Experimental Investigation Of A Breadboard Model Of A Carbon Dioxide U.S. Army Environmental Control Unit

M. Connaghan
U. S. Army Communications and Electronics Command

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EXPERIMENTAL INVESTIGATION OF A BREADBOARD MODEL OF A CARBON DIOXIDE U.S. ARMY ENVIRONMENTAL CONTROL UNIT

Michael Connaghan, Mechanical Engineer, U.S. Army, 10125 Gratiot Road, Suite 100,
Fort Belvoir, VA 22060, USA; Tel.: 703/704-2007; Fax: 703/704-2005
E-Mail: mconnaghan@belvoir.army.mil

ABSTRACT

Carbon dioxide is being investigated as a possible refrigerant for future U.S. Army Environmental Control Units (ECUs), which provide air cooling and heating in a range of capacities for Army shelters and tents. The anticipated benefits of using CO_2 include reduced logistics burden and improved heating performance. This paper reports test results of a breadboard air-conditioner/heat pump system that uses CO_2 as its refrigerant. The tested system consists of an evaporator, a semi-hermetic reciprocating compressor, a gas cooler, and two expansion valves connected in parallel. These are the first set of tests performed in a new facility consisting of separate air duct loops representing the heat source and heat sink. This basic system's primary purpose is to establish a baseline for future system improvements. The tests are performed under the following nominal conditions: The air temperature entering the gas cooler ranges from 32.2°C to 51.7°C (90°F to 125°F). Nominal air conditions entering the evaporator are 32.2°C (90°F) and 50% relative humidity. Discharge pressures range from 10 to 13 MPa when stable conditions are maintained. Higher discharge pressures increase evaporator capacity at all conditions tested and generally increase system efficiency. The greatest evaporator capacity measured is 11.3 kW.

NOMENCLATURE

COP: Coefficient of Performance = Q_{EV}/W
ECU: Environmental Control Unit
EV: Evaporator
GC: Gas Cooler
Q_{EV}: Evaporator capacity, total
W: Power input to compressor

INTRODUCTION

Previous studies indicate that carbon dioxide has somewhat surprising promise as a refrigerant, in particular when size is significantly constrained (for example, Robinson). The potential logistical savings associated with the elimination of refrigerant recovery and recycling requirements make it appealing from a life cycle cost point of view. Other authors have written more detail about the role of CO_2 in the U.S. Army's environmental control research program (Manzione and Terrell) and prototype development (Manzione & Calkins 2001, 2002).

With a relatively immature technology development status, CO_2 air-conditioning and heat pump systems have a number of questions yet to be addressed. While cycle energy efficiency is only one of these, it is perhaps the driving one for most applications. Even as considerations are made to address economical manufacture, reliability, safety, and so on, the final product must perform with low power input to compete with other designs. Therefore as each new prototype is produced, it is tested to determine its effect on system efficiency and to find clues for design improvements.

With this in mind, a laboratory has been constructed to provide additional in-house testing capabilities to complement those of cooperating institutions. This laboratory allows air-conditioners and heat pumps to be tested in a breadboard configuration, so that components may be individually replaced and tested as they become available. The performance of the individual components and the effect on the overall system may be examined. The first system tested is a basic CO_2 cycle, consisting of a compressor, gas cooler, expansion valves, and evaporator. Future systems will feature system enhancements. This paper reports on this first system tested in this laboratory with emphasis on overall performance.

DESCRIPTION

Laboratory Layout

The laboratory's main features are two closed air duct loops, one for the evaporator and one for the gas cooler (see Figure 1). Airflow is generated with two blowers at opposite corners of the loop to produce relatively constant air pressure throughout the ducts and small pressure differences between the room and any point in the duct. The other corners have turning vanes installed to help maintain uniform airflow profiles. The gas cooler air inlet dry bulb temperature is maintained by removing heat with a R-134a heat exchanger in the upper loop section. The R-134a system has two compressors installed in parallel, each independently belt-driven by a dedicated frequency-controlled motor to provide flexibility and control. To maintain dry bulb and wet bulb temperatures at the test evaporator inlet, the evaporator duct has a heater and a steam generator, each of which has an automatic temperature controller. The humidifier is located downstream of the heater so that the water is more easily absorbed and mixed in the warmer air.

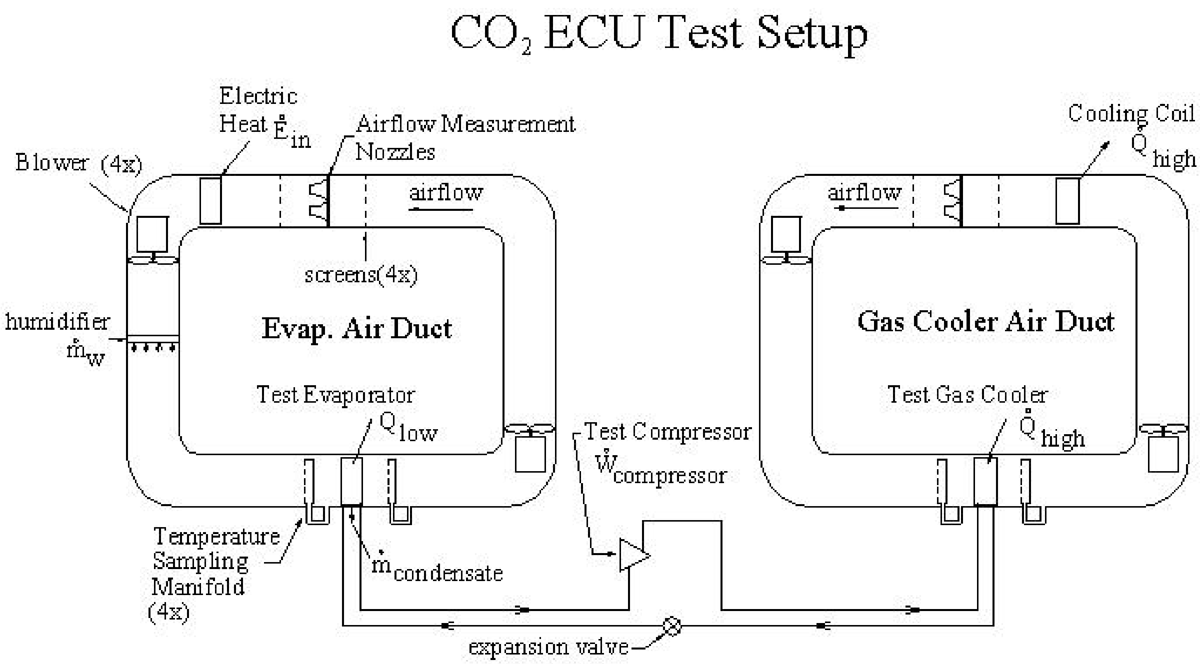

Figure 1: Laboratory schematic

The test CO_2 heat exchangers are placed in the lower section of their respective ducts. The dry bulb and wet bulb air temperatures entering and leaving the heat exchangers are measured with the use of sample trees and small blowers, as specified in ASHRAE Standard 37. The temperature sensors for the controllers are located in these sampling loops.

Airflow is measured with nozzles in accordance with ASHRAE Standard 37. Each duct has four nozzles. The nozzles in the evaporator duct all have throat diameters of 102mm (4 in.). In the gas cooler duct, two nozzles throat diameters are 152mm (6 in.) and two are 102mm (4 in.). This design was chosen to allow some nozzles to be blocked off in case lower airflow rates are desired. Air pressure differences are measured across the nozzles and temperature measurements are made at the throat, enabling airflow calculations.

CO_2 Test Equipment

The CO_2 compressor is a semi-hermetic reciprocating compressor with 2 cylinders. It has a nominal suction displacement rate of 0.001 m^3/sec. The lubricant used was Mobil EAL POE 100. Input electrical power is measured with a watt transducer.

The heat exchangers consist of identical brazed aluminum slabs of multi-port extrusion flat tubes, louvered fins, and manifolds. Each slab has 50 multi-port flat tubes for refrigerant flow. Baffles in the manifolds divide each slab into 4 sections, each with 12 or 13 flat tubes and its own 12mm outer diameter inlet and outlet tubes welded to the manifolds. For this series of tests, 2 slabs were plumbed to form the evaporator and 3 were used for the gas cooler. The slabs were stacked so that they were in series with respect to the airflow. Union tees divided the refrigerant flow among the 4 sections of the first slab. The refrigerant outlet from each section was connected to the inlet of the neighboring section in the following slab. The outlets of the final slab were joined with union tees. To reduce inefficiencies associated with undesirable heat transfer between slabs, small spacers were used to separate them by about 1 or 2 mm.

The expansion valves are two manually adjusted needle valves connected in parallel. These provide excellent adjustability and stable control for the lab, which the operator must adjust by trial and error while establishing the test conditions.

CO_2 Measurement Equipment

The CO_2 pressure is measured with pressure transducers at four points in the cycle, immediately before and after each heat exchanger. Although there is some pressure drop between the compressor and the pressure transducers before and after it, this is assumed to be fairly small, so the gas cooler inlet pressure is also called the discharge pressure here. In addition, the differential pressure is directly measured across each heat exchanger with independent transducers. These measurements always showed excellent agreement with the difference between the measured absolute pressures.

The CO_2 mass flow rate and temperatures at six different locations were also recorded. However, it is felt that additional validation of these measurements is required, so discussion of their results and analysis is reserved for a future report.

Data Acquisition

Data was collected using an Agilent 34970A Data Acquisition/Switch Unit with 3 HP 34901A 20-channel armature multiplexer cards. The software program managing the data acquisition was HP Benchlink. The collected data was then imported into Excel spreadsheets for calculations and consolidation.

After the test system appeared stable, data was taken at 10-second intervals. The minimum number of data points used in calculations for a given condition and discharge pressure was 61, representing 10 minutes of run-time. The maximum number of points was 267 with an average of 172. There were a total of 1,895 data points taken. For a given test, all of the directly measured data was averaged. Calculations were performed using the averaged data.

Test Conditions

The target test conditions are shown in Table 1.

Table 1: Cooling Test Conditions

Cooling condition number	GC Air Inlet Temperature [°C (°F)]	EV Air Inlet Temperature [°C (°F)]	EV Air Inlet Wet Bulb Temperature [°C (°F)]	Discharge Pressures [MPa]
1	51.7 (125)	32.2 (90)	23.9 (75)	12, 13
2	48.9 **(120)**	32.2 **(90)**	23.9 **(75)**	11, 12, 13
3	43.3 (110)	32.2 (90)	23.9 (75)	10, 11, 12, 13
4	37.8 (100)	32.2 (90)	23.9 (75)	10, 11

The condition #2 (in **bold)** above represents a condition in the military specification MIL-A-52767. Each type and size of military ECU has a certain minimum net cooling capacity at this condition. For future ECUs, this will be replaced by condition #1 with new capacity standards. For each cooling condition, separate tests were performed for each discharge pressure noted in the table. Note that the target evaporator air inlet condition is constant throughout all tests. The average airflow rates across all tests were 0.766 kg/sec (minimum 0.757, maximum 0.773) for the evaporator and 1.037 kg/sec (min. 1.016, max. 1.053) for the gas cooler. For this round of testing, the mass of refrigerant charged into the system was not optimized. The evaporator capacity was calculated in accordance with ASHRAE Standard 37. In this paper, Coefficient of Performance (COP) is defined as:

$$COP = Q_{EV}/W$$

Note that W is the power input to the compressor only, and does not include power to operate fans or any other ancillary equipment.

While air ducts and refrigerant lines have been insulated, there will always exist temperature differences between various internal parts of the system and the room. Even if the room is conditioned to match parts of the system, there will be other parts of the system at different temperatures. In particular, for these tests, the compressor shell was exposed to room air. While part of the compressor surface was hot (therefore transferring heat to the surrounding air), there was a portion that was in close contact with the suction gas that was relatively cool, therefore absorbing heat from the surroundings. This compressor could not be completely insulated, because it requires external cooling.

RESULTS AND DISCUSSION

The test results are shown in Figures 2-4. Lines connect data points of tests with nominally constant discharge pressure. The laboratory apparatus appears to have upper and lower limits of capacity beyond which it cannot maintain stable conditions. Efforts to understand and extend these limits are ongoing. It is also noted that at the two points shown here with the highest capacity, the air inlet wet bulb temperature was about 2.5°C below the target value.

Figure 2 shows the evaporator capacity versus gas cooler air inlet temperature. As may be expected, for constant discharge pressure, capacity falls as gas cooler temperature rises. For a given gas cooler temperature, capacity rises with discharge pressure. At condition #1, the highest capacity was 6.3 kW. At condition #2, the highest capacity was 8.9 kW.

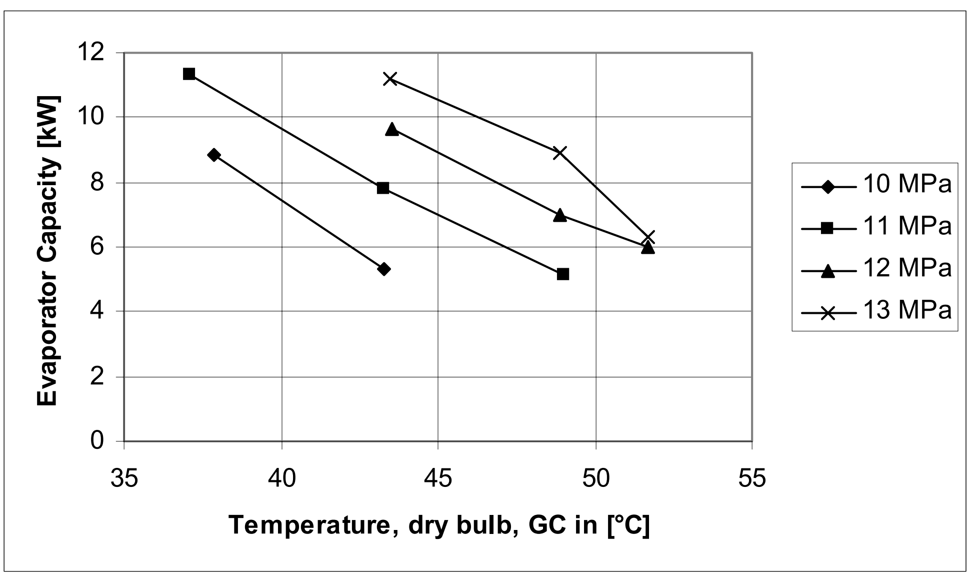

Figure 2: Evaporator capacity

Figure 3 shows compressor electrical power. It is seen that power varies much less than capacity. For a given discharge pressure, power is nearly constant. As discharge pressure and pressure ratio increase for a given gas cooler inlet temperature, the greater power required is partially counteracted by a smaller refrigerant mass flow rate. The maximum power required was 8.5 kW at condition #1 at 13 MPa.

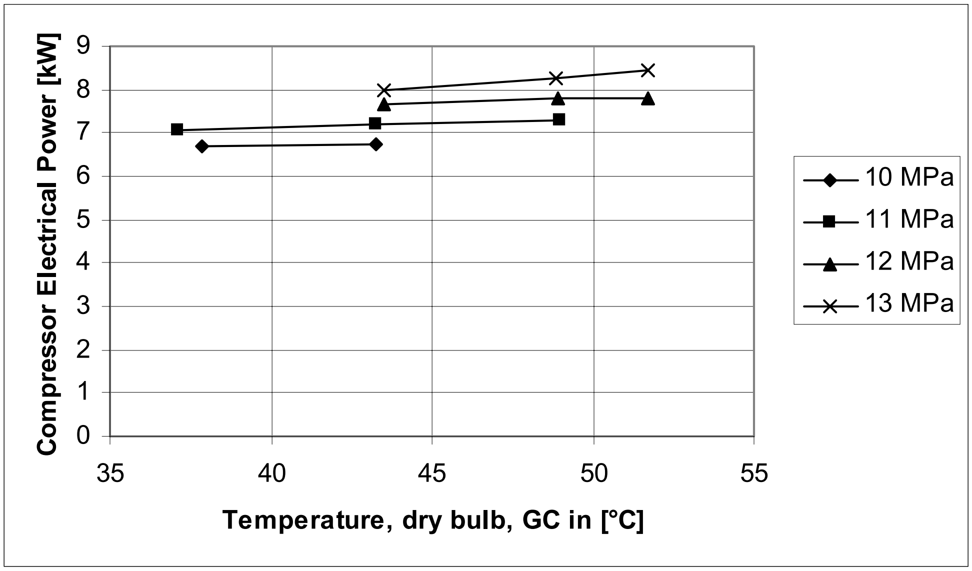

Figure 3: Compressor electrical power

The Coefficient of Performance (COP) is shown in Figure 4. Note that the power is only that of the compressor. Since power does not vary much, this graph has a very similar appearance to that of Figure 2. At condition #1, the maximum COP was 0.78 (12 MPa discharge pressure). At condition #2, the maximum COP was 1.10 (13 MPa).

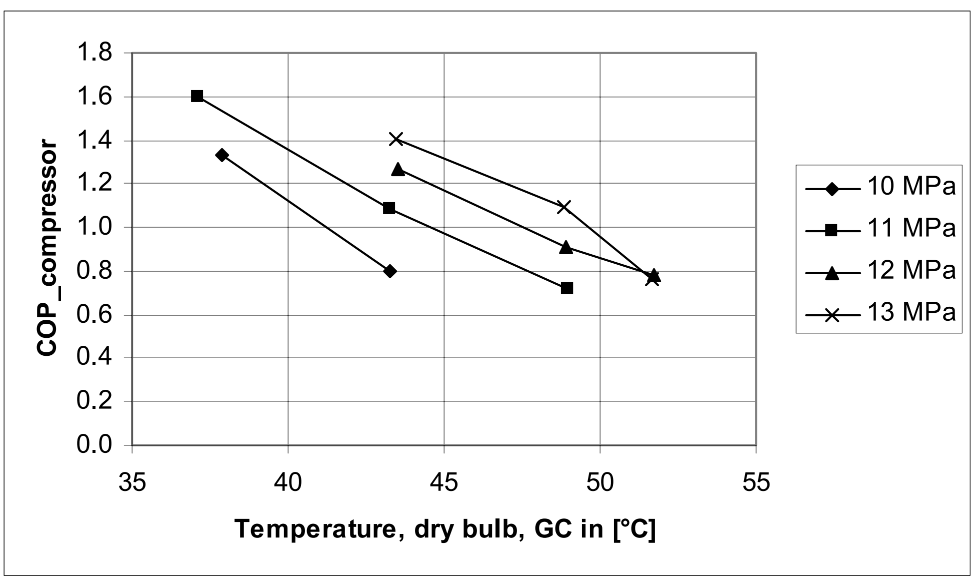

Figure 4: Coefficient of Performance

It is emphasized that these results are for a baseline system that has not been optimally charged and that lacks system enhancements such as an internal heat exchanger or work recovery device. These and other improvements shall be explored in detail in the future.

CONCLUSIONS

This series of experiments represents the first set of data taken in the new testing facility. This basic tested system will serve as a baseline for future tests, which will include system improvements such as an internal heat exchanger and modifications or replacements of existing components. At conditions #1 and #2, maximum cooling capacities achieved were 6.3 kW and 8.9 kW and COPs were 0.78 and 1.10, respectively.

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