

DEVELOPMENT OF CO₂ VENDING MACHINE FOR HEATING AND COOLING OF BEVERAGE

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ABSTRACT

A CO₂ prototype system for heating and cooling of beverage in a three compartment Japanese type vending machine has been designed, installed and tested in a calorimetric test chamber. Experimental tests have been performed in three different operational modes: Pull-down (Cooling of beverage), pull up (heating of beverage) and power consumption test (keeping beverage at temperatures of 52°C and 4°C for 24hours).

For comparison of the CO₂ prototype system, a standard baseline system with a R407C cooling system and resistance heaters for heating, was tested in the same calorimetric chamber at equal operational conditions and test modes.

The CO₂ system was first equipped with round tube heat exchangers in all compartments and also as exterior heat exchanger. Later, two of the compartment heat exchangers were replaced by Microchannel Heat Exchangers, which improved the performance. All heat exchangers were designed by an advanced in-house calculation tool - HXsim.

Experimental results show that CO₂ performed significantly better than the baseline system at all operational modes. In pull-down, an energy saving of 33 to 36% was achieved, in pull-down 35% and in the important power consumption test: 27%.

Simulations, experiments and observations indicate that the performance of the prototype CO₂ system can be further improved. Main options are: Optimize air circulation in the compartments, reduce oil circulation, use only MPE heat exchangers and improve compressor efficiency.

1 INTRODUCTION

It is common in Japan to both cool and heat beverage in the same vending machine, which contains separate chambers allowing heating and cooling to occur simultaneously. Heating is normally performed by resistant heaters, and cooling by a conventional HFC cooling process with one evaporator for each chamber.

Earlier studies have shown that the use of carbon dioxide (CO₂) for cooling of beverage is an energy efficient alternative, e.g. DeAngelis & Hrnjak (2005) and Veje & Süß (2004). The purpose of present work was to study the potential of using CO₂ as working fluid for both heating and cooling in a hot-cold vending machine. By utilizing the warm side of the process, beverage could possibly be heated without the use of resistant heaters. Based on this idea, the present project was initiated in September 2004.

A CO₂ prototype system has been designed, built, installed and tested in a calorimetric test chamber. Experimental tests have been performed in three different operational modes: Pull-down (Cooling of beverage), pull up (heating of beverage) and power consumption test (keeping beverage at temperatures of 52°C and 4°C for 24hours). The prototype system was reconstructed during the project period by replacing two round-tube heat exchangers with Microchannel heat exchangers.

For comparison, a standard baseline system with a HFC-407C cooling system and resistance heaters, was tested in the same calorimetric chamber at equal test conditions.

2 SYSTEM DESCRIPTIONS

2.1 Vending machine

The vending machine used in both R407C and CO₂ experiments had 30-racks divided by insulated walls into three compartments (A (main), B and C), where the temperatures could be controlled independently from each other. The fans, heat exchangers and electrical heaters were mounted in the bottom of each compartment, see Figure 1. The exterior heat exchanger and the compressor were located below the conditioned chambers.

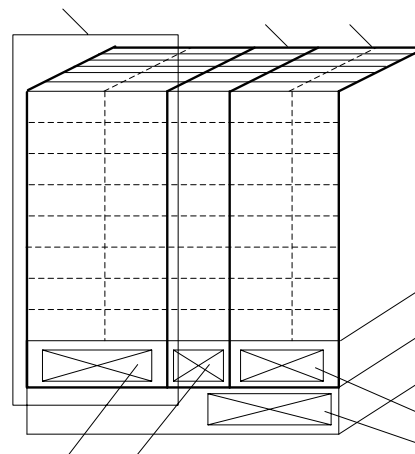


Figure 1: Principle sketch of three compartment vending machine

2.2 Baseline system

The baseline system consisted of three evaporators in parallel mounted in the compartments. Two of the chambers (B and C) had resistant heaters. The evaporator capacities were regulated by cycling of solenoid valves. Capillary tubes were used for throttling.

2.3 CO₂ prototype

The prototype system was designed based on principles suggested by Lorentzen & Pettersen (1993) for optimum control and operation of the trans-critical CO₂ process. The cycle includes a low pressure receiver (LPR) and an internal heat exchanger (IHX).

Design of heat exchangers

Advanced in-house simulation programs, HXsim (Skaugen, 2000) for heat exchangers and CSIM (Skaugen et al., 2002) for system simulations, were used to design of heat exchangers and system. The prototype had a total number of four air handling heat exchangers, equivalent to the baseline system. The difference was that the heat exchanger in compartment B and C were designed for reversible operation adapted to the beverage heating process. In addition, the exterior gas cooler was also designed for reversed operation, working as an evaporator in parts of the heating period, see below. All reversible heat exchangers had a counter-flow design. Both round tube in fin heat exchangers and microchannel heat exchangers were designed and evaluated. The baseline fans were used. See Table 1 for component specifications.

Due to time constraints in the project and availability of components, the first set of CO₂ heat exchangers were made as round tube in fin. The core volumes were equal to the R407C ones, but the fin density was a slightly higher for the CO₂ heat exchangers, increasing the air side surface by 33 to 57%, see Table 1. Continuous fins?

Also microchannel heat exchangers were designed for compartments B and C. Design point were gas cooler operation in pull-up mode. In order to avoid heat conduction, the fins were made non-continuous. Originally, the fins were specified as “wavy”, but they were delivered as “plain”.

The capacity of the IHX was optimized by use of CSIM for the situation of cooling compartment A and heating the two others (Pull-up).

Table 1: Component specification of commercial R407C system and Prototype CO₂ system

| | | R407C | R744 |
|--------|-------------------|--|---|
| System | Type | Commercial Cooling/heating system in 3-compartment vending machine | Prototype 3 compartment system in 3-compartment vending machine |
| | Expansion devices | 3 Capillary tubes | 4 Manually adjusted needle valves |

| | | | | |
|---------------|---------------------|--|--|--|
| Compartment A | Compressor | Horizontal hermetic | Hermetic 2-stage rolling piston. Displ.: 1.5cm ³ /rev | |
| | Operation | Dedicated cooling | Dedicated cooling | |
| | HX description | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | |
| | Fin spacing | 5.1 mm | 3.2 mm | |
| | Face area | 0.0494 m ² | 0.0494 m ² | |
| | Core depth | 0.038 m | 0.038 m | |
| | Fin air side area | 0.74 m ² | 1.18 m ² | |
| Compartment B | Operation | Cooling and heating | Cooling and heating | |
| | HX description | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | 3-row vertical serpentine, 10 channel x 1mm MPE tubes in Al fins, 2 circuits |
| | Fin Spacing | 4.2 mm | 3.1 mm | |
| | Face area | 0.0147 m ² | 0.0147 m ² | 0.0143 m ² |
| | Core depth | 0.038 m | 0.038 m | 0.06 m |
| | Fined air side area | 0.27 m ² | 0.36 m ² | 0.487 m ² |
| | Heating device | 300 W Electrical resistance heater | Reversed operation | |
| Compartment C | Operation | Cooling and heating | Cooling and heating | |
| | HX description | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | 6 Vertical*2 Horizontal Cu-tubes in Al-fins, 1-circuit | 3-row vertical serpentine, 10 channel x 1mm MPE tubes in Al fins, 2 circuits |
| | Fin Spacing | 5.1 mm | 3.2 mm | |
| | Face area | 0.0325 m ² | 0.0325 m ² | 0.0317 m ² |
| | Core depth | 0.038 m | 0.038 m | 0.06 m |
| | Fined air side area | 0.49 m ² | 0.77 m ² | 0.95 m ² |
| | Heating device | 300 W Electrical resistance heater | Reversed operation | |
| Exterior HX | Operation | Heat rejection | Heat rejection and absorption | |
| | HX description | 8 Vertical*3 Horizontal Cu-tubes in Al-fins, 1-circuit | 8 Vertical*3 Horizontal Cu-tubes in Al-fins, 1-circuit | |
| | Fin Spacing | 4.2 mm | 3.2 mm | |
| | Face area | 0.0655 m ² | 0.0694 m ² | |
| | Core depth | 0.055 m | 0.055 m | |
| | Fined air side area | 1.70 m ² | 2.4 m ² | |
| | Heating device | - | Reversed operation | |

3 TEST CONDITIONS AND EXPERIMENTAL FACILITIES

Three different test modes were examined, according to specifications of The Coca Cola Company, see Table 2. I) Pull-down: Beverage in all compartments is cooled down. II) Pull-up: Beverage in compartment A is kept cool and beverage in compartment B and C is heated up. III) Power consumption test: Beverage is kept at constant temperature for 24h, cold in compartment A and B, hot in compartment C.

Table 2: Test conditions

| Test | Compartment A, B and C settings*** | Ambient temperature | Ambient RH | Start of test cold compartment temperature | Start of test hot compartment temperature | End of test cold compartment temperature | End of test hot compartment temperature |
|----------------|------------------------------------|---------------------|------------|--|---|--|---|
| | A – B – C | [°C] | [%] | [°] | [°C] | [°] | [°C] |
| Pulldown | C – C – C | 32 ± 1.1 | 65 ± 5 | 32 ± 1.1 | - | 5 * | - |
| Pulldown | C – C – C | 24 ± 1.1 | 45 ± 5 | 24 ± 1.1 | - | 5 * | - |
| Pullup | C – H – H | 5 ± 1.1 | 45 ± 5 | 0 < T *** < 5 | 5 | 0 < T *** < 5 | 52 ** |
| Power consump. | C – C – H | 15 | 50 ± 5 | 0 < T *** < 5 | T *** > 52 | 0 < T *** < 5 | T *** > 52 |

* Test ended when average first available product was below this value

** Test ended when average first available product was above this value

*** “Steady state” value during test

**** C = Cold, H = Hot

In order to cover the cooling demand and heating demand for the 24h energy test, it was necessary to operate the system in two different modes:

- Mode 1 was covering cooling demand in compartment A and B and heating in compartment C. Exterior heat exchanger was operated as gas cooler.
- Mode 2 was covering only heating of compartment C. Exterior heat exchanger was used as evaporator.

From achieved results in mode 1 and 2, the total energy consumption was calculated according to COP and operation time at each mode.

The vending machine was placed inside an insulated chamber where temperature and relative humidity were controlled. Temperatures were measured by thermocouples type T, pressures by Druck pressure sensors, and the mass flow by a Rheonik coriolis mass flow transducer. Refrigerant temperatures were measured on the outer tube wall, whereas the beverage temperatures were measured inside the cans by cannula tubes.

4 INITIAL SYSTEM SIMULATIONS – PULL UP

The pull-up situation was from the beginning regarded as the critical mode for the CO₂ system, since the air inlet temperatures to the compartment gas coolers would reach around 40°C at the end of the heat up period. Initial system simulations were performed where all heat exchangers were replaced by either round tube in fin heat exchangers (wall thickness of 0.038”), or initial generic microchannel heat exchangers. All heat exchangers were constrained to the same core volumes as the baseline heat exchangers.

Simulation results in Figure 2 show that the heating capacity is above baseline capacity at all air inlet temperatures for both round tube and microchannel heat exchangers. The COP decreased as expected with increasing air inlet temperatures, but results indicated well above 1 also at heat up termination.

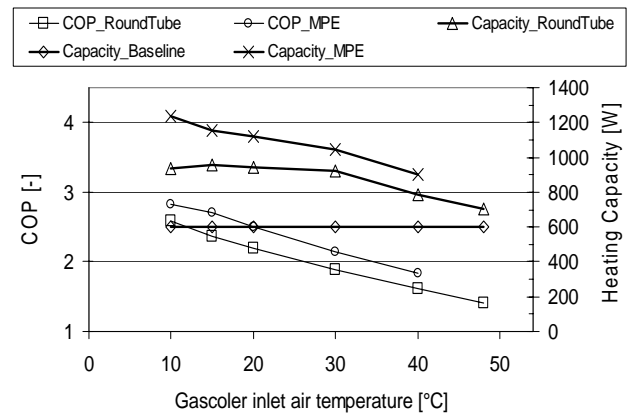


Figure 2: Initial simulation results of CO₂ prototype performance in pull-up mode

5 TEST RESULTS, SYSTEM COMPARISON AND DISCUSSION

Main experimental results for the R407C baseline system and the CO₂ prototype with tube-in-fin and microchannel heat exchangers are presented in the present chapter.

5.1 Pull-up tests

Results in Figure 4 show that the prototype CO₂ “tube in fin” system heats the cans faster at lower beverage (or compartment) temperatures than the baseline system. At higher beverage temperatures, the heating capacity of the CO₂ system dropped considerably, and the target beverage temperature of 52°C was not reached. The electrical heaters of the baseline system outperformed the tube-in-fin system. From heating capacity measurements shown in Figure 4, it is evident that the tube in fin system had significantly lower capacity than initial simulations indicated, see Figure 2. The main reason was that the delivered tubes had lower wall thickness than in the calculations. It was assumed to be 0.038”, but was delivered with 0.016”, increasing the cross section area by 47%.

According to the initial simulations above, microchannel heat exchangers would improve the performance. It was decided to replace the two most critical heat exchangers. Optimized microchannel heat exchangers were designed for compartment B and C. Data are given in Table 1.

With the new heat exchanger configuration, the prototype CO₂ heating capacity increased, and the performance was significantly better than both the baseline system and tube-in-fin configuration. The target beverage temperature of 52°C was reached around 200 minutes faster than the baseline system. The power consumption was slightly higher, but the energy consumption was 35% lower.

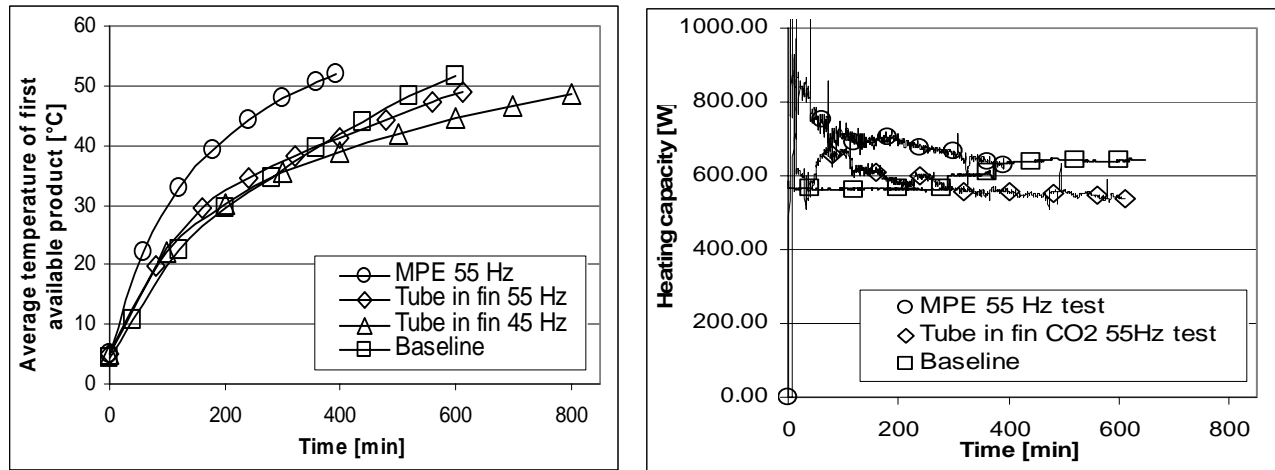


Figure 4: Measured performance of pull-up tests. CO₂ compressor speed in Hz.

Since the air inlet temperatures to gascoolers necessarily became high as the heating progressed, it was of vital importance to achieve as low temperature approaches as possible. Results in Figure 5 show a comparison of the gas cooler temperature approaches for the two 55 Hz CO₂ tests with tube in fin and microchannel heat exchangers. It is evident from the results that the introduction of MPE heat exchangers gave a significant improvement. The temperature approaches of heat exchanger B and C were reduced by approximately 10K and 5K respectively.

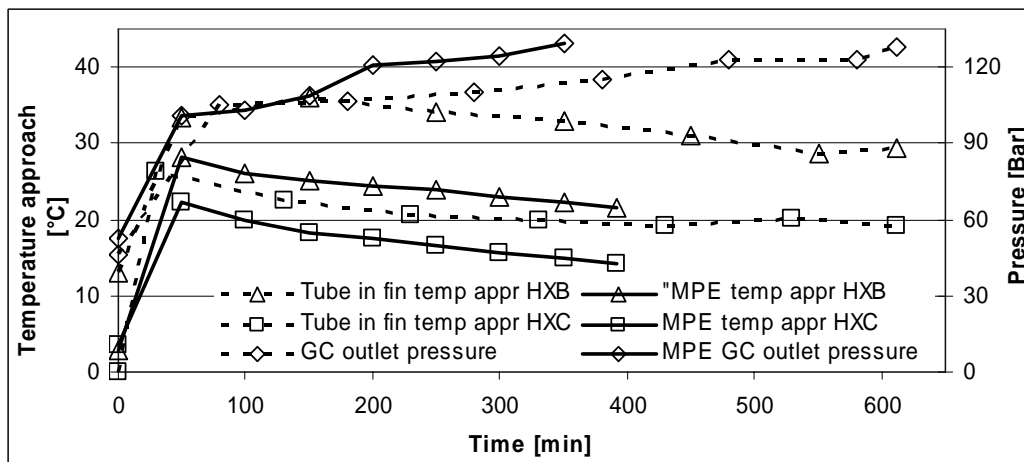


Figure 5: CO₂ gas cooler temperature approach comparison for the pull-up tests

In the pull-up tests, it was important to conserve the heat and reject it at useful location. Refrigerant line heat losses caused the performance to be poorer than necessary. Calculations for the tube in fin tests based on measured data revealed a discharge line heat loss of 91W for the 55 Hz test, which was significant. Better insulation had improved the pull-up tests.

High oil circulation did most likely contribute to a poorer performance than simulated. Cold oil entering the compressor reduces the discharge temperature, since an incompressible fluid such as oil, do not increase its temperature in an adiabatic compression process, and oil will therefore be heated

with energy from the compressor discharge gas. Reduced temperature into the gas cooler decreases the LMTD and thereby the ability to reject heat over the same heat exchanger surface

5.2 Pull-down tests

As Figure 3 shows, the prototype CO₂ system performs better than the baseline R407C system in both the pull-down tests, especially when microchannel heat exchangers were applied in compartment B and C. The energy consumption was then 36% and 33% lower for the 24°C and 32°C tests respectively, compared to the baseline tests. The pull-down time was reduced with over 40% for both the tests.

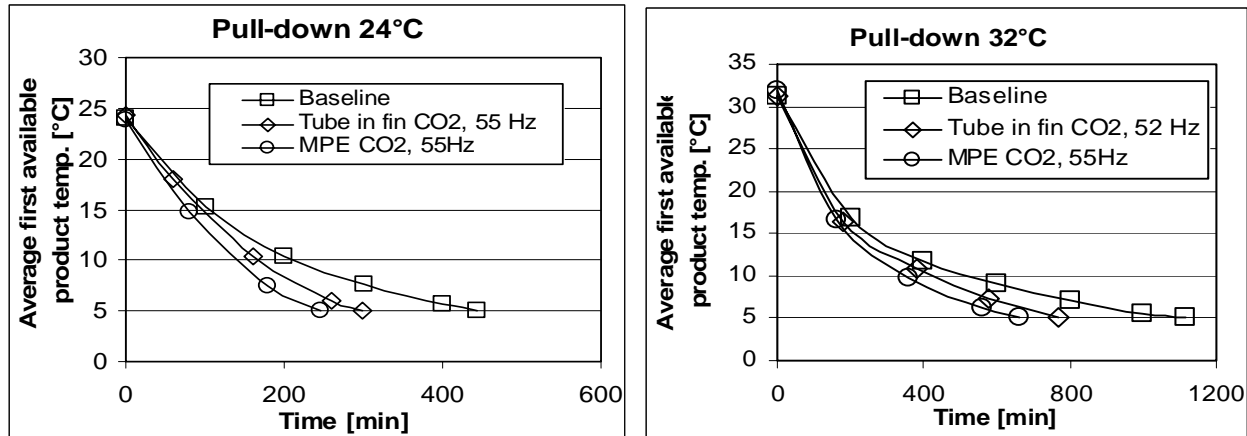


Figure 3: Measured performance in pull-down test at 24°C and 32°C. Compressor speed: 55Hz

In the tube-in-fin tests, the beverage in compartment A was cooled down faster than the other compartments, especially compared to compartment B. The opposite was observed after switching to microchannel heat exchangers in compartment B and C. Then, compartment B had the fastest pull-down, and was significantly faster than compartment A, which still had the tube-in-fin heat exchanger. If a microchannel heat exchanger had been installed in compartment A as well, an even larger performance improvement would most likely have been achieved.

5.3 Power consumption test

The aim of the power consumption test was to reduce the power consumption by 30% compared to the baseline system. As explained in Chapter 3, the test consisted of two operational modes: Mode 1: Combined heating and cooling, and Mode 2: Heating only.

5.3.1 Combined heating and cooling

With combined heating and cooling the CO₂ systems performed better than the baseline system. As can be read from Table, the test with the MPE had the highest combined capacity combined with the lowest power consumption, 4% higher combined heating and cooling capacity and 30% lower power consumption.

5.3.2 Heating only operation

Only compartment C was heated, and for the baseline system this was carried out by resistant heaters. When the CO₂ system was used for heating, it was switched on and off to achieve correct average beverage temperature. The average baseline power consumption was 21% lower than the CO₂ system with tube-in-fin heat exchangers, whereas 5% higher after switching to MPE heat exchangers. The gain by using the CO₂ system as heat pump was still limited.

However, by increasing the air flow over the heat exchanger by roughly 40 to 50% (measured by handheld propeller), the performance improved considerably. Measured power consumption was 14% lower than the baseline system. The fan power was increased minimally in absolute numbers; from 4.6W to 6.9W. Improved heat transfer on the air side gave around 5K lower temperature

approach. Further improvements can be expected by optimizing the compartment airflows for the CO₂ system. The compartment airflows for the R407C system were just adopted when switching to CO₂.

Total energy saving for the CO₂ prototype during 24h operation was 27% with microchannel heat exchangers in compartment B and C, and with increased airflow in compartment C.

Table 3: Heating and cooling results

| Test | | Unit | Value |
|----------------|------------------------------|------|-------|
| Baseline | Heating and cooling capacity | W | 532 |
| | Power consumption | W | 571 |
| Tube in fin HX | Heating and cooling capacity | W | 550 |
| | Power consumption | W | 421 |
| MPE HX | Heating and cooling capacity | W | 554 |
| | Power consumption | W | 401 |

6 SIMULATION OF IMPROVEMENT ALTERNATIVES

Both by comparing the initial simulations with analysis of the tests in various quasi-steady-state operating points, it became evident that there is still room for significant improvements. Focus was put on conditions at the end of a pull-up cycle, where the CO₂ system had the most challenging working conditions. The in-house simulation program CSIM was used to calculate the effect of the improvement alternatives.

An average air inlet temperature in compartment B/C heat exchangers of 35°C was selected. The heat source temperature was 5°C. In the basecase, CSIM was calibrated by tuning the refrigerant line heat loss, compressor heat loss and efficiencies and air side heat transfer coefficients at equal evaporating temperatures and refrigerant flow. Next, characteristics of the gascooler B/C air side heat transfer coefficient and fan power consumption as a function of air face velocity were generated, and used as part of the optimization scenario.

The high side pressure in the basecase test was 124 bar. The optimization scenarios were (accumulated): Optimize the high side pressure (1), optimize the compartment B/C air face velocities (2), optimized with improved compressor efficiencies (3) and compressor speed (4). The results from the simulations are shown in Table 4 below.

Table 4: Simulation of potential for improvement in pull-up mode

| | Uface [m/s] | Php [bar] | Speed [RPM] | Qheat [W] | Pcomp [W] | PfanEV [W] | PfanGC [W] | System COP |
|----------------------------------|----------------|--------------|----------------|--------------|--------------|---------------|---------------|---------------|
| Basecase | 0.8 | 124 | 3300 | 675 | 698 | 30 | 9.3 | 0.916 |
| Optimum high side pressure | 0.8 | 130 | 3300 | 692 | 702 | 30 | 9.3 | 0.935 (2.1) |
| Optimum air face velocity | 1.31 | 130 | 3300 | 740 | 699 | 30 | 35.9 | 0.968 (5.7) |
| Improved compressor efficiencies | 1.16 | 130 | 3300 | 678 | 518 | 30 | 26.8 | 1.180 (28.9) |
| Optimum compressor speed | 1.01 | 130 | 2907 | 600 | 458 | 30 | 17.8 | 1.187 (29.7) |

In the basecase, the heating capacity and compressor powerconsumption is 675 and 698 Watt as measured. The compartment B/C fan powerconsumption was 9.3 W with a face velocity of approximately 0.8m/s. The system COP ($Q_{\text{heat}}/(P_{\text{comp}}+P_{\text{fanEV}}+P_{\text{fanGC}})$) was 0.916. By increasing the pressure from 124 to 130 bar at equal airflow rate, the COP could be increased by 2.1% to 0.935. By also increasing the gas cooler airflow rates to 1.31 m/s, the COP could be increase by 5.7%.

The compressor efficiencies was in these tests unexpectedly low, giving a high power consumption, meaning the COP is not so sensible to the power consumptions for the fans. As seen from Table 4, this has increased from 9.3 to 35.9 Watt. By using a more efficient compressor, (the isentropic and volumetric efficiencies increased from around 0.4 to around 0.6), the COP is expected to be 28.9% higher. Then it is also more advantageous to reduce the airflow again – to 1.16 m/s. With higher

volumetric efficiency of the compressor, the refrigerant pressure drop is increased, so the optimum compressor speed is found to be 2907 RPM at required heating capacity of 600 W.

7 CONCLUSIONS

A CO₂ prototype system for heating and cooling of beverage in a three compartment Japanese type vending machine has been designed, installed and tested in a calorimetric test chamber. Experimental tests have been performed in three different operational modes. The prototype was initially equipped with round tube heat exchangers. Later, two of the compartment heat exchangers were replaced by microchannel heat exchangers. Main experimental results are shown in the table below.

Table 3: Test results comparison

| | R407C | | CO ₂ tube in fin HX | | CO ₂ MPE HX | |
|------------------------------|----------|--------------|--------------------------------|--------------|------------------------|--------------|
| | Time [h] | Energy [kWh] | Time [h] | Energy [kWh] | Time [h] | Energy [kWh] |
| Pull-down 32 °C | 18.6 | 10.5 | 12.8 | 7.6 (-28%)* | 11.1 | 7.0 (-33%) |
| Pull-down 24 °C | 7.4 | 3.6 | 5.0 | 2.5 (-31%)* | 4.1 | 2.3 (-36%) |
| Pull-up 5 °C | 10.2 | 6.75 | | | 6.5 | 4.4 (-35%) |
| Power consumption test 15 °C | 24 | 6.8 | 24 | 6.2 (-9%)* | 24 | 5.2 (-24%) |
| | | | | | 24 | 4.9 (-27%)* |

* Test with increased compartment C air flow

The CO₂ unit with MPE heat exchangers in compartment B and C gave significantly better results compared to both the baseline and tube-in-fin units for all the tests so further improvements are possible letting microchannel heat exchangers replace all round tube heat exchangers. Simulations show that with an improved compressor 20-25% improvement in efficiency can be expected, Further improvements is possible by optimizing the compartment air flow rates. During the limited time in the present project, it was not possible to do enough experiments and simulations to reveal an optimum regulation strategy covering all operational configurations, compartment temperatures and ambient conditions.

8 ACKNOWLEDGEMENT

We are grateful to Mr. Christopher Richard and Mr. Darren Simmons from The Coca Cola Company who initiated and funded this project.

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