PREDICTION OF THE PRESSURE DROP OF CO_2 IN AN EVAPORATOR USED FOR AIR COOLING

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ABSTRACT

A well instrumented experimental set-up for testing heat exchangers in a capacity range of up to 10 kW was built according to ASHRAE standards and has been used to study CO_2 as the coolant in a secondary loop for applications in low temperature display cases for supermarkets. Tests were conducted with a commercial staggered air refrigeration coil.

Excellent stability is obtained when operated with CO_2 recirculation. Energy balances on air and on CO_2 correlate within ± 5%. The pressure drop in the coil was measured at different operating conditions. None of the available correlations predicting the pressure drop in coil gave acceptable results, however, the experimental values were fitted using an equation having the form $DP = a \cdot (x)^b \cdot (G_{CO2})^c$ with a mean difference of 2.8%. The presented experimental data and empirical correlation can be useful for validating theoretical models.

1. INTRODUCTION

The use of carbon dioxide (CO_2) as secondary refrigerant for freezing temperature applications in supermarkets is already a reality in Western Europe [1] and is at the demonstration stage in North America. In support to the commercialization of this technology, a well instrumented test bench was built in view to provide experimental data for the validation of a design tool that would predict the operating characteristics such system (capacity, pressure drop, etc…).

In this paper will be presented a description of the experimental set-up, the results of some tests and a correlation allowing to predict the pressure drop in the evaporator for a wide range of carbon dioxide flow and outlet quality.

2. EXPERIMENTAL SET-UP AND PROCEDURE

The experimental set-up, built according to the ASHRAE standard for forced circulation air cooling and heating coils [2], is showed in Figure 1. The CO_2 loop includes a circulating centrigugal pump, two mass flow and density sensors, a CO_2/air corrugated fin coil, a CO_2 brazed plate condenser and a CO_2 tank having a volume of eighteen liters. The head of the CO_2 pump is 300 kPa. The loop is also equipped with several temperature (RTD) and pressure sensors. The CO_2 is condensed using a brine loop (potassium formate), allowing good flexibility and control of the condensing temperature and capacity. A CO_2 level sensor is used to monitor the level of CO_2 in the tank.

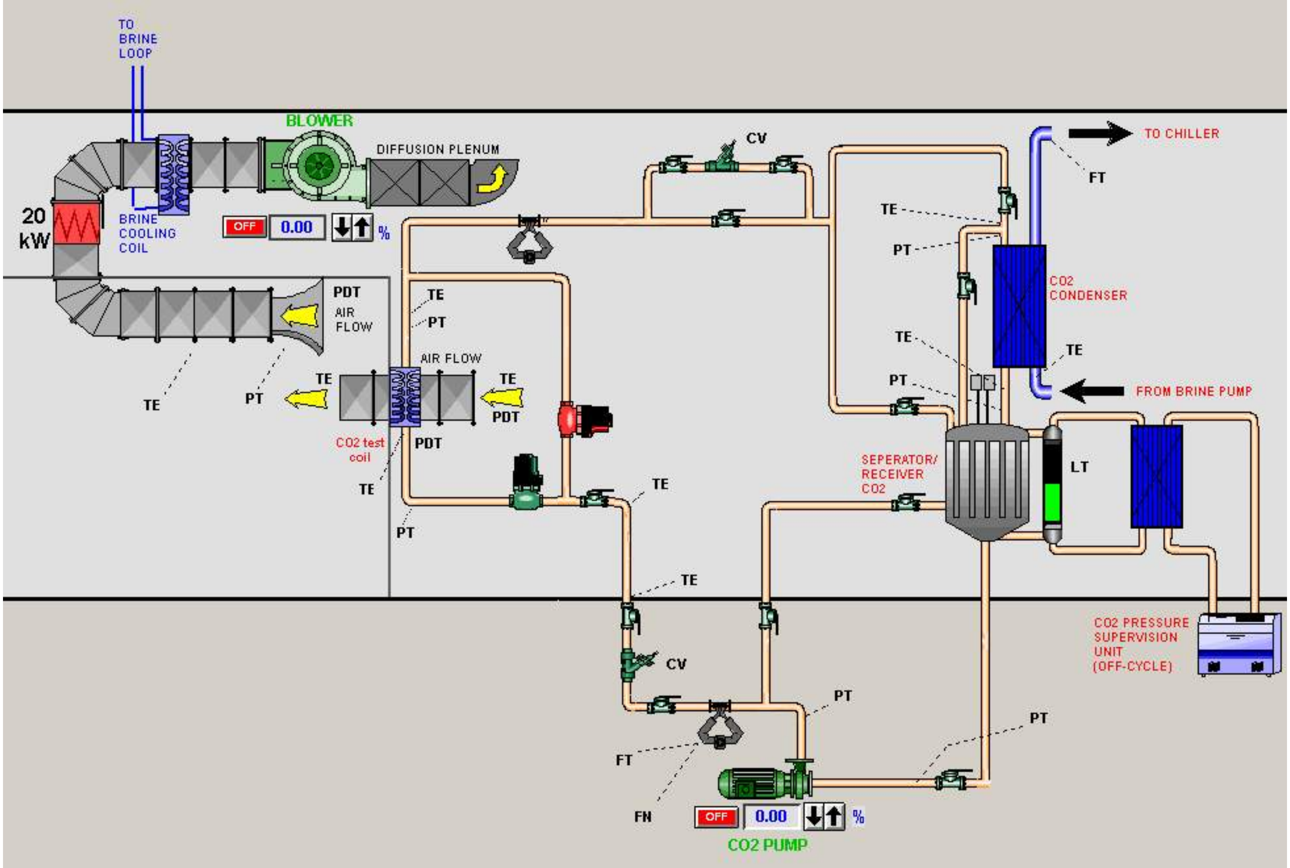

Figure 1. Simplified scheme of the experimental set-up (CV: electronic valve, FN: fluid density sensor, FT: flow transmitter sensor, LT: level transmitter sensor, PDT: pressure differential sensor, PT: pressure sensor, TE: temperature sensor).

The air loop is made of a small room enclosed in a larger room. The air flows from the main room to the small one through the CO_2 evaporator and air returns to the main room through a duct, pull by a centrifugal blower. In the duct, a corrugated fin coil can be used to cool down the air using the cold brine loop and an electric heater can be used to warm the air, increasing the refrigeration charge. The two rooms are equipped with temperature (RTD) and dew point temperature sensors. The evaluation of air flowrate is done using a nozzle at the inlet of the duct, as described in ASME MFC-3M-1989 standard [3]. The pressure drop across the nozzle is measured using a differential pressure sensor. The rooms are well insulated from the outside, minimizing air and moisture infiltration, and consequently very few frost is formed on the evaporator during tests. The room is also equipped with a CO_2 sensor for safety.

All tests reported in this paper were conducted with an evaporator made of two commercial refrigeration staggered coil of 0.61 meter wide and 0.32 meter high, for a corresponding face area of 0.1936 m^2. Each coil has four rows of ten tubes per row, and four corrugated fins per inch. The evaporator thus have 80 copper tubes of $^3/_8$" O.D. (i.d. of 8.712 mm) arranged in one circuit, as shown in Figure 2. Tube spacing are 31.75 mm vertically and 27.48 mm horizontally. Fin thickness is 0.254 mm and fin length is 110 mm for each coil of four rows, giving a total fin length of 220 mm, with and empty space of 50 mm between the two coils. The total external heat transfer surface (fins + tubes) is 13.64 m^2 and the internal tube heat transfer surface is 1.34 m^2. Finally, in this evaporator, air and CO_2 are circulating counter-flow.

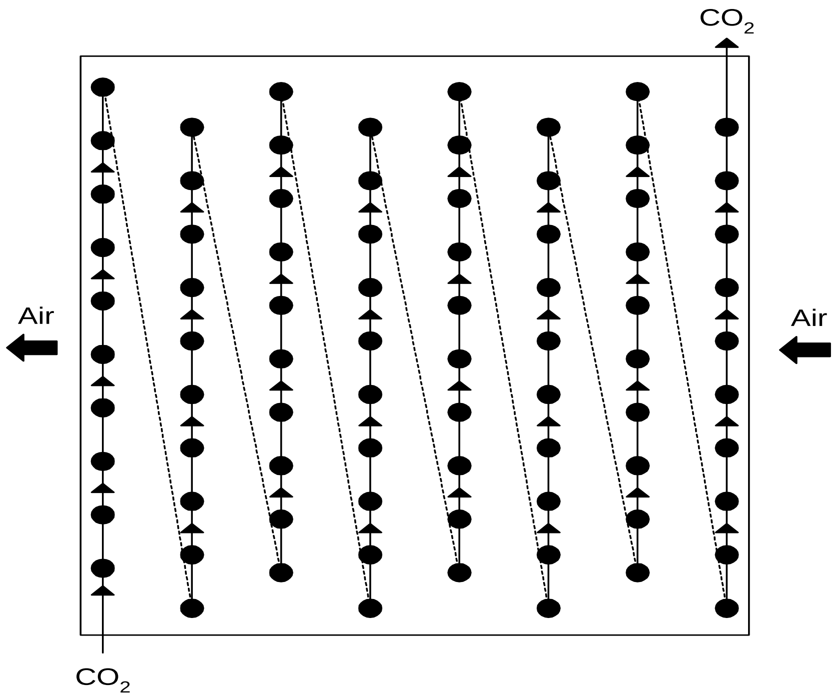

Figure 2. Configuration of the air/CO_2 evaporator.

The temperature sensors (RTD) were periodically calibrated and the accuracy was always better than 0.05°C. The pressure sensors (PT), differential pressure sensors (PDT) and CO_2 flow sensor were calibrated by the manufacturer and their accuracy were ± 1 kPa, ± 1 Pa and ± 0.1% respectively. The accuracy of the air flow measured using the nozzle was ± 2% [3]. Moreover, the use of several sensors at critical location (e.g. air temperature sensors before and after the evaporator) and the correlation between the saturated pressure and temperature for CO_2 allow to rapidly identify possible inaccurate sensors. A data acquisition system is used to continuously record the value of all sensors and set-points four times per second.

Prior to the realization of a test, the room is cooled down using the air/brine coil located in the duct. When the required room temperature is reached, the brine stream to the air/brine coil is stopped and liquid CO_2 is circulated to the CO_2 evaporator. The CO_2 flow is adjusted using the variable frequency drive of the CO_2 pump and the electronic valves (CV). The CO_2 temperature range for this test bench is from -18°C to -35°C, corresponding to pressure in the range of 1200 to 2100 kPa. Stable CO_2 mass flow velocity is obtained in a wide range going from 0.002 kg/s to 0.100 kg/s, however the maximum flow is lower when pressure drop in the evaporator is high. Air velocity is adjusted using the variable frequency drive of the blower and the nozzle diameter; its range is from 0.047 m^3/s to 1.15 m^3/s.

3. RESULTS AND DISCUSSION

3.1. Stability and energy balance

Figure 3 is showing the air and CO_2 temperatures and the CO_2 flow during a typical test. CO_2 total flow was quite stable at about 0.0275 kg/s and recirculation ratio was around four to one. Air mass flow was 3.13 kg/m^2·s, corresponding to an air velocity of 2.23 m/s. During this test, CO_2 was at saturation conditions at both inlet and outlet of the coil, and the CO_2 temperature was lower at the outlet (-25.6°C) than at the inlet (-24.1°C) because of the CO_2 pressure drop in the evaporator. The air was cooled by about 2.5°C (from around -20.3°C to -22.8°C). An energy balance on the air side gave a capacity of 1.54 kW (at ± 6%).

In some other tests, experimental conditions were ajusted to obtain superheated CO_2 at the exit of the coil. This allowed to compare energy balances on air and on CO_2, and both values were in good agreement, the difference always being lower than ± 5%, which is within the experimental error.

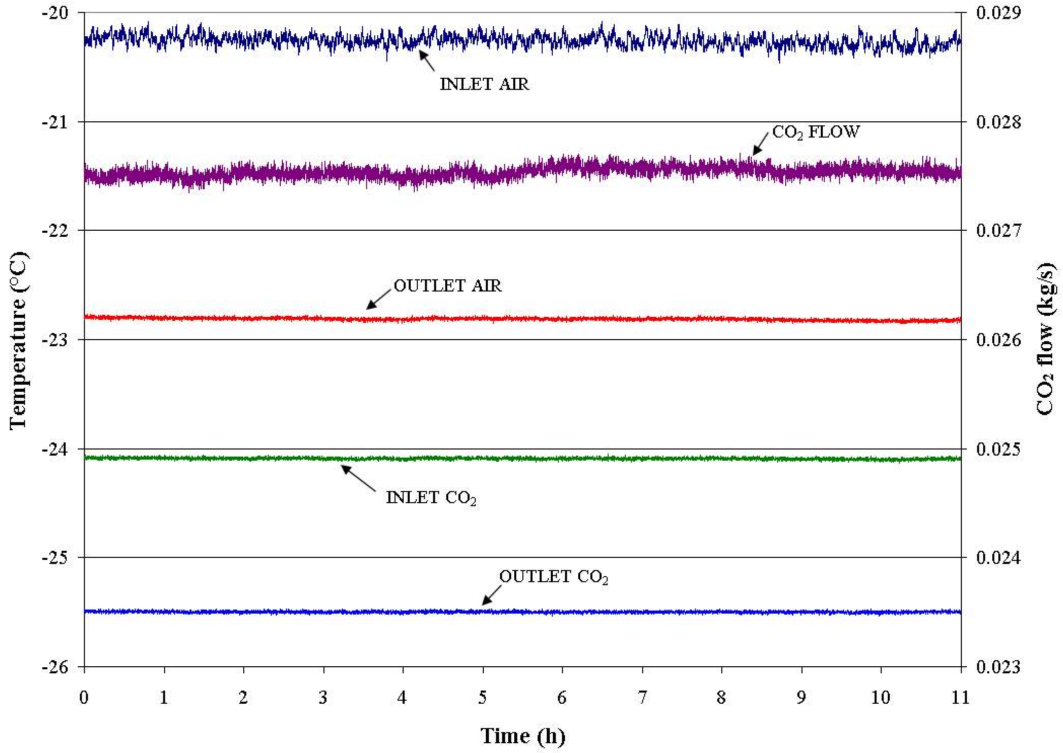

Figure 3. Typical results showing the stability of the system.

3.2. Pressure drop in the evaporator

The test bench was operated at different conditions over a period of 10 months. During that period, all tests were conducted in the absence of frost on the fins. In Table 1 are presented some of the results of tests identified by the corresponding date and time. These results were taken during periods of stable operation and are average values over a five-minute period (1200 measurements). Typically, the standard deviation for temperature values was 0.045°C.

The data reported in Table 1 were used to calculate the mass flow of CO_2 per unit flowing area (G_{CO2}). Data in Table 1 also allowed to perform energy balance on air, which led to the evaporator capacity and the CO_2 quality at the outlet of the evaporator.

In a recent paper, Ould Didi et al. [4] discussed and compared several models developed to predict the pressure drop (DP) in coil for different refrigerants. Those models were investigated but they all failed to predict the pressure drop observed in the present work (DP values in Table 1). However, it was found that the pressure drop measured in the evaporator can be well represented by a function of both the CO_2 flow (G_{CO2}) and the outlet quality (x), having the form:

$$DP = a \cdot (x)^b \cdot (G_{CO2})^c \tag{1}$$

Table 1. Experimental results for several tests.

Date	Time	$(T_{air})_{in}$	dT_{air}	$(T_{CO2})_{in}$	$(T_{CO2})_{out}$	Flow air	Flow CO_2	dP	G_{CO2}	Capacity	x
		°C	°C	°C	°C	kg/s	kg/s	kPa	kg/m^2.s	kW	
November 20, 2004	8h30	-17.91	2.42	-21.33	-23.20	0.592	0.0407	106	683	1.44	0.137
November 19, 2004	5h25	-16.58	2.41	-20.03	-21.73	0.586	0.0397	101	666	1.42	0.138
August 16, 2005	5h03	-21.53	2.72	-25.47	-27.27	0.585	0.0305	90	511	1.60	0.189
December 8, 2004	15h00	-20.32	2.52	-24.13	-25.59	0.600	0.0286	81	480	1.52	0.202
December 10, 2004	7h15	-20.32	2.49	-24.09	-25.50	0.603	0.0275	78	463	1.51	0.199
August 10, 2005	16h26	-20.53	2.81	-25.21	-27.54	0.836	0.0302	119	507	2.36	0.280
August 9, 2005	8h10	-20.91	2.67	-25.48	-27.59	0.850	0.0268	106	450	2.28	0.300
August 16, 2005	14h20	-17.56	5.13	-25.06	-27.67	0.585	0.0304	138	510	3.02	0.352
August 5, 2005	17h10	-18.95	3.62	-24.77	-27.36	0.839	0.0298	133	500	3.05	0.363
December 7, 2004	6h30	-20.31	2.43	-24.13	-24.90	0.606	0.0147	44	247	1.48	0.350
December 10, 2004	15h45	-20.25	2.51	-24.17	-24.94	0.600	0.0145	45	244	1.51	0.362
November 29, 2004	17h10	-20.05	3.08	-24.78	-25.89	0.611	0.0183	61	307	1.89	0.359
December 9, 2004	3h20	-20.19	2.52	-24.15	-24.93	0.606	0.0147	45	248	1.54	0.362
May 12, 2005	8h00	-10.08	9.04	-24.76	-28.87	0.587	0.0442	304	741	5.33	0.434
November 18, 2004	11h45	-17.38	2.46	-21.21	-21.75	0.587	0.0112	34	187	1.45	0.458
May 5, 2005	10h15	-12.66	7.83	-25.26	-28.68	0.598	0.0341	223	573	4.71	0.487
November 29, 2004	14h55	-20.04	4.06	-26.24	-27.62	0.611	0.0173	73	291	2.50	0.494
August 9, 2005	13h05	-17.51	4.20	-24.58	-27.12	0.836	0.0236	131	396	3.54	0.523
November 22, 2004	15h55	-20.50	2.57	-24.46	-25.05	0.593	0.0103	34	173	1.53	0.510
August 10, 2005	1h30	-17.39	4.27	-24.55	-27.09	0.838	0.0232	132	390	3.60	0.542
August 11, 2005	13h03	-14.56	5.85	-25.43	-28.33	0.845	0.0314	224	527	4.97	0.552
August 4, 2005	15h15	-15.88	6.39	-24.46	-27.03	0.592	0.0238	133	399	3.80	0.558
August 11, 2005	16h38	-13.72	6.17	-25.12	-28.03	0.843	0.0303	226	509	5.23	0.601
August 5, 2005	13h02	-12.20	6.39	-23.49	-26.39	0.835	0.0288	207	482	5.37	0.656
November 29, 2004	1h20	-19.93	5.09	-27.67	-29.35	0.603	0.0161	85	270	3.09	0.650
December 9, 2004	15h20	-20.03	2.48	-24.08	-24.54	0.606	0.0076	28	127	1.51	0.688
December 6, 2004	6h00	-20.06	2.46	-24.10	-24.55	0.610	0.0076	28	128	1.51	0.682
August 5, 2005	9h37	-9.21	7.62	-22.62	-25.71	0.825	0.0286	234	480	6.32	0.779
May 6, 2005	14h45	-15.67	7.29	-26.94	-29.62	0.597	0.0188	132	316	4.38	0.792
August 5, 2005	4h10	-8.57	9.84	-23.02	-26.31	0.576	0.0228	175	382	5.70	0.875
August 11, 2005	5h08	-7.59	9.01	-23.76	-27.14	0.825	0.0281	273	472	7.48	0.928

In a first step, the values of DP were plotted as a function of $(x)^n(G_{CO2})^m$ for different values of n and m. The best correlation was obtained for values of n = 1 and m = 1.981. Thus next step was to assume $F = x \cdot G_{CO2}^{1.981}$ and to realize a curve fit on the data (see Figure 4), which led to the following equation:

$$DP = 0.0169 \cdot F^{0.799} \quad (R^2 = 0.998) \tag{2}$$

And replacing F by $x \cdot G_{CO2}^{1.981}$ gave the final form of a correlation that predicts quite accurately the pressure drop in the evaporator for the test bench:

$$DP = 0.0169 \cdot (x)^{0.799} \cdot (G_{CO2})^{1.583} \tag{3}$$

Using this correlation to recalculate the DP for each of the tests of Table 1 showed a mean difference of 2.8% and a maximum difference of 6.1%.

The above correlation, as it is, only apply to the coil used in this work. It was validated for a G_{CO2} range from 125 to 750 kg/m^2·s, for all value of quality.

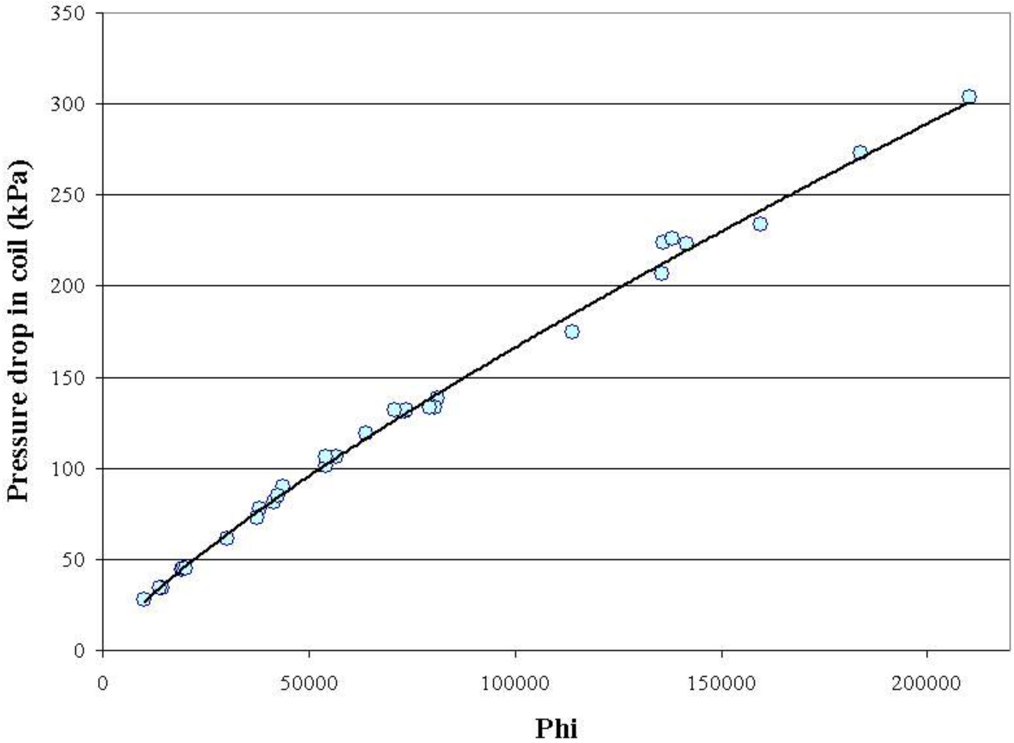

Figure 4. Pressure drop in the evaporator as a function of the CO_2 mass flow and outlet quality.

4. CONCLUSION

A well instrumented test bench was built in view to provide experimental data for the validation of a design tool that would predict the operating characteristics of refrigeration systems for applications in low temperature display cases for supermarkets using CO_2 as secondary refrigerant.

As a first result, an emperical correlation allowing to predict the pressure drop of CO_2 in the evaporator was obtained from a set of experimental results covering a given range of CO_2 flow and outlet quality.

Further tests will be done in view to enlarge the range of operating conditions for which this correlation is valid. Also, emperical correlations predicting other characteristics such as air and CO_2 outlet temperatures, evaporator capacity and CO_2 heat transfer coefficient will be developed.

NOMENCLATURE

dP and DP	pressure drop	(kPa)
dT	difference of temperature	(°C)
fn	function	
G_{CO2}	mass flow of CO_2 per unit flowing area	($kg/m^2 \cdot s$)
i.d.	inside diameter	
O.D.	outside diameter	
Phi and F	used in place of $x \cdot G_{CO2}^{1.981}$	
R^2	Variance	
T	temperature	(°C)
x	quality of the CO_2 at the outlet of the coil	

Subscripts

in	inlet
out	outlet

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