areas. A leak at State 2 caused the internal heat exchanger and evaporator exergetic losses to decrease, while losses increase for the compressor and gas cooler. As expected, the exergetic efficiencies increased for the internal heat exchanger and the evaporator, while decreasing for the compressor and gas cooler. The observed correlation between exergy loss and exergetic efficiency is expected.

Previous research explored examples of the ways in which leaks are detected within the automotive refrigeration application and the physical changes that must be made to the system for it to withstand the pressures and temperatures produced by the system. A number of CO_2 air-conditioning systems are modeled in full. Brown (2003) creates a transcritical CO_2 refrigerant cycle model and applies it successfully to a CO_2 based automotive AC system. This model includes a baseline system as well as the ability to incorporate additional hardware. Filho et al. (2007) creates another model applicable to steady state situations at high ambient temperatures. This model produces results comparable to experimental data.

3.1 Compressor Models

Compressor models for the CO_2 automotive AC system have been examined (refer to States 1 and 2 in Figures 1 and 2 for the compressor). Hwang (1997) presents two steady state compressor simulation models: a thermodynamic model and a heat transfer model. These two models rely solely on motor and mechanical efficiency. Also included in the models is an internal refrigerant leak that was utilized in the mass and energy balances, which had not been addressed previously in literature. These models are validated within 5% error through experimentation. The largest source of error occurred because of the compressor chosen to make the experiment more practical; an electric compressor is used as opposed to a non-hermetically sealed compressor, which would be more realistic.

Other systems are developed and tested using CO_2 as the refrigerant. Yoshioka et al. (2000) adopt a scroll compressor from the HFC-134a system because the low volume change rate during the compression cycle helps reduce the discharge pulsation, thus allowing for low noise and vibration. Hagita et al. (2002) improve upon this scroll compressor design by having the orbiting scroll pushed to the fixed scroll. The majority of losses coming from the scroll compressor are attributed to the leaks; it is also observed that the new configuration controls the leakage from the scroll tip surface. Mu et al. (2003) incorporate a reciprocating compressor into their prototype system because of the ease with which it controls leakage between the piston and cylinder. Liu et al. (2005) use a swash plate design with a high side pressure control in order to maintain evaporator pressure under the increased pressure of a CO_2 based system. Maintaining evaporator pressure optimizes system efficiency.

3.2 Heat Exchanger Models

Hwang (1997) also presents a heat exchanger model for the evaporator and gas cooler of the system. The heat exchanger model assumes the configuration to be counterflow, shell and tube type, and installed horizontally. Heat loss from the heat exchanger is neglected and the evaporation and gas cooling processes are regarded as isobaric, except for pressure drops along the tubes of the heat exchanger. These models are verified through experimentation with an error of 2%. Although Hwang's experimental set up is practical for experimentation, it does not reflect real life situations. Hwang also used water as a cooling medium for simulation and experimental models; air is a more realistic fluid.

Along with compressor models, Yoshioka et al. (2000) develops several heat exchangers for the gas cooler (refer to States 2 and 3 in Figures 1 and 2) and evaporator (refer to States 5 and 6 in Figures 1 and 2). A corrugated louvered fin and flat tube made of aluminum is adopted for the gas cooler; optimum heat exchanger efficiency occurs with 1 mm diameter tubes. With the higher pressure and density involved in using CO_2 as a refrigerant, it is possible to have a larger pressure loss in the gas cooler than the loss produced by HFC-134a, resulting in a smaller hole diameter in the flat tube. Mu et al. (2003) adopt a tube-fin type heat exchanger for the gas cooler and a reversed-flow tube-in-tube internal heat exchanger. Liu et al. (2005) incorporates the tube-fin type heat exchanger with a 2 mm diameter tube for the gas cooler and evaporator. An accumulator cylinder with installed high-pressure heat transfer tubes is used for the internal heat exchanger.

Efforts have also been made to use heat dissipated during dehumidification in the AC system. Current HFC-134a systems have an auxiliary electric heater to dissipate this heat outside the car. Tamura et al. (2005) incorporate a water-refrigerant heat exchanger to transfer heat released from dehumidification to engine coolant, bringing the coefficient of performance (COP) to 1.31 times the current HFC-134a system. Tamura et al. also integrates two expansion valves around the outdoor heat exchanger. These valves regulate and optimize the amount of refrigerant depending on whether the system is in a heating or cooling cycle. More refrigerant is needed during the heating cycle because the ambient temperature outside the vehicle is generally lower than with the cooling cycle.

4. Assumption Clarification

After understanding how to construct a CO_2 AC system that reflects a realistic prototype, it is necessary to discuss the assumptions of the simulation model in order to make it more representative of a real life cycle. Throughout the studies, steady state operation is assumed, thus neglecting transient operations such as initial start up and acceleration. This assumption reflects the data collected by McEnaney et al. (1999) while operating a prototype CO_2 AC system to validate the refrigerant's feasibility. McEnaney et al. also developed a prototype to be nearly identical to the commercially available HFC-134a system. Various compressor speeds were used during testing, but data was only collected once the system reached steady state operation.

The steady state condition refers to the operation of the AC system assuming constant speed of the compressor motor and constant ambient temperature, while also being subjected to a constant mass flow loss rate. The mass balances are satisfied over a time interval where the mass before the leak is equal to the mass in the system after the leak plus the mass leaked. The cycle is run only once with an applied leak, and then appropriate properties calculated. These data are tabulated, and then the system is run again assuming a new mass flow rate based on an applied leak for a given time interval. This is continued in order to establish a trend in the data. The pressure drops are assumed to remain constant for each cycle, because more detailed information about the prototype vehicle pressure drops is unavailable and the current assumption seemed appropriate.

In developing the heat exchanger models for the system, which includes the gas cooler and evaporator, the authors assumed constant pressure drops for both the air and refrigerant sides. Table 1 provides a tabulated form of those pressure drops and also the flow rates that correspond to each fluid.

Data collected by McEnaney et al. is used by Fartaj et al. (2004) for a second law analysis of the transcritical CO_2 system. Fartaj et al. chose to use data corresponding to a medium compressor speed of 1800 RPM. In the current work represented here, the authors have used the same set of data and Engineering Equation Solver (EES) software in order to validate the healthy air conditioning system. Also, Fartaj et al. used the thermophysical property tables given by Vargaftik (1975) to determine enthalpies and entropies needed to calculate exergy losses and exergetic efficiencies. Only a slight change was found between values determined by Fartaj et al. and the authors when the thermophysical property tables provided by EES were used.

Table 1. Heat Exchanger Information.

		Flow Rate (kg/min)	**ΔP (kPa)**
Gas Cooler	**Air**	29.57	0.061
	Refrigerant	2.64	139
Evaporator	**Air**	8.42	0.118
	Refrigerant	2.64	131

The following assumptions have been altered for the current work from previous research to create more realistic models.

4.1 Leak Rate (previously 0.005 kg/min)

Identifying a leak rate that corresponds to a common leak in AC systems is important. Current technology used on conventional systems can detect leaks as small as 5 kg/yr, or 9.5×10^{-6} kg/min. A leak of this size correlates to a constant system failure that will cause the system to completely lose charge in up to 100 years. This type of leak is not within the scope of this project.

Comprehensive studies on CO_2 AC systems have been done (Hwang, 1997); a continuous leak rate of 9% of the mass flow rate around the compressor has been observed. This leak is a reflection of a catastrophic failure that will cause the system to lose charge within two minutes. For the research conducted here, the leak rate of 0.005 kg/min, only 0.2% of the mass flow rate, will cause the system to lose charge completely in 100 min. This leak can also be considered catastrophic, because it causes complete failure in less than two hours. These catastrophic failures would not necessarily require leak detection because the decrease in cooling capacity would immediately alert the passenger to possible system failure. Reverse process is used in order to develop a more realistic leak rate within the scope of this project. Starting with a time frame of 24 hrs, or 1440 min, for complete failure, a leak rate of 3.5×10^{-4} kg/min, 0.01% of the initial mass flow rate, is calculated.

The current OSHA (Occupational Safety and Health Administration) (2001) standard for CO_2 exposure is a concentration of 5000 ppm (9000 mg/m^3) for a period of 30 minutes; any exposure more severe than this can lead to health issues. A leak correlating to less than 2 ppm/s (120 ppm/min) would require almost 7 hours of continuous operation to reach an unhealthy limit within a vehicle cabin. Currently, CO_2 gas sensors have been developed that can detect leaks as small as 2000 ppm/s (Frodl and Tille, 2006), which would require a leak 1000 times greater than what this methodology requires for leak detection.

A major change from previous research (Canfield and Bailey, 2005) is that the leaks applied to the system will be continuous until the system completely fails, indicated by the system losing charge. Before, only one leak was applied in order to simplify the simulation. Now, the simulation can integrate parametric tables in order to collect data as the leak occurs. The collected data is exported to observe trends, which are used to create the FDD methodology.

4.2 Compressor Power Consumption (previously constant)

Compressor power consumption will continue to increase while the system is failing. How to handle this phenomenon was originally unknown. After research on a variety of compressors, it is decided to assume that the compressor power consumption increases at the same percent rate as the mass flow rate decreases. The compressor power consumption is based on the percentage of the refrigerant mass flow rate loss and the percentage of the enthalpy difference. Eqs (1)-(3) reflect the theory behind this assumption.

$$ {}_1\dot{W}_2 = \dot{m}(h_2 - h_1) \tag{1} $$

It is observed through the EES simulation model that the quantity $(h_2 - h_1)$ is decreased by 2% and the variable ΔH is used to represent the percentage change. The mass flow rate, $\dot{m}$ is manually decreased by 0.01% of the mass flow rate with each run of the cycle during faulty operation. Eq. (2) reflects the parameter changes.

$$ {}_1\dot{W}_2(1+\Delta\dot{W}) = \left[\dot{m}(1-\Delta\dot{m})\right]\left[(h_2 - h_1)(1-\Delta H)\right] \tag{2} $$

By canceling out Eq. (1) in Eq. (2), the following remains for Eq. (3):

$$ (1+\Delta\dot{W}) = (1-\Delta\dot{m})(1-\Delta H) \tag{3} $$

After solving the equation for the change in power consumption, and substituting in the values for $\dot{m}$ and ΔH, $\Delta\dot{W}$ is found to increase by 2% with every run of the cycle during faulty operation. Although this assumption is not necessarily the exact result of an actual compressor leak, it is closer to reality than previous assumptions.

4.3 Compressor Isentropic Efficiency (previously constant)

Research has shown that assuming constant isentropic efficiency during normal operation is accurate within 2% error of experimental data (Hwang, 1997). When the compressor is under faulty operation, its power consumption will continue to increase, and it becomes apparent that neither the isentropic nor the thermodynamic efficiency of the compressor remains constant. If the power consumption of the simulation model is constantly increased by 2%, then the isentropic efficiency of the compressor must decrease by the same amount, which is reflected in Eq. (4):

$$ \eta_c = \frac{\dot{W}_s}{{}_1\dot{W}_2} \tag{4} $$

Eq. (5) reflects the changes that occur for both η_c and ${}_1\dot{W}_2$, while $\dot{W}_s$ remains constant because it is an ideal parameter.

$$ \eta_c(1-\Delta\eta) = \frac{\dot{W}_s}{{}_1\dot{W}_2 * (1+\Delta\dot{W})} \tag{5} $$

By canceling out Eq. (4) in Eq. (5), and substituting the value for $\Delta\dot{W}$, the change in η is equal to the reciprocal of the change in $\Delta\dot{W}$, which is to be expected. Therefore, in the EES simulation, the isentropic efficiency is constantly decreased by 0.67%.

4.4 Internal Heat Exchanger Effectiveness (previously constant)

The heat exchanger cannot be determined without heat exchanger dimensions, because there are no means to verify the results. It was therefore assumed in previous models that the effectiveness was constant. This research has removed the heat exchanger effectiveness assumption by using new parameters that do not include the heat exchanger effectiveness.

With the assumptions more fully understood, a more realistic simulation model is developed; the new simulation model encompasses the new leak rate as well as the new compressor power consumption assumption. Because some of the assumptions, such as the isentropic efficiency, were not all clarified, it is impossible to fully develop a realistic simulation model. Holes are left in the model where knowledge is lacking about the system response to leaks. In discussions with Delphi-Harrison, Inc. it was verified that developing a model on any AC system is difficult because of the variations that occur throughout the system, even while the system is stabilized.

5. Simulation Model and Validation Plan

The First and Second Laws of Thermodynamics are used as described in the following section to verify normal operational data and performance parameters obtained from past experimental and analytical work.

Both a first law based (energetic) efficiency as well as a second law based (exergetic) efficiency can be found in order to evaluate the system's performance. The concept exergy comes from using the Second Law of Thermodynamics in conjunction with the conservation of energy laws. Exergy can be defined as the maximum theoretical work obtainable as a system interacts with its reference environment to reach equilibrium (Moran and Shapiro, 2000). The use of exergy in thermodynamic analyses provides more meaningful results than simply using the first law of thermodynamics. The loss or destruction of exergy determined through an exergy analysis indicates where and how losses occur in the system. With exergetic efficiency, the examiner can gauge the potential for improvement in the performance of a system (Moran and Shapiro, 2000).

The exergy of a pure substance is given by Eq. (6):

$$ e = (h - h_0) - T_0(s - s_0) \tag{6} $$

where the enthalpy (h) and entropy (s) of the given state is compared to the reference, or dead state (depicted by subscripts of '0'). An environment is assumed to be a simple compressible system that is large in extent and uniform in both temperature and pressure. This reference state is typically the ambient temperature and pressure for the system, such as 298 K and 101.325 kPa, respectively. The exergy loss (Δe) is calculated with Eq. 7 (Moran and Shapiro, 2000):

$$ \Delta e = \text{exergy supplied} - \text{exergy recovered} \tag{7} $$

By calculating the exergy loss, components of the system can be ranked based upon their contribution to the overall exergy destruction of the system. An energy balance and an exergy balance are done around every component of the system (gas cooler, compressor, evaporator, internal heat exchanger, and expansion valve). The energy balances produce the total system coefficient of performance (COP) and the exergy balances produce the corresponding exergetic efficiency of each system component. The advantage of calculating the exergetic efficiency at the component level, rather than looking at the total system COP, is the ability to detect the leaks at their source. If only the COP is used for detecting failure in the system, more calculations and possibly more measurements are required to find the source of failure. The exergetic efficiency of each component is given by Eq. (8) (Moran and Shapiro, 2000):

$$\xi = \text{exergy recovered} / \text{exergy supplied} \qquad (8)$$

Within EES, a "Diagram Window" is presented as the user interface, where the AC system temperature and pressure of each state point are required input data. From the user input, the software utilizes an extensive database to look up various thermodynamic properties, in this case, enthalpy and entropy, at each state point. Initially, a first law analysis is performed in EES, and the respective heat transfers and power consumptions are calculated. In addition, a second law analysis is performed and the exergy losses and exergetic efficiencies are calculated at each state point. To incorporate a leak, an $\dot{m}$ loss is introduced one at a time to States 2, 5 and 6, as shown in Figure 1.

Trends are developed from the exergy loss and exergetic efficiency variations caused by the introduced leaks. These plots can be used as the method for identifying and therefore diagnosing where the system fault is occurring. If the AC system is operating at conditions that reflect the trends produced in the simulation model, the leak can be effectively detected and diagnosed.

The previously developed simulation model has been enhanced through the aforementioned assumption clarifications. A leak rate of 3.5×10^{-4} kg/min is utilized to simulate system leaks both around the evaporator (states 5 and 6) and at the discharge of the compressor (state 2). The new simulation model also incorporates a compressor power consumption increase of 2.03% after each cycle, which correlates to the leak rate. A cycle is when the refrigerant makes one complete pass through the entire AC system. The isentropic efficiency will also reflect the leak by a steady decrease of 0.67% after each cycle.

In order to implement this leak detection and diagnosis methodology into real systems, pressure and temperature sensors are required at each of the state points depicted in Figure 1. These measurements are used to identify the enthalpy and entropy at each respective state. The measured and calculated state properties are then used to calculate the exergy losses and exergetic efficiency.

When considering an experimental setup for validation of the simulation model, deciding on whether to setup the entire system or components of the system is necessary. Component selection is limited by industry availability. Cross flow heat exchangers represent the gas cooler and evaporator of a CO_2 AC system, while the compressor is a non-hermetically sealed type. Also required for complete validation is equipment to measure and log the temperature and pressure data for each state point. High sensitivity and resolution are required in the measuring instrumentation for a more accurate second law analysis.

With the wide range of temperature and pressure differences for the various leak locations, finding suitable sensors to accommodate these needs is challenging. Currently, thermocouples are a cheap and fairly accurate for temperature measurements. Thermocouples use metal properties as the temperature changes to determine the temperature being measured. K-type thermocouples have a range of -200°C – 1350°C, with a sensitivity of ±0.5°C, which is suitable for the validation plan. Pressure transducers are the currently used measuring tool for pressure in FDD systems, with the simplest construction utilizing strain gages. These devices can measure pressure from 0 – 250 bar, with an accuracy of ±0.25%, even in high temperatures. Sensitivity ranges of the sensors are chosen by noticing how effective temperature and pressure differences are on the trend lines produced during leak simulations. If the sensitivity is too large, the fault may not be detected, but on the other hand, if the sensitivity range is too small, then the system could cause a false alarm.

6. Results

Figures 3 through 6 show how each component in the system reacts with regards to exergetic efficiency and exergy losses as the various leaks are applied. The "Leak #" refers to a location in Figure 1 where the leak was applied and the "EE" or "EL" refers to exergetic efficiency or exergy loss, respectively. The exergetic efficiencies reflect a completely opposite response to the exergy losses, which is to be expected. As the exergetic efficiency for a component increases with the loss of mass flow rate of CO_2, the exergy loss decreases. This correlation can be seen in all of the components. Because of this opposing reaction of exergetic efficiencies and exergy losses, it may only be necessary to calculate one of the phenomena for diagnostic purposes.

For the compressor and gas cooler, a leak at either States 2, 5 or 6 cause a decrease in exergetic efficiency (refer to Figures 3 and 4). This reveals a trend that if a leak occurs within the compressor, its efficiency decreases, and therefore works less effectively. Conversely, the internal heat exchanger and evaporator experience an increase in exergetic efficiency for all three of the leak scenarios (refer to Figures 5 and 6). Through these trends, it is observed that knowledge of these trends is required to accurately assess a leak.

The gas cooler, see Figure 3, produced the most obvious trends for each leak scenario. It may therefore be possible to use only the data around the gas cooler to sufficiently diagnose the leaks. The improved model produced differing results from the simplistic model for all components except the gas cooler. The apparent sensitivity of the gas cooler to the effects of system leaks implies that it should be examined further to see what other significance it may have in accurately diagnosing leaks. At the beginning of each trend for each system component is a distinct transient behavior, which stabilizes into a steady trend as the leak continues. This behavior implies that many samples are required to effectively detect and diagnose a leak.

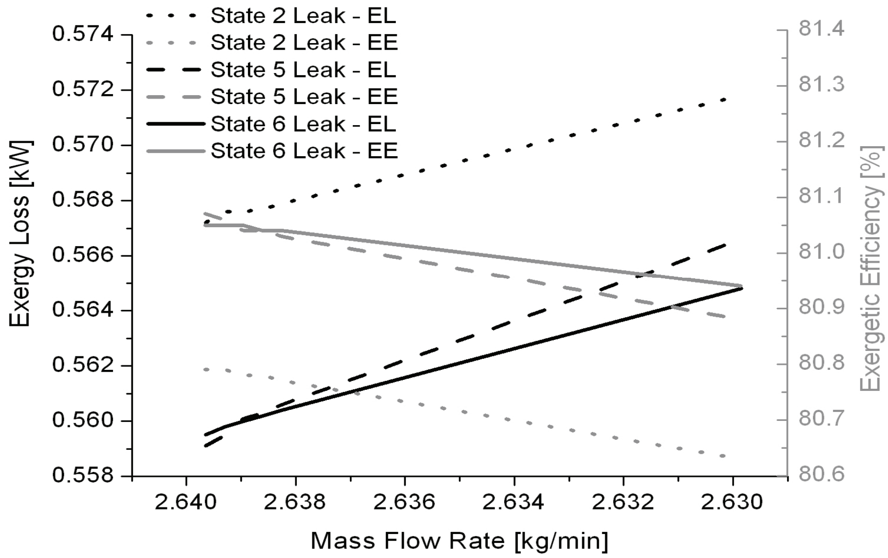

Figure 3. Comparison of Exergy Losses and Exergetic Efficiencies for the Compressor.

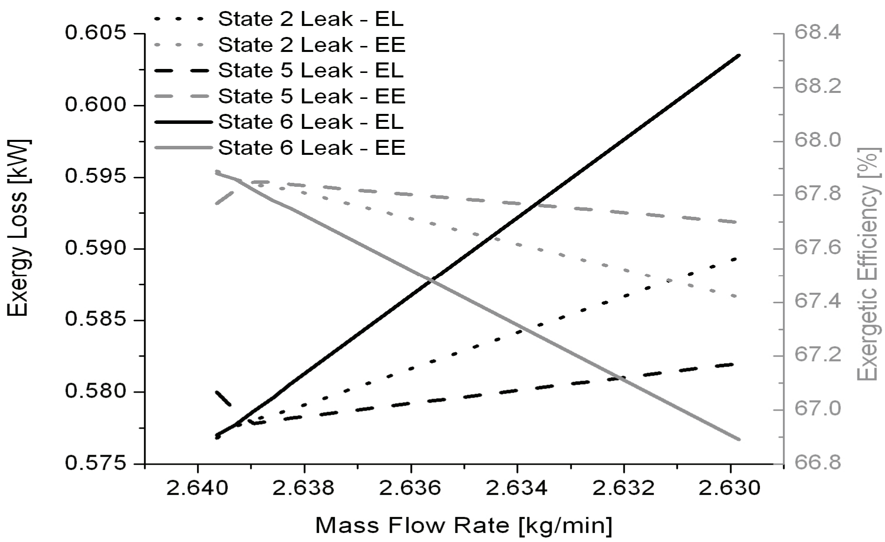

Figure 4. Comparison of Exergy Losses and Exergetic Efficiencies for the Gas Cooler.

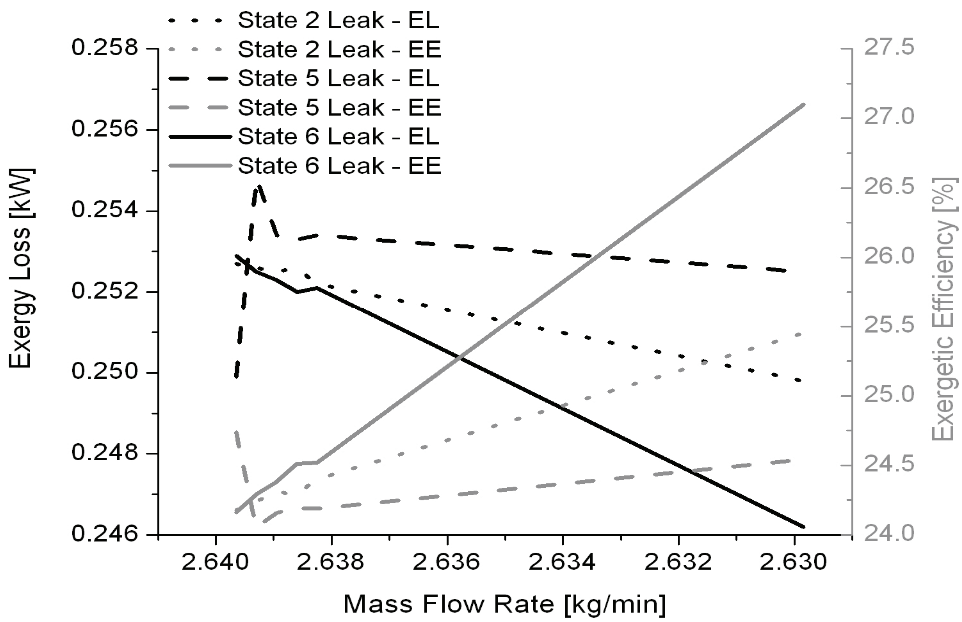

Figure 5. Comparison of Exergy Losses and Exergetic Efficiencies for the Internal Heat Exchanger.

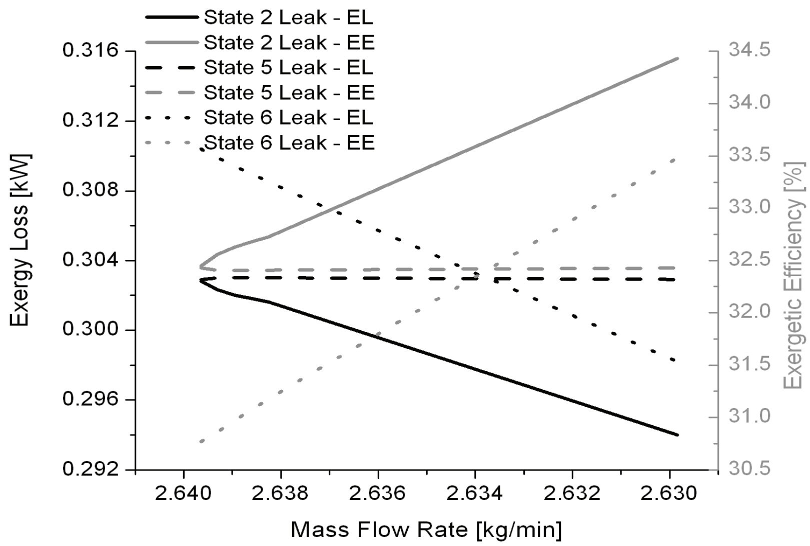

Figure 6. Comparison of Exergy Losses and Exergetic Efficiencies for the Evaporator.

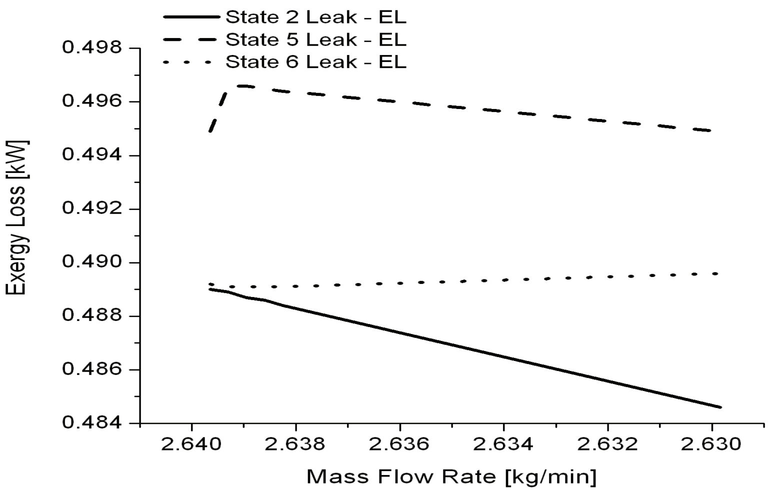

Figure 7. Comparison of Exergy Losses and Exergetic Efficiencies for the Expansion Device.

7. Conclusion

A basis for a more realistic automotive CO_2 AC system is developed. The previously developed EES simulation model is enhanced by justifying and changing gross assumptions. Refrigerant leaks are simulated as mass flow rate losses and are applied to three areas prone to leakage. Trends in the exergetic efficiency and exergy losses are developed in order to create a leak detection and diagnosis system for the CO_2 AC system. When looking at the trends, differences can be seen between how each component reacts to the proposed leaks. Through these trends, it is observed that only one mechanism need be calculated in order to effectively detect and diagnose leaks. Future work includes developing an experimental plan, along with an instrumentation package for a prototype CO_2 AC system, in order to validate the simulation model.

Further research into the implications of the gas cooler results on effectively detecting and diagnosing leaks is recommended. A more in-depth analysis, which includes the consideration of off-design system operation, the implementation of compressor performance curves, and the effect of error in measurements on results, is required for a better understanding of the effectiveness and capability of using an exergetic based leak detection and diagnosis methodology. The construction and implementation of the AC system model presented is required to validate the leak detection and diagnosis methodology.

Acknowledgments

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Nomenclature

e	Exergy [kJ/kg]
Δe	Change in exergy [kJ/kg]
h	Enthalpy [kJ/kg]
ΔH	Change in enthalpy difference [kJ/kg]
$\dot{m}$	Refrigerant mass flow rate [kg/s]
$\Delta\dot{m}$	Change in refrigerant mass flow rate [kg/s]
ΔP	Change in pressure [kPa]
s	Entropy [kJ/kg-K]
T	Temperature [K]
${}_1\dot{W}_2$	Compressor power consumption [kW]
$\dot{W}_s$	Ideal isentropic power consumption [kW]
$\Delta\dot{W}$	Change in compressor power consumption [kW]

1, 2, 3	Cycle path designations
ξ	Exergetic efficiency
η_c	Isentropic efficiency
$\Delta\eta$	Change in isentropic efficiency

Subscripts

c	Compressor
s	Isentropic
0	Environmental or dead state

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