Theoretical Performance Evaluation of a Carbon Dioxide based Environmental Control Unit (ECU) with Microchannel Heat Exchangers

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Abstract

Environmental control units (ECUs) are used by the U.S. Army to provide air conditioning for field installation and personnel. Currently, these units use R-22 as the refrigerant. Based on the upcoming phase out of R-22, an initiative is under way by the U.S. Army to find a suitable replacement technology. A performance comparison study is reported here based on the simulation results of a carbon dioxide based ECU and the experimental results of a standard R-22 based ECU. The simulation of the carbon dioxide based ECU was performed using ACCO2, a computer code that simulates transcritical carbon dioxide based air conditioning and refrigeration systems, which was developed by the authors in earlier studies. The predictions of ACCO2 were validated using experimental results of a carbon dioxide based breadboard ECU system that was constructed and tested at the Ray W. Herrick Laboratories at Purdue University.

The results indicate that the carbon dioxide based ECU demonstrates a higher cooling COP and a higher cooling capacity compared to the R-22 based ECU. Both the cooling and heating performance of the carbon dioxide based ECU can meet the requirements for the next generation of ECU as stipulated by the U.S. Army.

Introduction

Military Standard Environmental Control Units (MIL-STD ECUs) are used by the U.S. Army to provide an air-conditioned environment for their equipment and personnel in the field. MIL-STD ECUs are similar in style to packaged unitary systems, but have unique requirements that make them different from commercial air conditioners (Calkins 2000). They are designed to meet both high stresses and environmental extremes. They must meet high vibration and impact requirements resulting from military off-road and rail transport requirements. MIL-STD ECUs are performance rated at a 48.8°C condenser ambient and 32.2°C, 50% relative humidity evaporator ambient. Currently, they do not operate as heat pumps due to the poor performance of R-22 at low ambient temperatures. Because military systems are mobile, the primary design concerns for ECUs are weight and size. These concerns constrain the system to have smaller coils than commercially available packaged systems. Heavier fan assemblies are required to handle the increased pressure drop that results from the smaller coils and to provide additional airflow. In addition, ECUs are designed to minimize the maximum power draw. As a result they are not optimized for efficiency over the range of operating conditions.

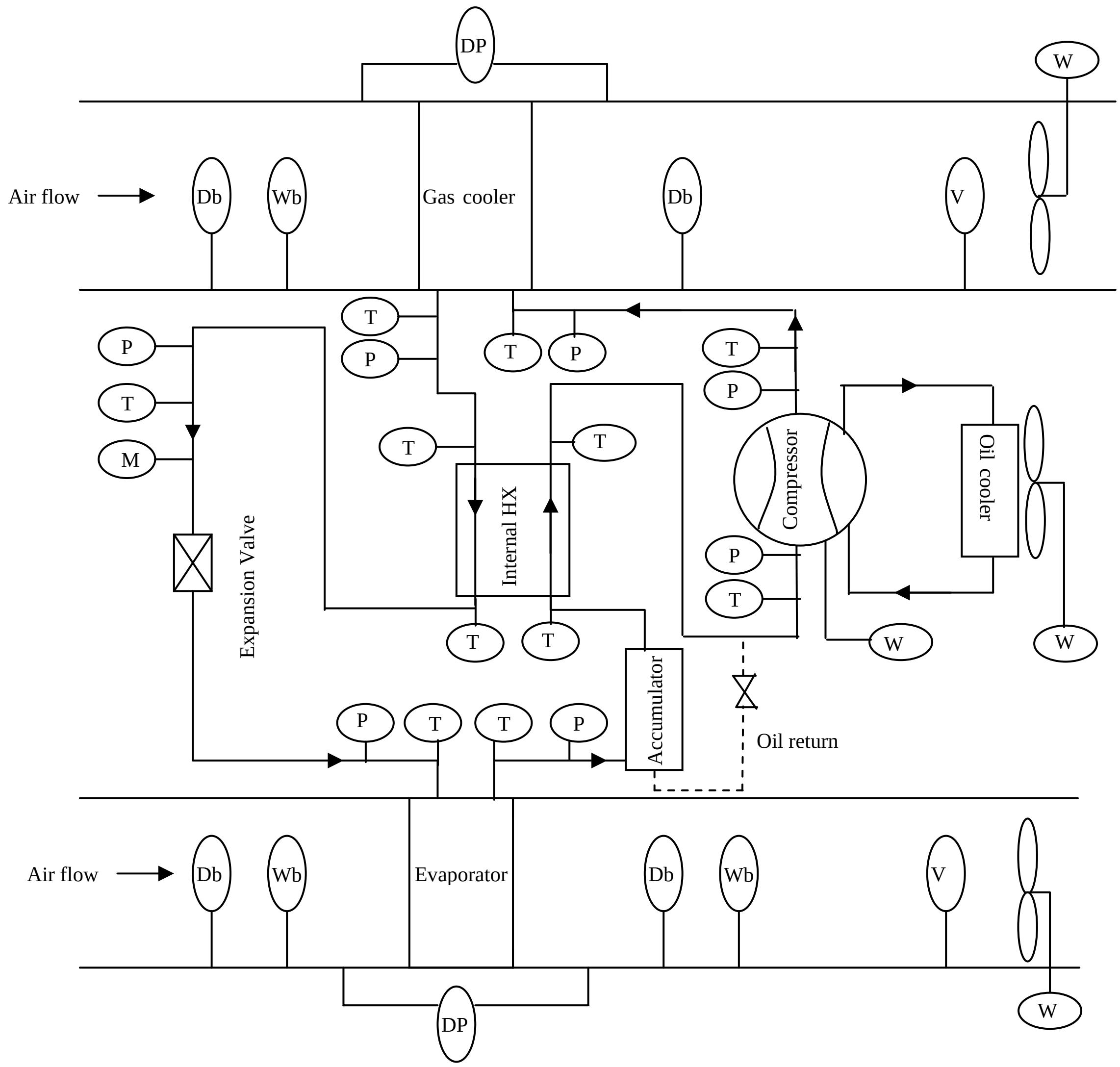
Currently, ECUs use R-22 as the refrigerant. Due to the Clean Air Act, R-22 will be phased out in the future and the U.S. Army is seeking an alternative refrigerant to replace R-22 in their ECU applications. Carbon dioxide is one of the options considered because it has minimal direct environmental impact compared to other refrigerants and some of its characteristics make it an especially attractive refrigerant to the ECU application (Manzione 1998). A previous simulation study by Robinson and Groll (2000) showed that a carbon dioxide based ECU may have a similar performance to the R-22 based ECU if the carbon dioxide based ECU is designed such that the heat exchangers are of the same packaged volume size as the R-22 based ECU. The heat exchangers used in this simulation study were fin-and-tube type heat exchangers.

To investigate the potential of a carbon dioxide based ECU to have a better performance than the standard R-22 based ECU, a simulation study was conducted in which the fin-and-tube heat exchangers were replaced by microchannel heat exchangers. However, in the first step, it had to be shown that the modified simulation model ACCO2 is capable to accurately predict the performance of transcritical carbon dioxide based air conditioning systems with microchannel heat exchangers. To this end, the experimental results of a carbon dioxide based breadboard ECU system with microchannel heat exchangers were used to validate the predictions by ACCO2. Once the model predictions were validated, the model inputs were varied to reflect the design of an actual carbon dioxide based ECU. To this end, it was assumed that the microchannel gas cooler and the microchannel evaporator of the CO_2 based ECU have the same frontal areas and volumes as the fin-and-tube condenser and evaporator of the R-22 based ECU, respectively. Furthermore, the compressor performance of the CO_2 based ECU was based on the test results of a prototype CO_2 semi-hermetic reciprocating compressor as presented by Hubacher and Groll (2002). In addition, a liquid-to-suction line heat exchanger was employed in the CO_2 based ECU. After the new model inputs were generated, detailed simulations of the CO_2 based ECU were performed for both cooling and heating operating conditions to verify whether the unit can meet the performance of newly proposed U.S. Army standards for ECUs. Finally, the cooling performance of the CO_2 based ECU was also compared to the experimentally measured performance of an R-22 based ECU.

Validation of ACCO2

ACCO2 is a computer simulation software tool that predicts the performance of transcritical carbon dioxide air conditioning and heat pump systems. It was first developed in Fortran (Robinson, 2000). It was recently converted to C++ and modified to include the capability to simulate microchannel heat exchangers (Ortiz et al., 2003).

To validate the ACCO2 simulation model, a carbon dioxide based breadboard ECU system was constructed and tested in the psychrometric chambers of the Ray W. Herrick Laboratories at Purdue University. The system and the test setup are shown in Figure 1. The breadboard system consists of a gas cooler, an evaporator, a compressor with oil cooler, an internal heat exchanger, an expansion valve, and a suction accumulator. The gas cooler has accordion-style fins in between single microchannel tubes that are soldered together to make a tube-bank with a face area of 625 mm x 410 mm, and a depth of eight soldered tubes directly next to each other. The tubes have an outside diameter and inside diameter of 3.175 mm and 1.905 mm, respectively. The evaporator is a microchannel heat exchanger with accordion-style fins with a face area of 225 mm x 150 mm and a depth of three multi port extruded (MPE) tube bands. Each MPE tube band has twenty microchannels (total of sixty) with an inside diameter of 0.79 mm. Schematics of the gas cooler and the evaporator configurations are shown in Figure 2 and Figure 3. The compressor is a semi-hermetic, two-piston reciprocating compressor. A fan-coil oil cooler is used to reduce the oil temperature. The internal heat exchanger is a stainless steel coaxial heat exchanger. A manual adjusting back-pressure regulating valve is used as the expansion valve.



Legend of sensors: T - temperature; P - pressure; Db - dry bulb temperature; Wb - wet bulb temperature; DP - differential pressure; M - mass flow rate; V - volume flow rate; W - power

Figure 1: Schematic of CO_2-based ECU breadboard test setup

The test setup was installed in two psychrometric chambers that are located side-by-side. The test arrangement that was used is the room air-enthalpy method. To implement this method, one psychrometric chamber simulates indoor conditions while the other psychrometric chamber simulates outdoor conditions. The evaporator, the accumulator, the internal heat exchanger, and the expansion valve are located inside the first psychrometric chamber. Preconditioned air from the chamber is drawn into the inlet plenum of the air coil, and flows through the air coil into the air-measuring device, which is attached to the air coil outlet. After flowing through the air-measuring device, the air is returned to the psychrometric chamber. The gas cooler, the compressor, and the oil cooler are located in the other psychrometric chamber. Preconditioned air from the chamber is drawn into the gas cooler air coil, flows through the air coil into the air measuring devices, and is discharged back into the chamber. During the cooling (air conditioning) tests, the first psychrometric chamber containing the evaporator simulated indoor conditions and the second psychrometric chamber containing the gas cooler simulated outdoor conditions. Both chambers can be controlled independently to obtain various indoor and outdoor environmental conditions.

T-type thermocouples were inserted in the refrigerant flow to measure the inlet and outlet temperatures of the gas cooler, the evaporator, the compressor suction and discharge, and before and after the internal heat exchanger. Absolute pressure transducers were installed to measure the pressure at the inlet and outlet of the gas cooler and evaporator, the suction and discharge pressures of the compressor and the pressures before and after the expansion valve.

A coriolis-effect mass flow rate meter was installed before the expansion valve to measure the mass flow rate of carbon dioxide. The refrigerant-side cooling and heating capacities were determined from the refrigerant enthalpy change and the measurement of the mass flow rate. The enthalpy changes of the refrigerant were determined from measurements of the entering and leaving evaporator and gas cooler pressures and temperatures, and the temperature and pressure before the expansion valve.

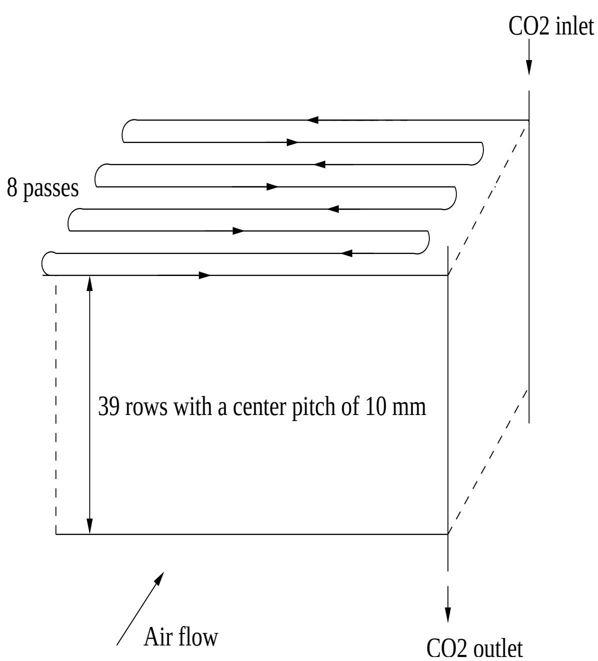


Figure 2: Configuration of the gas cooler

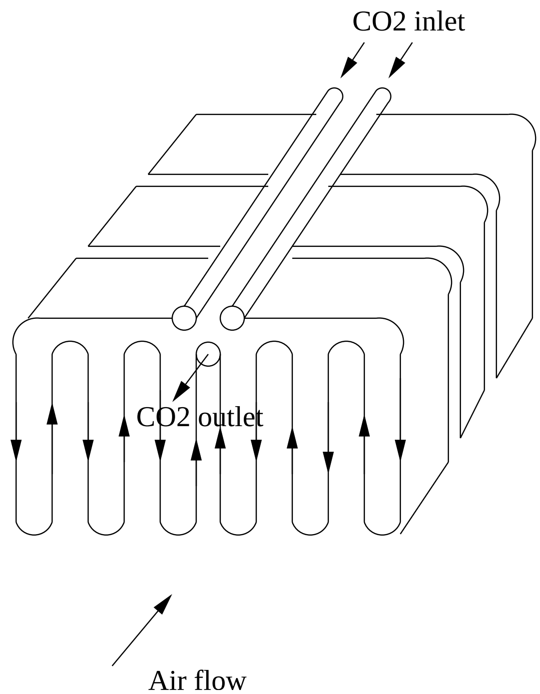


Figure 3: Configuration of the evaporator

The air side measurements were as follows. A grid with eleven T-type thermocouples was installed in the gas cooler duct to measure the dry bulb temperature of air after the gas cooler. A grid with seven T-type thermocouples was installed in the evaporator duct to measure the dry bulb temperature of air after the evaporator. A dew point meter was used to obtain the dew point temperature of air after the evaporator. The inlet air temperatures and humidities of the gas cooler and evaporator were obtained from the sensors installed in the psychometric rooms. The pressure drop across the evaporator and the gas cooler were measured with two differential pressure transducers that have a range of 0 to 63.5 mm of water. The air volume flow rate in the evaporator duct was measured with a 76.2 mm nozzle device and the air volume flow rate in the gas cooler duct was measured with a 152.4 mm nozzle device. Both nozzle devices comply with ANSI/ASHRAE Standard 37-1988.

An electronic data acquisition system was used to convert the incoming voltages from the measuring instrumentation to digital signals, which are transferred to a personal computer for further data analysis. A series of 12 air conditioning tests following the operating conditions shown in Tables 1 were carried out. Each test was started after the desired steady-state operating conditions were reached.

Table 1: Test matrix of CO_2 –based ECU cooling tests

Indoor Temperature & Relative Humidity	**Outdoor Temperature**	**High-side Pressure**	**Test No.**
32.2°C and 50% (90°F and 50%)	35°C (95°F)	8.0 MPa (1160 psia)	1
		10.0 MPa (1450 psia)	2
		12.0 MPa (1740 psia)	3
	40.6°C (105°F)	8.0 MPa (1160 psia)	4
		10.0 MPa (1450 psia)	5
		12.0 MPa (1740 psia)	6
	46.1°C (115°F)	8.0 MPa (1160 psia)	7
		10.0 MPa (1450 psia)	8
		12.0 MPa (1740 psia)	9
	51.7°C (125°F)	8.0 MPa (1160 psia)	10
		10.0 MPa (1450 psia)	11
		12.0 MPa (1740 psia)	12

During the air conditioning tests, the sensible, latent and total cooling capacities based on the air-side evaporator test data were calculated using Equations (1) to (3):

- Sensible air-side cooling capacity: $\dot{Q}_{sen} = \rho_{air,evap}\dot{V}_{air,evap}c_{p,air}(T_{air,in} - T_{air,out})_{evap}$ (1)
- Latent cooling capacity: $\dot{Q}_{lat} = \rho_{air,evap}\dot{V}_{air,evap}(W_{air,in} - W_{air,out})_{evap}h_{fg,water}$ (2)
- Total air-side cooling capacity: $\dot{Q}_{c,air} = \dot{Q}_{sen} + \dot{Q}_{lat}$ (3)

where,

$\rho_{air,evap}$ = Air density at nozzle exit in evaporator duct

$\dot{V}_{air,evap}$ = Air volume flow rate at nozzle exit in evaporator duct

$c_{p,air}$ = Air specific heat

$T_{air,in}$ = Air dry bulb temperature before evaporator

$T_{air,out}$ = Air dry bulb temperature after evaporator

$W_{air,in}$ = Air humidity ratio before evaporator

$W_{air,out}$ = Air humidity ratio after evaporator

$h_{fg,water}$ = Latent heat of evaporation for water

The cooling COP was calculated based on the evaporator capacity and the compression work of the compressor, which was determined by the mass flow rate of the refrigerant and the enthalpy difference across the compressor, as shown in Equation (4):

$$COP = \frac{\dot{Q}_c}{\dot{m}_{ref}\,(h_{o,comp} - h_{i,comp})} \tag{4}$$

where,

$h_{o,comp}$ = Refrigerant enthalpy after the compressor

$h_{i,comp}$ = Refrigerant enthalpy before the compressor

The comparisons of the cooling capacity and the cooling COP between the predicted data and the test data for test runs No. 1 to 12 are shown in Figure 4 and Figure 5, respectively.

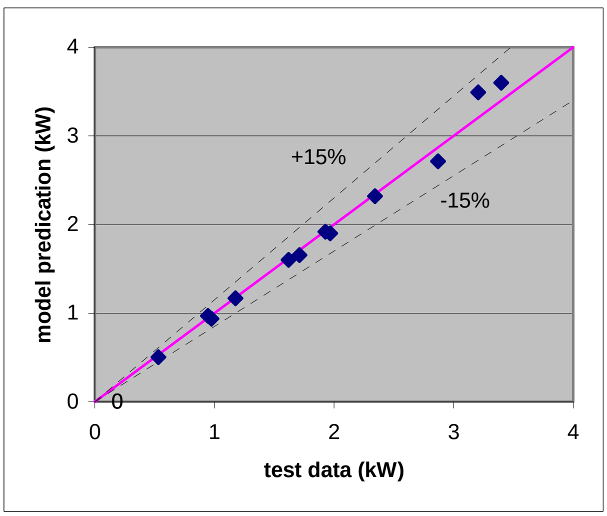


Figure 4: Comparison of cooling capacity between model predication and test data

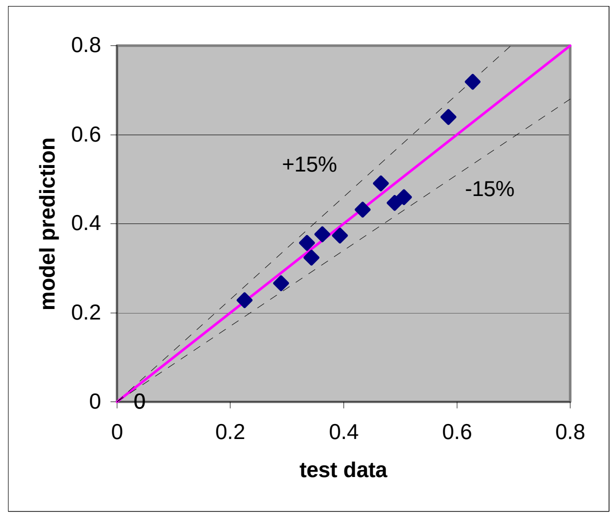


Figure 5: Comparison of cooling COP between model predication and test data

It can be seen that the deviation between the model prediction and the experimental results for both the cooling capacity and COP are within ±15% for all 12 test points and within ±6% for 8 test points. Additional model validation was conducted by Ortiz et al. (2003). Based on these results, it was concluded that ACCO2 can be used to predict the performance of the carbon dioxide based air conditioning system with microchannel heat exchangers.

Simulation of Carbon Dioxide based ECU with Microchannel Heat Exchangers

The computer model ACCO2 was used to simulate the performance of a carbon dioxide based ECU with microchannel heat exchangers for both cooling and heating operating conditions to investigate if its performance can meet the new U.S. Army performance standards. For this purpose, new evaporator and gas cooler configurations for a prototype carbon dioxide based ECU were specified and are shown in Table 2. Both heat exchangers have the same geometry dimensions in the width, height and depth as the ones of the R-22 based ECU that is currently used by the U.S. Army. Typical microchannel heat exchanger with Multiple-Port-Extruded (MPE) tubes and accordion-style fins in between are used for both the evaporator and the gas cooler. The compressor model is based on the measured performance of a prototype semi-hermetic carbon dioxide compressor (Hubacher and Groll 2002). The performance requirements as stipulated by the U.S. Army for the next generation of ECUs are shown in Table 3. The cooling capacity is rated at a 48.8 °C outdoor ambient and 32.2 °C, 50% relative humidity indoor ambient. The heating capacity is rated at a 15.6 °C indoor ambient and -8.3 °C outdoor ambient.

Table 2: Carbon Dioxide Based ECU's Heat Exchanger Configurations

	Width (mm)	Height (mm)	Depth (mm)	MPE Tube Rows high	MPE Tube Rows deep	Inside Diameter (mm)	Flow Circuit
Evaporator	685.8	304.8	114.3	30	6	0.79	3 circuits of 60 MPE tubes
Gas Cooler	488.95	628.65	76.2	60	4	0.79	6 circuits of 40 MPE tubes

Table 3: Performance Requirements for New U.S. Army ECUs

Operating temperature range	Cooling: -31.7 to 48.9 °C Heating: -45.6 to 26.7 °C
Rated cooling capacity[1]	12 kW
Rated heating capacity[2]	9.1 kW
Max power consumption	13.5 kW
Conditioned air flow at 0 inches wg	0.65 m^3/s

[1]48.8 °C outdoor ambient and 32.2 °C, 50% relative humidity indoor ambient
[2]15.6 °C indoor ambient and -8.3 °C outdoor ambient

The cooling performance of the carbon dioxide based ECU based on the simulation results are shown in Figure 6 and Figure 7. It can be seen from Figure 6 and Figure 7 that both the cooling capacity and the cooling COP drop with an increase of the outdoor temperature. A higher indoor temperature leads to a higher cooling capacity and cooling COP. The indoor humidity has little effects on the cooling capacity and cooling COP.

The heating performance of the carbon dioxide based ECU based on the simulation results are shown in Figure 8 and Figure 9. The heating performance predictions of the carbon dioxide based ECU were performed with the gas cooler specified in Table 2 operating as the evaporator and the evaporator specified in Table 2 operating as the gas cooler. The air flow rates for both the indoor heat exchanger and the outdoor heat exchanger are still the same as the ones during the cooling mode. The indoor air relative humidity is assumed at 50%. The CO_2 pressure in the heating mode gas cooler is still higher than the critical pressure and thus, the system maintains its operation as a transcritical cycle.

It can be seen from Figure 8 and Figure 9 that the heating capacity and heating COP increase with an increase of the outdoor temperature. The heating capacity and heating COP decrease with an increase of the indoor temperature at outdoor temperatures of -8.3 °C and 8.3 °C. There is little difference in heating capacity and heating COP for the two indoor temperatures when the outdoor temperature is -23.3 °C.

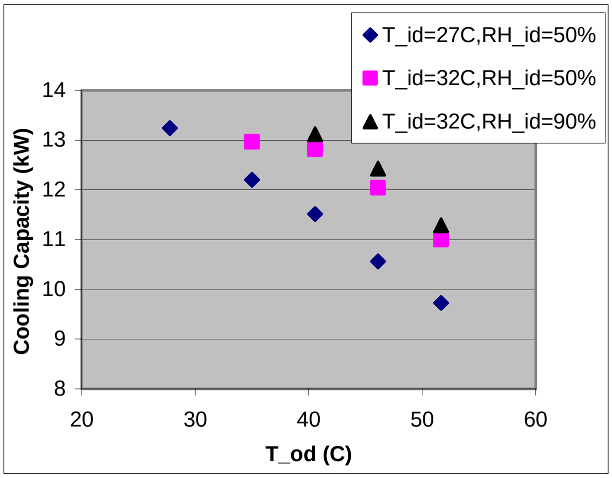


Figure 6: Cooling capacity of carbon dioxide based ECU

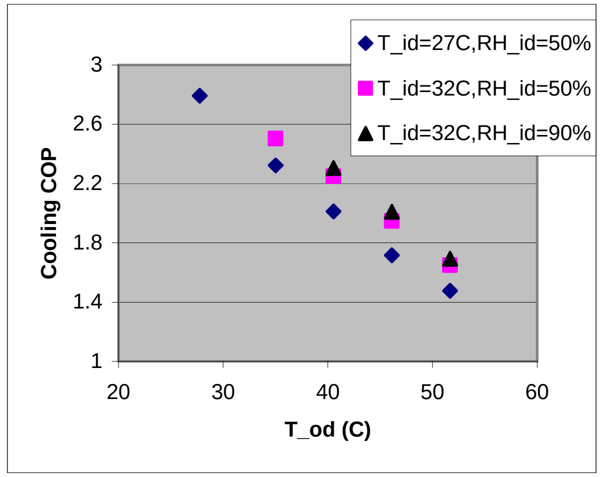


Figure 7: Cooling COP of carbon dioxide based ECU

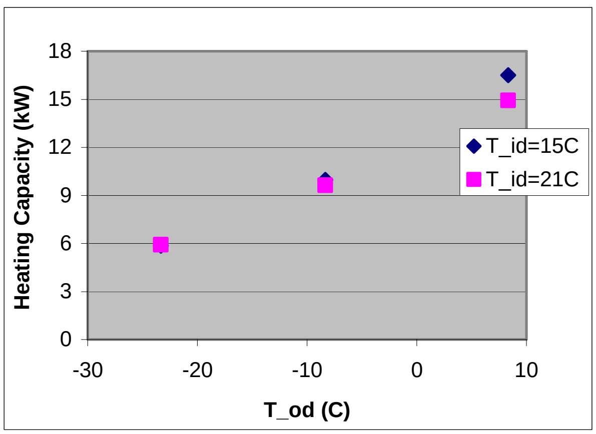


Figure 8: Heating capacity of carbon dioxide based ECU

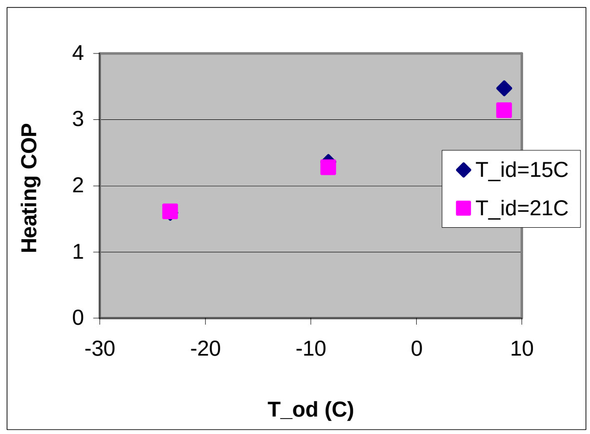


Figure 9: Heating COP of carbon dioxide based ECU

Based on the results presented in Figures 6 through 9, it can be seen that the required rated cooling capacity of 12 kW and heating capacity of 9.1 kW can be met by the carbon dioxide based ECU as specified here.

The simulation model was also exercised to compare the performance of the carbon dioxide based ECU with the one of the R22-based ECU that is currently used by the U.S. Army. For this purpose, the test data reported by Calkins (2000) were used here to represent the performance of the R22-based ECU. Figures 10 and 11 presents a comparison of the cooling capacity and cooling COP of the carbon dioxide based ECU and the R22-based ECU at the same operating conditions, respectively. From Figures 10 and 11, it can be seen that the carbon dioxide based ECU has a higher cooling capacity and higher cooling COP than the ones of the R-22 based ECU at all operating conditions. However, it should be noted that a liquid to suction line heat exchanger is used in the carbon dioxide based ECU and the simulation results of the carbon dioxide based ECU's

COP did not include the fan power consumption. The fan power accounts for approximately 10% of the total power consumption of the R-22 based ECU. Thus, even if the fan power is included in the cooling COP of carbon dioxide based ECU the conclusion drawn from Figure 11 will not change.

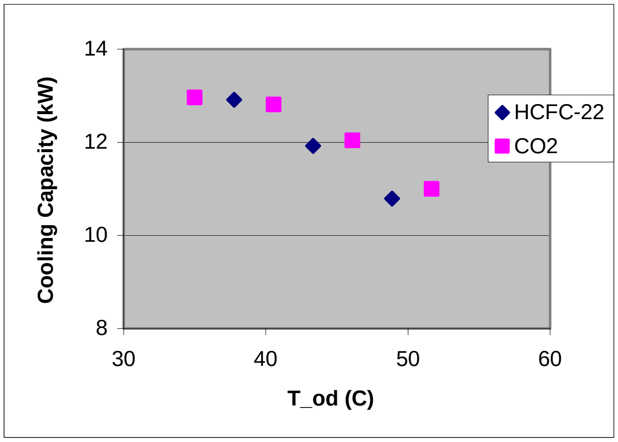


Figure 10: Comparison of cooling capacity between CO2 based ECU and R-22 Based ECU

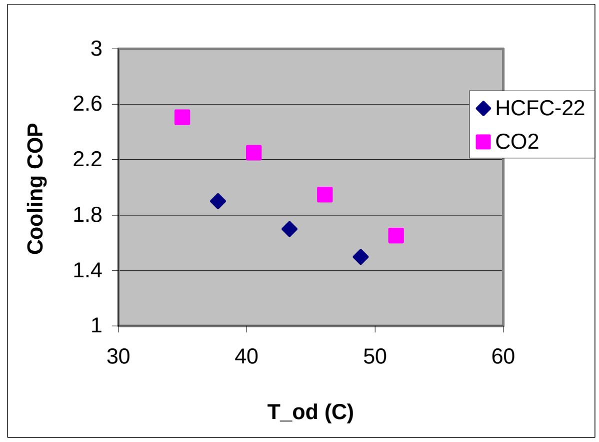


Figure 11: Comparison of Cooling COP between CO2 based ECU and R-22 Based ECU

Conclusions

The simulation model ACCO2 was used to predict the performance of a prototype carbon dioxide based ECU that uses microchannel heat exchangers of the same size (occupied volume) as the fin-and-tube heat exchangers that are currently used in R-22 based ECUs. The simulation results demonstrate that the carbon dioxide based ECU can meet the new performance requirement of the U.S. Army Standard for ECUs without increasing the geometrical size of the heat exchangers. The comparison between the predicted performance of the carbon dioxide based ECU and the experimental results of the R-22 based ECU shows that a better performance can be achieved for all operating conditions considered.

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