R-744 COMPARED TO R-290 IN SMALL FREEZER APPLICATIONS

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ABSTRACT

An experimental and analytical comparison between R-744 (CO_2) and R-290 (Propane) as refrigerant for small capacity freezer applications is described. A commercially available plug-in freezer cabinet for sales of frozen foods or ice cream, fitted with a R-290 (8,1 cm^3) hermetic compressor, was retrofitted to R-744 using standard components and a state of the art R-744 compressor. The results of energy consumption and storage temperature testing on both systems are presented as well as a discussion on the system modifications required for using R-744. Both refrigerant loops are analysed and the deviations with the thermodynamically ideal loops are discussed. The analysis is further supported by calorimetric tests of the R-744 compressor at low evaporation temperatures. Theoretical comparison of the ideal loops, using performance data from the compressors used in the experiments, shows that the R-744 circuit is 31% less efficient than the R-290 circuit, at normal operating conditions for frozen foods or ice cream. Comparison of the experimental results from the almost fully optimised R-744 freezer with a standard R-290 freezer at an ambient of 30ºC, shows that the R-744 freezer is 23% less efficient than the R-290 freezer, based on the ISO 23953-2 methodology (warmest product temperature of -18ºC). The R-744 is 13 % less efficient than the R-290 freezer at an average product temperature of –23°C. Finally, costs of major components of both systems are evaluated. This shows that the difference in production costs between a R-744 and a R-290 freezer is mainly driven by the compressor.

1 INTRODUCTION

This paper presents the final results of an experimental and analytical comparison between R-744 (CO_2) and R-290 (Propane) as refrigerant for small capacity freezer applications performed during 2006 and 2007. The study is based on comparing appliance performance of a commercially available Ice Cream Cabinet retrofitted to R-744 with the original R-290 based system. Originally the cabinet was equipped with a standard reciprocating 8.1cm^3 R-290 compressor (ACC NL80FB). Based on commercial availability and capacity estimations a reciprocating R-744 compressor with a swept volume of 1.54 cm^3 was selected (Danfoss TN 1410) for retrofitting. A picture of the appliance evaluated is presented in Figure 1. The objective of the study was to evaluate the potential of R-744 in comparison to R-290, both technically and commercially, for small capacity freezer applications. The main technical evaluation parameters were the energy consumption at climate class 4 (ambient at 30°C and a relative humidity of 55%) and temperature performance at climate class 7 (ambient at 35°C and a relative humidity of 75%). The cabinet tests were performed according ISO 23953-2.

Figure 1. Appliance evaluated

2 RETROFITTING

The cooling system of the R-290 based system is made up of a small pre-cooler (1.9 meter of copper tube with an outer diameter of 6 mm), a skin condenser, a standard filter dryer, skin evaporator, and a capillary tube as expansion device. The capillary exchanges heat with the suction tube to increase system efficiency. The operating pressures for R-744 are typically much higher than the operating pressures for R-290; therefore the filter and pre-cooler of the original R-290 system needed to be exchanged for higher pressure resistant versions. To establish a fair comparison, both cost and technical wise, the foamed in parts (both condenser and evaporator) were not modified in order to keep the insulation quality of the cabinet unchanged. As well as changing the pre-cooler and filter dryer, the R-744 system was also fitted with an accumulator and an after cooler. The accumulator was only added as an additional safety provision for testing purposes, mainly during charge determination. The after cooler was implemented to decrease the gas cooler exit temperature, thereby increasing system efficiency. Since operating pressures and refrigerant flow are quite different in a R-744 configuration, the capillary tube and capillary heat exchanger were extended (total flow of 2.95 dm^3/min of N2 at 10 bar pressure difference). After retrofitting, charge tests were performed on the R-744 system. These tests showed that the appliance was properly filled with a refrigerant charge of 433 grams. At this condition the evaporation temperature was close to the temperature obtained in the R-290 system and the discharge pressure was close to optimal (pressure corresponding to maximum efficiency). The system configurations of both the original R-290 and retrofitted R-744 system are presented in Figure 2.

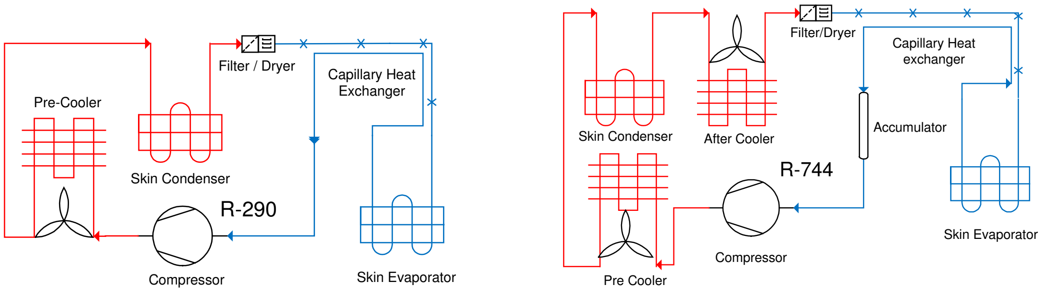

Figure 2. System configurations

3 TEST RESULTS

Energy consumption and storage temperature tests according ISO 23953-2 were performed on both systems at respectively climate class 4 and climate class 7. The main results of these tests are presented in Table 1 and the loading scheme of the appliance is presented in Figure 3. In order to quantify system performance, the energy consumption was measured using a Yokogawa WT130 power meter and the package temperatures were measured using T type thermocouples and a Fluke Helios I 2289A data acquisition module. Additional thermocouples were mounted on the pre-cooler, condenser, evaporator, suction line of the compressor, and directly after the suction gas heat exchanger. The R-744 system was further fitted with two pressure transducers respectively recording the suction pressure using a Druck PDCR 961 with a range of 0 to 20 bar and the discharge pressure using a Kulite IPT 1000 with a range of 0 to 345 bar.

Table 1. Results energy consumption testing at CC4. *Uncertainty presented results from; ± 0.15 K uncertainty thermocouple reading, ± 0.3K uncertainty reproducibility (placing of packages) and ± 0.054 kWh/24h uncertainty power meter reading.*

		Warmest package [°C]	Number of warmest package	Energy consumption [kWh/24h]	Running time [%]
R-290	Test 1	-18.7	20	3.86	66.0
	Test 2	-17.4	20	3.80	63.6
	Interpolated	-18.0		3.83 ± 1.5%	64.7
R-744	Test 1	-17.6	3	4.60	60.6
	Test 2	-19.1	3	5.01	67.1
	Interpolated	-18.0		4.71 ± 2.3%	62.3

Table 2 Results storage temperature testing. U*ncertainty presented results from; ±0.15 K uncertainty thermocouple reading and uncertainty of ± 0.3K reproducibility (package placement).*

	Warmest package [°C]	Number of warmest package	Running time [%]
R-290	-19.4 +/- 0.45	20	100
R-744	-21.3 +/- 0.45	3	100

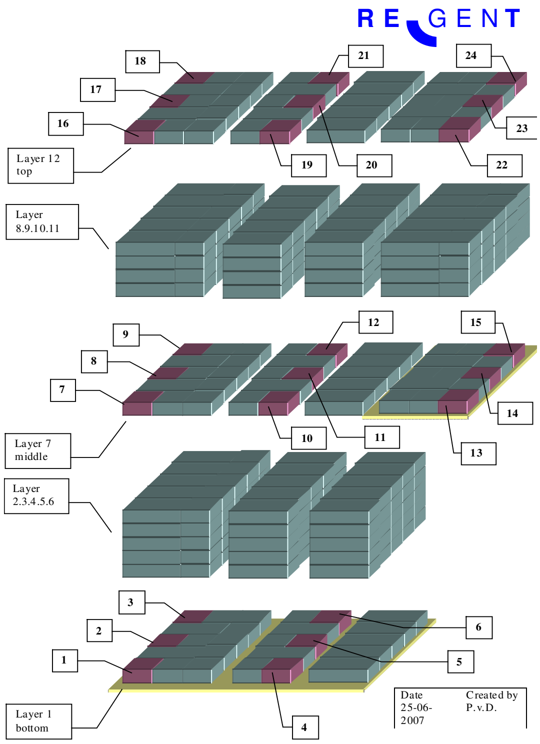

Figure 3. Loading scheme appliance

The tests showed that the energy consumption of the R-744 system was approximately 23% higher than the energy consumption of the R-290 system. The storage tests resulted in a difference in warmest package temperature of 1.9°C between the R-290 and the R-744 configuration. From this and the difference in compressor running time percentage during the energy consumption test it was concluded that the cooling capacity of the R-744 system is somewhat higher than the cooling capacity of the original system. The test also showed that the location of the warmest package changed from the top centre position in the R-290 system to a package placed at the bottom of the cabinet in the R-744 system. This indicates that the two systems have a different thermal resistance between the packages and the evaporating refrigerant inside the evaporator. The evaporator of the freezer is a tube mounted against the inner liner of the cabinet (skin evaporator). At the top part of the cabinet, the distance between the tubes is quite small and increases towards the bottom. At the bottom plate of the cabinet there are no tubes at all. The cooling of the packages located at the bottom region mainly results from air circulation as a result of natural convection combined with conduction through the inner liner into the upper tubes. An estimation of the heat resistances based on the equations as presented by Gungor and Winterton (1987) indicated that the thermal resistance between the evaporator pipe and the evaporating refrigerant is approximately 24% larger for the R-744 configuration compared to the R-290 configuration. From this it was concluded that the original design of the evaporator is not optimal for the R-744 configuration and that the performance could be improved by changing the pitch and the internal diameter of the evaporator tubing. Due to the change in the position of the warmest zone, the heat load will be different between the two systems when the evaluation is based on the warmest package. Since comparing the appliance performance at equal heat load is more correct, the appliance performance was also evaluated at average package temperature next to the ISO 23953-2 methodology of using warmest package. Both comparisons are presented in Table 3.

Table 3 Energy consumption comparison. *Uncertainty presented results from, ±0.15 K uncertainty thermocouple readings, ±0.3K uncertainty reproducibility (placing of each packages) and ±0.054 kWh/24h uncertainty power meter reading.*

	Average package		Warmest package	
	Temperature [°C]	Energy consumption [kWh/24h]	Temperature [°C]	Energy consumption [kWh/24h]
R290	-23.0	3.80 ± 1.4%	-18.0	3.83 ± 1.5%
R744	-23.0	4.30 ± 1.3%	-18.0	4.71 ± 2.3%

At equal average package temperature the energy consumption measured in kWh/24h of the R-744 configuration is 13.2% (± 2.2)% larger than the energy consumption of the R-290 configuration . Based on the ISO 23953-2 methodology (warmest package) the difference in measured energy consumption is 23.0 (± 2.8)%.

4 REFRIGERANT LOOPS

In Figure 4 the measured and the estimated ideal loops for the R-290 and R-744 system are presented. The loops were constructed in a spreadsheet based model set up in Excel. The model calculates the refrigerant properties at specified conditions using the refrigerant property program REFPROP 6.01 (McLinden et al, 1998). The model uses the isentropic and volumetric efficiency, as defined in eq. 1 and 2, respectively, to characterise the compressor and to determine the cooling capacity, energy consumption and resulting C.O.P. of the configuration.

$$\eta_i = \frac{\dot{m}\,\Delta h_{isen}}{P_{in}} \tag{1}$$

$$\eta_v = \frac{\dot{m}}{\rho_{suc} V_{swept} f} \tag{2}$$

$$\dot{m} = \frac{Q_{cool}}{\Delta h_{cool}} \tag{3}$$

$$\Delta h_{cool} = h_{subcool} - h_{superheat} \tag{4}$$

$$C.O.P = \frac{Q_{cool}}{P_{in}} \tag{5}$$

Where $\dot{m}$ is the massflow in kg/s, Δh_{isen} the enthalpy increment of the fluid in J/kg at isentropic compression from the suction to the discharge pressure, P_{in} the energy consumption in W of the compressor, ρ_{suc} the density of the suction gas in kg/m^3, V_{swept} the swept volume of the compressor in m^3, f the operating frequency of the compressor in Hz, Q_{cool} the cooling capacity of the system, $h_{subcool}$ the enthalpy in J/kg of the medium prior to the expansion device and $h_{superheat}$ the enthalpy of the refrigerant after the suction gas heat exchanger.

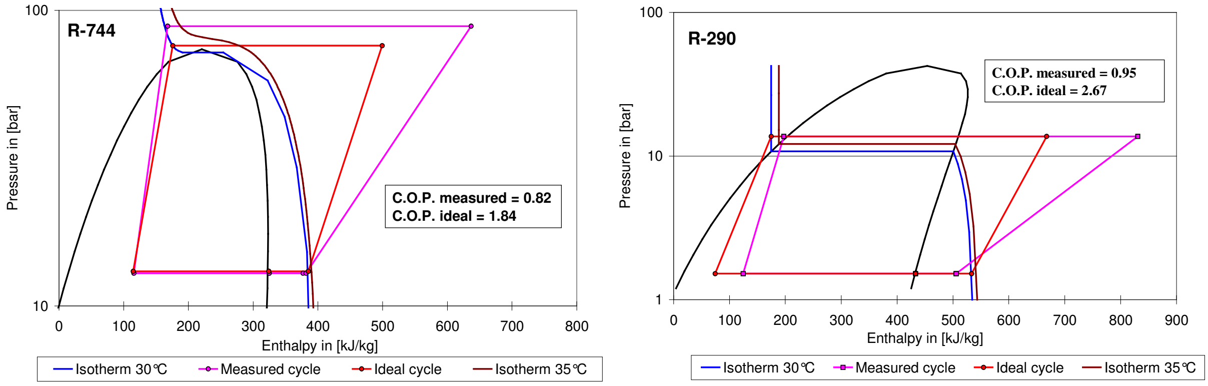

Figure 4. Measured and theoretical ideal loops

The theoretical loops are based on ideal compression, ideal heat exchange of the capillary heat exchanger and a subcooling temperature equal to the ambient (30°C). The evaporation temperature was taken from the measurement with the R-290 configuration (-32.5°C). For the R-290 loop the condensation temperature was set to the measured 40°C and in the R-744 loop the discharge pressure was fitted to obtain maximum C.O.P. at the conditions specified. The "measured" loops were based on the recorded system temperatures and pressures (e.g. pre-cooler, condenser, evaporator, suction line and subcooling) during the energy consumption measurements. As input for the model the average values during a single on cycle of the compressor at stable conditions according ISO 23953-2 were used. The isentropic and volumetric efficiencies were fitted on the measured compressor energy consumption and calculated appliance heat load. The heat load of the appliance (135W+/- 10%) was determined with the model by calculating the isentropic and volumetric efficiency of the R-290 compressor from commercially available manufacturer data at conditions corresponding to the energy consumption tests performed with the R-290 system (evaporation temperature of –32.5°C, condensation temperature of 45°C, suction gas temperature of 32°C).

Visible from Figure 4 is that the ideal R-744 loop is approximately 31% less efficient than the ideal R-290 loop. Also visible from the figure is that this deviation is largely compensated in the configurations built, as the efficiency difference resulting from the measurement is estimated at approximately 13.7%.

In Table 4 and Table 5 the deviations between the "measured" and the ideal loop have been presented for respectively the R-290 and the R-744 configuration.

Table 4. Deviation from ideal loop R-290

R-290	Isentropic efficiency [-]	Subcooling temperature [°C]	Capillary heat exchanger efficiency [-]	Evaporation temperature [°C]	Condensation temperature [°C]	C.O.P [-]	C.O.P. deviation [%]
Ideal	1	30	1	-32.5	40	2.67	0%
Isentropic efficiency	0.39	30	1	-32.5	40	1.04	- 61 %
Subcooling temperature	1	38	1	-32.5	40	2.53	- 5 %
Heat exchanger efficiency	1	30	0.66	-32.5	40	2.59	- 3 %
Measured Loop	0.39	38	0.66	-32.5	40	0.95	-64 %

Table 5. Deviation from ideal loop R-744

R-744	Isentropic efficiency [-]	Gas cooler exit [°C]	Capillary heat exchanger efficiency [-]	Evaporation temperature [°C]	Discharge pressure [°C]	C.O.P [-]	C.O.P. deviation [%]
Ideal	1	30	1	-32.5	76	1.84	0
Isentropic efficiency	0.49	30	1	-32.5	76	0.90	- 51
Gas cooler exit	1	31.1	1	-32.5	76	1.75	-5
Heat exchanger efficiency	1	30	0.85	-32.5	76	1.79	-2
Evaporation temperature	1	30	1	-32.9	76	1.82	-1
Discharge pressure	1	30	1	-32.5	88.4	1.77	-4
Measured Loop	0.49	31.1	0.85	-32.5	88.4	0.82	-55

From the tables it can be seen that the deviations from the ideal loops are mainly resulting from the deviation in isentropic efficiency of the compressor. Also visible is that the isentropic efficiency of the R-744 compressor is estimated to be 26% larger than the isentropic efficiency of the R-290 compressor. This difference in isentropic efficiency is the main reason for the reduction in efficiency deviation from 31 to 13.2% between the ideal and the measured loops. The isentropic efficiency is strongly related to the electric motor and the compression process. The manufacturer of the R-744 compressor made it clear that the compressor is fitted with a standard induction motor using a start and run capacitor (CSCR motor). The R-290 compressor is only fitted with a start capacitor (CSIR), which explains a part of the deviation. The efficiency increase of the CSCR motor over the CSIR motor ranges from about 8 to about 10 percent (Arthur D. Little, Inc, 1999). R-290 compressors with isentropic efficiency similar to the R-744 compressor are commercially available. For example the ACC NLY80LA_b is a R-290 compressor with equal swept volume as the NL80FB and approximately 25% higher efficiency than the NL80FB.

The compression chamber of the R-744 compressor is located on the compressor shell as visible in Figure 5. Therefore the R-744 compressor has some principle efficiency benefits in comparison to the standard configuration used for the R-290 compressor. The heat absorption of the suction gas entering the compression chamber of the R-744 compressor is relatively low as the gas does not pass through the compressor shell. The shell of the R-744 compressor is quite thick and directly connected to the compression chamber, thereby cooling the gas during the compression process. This effect is further enhanced by the higher surface to volume ratio of the R-744 compressor (1.5cm^3) compared to the R-290 compressor (8.1cm^3). Also the higher absolute pressures have a positive impact on the efficiency, as relatively the work required for opening the valves decreases with absolute pressure. The effect of these parameters has not been further evaluated in the study.

The cycle evaluation indicates that the second largest deviation between the measured and the ideal loop results from the temperature difference between the medium prior to expansion and the ambient. The effect on efficiency of this temperature is much larger for R-744 compared to R-290. This can easily be seen in Figure 4, from which it is visible that expansion from 35°C instead of the drawn 30°C yields a very large reduction in cycle efficiency for R-744 while the reduction in cycle efficiency for R-290 is much smaller.

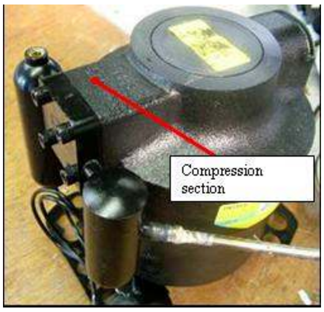

Figure 5. R-744 Compressor

4 COMPRESSOR PERFORMANCE

Compressor performance data of the R-290 compressor was commercially available. Calculations of the isentropic efficiency using eq. 1, 3 and 4 and the rated operating conditions resulted in isentropic efficiencies of 0.38 and 0.40 at evaporation temperatures of respectively –35 and –30°C, a condensation temperature of 45°C and a superheating temperature of 32°C. These isentropic efficiencies were used to determine the appliance heat load as presented in section 3. Further modelling of the measured conditions showed a good agreement (+/- 3%) with these estimated efficiencies. At the time of performing the evaluations no compressor data was available for the R-744 compressor at low back pressure conditions. Therefore the R-744 compressor was evaluated on the Re/genT compressor calorimeter. This is a “secondary fluid type” calorimeter which automatically measures the compressor performance according ISO 917 (method A). Based on performance measurements compressor performance regression lines were constructed. These regression lines were used to calculate the compressor performance at the pressures obtained during the measurement on the cabinet. In Figure 6 the measured compressor performance is compared with the compressor performance estimated from the cabinet tests. The accuracy of the results is represented by the error bars in the figures. The accuracy of the calorimetric testing was within 5% on volumetric efficiency and within 1% on input power, the accuracy of the performance estimates based on the practical testing with the cabinets was much lower due to the use of average values (e.g. evaporation temperature decreases over the compressor on cycle, and hence all other parameters also change over time).

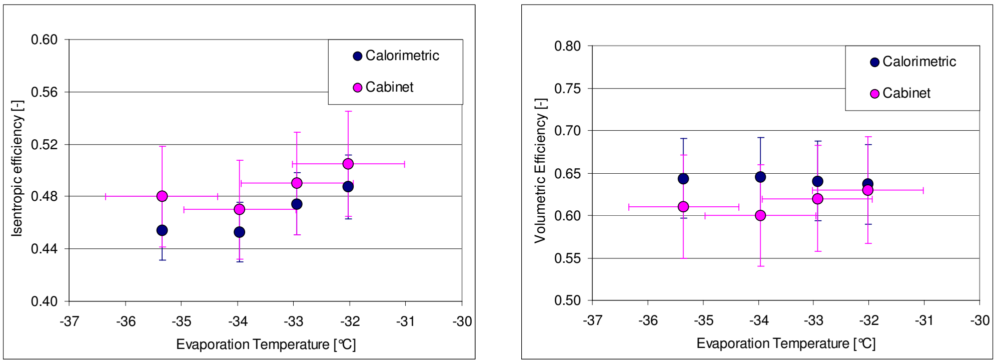

Figure 6. Compressor performance measured on calorimeter and estimated from measurement

Visible from Figure 6 is that the compressor performance resulting from the calorimetric compressor tests corresponds quite closely with the compressor performance estimated from the measurement.

5 POTENTIAL OF R-744 IN SMALL FREEZER APPLICATIONS

The main advantage of R-744 as refrigerant in comparison to R-290 is that it is non-flammable. However, for small commercial systems like ice cream cabinets, the use of R-290 below a quantity of 150 gram per refrigerant circuit is accepted by all applicable ISO and EN standards, only requiring limited design changes (e.g. spark proof electrical components), with negligible cost impact. When using R-290 in larger quantities, specific provisions may be required to ensure safe operation. Using R-744 as refrigerant in small capacity freezer applications has three major disadvantages in comparison to R-290, namely the much higher pressure levels, the lower energy efficiency and the higher equipment costs.

The operating pressures for R-744 are much higher than those obtained with R-290. These high operating pressure levels result in higher construction pressures. To cope with these high construction pressures the compressor and several other components (e.g. filters, pre- and after coolers and condenser and evaporator tubing) need to be designed much stronger than for R-290, which has its impact on cabinet production costs, see Table 6, where an estimate of the price increment is presented. The price difference between R-290 and R-744 will be largely driven by the difference in compressor costs, as reducing the tubing diameter and switching from copper to steel for R-744 can compensate for the higher specifications of the tubing. A disadvantage of this diameter reduction is that the heat transfer area between the refrigerant and the pipe decreases, so more piping length could be required. An estimate of possible future cost difference, assuming highly efficient high capacity production lines for both compressors was made by comparing the weight ratio of the R-744 compressor (15.5 kg) and the R-290 compressor (10.9 kg). Next to the compressor also the filter/dryer will be a more expensive component in a R-744 configuration but as the absolute costs of this component are much lower the effect on the total pricing will be less severe.

Table 6; Estimated relative cost increment

Component costs	R290	R744
Compressor	1	1.3 to 1.5
Pre-cooler	1	1.0 to 1.1
Condenser Tubing	1	1.0 to1.2
Filter/dryer	1	2 to 3
Capillary tube	1	1.5 to 2
Evaporator tubing	1	1.0 to1.2
Cabinet	1	1

6 DISCUSSION

The R-744 system as tested represents an almost optimal configuration using a state of the art R-744 compressor combined with low cost components. Namely the approach temperature, discharge pressure and evaporation temperature were close to the theoretical ideal loop. A drawback of the system tested is that the evaporator and gas cooler tubing are designed for R-290, which resulted in a slight increase of the thermal resistance between the bottom packages in the cabinet and the evaporator tubing in comparison to the original R-290 system. By comparing the test results based on the average product temperature instead of the warmest package temperature, this effect has been taken into account. If the ice cream cabinet would have been specially built for R-744 the evaporator and condenser tubing would be different. A system specially designed for R-744 would operate at the optimum discharge pressure (discharge pressure corresponding to maximum efficiency) and an evaporation temperature approximately equal to that obtained with the R-290 system. At such condition the C.O.P. of the R-744 system would be 0.84 at climate class 4 which is 10% lower than the evaluated R-290 system. It should be noted that the R-744 system was fitted with an after cooler to decrease the approach temperature as the approach temperature has a large effect on the system efficiency. If similar measures would have been implemented on the R-290 configuration the subcooling temperature would approach the ambient (or the condensation temperature would reduce) and the system C.O.P would increase from 0.95 to 1.0.

Climate Class 4 (30°C/55%) is used as the main rating condition. A calculation of the ideal cycle efficiencies at an ambient of 20°C (28ºC condensation temperature, -30ºC evaporation temperature, heat exchanger efficiency and isentropic efficiency of 1, resulted in a C.O.P of 3.51 and 2.50 for respectively the R290 and the R744 cycle. This is quite similar to the 31% efficiency difference obtained at climate class 4, so it is not expected that energy consumption comparison based on average operational conditions will result in significantly different conclusions.

7 CONCLUSION

From the study it is concluded that a R-744 system in principle will be more expensive and less efficient than a R-290 configuration when implemented in a low capacity ice cream cabinet. Based on the estimated production price it ought to compete with high efficiency R-290 compressors hence increasing the efficiency gap. The only disadvantage of R-290 compared to R-744 is its flammability. In small commercial applications (as in an ice cream freezer), below a quantity of 150 grams of R-290 per refrigerant circuit, the use of spark proof components is sufficient, whilst in larger commercial systems additional safety precautions may be required.

All tests have been performed with two single compressors. Using different compressors of the same type (since the R744 compressor is still a hand made compressor, variation up to 5% at the rating condition is common between different samples), could result in slightly different efficiency figures but it is expected that the main conclusions will not be affected.

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REFERENCES

Arthur D. Little, Inc, 1999, Opportunities for Energy Savings in the Residential and Commercial Sectors with High-Efficiency Electric Motors. *Report 35495-14, Prepared for U.S. Department of Energy*

Gungor K, Winterton R, 1987, Simplified general correlation for saturated flow boiling and comparisons of correlations with data, *Chem Eng Res Des, Vol. 65, March 1987*

McLinden M, Lemmon E, Klein S, Peskin A, 1998, Thermodynamic and Transport Properties of refrigerants and Refrigerant Mixtures, *NIST Standard National Database 23-Version 6.01*