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An experimental analysis of the effect of refrigerant charge level on an automotive refrigeration system

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Abstract — The performance of an automotive refrigeration system is dependent on the refrigerant charge level. Due to inevitable leaks in the system, the amount of refrigerant will decrease over time and thus ultimately reduce the system's performance. A reduction in the amount of refrigerant charge results in excessive compressor cycling, a lower condenser pressure, a higher refrigeration temperature, and an increase in the amount of superheat. This paper identifies and quantifies the individual component losses in an automotive refrigeration system as a function of the refrigerant charge level. A second law analysis, based on nondimensional entropy generation, is carried out to quantify the thermodynamic losses. A passenger vehicle with a cycling-clutch, orifice tube refrigeration system was instrumented to measure various temperatures and pressures, and relative humidity. The data were collected at idle conditions. Thermodynamic equations, which are used to determine the system's thermal performance, are presented. The system's second law efficiency increases 26 % as the amount of refrigerant charge decreases by 44 %. Also the individual component losses are quantified as a function of the refrigerant charge level. The compressor and the condenser losses account for the largest percentage of the total losses, and are of similar magnitude. The evaporator–accumulator and the orifice tube losses account for a smaller percentage of the total losses, and are also of similar magnitude. With a reduction in the refrigerant charge level of 44 %, the losses in the compressor, the condenser, the evaporator–accumulator, and the orifice tube decrease 13 %, 8 %, 10 %, and 33 %, respectively. © 2000 Éditions scientifiques et médicales Elsevier SAS

automotive / refrigeration / charge / entropy / generation

Nomenclature

A	orifice tube cross-sectional area	m^2
C	orifice tube discharge coefficient	
COP	coefficient of performance	
c_P	specific heat at constant pressure	$kJ \cdot kg^{-1} \cdot K^{-1}$
h	specific enthalpy	$kJ \cdot kg^{-1}$
M	refrigerant charge	kg
$\dot{m}$	mass flowrate	$kg \cdot s^{-1}$
P	pressure	$N \cdot m^{-2}$
Q	heat transfer	kJ
$\dot{Q}$	heat transfer rate	kW
R	gas constant	$kJ \cdot kg^{-1} \cdot K^{-1}$
S	entropy transfer	$kJ \cdot K^{-1}$
s	specific entropy	$kJ \cdot kg^{-1} \cdot K^{-1}$
$\dot{S}$	entropy transfer rate	$kW \cdot K^{-1}$
T	temperature	K
t	time	s
$\dot{V}$	compressor volumetric flowrate	$m^3 \cdot s^{-1}$
W	work	kJ

Greek symbols

ϕ	relative humidity	
η_V	compressor volumetric efficiency	
ρ	density	$kg \cdot m^{-3}$
ω	humidity ratio	$kg(v) \cdot kg(a)^{-1}$

Subscripts

a	dry air
air	moist air
c	compressor
Carnot	Carnot
cond	condenser
evap	evaporator
f	flux
g	saturated water vapor
gen	generation
H	high
L	low
ot	orifice tube

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tot	total
v	water vapor
1, 2, ...	system locations (see *figure 1*)

1. INTRODUCTION

An automotive refrigeration system may lose refrigerant over time due to small leaks or due to permeable seals and hoses in the refrigerant lines. A loss of refrigerant can have detrimental effects on the system's performance. In particular, the observable effects are an increase in compressor cycling, a decrease in refrigeration capacity, a decrease in condenser pressure, an increase in refrigeration temperature, and an increase in the amount of superheat. Design engineers normally establish the amount of refrigerant charge so that the system can operate across a wide range of evaporator heat loads and vehicle operating conditions. This requirement makes tradeoffs necessary since there is an optimum charge level for a given operating condition.

Due to ever increasing desires to improve vehicle fuel economy and to reduce vehicle emissions, it is becoming increasingly more important to determine, and more importantly, to reduce the amount of power required to operate the refrigeration system for a fixed amount of cooling. To this end, system losses need to be identified and reduced. Thus, thermodynamic analyses that quantify the system efficiency are needed.

An analysis that is based on the second law of thermodynamics is particularly useful in identifying and quantifying the thermodynamic losses that are related to the amount of refrigerant charge. Other researchers [1–5] have shown the usefulness of the second law of thermodynamics in quantifying system losses. It is the objective of this paper to separate and quantify the significant losses in an automotive refrigeration system as a function of the refrigerant charge level. In particular, a second law analysis, based on entropy generation, will be carried out for various refrigerant charge levels.

This analysis is an extension of previous work by Ratts and Brown [1]. They quantified the entropy generation distribution in an automotive refrigeration system as a function of the compressor cycling frequency. The instrumented vehicle that they used and the second law analysis that they developed are the basis for this work. This paper looks specifically at the effects of the refrigerant charge level on the distribution of the system's losses at idle conditions. The system's response to the loss of refrigerant charge is known as a function of various system temperatures and pressures and also in terms of the compressor cycling rate. The major aim of this paper is to provide a framework for vehicle climate control engineers to quantify, and more importantly, to reduce the thermodynamic losses associated with loss of refrigerant charge.

First, we describe the refrigeration system used in this study and explain its operation. This is followed by a brief presentation of the thermodynamic equations used to evaluate the refrigeration system's performance. We then discuss the experiments and present their data. Finally, we present some conclusions.

2. BACKGROUND

Figure 1 shows a simplified flow schematic of the vehicle air conditioning system used in this study. The refrigeration system is a mechanically driven vapor compression system consisting of a compressor, a condenser, an orifice tube, an evaporator, and a charge reservoir/dehydrator (otherwise known as an accumulator). In this experiment, the passenger compartment airflow is continuously recirculated, that is, the air that flows across the evaporator is drawn from the passenger compartment, and not from outside of the vehicle.

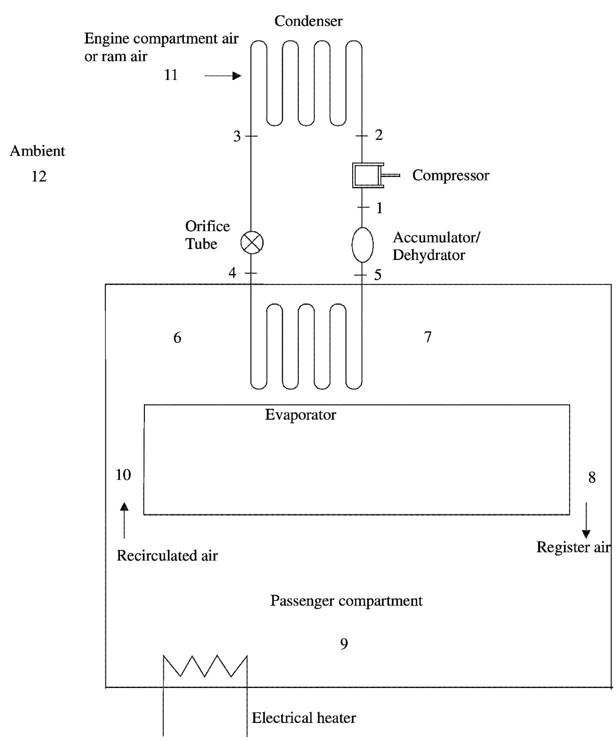

Figure 1. Flow schematic of the air conditioning system.

The refrigeration system was designed to operate under a wide range of heat load conditions. Vehicle designers chose the amount of refrigerant charge so that the system can operate satisfactorily when it is subjected to the largest heat loads. Again, since the system must operate under a wide range of conditions, the capacity of the fixed volume compressor is larger than needed under most operating conditions. The capacity of the compressor is reduced by cycling the compressor on and off based on the low-side refrigerant pressure. The compressor is shut off when the pressure in the accumulator falls below a preset value (chosen to ensure that condensate does not freeze on the evaporator). After the compressor shuts off, there persists a pressure difference across the orifice tube that forces refrigerant to flow from the condenser to the evaporator. As the evaporator fills with refrigerant, its pressure increases. Once the low-side refrigerant pressure reaches the preset level, the compressor restarts. The compressor is continuously turned on and off in this manner; hence, the name cycling-clutch, orifice-tube (CCOT) refrigeration system.

Reducing refrigerant charge affects the system properties. With a smaller amount of refrigerant charge in the evaporator, under favorable ambient conditions, the amount of time required for the evaporator pressure to drop to its threshold value will be less, and hence there will be an increase in the compressor cycling frequency. With a smaller amount of refrigerant charge in the condenser, the maximum condenser pressure will also be lower. The net result will be a reduction in the maximum compressor pressure ratio. This will reduce the required compressor work, and thus will ultimately lead to a reduction in the refrigeration effect. Depending on how these two quantities change relative to one another, the cycle thermal efficiency can either increase or decrease. Also, when the amount of refrigerant charge is reduced, for a fixed evaporator heat load, the superheat at the evaporator outlet will increase. Superheat tends to reduce the compressor inlet density, and thereby tends to reduce the refrigerant mass flowrate. As the refrigerant mass flowrate drops, there will be a reduction in the required compressor power and thus ultimately a reduction in refrigeration capacity. Again, depending on how these two quantities change relative to one another, the cycle thermal efficiency can either increase or decrease. As mentioned earlier, the frequency of compressor cycling also increases as the refrigerant charge level is decreased. Krause and Bullard [6] and Coulter and Bullard [7] have shown that cycling losses can degrade the system performance. Therefore, compressor cycling tends to reduce the cycle thermal efficiency. Thus, tradeoffs need to be made between the above described advantages and disadvantages. These tradeoffs will ultimately decide whether there will be an improvement or degradation in the overall cycle thermal performance. This paper investigates the results of these inevitable tradeoffs.

3. SECOND LAW THERMAL EFFICIENCY

In this section, a second law thermal efficiency is described which is used to compare refrigeration systems that operate between different thermal reservoirs. Ratts and Brown [1] provide the detailed development. Here, their analysis is restated in a condensed form.

The most common way to measure the performance of a refrigeration system is to use a first law efficiency, or coefficient of performance (*COP*), which is a measure of how well the system uses energy. However, when comparing the performance of different systems, or even the same system operating between different thermal reservoirs, one cannot directly compare *COP*s. One way to overcome this shortcoming is to compare the system's performance to that of a reversed Carnot cycle operating between the same thermal reservoirs.

However, a better way to compare systems operating between different thermal reservoirs is to use the concept of entropy generation. A refrigeration system can be thought of as an entropy pump whose task it is to transfer the entropy load from the low-temperature thermal reservoir (the passenger compartment) to the high-temperature thermal reservoir (the ambient). Ideally, the entropy pump should transfer the entropic load without generating any entropy due to dissipative effects. That is, for a given temperature lift, the system efficiency will be greater if it generates a smaller amount of entropy during the transfer of the entropic load. Conversely, for a given amount of entropy generation, the system efficiency will be greater if the temperature lift is smaller. However, when comparing systems that operate between different thermal reservoirs, one cannot directly compare the amounts of entropy generation. Fortunately, one way to overcome this shortcoming is to compare the entropy generation to the entropic load.

Ratts and Brown, after combining a refrigeration system entropy balance, the definition of the *COP*, and the *COP* equation for a reversed Carnot cycle, showed that

$$\frac{S_{\mathrm{gen,tot}}}{S_{\mathrm{L}}} = \frac{1/COP + 1}{1/COP_{\mathrm{Carnot}} + 1} - 1 \tag{1}$$

where $S_{\mathrm{gen,tot}}$ is the total entropy generation for the system and S_{L} is the entropic load. Equation (1) shows the equivalence of the ratio $S_{\text{gen,tot}}/S_{\text{L}}$ to COP/COP_{Carnot}. Since COP/COP_{Carnot} is used as a standard measure of system performance, then so also can the entropic generation ratio be used as a standard measure of system performance. Equation (1) is used in this paper to compare overall system performances.

After it has been identified that one system is better than another is, it is insightful to identify, to quantify, and more importantly, to reduce the thermodynamic losses. Quantifying and comparing thermal and mechanical dissipation losses through an energy analysis, however, is difficult to accomplish. On the other hand, an entropy generation analysis is a convenient way to compare thermal and mechanical dissipation on an even footing.

Equation (1) gives the basis for comparing individual component losses in systems that operate between different thermal reservoirs. Total entropy generation is an additive quantity; that is, the total entropy generation is simply the sum of the entropy generation in each of the individual components. Therefore, if the ratio of the entropy generation to the entropic load can be compared for two systems that operate between different thermal reservoirs, then so also can the ratio of component entropy generation to the entropic load be compared for two systems that operate between different thermal reservoirs. This is the fundamental and thermodynamic basis for the analysis presented in this paper.

4. SYSTEM ANALYSIS

Again for completeness of this paper, we present the system equations, developed by Ratts and Brown, needed to determine the refrigeration system performance. They developed their equations by applying the principles of mass and energy conservation, and the increase in entropy principle.

The first step in the analysis is to ensure conservation of mass over a cycle. First of all, the mass flux is defined in this paper as the integrated amount of mass that passes through a point in the system over a cycle. For cyclical steady state, the mass flux at any point in the system must be constant from cycle to cycle, and it must also be equal to the mass flux at every other point in the system. It can be expressed mathematically as

$$\oint \dot{m}\,\mathrm{d}t = m_{\mathrm{f}} \tag{2}$$

where $\dot{m}$ is the instantaneous mass flowrate, $\mathrm{d}t$ is a differential amount of time, and m_{f} is the mass flux over the cycle. The instantaneous mass flowrate through the compressor, $\dot{m}_{\mathrm{c}}$, is given by

$$\dot{m}_{\mathrm{c}} = \rho_1 \dot{V}_{\mathrm{c}} \eta_{\mathrm{V}} \tag{3}$$

where ρ_1 is the refrigerant density at the compressor inlet (state 1 in *figure 1*), $\dot{V}_{\mathrm{c}}$ is the ideal volumetric flowrate (compressor displacement volume multiplied by the compressor speed) through the compressor, and η_{V} is the compressor volumetric efficiency.

The instantaneous mass flowrate through the orifice tube $\dot{m}_{\mathrm{ot}}$ is calculated by

$$\dot{m}_{\mathrm{ot}} = CA\sqrt{\rho_3(P_3 - P_4)} \tag{4}$$

where C is the orifice tube discharge coefficient, A is the minimum cross-sectional area of the orifice tube, ρ_3 is the refrigerant density at the orifice tube inlet, and P_3 and P_4 are the pressures at the inlet and outlet of the orifice tube, respectively. Substituting equations (3) and (4) into equation (2), the relationship between the effective orifice tube discharge coefficient and the effective volumetric flowrate is

$$\eta_{\mathrm{V}} \oint \rho_1 \dot{V}_{\mathrm{c}}\,\mathrm{d}t = CA \oint \sqrt{\rho_3(P_3 - P_4)}\,\mathrm{d}t \tag{5}$$

The second step in the analysis is to ensure conservation of energy; that is, energy must be conserved over a cycle. The cycle energy balance for the system is

$$W_{\mathrm{c}} = Q_{\mathrm{cond}} - Q_{\mathrm{evap}} \tag{6}$$

where W_{c} is the compressor work per cycle, Q_{cond} is the condenser heat transfer per cycle, and Q_{evap} is the evaporator heat transfer per cycle.

The energy absorbed by the refrigerant in the evaporator is equal to the energy removed from the air, that is,

$$Q_{\mathrm{evap}} = \oint \dot{Q}_{\mathrm{evap}}\,\mathrm{d}t = \oint \rho_6 \dot{V}_{\mathrm{a}}(h_6 - h_7)\,\mathrm{d}t \tag{7}$$

where $\dot{Q}_{\mathrm{evap}}$ is the instantaneous heat transfer rate from the air to the refrigerant in the evaporator, ρ_6 is the density of the air at the evaporator inlet, $\dot{V}_{\mathrm{a}}$ is the steady volumetric flowrate of the air, and h_6 and h_7 are the moist air enthalpies at the evaporator inlet and outlet, respectively. The enthalpy of the moist air, h_{air}, is given by

$$h_{\mathrm{air}} = h_{\mathrm{a}}(T) + \omega h_{\mathrm{v}}(T) \tag{8}$$

where T is the air temperature, $h_{\mathrm{a}}(T)$ is the dry air enthalpy, ω is the humidity ratio, and $h_{\mathrm{v}}(T)$ is the water vapor enthalpy.

The energy absorbed by the refrigerant in the evaporator based on the refrigerant mass flowrate is

$$Q_{\mathrm{evap}} = \oint \dot{m}_{\mathrm{c}} h_1 \,\mathrm{d}t - \oint \dot{m}_{\mathrm{ot}} h_4 \,\mathrm{d}t \tag{9}$$

where h_1 and h_4 are the refrigerant enthalpies at the outlet of the accumulator and the inlet to the evaporator, respectively. The accumulator is assumed to be adiabatic. If equations (3)–(5) are substituted into equation (9), then the energy absorbed by the refrigerant can be expressed as

$$Q_{\mathrm{evap}} = \eta_{\mathrm{V}} \left\{ \oint \rho_1 \dot{V}_{\mathrm{c}} h_1 \,\mathrm{d}t - \frac{\oint \rho_1 \dot{V}_{\mathrm{c}} \,\mathrm{d}t}{\oint \sqrt{\rho_3 (P_3 - P_4)} \,\mathrm{d}t} \oint \sqrt{\rho_3 (P_3 - P_4)}\, h_4 \,\mathrm{d}t \right\} \tag{10}$$

Equations (7) and (10) can be set equal to one another to determine the average compressor volumetric efficiency. Once the compressor volumetric efficiency is determined, then the instantaneous compressor mass flowrate, the instantaneous orifice tube mass flowrate, and the average orifice tube discharge coefficient can be calculated from equations (3), (4), and (5), respectively. In addition, the heat transfer from the refrigerant in the condenser to the air flowing across the condenser is given by

$$Q_{\mathrm{cond}} = \oint \dot{m}_{\mathrm{c}} h_2 \,\mathrm{d}t - \oint \dot{m}_{\mathrm{ot}} h_3 \,\mathrm{d}t \tag{11}$$

The third step in the analysis is to write an entropy balance for the system. The amount of entropy generated by the system is simply the difference between the entropic load and the entropy transferred to the high-temperature thermal reservoir. Mathematically this can be expressed as

$$\frac{S_{\mathrm{gen,tot}}}{S_{\mathrm{L}}} = \frac{S_{\mathrm{H}}}{S_{\mathrm{L}}} - 1 \tag{12}$$

where S_{H} is the entropy transfer per cycle from the refrigerant in the condenser to the ambient air. Note that the ratio $S_{\mathrm{gen,tot}}/S_{\mathrm{L}}$ is given by equation (1). The ratio is determined experimentally from the entropic load, S_{L}, and the condenser entropy transfer, S_{H}.

The entropy load is given by

$$S_{\mathrm{L}} = \oint \rho_6 \dot{V}_{\mathrm{a}} (s_6 - s_7) \,\mathrm{d}t \tag{13}$$

where s_6 and s_7 are the specific entropies of the moist air at the inlet and outlet of the evaporator, respectively. The specific entropy of the moist air, s_{air}, is given by

$$s_{\mathrm{air}} = c_{P,\mathrm{a}} \ln(T) + \omega \left[s_{\mathrm{g}}(T) + R_{\mathrm{v}} \ln(\phi) \right] \tag{14}$$

where $c_{P,\mathrm{a}}$ is the dry air's specific heat at constant pressure, T is the moist air temperature, $s_{\mathrm{g}}(T)$ is the saturated water vapor specific entropy, R_{v} is the water vapor gas constant, and ϕ is the moist air relative humidity. The entropy transfer from the condenser to the high-temperature thermal reservoir is

$$S_{\mathrm{H}} = \oint \frac{\dot{Q}_{\mathrm{cond}}}{T_{12}} \,\mathrm{d}t = \frac{Q_{\mathrm{cond}}}{T_{12}} \tag{15}$$

where $\dot{Q}_{\mathrm{cond}}$ is the instantaneous heat transfer rate from the refrigerant in the condenser to the ambient air. The heat transfer from the condenser is given by equation (11).

The entropy generation in each of the system components can now be calculated. The entropy generation in the evaporator is

$$S_{\mathrm{gen,evap}} = \oint \left(\dot{m}_{\mathrm{c}} s_1 - \dot{m}_{\mathrm{ot}} s_4 \right) \mathrm{d}t - S_{\mathrm{L}} \tag{16}$$

The entropy generation in the compressor is

$$S_{\mathrm{gen,c}} = \oint \dot{m}_{\mathrm{c}} (s_2 - s_1) \,\mathrm{d}t \tag{17}$$

The entropy generation in the orifice tube is

$$S_{\mathrm{gen,ot}} = \oint \dot{m}_{\mathrm{ot}} (s_4 - s_3) \,\mathrm{d}t \tag{18}$$

The entropy generation in the condenser is

$$S_{\mathrm{gen,cond}} = \oint \left(\dot{m}_{\mathrm{c}} s_2 - \dot{m}_{\mathrm{ot}} s_3 \right) \mathrm{d}t - S_{\mathrm{H}} \tag{19}$$

5. EXPERIMENTAL METHOD

The air conditioning system of a passenger vehicle was instrumented to measure the refrigerant temperature and pressure at several locations and the air temperature and relative humidity at several locations. The refrigerant temperature and pressure were measured at locations 1–5 as defined in *figure 1*. The accuracy of the pressure measurements was ±0.034 bar with temperature compensation over the range from 1 °C to 85 °C. The refrigerant temperatures were measured with T-type thermocouples in stainless steel thermowells. The thermocouples were calibrated to provide an accuracy of ±0.5 K. The engine speed was monitored using a tachometer with an accuracy of ±5 rpm. The air temperatures were measured at locations 6–12 as defined in *figure 1*. They were measured with exposed soldered-joint, T-type thermocouples. The thermocouples were calibrated to provide an accuracy of ±1.0 K. The relative humidity (rh) of the air was measured at locations 6, 8, and 11 as defined in *figure 1*. The accuracy of the relative humidity measurements was ±2 % rh over the range from 5 % rh to 95 % rh. The volumetric flowrate of the air was determined by averaging velocity measurements taken downstream of the evaporator using a hot-wire anemometer.

The vehicle was tested with different amounts of refrigerant-134a placed into the refrigeration system. Between vehicle tests, the refrigeration system was completely evacuated. Firstly, any oil removed during the evacuation procedure was added back into the system, and then finally the desired amount of refrigerant was added to the system, which will be referred to as the refrigerant charge M in the remainder of the paper. The air conditioning system was operated in the recirculation mode; that is, no fresh air was introduced into the passenger compartment. All vehicle tests were performed at idle conditions. An electric heater placed in the passenger compartment simulated the vehicle heat load. Measurements were taken only after the refrigeration system had reached a cyclical steady-state operating condition.

6. RESULTS AND DISCUSSION

Experimental data were recorded and processed so that the system losses could be evaluated as a function of the refrigerant charge level. The raw data were processed using the equations developed in the System Analysis section. The R-134a properties were determined with the software CATT [8].

Before the processed results are presented, we will provide a brief description of some of the raw temperature data. *Figures 2a*, *2b*, and *2c* are plots of the instantaneous temperatures measured at refrigerant charge levels of 0.77 kg, 0.52 kg, and 0.43 kg, respectively (collectively these tests will be referred to as trial #1). The figures include the bulk interior air temperature, the ambient air temperature, and the air temperature at the register exits. The figures also show refrigerant temperatures at the condenser and orifice tube exits. The refrigerant temperature at the condenser exit reflects the degree of compressor cycling. If *figures 2a*, *2b*, and *2c* are compared, we see that a decrease in the refrigerant charge level leads to an increase in the compressor cycling frequency. In *figure 2a* the compressor is not cycling; the system is running continuously. Also as the refrigerant charge level decreases, the condenser's saturation temperature (and pressure) also decreases. In the same way, as the refrigerant charge level decreases, both the register air temperature and the bulk average interior air temperature increases.

Quantitatively, *table I* identifies the effects of the refrigerant charge level on some of the system properties. The time-averaged temperatures in *figures 2a*, *2b*, and *2c* are shown in columns (2)–(6). A decrease in the refrigerant charge level from 0.77 kg to 0.52 kg resulted in an increase in the evaporator temperature of 4 K, an increase in the register air temperature of 5 K, and an increase in the bulk interior air temperature of 5 K. The condenser temperature decreased 11 K. A further decrease in the refrigerant charge level from 0.52 kg to 0.43 kg resulted in smaller changes in the time-averaged properties. The evaporator temperature had no significant change, the register air temperature increased 4 K, the bulk interior air temperature increased 3 K, and the condenser temperature decreased 1 K.

TABLE I
System average properties for trial #1.

(1) Refrigerant charge level [kg]	(2) Condenser refrigerant exit temp. [°C]	(3) Ambient air temp. [°C]	(4) Interior air temp. [°C]	(5) Register air temp. [°C]	(6) Evaporator refrigerant inlet temp. [°C]	(7) Compressor cycling rate [rad·s^{-1}]	(8) Entropic condenser refrigerant temp. [°C]	(9) Entropic evaporator refrigerant temp. [°C]	(10) Condenser refrigerant superheat temp. [°C]	(11) Evaporator refrigerant superheat temp. [°C]
0.77	44	23	12	12	−3	0	52	−3	24	−2
0.52	33	21	17	17	1	0.22	43	3	39	10
0.43	32	22	20	21	1	0.45	42	8	45	25

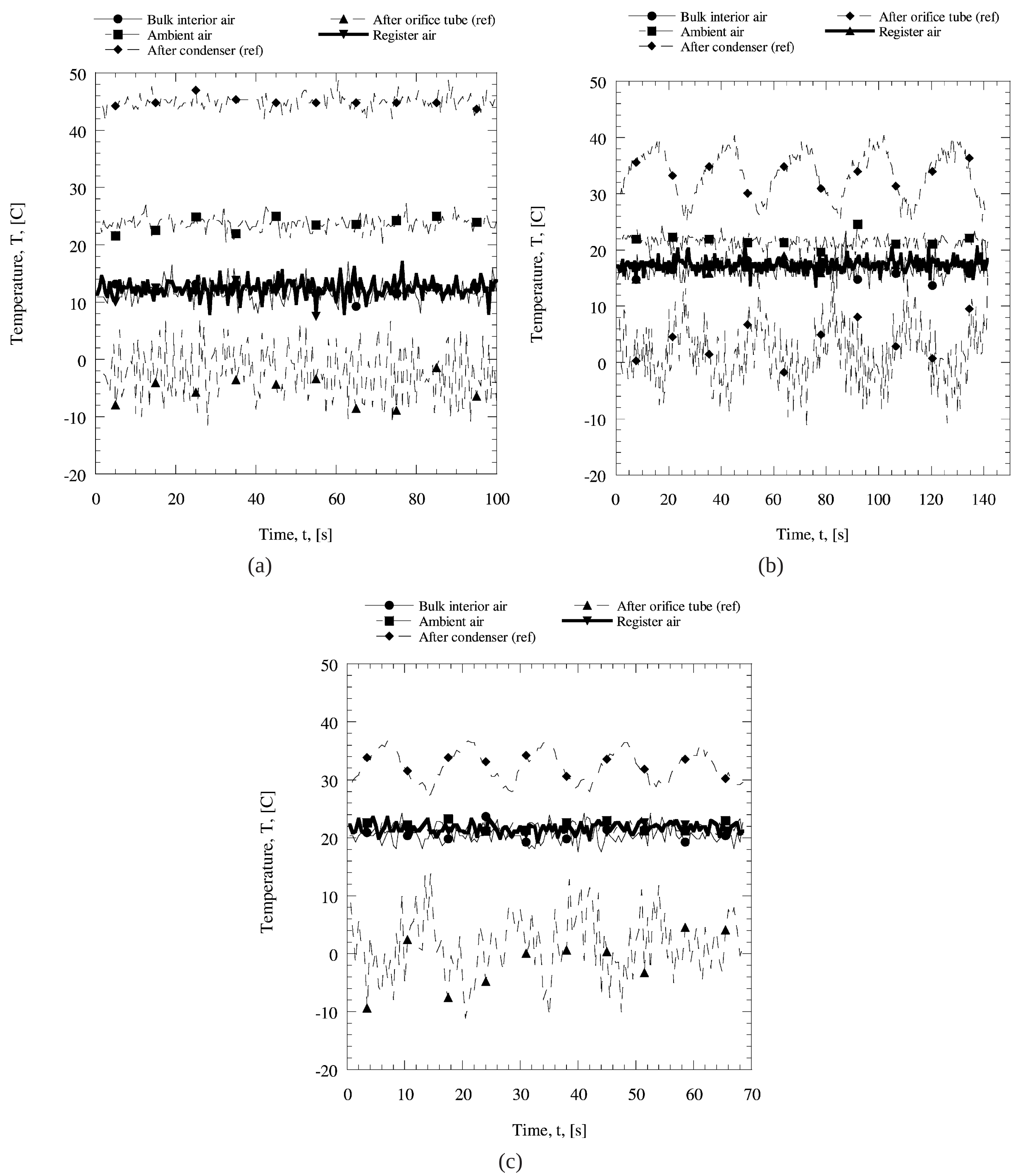

Figure 2. Measured temperatures (trial #1) (a) for the 0.77 kg case, (b) for the 0.52 kg case and (c) for the 0.43 kg case.

The most noticeable effect is the increase in the frequency of compressor cycling as the refrigerant charge level is reduced. For the 0.77 kg case, there is no compressor cycling. At this condition, the evaporator pressure is not lowered to a sufficient value by the compressor to induce cycling. In other words, the compressor did not "starve" the evaporator of refrigerant. For the 0.52 kg and 0.43 kg cases, the compressor cycling rate is 0.22 $rad \cdot s^{-1}$ and 0.45 $rad \cdot s^{-1}$, respectively (column (7)). If the compressor is operating at a reduced refrigerant charge level, the ratio of the net amount of mass drawn from the evaporator by the compressor to the net amount of mass left in the evaporator will increase. This reduces the time necessary for the evaporator pressure to reach a sufficient value to induce compressor cycling.

The condenser refrigerant properties are also of considerable interest. As the compressor cycling rate increases, the net amount of mass transferred into the condenser decreases, resulting in a decrease in the condenser pressure. The condenser pressure is directly related to the saturation temperature of the refrigerant in the condenser. Therefore, the condenser's maximum saturation temperature decreases as the refrigerant charge level decreases. This effect can be seen in the figures for the condenser refrigerant exit temperature. The exit state is not subcooled. Another interesting effect is the reduction in the time rate of change of the condenser exit temperature with respect to the refrigerant charge level. The instantaneous slope of the condenser exit temperature, just prior to when the compressor shuts off, is a measure of the system's nearness to steady state. If the slope is zero, the system is operating at steady state. As the amount of refrigerant charge is reduced, the instantaneous time rate of change of the condenser exit temperature, just prior to when the compressor shuts off, increases.

The time-averaged temperatures, however, are not truly representative of the system's performance. In actuality, energy transfers and entropy transfers are needed to determine the system performance. Therefore, if the temperatures are weighted by the energy and entropy transfers, they will be more representative of the actual system performance. Amrane and Rademacher [4] defined an entropic-averaged temperature as the ratio of the energy transferred to the entropy transferred. Column (8) (*table I*) shows the refrigerant entropic temperature for the condenser based on this definition. This temperature is simply the ratio of the energy transferred from the condenser per cycle to the entropy transferred from the condenser per cycle. Similarly, column (9) is the refrigerant entropic temperature in the evaporator. For the 0.77 kg case, the entropic condenser temperature of 52 °C is 8 K larger than the time-averaged condenser temperature of 44 °C. The reason for this difference is that the entropic temperature takes into account the superheated refrigerant in the condenser entrance region. The time-averaged superheat entering the condenser (column (10)) was approximately 24 °C, 39 °C, and 45 °C, for the 0.77 kg, 0.52 kg, and 0.43 kg cases, respectively. These time-averaged temperature values seem high because the temperature of each fluid element is weighed evenly when in fact they should have been weighed with respect to the amount of energy and entropy that they transferred. The entropic temperature does exactly this. For the 0.77 kg case, the entropic evaporator temperature of −3 °C is the same as the time-averaged evaporator temperature. This suggests, in contrast to the condenser, that the fluid in the evaporator–accumulator was not superheated, but was saturated. As the refrigerant charge level decreased from 0.77 kg to 0.52 kg, the entropic evaporator temperature increased 6 K from −3 °C to 3 °C. Then as the refrigerant charge level was decreased further from 0.52 kg to 0.43 kg, the entropic evaporator temperature increased 5 K from 3 °C to 8 °C. The entropic temperature takes into account the superheated fluid exiting the evaporator–accumulator. The refrigerant superheat at the accumulator exit was −2 °C, 9 °C, and 26 °C for the 0.77 kg, 0.52 kg, and 0.43 kg cases, respectively. Recall the earlier described changes in register air temperatures (or bulk interior air temperatures) of 5 K and 4 K. From this, we can conclude that the entropic temperature is indicative of the system's performance. Also note that the difference between the entropic temperatures, that is, the difference between the entropic condenser temperature and the entropic evaporator temperature, decreases as the refrigerant charge level decreases. For the 0.77 kg, 0.52 kg, and 0.43 kg cases, the temperature differences were 55 K, 40 K, and 34 K, respectively. As the entropic temperature difference decreases, the system efficiency increases; however, it increases at the expense of an increase in the bulk interior air temperature.

Another trial was performed to verify these trends. The temperature time-history plots for what will be collectively referred to as trial #2 are shown in *figures 3a*, *3b*, and *3c*. The refrigerant charge levels are 0.68 kg, 0.54 kg, and 0.41 kg, respectively. This trial had similar trends to the data of trial #1. In particular, as the refrigerant charge level decreases, the condenser pressure decreases, the register air temperature increases, and the frequency of compressor cycling increases. Also just as in trial #1, the instantaneous time rate of change of the condenser temperature increased, just prior to when the compressor shut off, as the refrigerant charge level decreased.

Table II presents the time-averaged and entropic-averaged temperatures for these figures. The results for the 0.54 kg case and for the 0.41 kg case are very similar. The 0.41 kg case has a discrepancy somewhere in the data of the bulk interior air temperature and/or the register air temperature. The register air temperature cannot be warmer than the bulk interior air temperature; there-

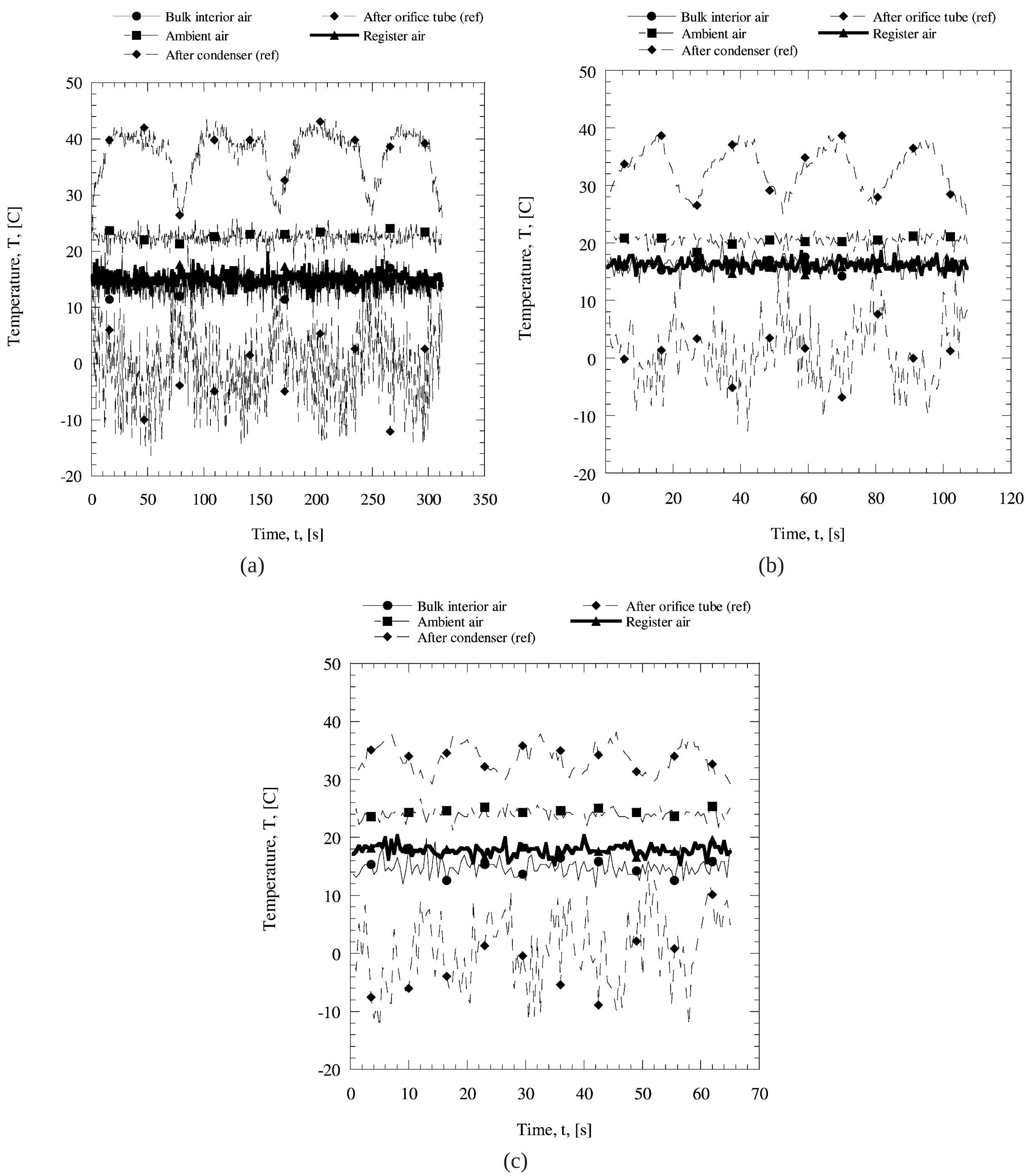

Figure 3. Measured temperatures (trial #2) (a) for the 0.68 kg case, (b) for the 0.54 kg case and (c) for the 0.41 kg case.

fore, this case will not be used in the following comparison. Other than this discrepancy, the data appears fine in terms of the trends seen. Thus, a comparison between the 0.68 kg case and the 0.54 kg case will be made to generalize the trends. As the refrigerant charge level was reduced from 0.68 kg to 0.54 kg, the evaporator temperature in-

TABLE II
System average properties for trial #2.

(1) Refrigerant charge level [kg]	(2) Condenser refrigerant exit temp. [°C]	(3) Ambient air temp. [°C]	(4) Interior air temp. [°C]	(5) Register air temp. [°C]	(6) Evaporator refrigerant inlet temp. [°C]	(7) Compressor cycling rate [rad·s^{-1}]	(8) Entropic condenser refrigerant temp. [°C]	(9) Entropic evaporator refrigerant temp. [°C]	(10) Condenser refrigerant superheat temp. [°C]	(11) Evaporator refrigerant superheat temp. [°C]
0.68	37	22	14	15	−1	0.07	46	−3	33	0
0.54	33	20	16	16	1	0.25	43	0	41	10
0.41	33	24	15	18	0	0.52	44	0	45	23

creased 2 K, the register air temperature increased 1 K, the bulk interior air temperature increased 2 K, and the condenser temperature decreased 4 K. The decrease in the condenser temperature is attributed to the decrease in the refrigerant charge level. Once again, the time rate of change of the condenser temperature decreased, just prior to when the compressor shut off, as the refrigerant charge level was decreased. For the 0.68 kg case, the system was able to reach a steady state operating condition. For the 0.54 kg case, the system never reached steady state. This led to a decrease in the condenser temperature. Also note that there is an increase in the frequency of compressor cycling from 0.07 rad·s^{-1} to 0.25 rad·s^{-1} as the refrigerant charge level decreases. The entropic condenser temperature decreased 3 K and the entropic evaporator temperature increased 3 K. Once again, when the entropic temperature differences are compared, the system with the lesser amount of refrigerant charge had a smaller entropic temperature difference.

The system efficiencies are compared using a second law analysis for these two trials as well as for a third trial. The overall system performances are shown in *figure 4*. As the refrigerant charge level decreases, the value of the second law efficiency parameter decreases, thus indicating that the system's efficiency increases. As the refrigerant charge level was decreased by 44 %, the thermodynamic losses decreased by 26 %. This increase in the system efficiency was gained at the expense of an increase in the bulk interior air temperature for a relatively constant thermal load.

Figure 5 shows the loss distribution for the individual components for the 0.54 kg and 0.68 kg cases. For the 0.68 kg case, the compressor accounts for the largest loss, followed by the condenser loss, the evaporator–accumulator loss, and finally the orifice tube loss. The condenser and compressor losses were of the same magnitude. The evaporator–accumulator and the orifice tube losses were also of a similar magnitude, but much

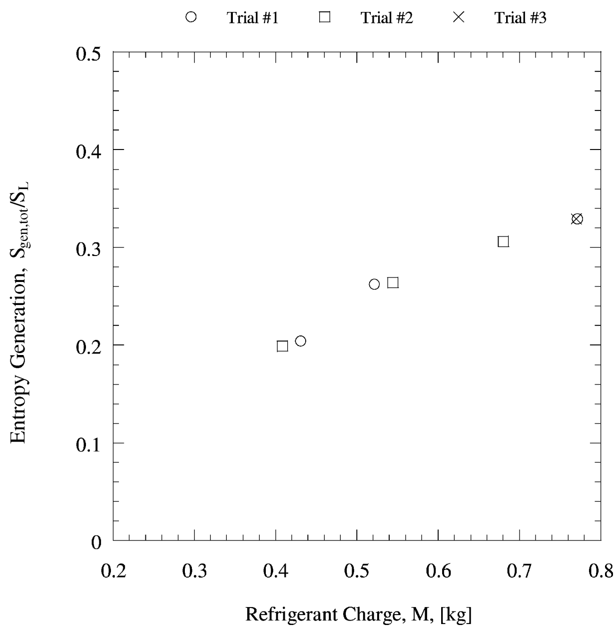

Figure 4. Total entropy generation.

smaller than the compressor and condenser losses. The losses of the compressor, the condenser, the evaporator–accumulator, and the orifice tube were 0.13, 0.098, 0.047, and 0.035, respectively. A decrease in the refrigerant charge level from 0.68 kg to 0.54 kg reduced the compressor, the condenser, the evaporator–accumulator, and the orifice tube losses by 13 %, 8 %, 10 %, and 33 %, respectively. All of the losses reduced by a nearly equal percentage, except for the percentage reduction in the orifice tube loss, which was of a much greater magnitude. The entropy generation within the orifice tube is strongly dependent on the inlet fluid state. As the average inlet temperature reduces so does the effective discharge coefficient and ultimately the entropy generation. Also as previously stated, the difference in the saturation temper-

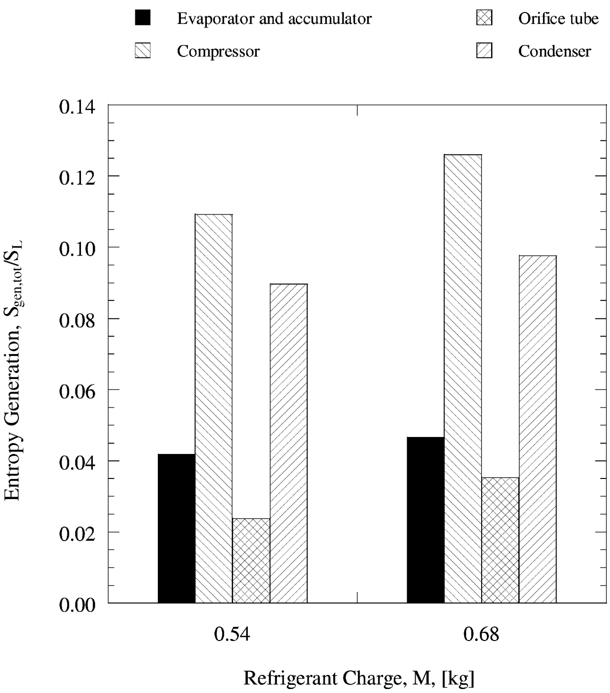

Figure 5. Effect of refrigerant charge on the component entropy generation distribution relative to the entropic load (trial #2).

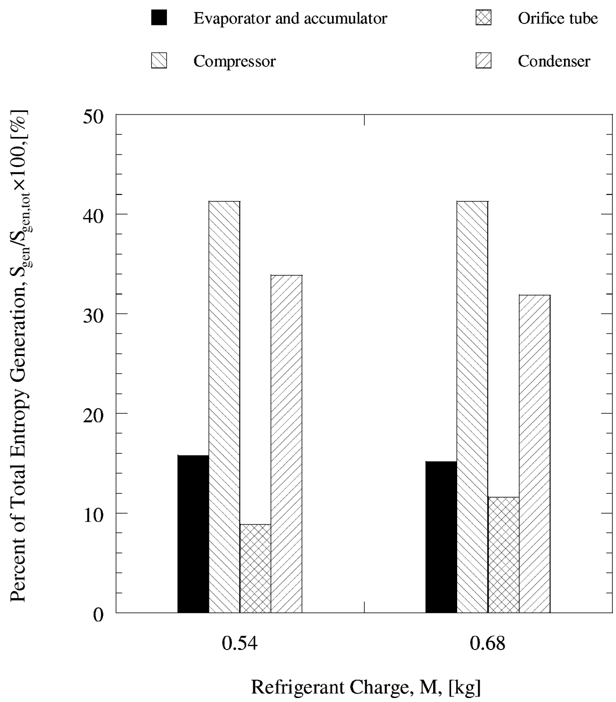

Figure 6. Effect of refrigerant charge on the component entropy generation distribution relative to the total entropy generation (trial #2).

ature in the condenser and in the evaporator is strongly dependent on the amount of refrigerant charge. Therefore, there is a strong dependence between the orifice tube loss and the refrigerant charge level. We also expected that the compressor losses would decrease more rapidly since the compressor losses are also dependent on the temperature ratio. This, however, was not the case. It is possible that there was a tradeoff between losses in the compressor that kept the percentage reduction lower than expected. *Figure 6* shows the same information as *figure 5*, but plots the results in terms of a ratio of the component entropy generation to the total system entropy generation. When the 0.68 kg and 0.54 kg cases are compared, there are no significant effects noted other than the decrease in orifice tube losses as the refrigerant charge level decreases.

The system capacity for trial #1 is shown in *table III*. The table lists the time-averaged mass flowrate, the time-averaged refrigeration load, and the time-averaged entropic load. As the refrigerant charge level decreases, both the refrigeration and entropic loads increase while the mass flow rate remains approximately constant. An electric heater imposed a thermal load in the passenger compartment. As the heater operated at the higher temperature levels, its thermal resistance increased dissipating more thermal energy into the passenger compartment. We expect that as the refrigerant charge level is reduced there will be a decrease in the refrigeration capacity. Since the thermal load was increased, the level of superheat had to adjust to meet the imposed thermal load. In this study, the refrigeration capacity is defined as the amount of latent heat transfer that occurs in the evaporator. Similarly, the entropic capacity is defined as the latent entropy transfer in the evaporator. *Figure 7* is a plot of the latent entropy transfer in the evaporator as a function of the refrigerant charge level. For the 0.77 kg case, there is no superheat. Thus, all of the entropy transfer was latent. As the refrigerant charge level decreases, there are

TABLE III
System loads for trial #1.

Refrigerant charge M [kg]	Refrigerant mass flowrate m^* [kg·h^{-1}]	Refrigerant load Q^*_L [kW]	Entropic load $S^*_L \cdot 10^3$ [kW·K^{-1}]
0.77	51	1.9	6.8
0.52	47	2.1	7.4
0.43	52	2.5	8.7

* Time-averaged.

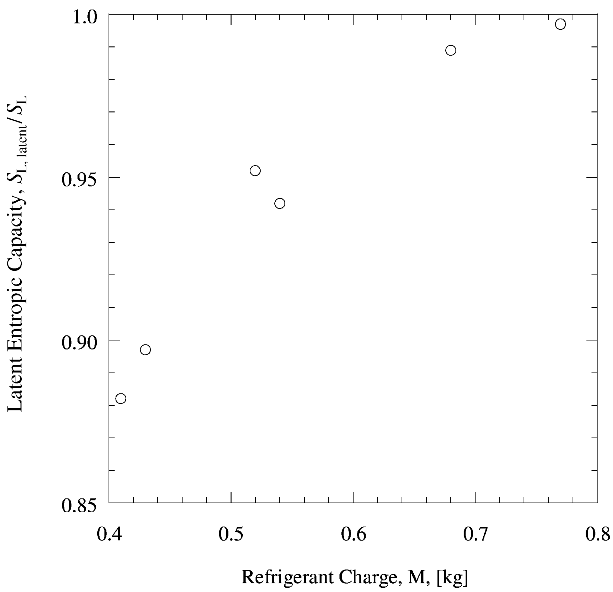

Figure 7. Effect of refrigerant charge on entropic capacity (trial #1).

increasing levels of superheat at the evaporator outlet. As was previously stated, an increase in the level of superheat leads to a reduction in the refrigeration capacity. At the lowest charge level, approximately 12 % of the total entropy transfer is sensible, resulting in a significant reduction in the refrigeration capacity.

7. CONCLUSIONS

Thermodynamic losses in a CCOT automotive refrigeration system, at idle conditions, have been quantified as a function of the refrigerant charge level. The losses are identified and quantified in terms of the ratio of entropy generation to the entropic load. A second law analysis provides the framework for discussing the system performance. The second law analysis identifies the location of the largest losses and the trends in the distribution of these losses as a function of the refrigerant charge level. The compressor and the condenser account for the largest percentage of the total losses, and are of a similar magnitude. The evaporator–accumulator and the orifice tube losses account for a smaller percentage of the total losses, and are also of similar magnitude.

The experimental results show that the system is more efficient as the refrigerant charge level decreases; however, this is accomplished at the expense of increased refrigeration temperature and decreased refrigeration capacity. The refrigerant temperature ratio decreases as the amount of refrigerant decreases, ultimately leading to an increase in the system efficiency. As the refrigerant charge level decreases, all of the losses, except for the orifice tube loss, decrease uniformly. The orifice tube loss decreases more rapidly than do the other losses because of its strong dependence on the refrigerant temperature ratio. Although compressor work is also strongly dependent on the refrigerant temperature ratio, the compressor losses do not exhibit nearly as strong of a dependence on temperature as do the orifice tube losses. There may be a tradeoff between the losses in the compressor that affects its net result. The temperature difference between the thermal reservoirs and the refrigerant in the heat exchangers decreases as the refrigerant charge level decreases. Also, the temperature difference between the bulk interior air temperature and the ambient air temperature decreases as the refrigerant charge level decreases. These decreases lead to smaller amounts of thermal dissipation in the heat exchangers. An interesting result is that the amount of superheat does not play a significant role in determining the system efficiency. As the refrigerant charge level decreases, the amount of superheat at the evaporator outlet and at the condenser inlet increases significantly. These are sources of irreversibility; however, the overall data trends show that the system is more efficient as the amount of refrigerant charge decreases.

The known variations in system properties with respect to reduced refrigerant charge levels are observed. In particular, the amount of compressor cycling increases, the refrigeration capacity decreases, the condenser exit temperature decreases, the refrigeration temperature increases, and the amount of superheat increases, as the refrigerant charge level decreases. In addition, we have shown that the time rate of change of the condenser pressure (saturation temperature) is useful in identifying low refrigerant charge. As the refrigerant charge level decreases, the time rate of change of the condenser exit temperature (saturation temperature), just prior to when the compressor shuts off, is a measure of how far the system is from a steady state (noncycling) operating point. The time rate of change, as well as the absolute magnitude, of the condenser saturation temperature can be used to identify low refrigerant charge levels, or can be used to identify how far away the system is from a steady state operating condition.

This paper also shows that entropy-based properties are more useful than time-averaged properties since time-averaged properties do not take into account energy and entropy transfers. Entropy-based properties allow one to account for superheating and cycling effects. In fact, we show that the entropic temperature can identify the reduction or improvement in the thermodynamic efficiency. The entropic temperature also helps quantify the amount of thermal dissipation in the heat exchangers.

The investigation presented in this paper suggests that active charge control may be one way to achieve control of register air temperature as well as control of system capacity.

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