INTEGRATED CO_2 REFRIGERATION AND AC UNIT FOR HOT CLIMATES

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ABSTRACT

An integrated refrigeration and AC system with CO_2 as the only working fluid for use in very hot climates (+45 °C) has been designed, modelled and simulated in Dymola. Based on the simulation results, its feasibility is proven. The system can achieve 3 kW of cooling capacity at the low evaporation temperature level of -28.6 °C, 10 kW at the medium temperature level (-3.2 °C) and 20 kW for AC at an evaporation temperature of +12.6 °C. The use of ejectors is essential for the system performance at these extreme ambient temperatures, and enables an, until now, impossible application for the natural, environmentally friendly refrigerant CO_2. The CO_2 system achieves a similar performance compared to existing solutions applying synthetic refrigerants with high GWP and ODP, which are typically applied nowadays in high ambient temperature regions.

1. INTRODUCTION

The average annual specific energy consumption of commercial buildings in India is 210 kWh m^{-2} including office buildings, warehouses and retail [1]. The value for supermarkets alone is up to 3 times higher [2]. This is primarily due to the refrigeration system, which usually covers 30 to 70 % of the electricity bill of a supermarket, while heating, ventilation and air conditioning (HVAC) systems cover between 15 and 25 %. Apart from the high energy demand of these installations, necessary to maintain food safety and comfort inside buildings, they additionally cause large *direct* greenhouse gas (GHG) emissions due to leakage of the currently applied synthetic refrigerants: hydro-chlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs). On a global basis, about 30 % of the refrigerant leaks out to the surroundings [3]. Common refrigerants like R134a and R410A have GWPs of respectively 1430 [4, 5], and 2088 [5] CO_2-equivalents that is, 1430 and 2088 times as high as for CO_2, and thus contributes strongly to global warming upon leakage. There is hence a huge demand for improvements, regarding both the energy efficiency and environmental impact of refrigeration and HVAC systems.

The number of individual retailers is around 12 million in India, which is the highest in the world (there are slightly more than 1 million retail stores in the entire Europe [6]). The Indian retail sector contributes with around 10-11 % of the gross domestic product, amounting to around US $ 180 billion. Indian supermarket and cold chain industry currently choose non-natural refrigerant alternatives (HCFCs and HFCs) due to lack of knowledge and experience with refrigeration systems based on natural working fluids and their performance in tropical conditions. India is further committed to eliminate the usage of ozone depleting HCFC and HFCs due to their high global warming potential (GWP) [7].

To replace HCFCs and HFCs, two options are available: natural working fluids or a fourth generation of short living synthetic working fluids known as hydrofluoroolefins (unsaturated HFCs, also incorrectly named HFOs). One new refrigerant being recommended by [8, 9] to replace the harmful HFCs is the synthetic HFC-

32, which is stated to be 15% better than existing R410A solutions and much more environmentally friendly with its GWP of 675 [4, 5, 8, 9]. However, this GWP value is still very high. Ways to phase out refrigerants with high GWPs are now considered and regulations banning refrigerants with GWP higher than 150 CO_2-eqv. are suggested [10]. This maximum limit is believed to be reduced even further in the future. In addition, CO_2 taxes, costs of emitting substances with a positive GWP, and proportional to its GWP, has been or will be introduced started in several European countries [10], for example, this was implemented from 1st of January 2016 in Spain [11]. This is a part of the phase out strategy, giving companies economic incentives to do so. Thus, installing a new system should aim for refrigerants that will remain legal also in the future. This will prevent the owner from having to buy a new solution within few years and paying high CO_2 taxes for refrigerant leakage. The proposed 'new' HFCs have very short atmospheric lifetimes, of a few days, leading by definition to a low GWP; however they are still synthetic working fluids – there exist no natural cycles to absorb the emissions, the decomposition products are highly toxic for water organisms even at low concentrations, i.e. their impacts on the environment cannot be completely predicted. Natural refrigerants have the advantage that they are part of the surroundings and nature already, and unlike synthetic ones, they have no unknown detrimental effects on the environment. An advisable strategy for the development of refrigeration technology would therefore be to apply only natural working fluids

Another suggested alternative to replace current working fluids is the natural refrigerant propane, or HC-290 [8, 9], which has a GWP of 3 CO_2-eqv. [4], contains no harmful fluorine and has natural breakdown mechanisms in the nature. The implementation of large capacity direct expansion systems with R-290 is unlikely, due to safety issues. On the other hand, CO_2 (R-744) has been used increasingly in the recent years as an alternative working fluid to replace HCFCs and HFCs in supermarkets in Europe and North America. CO_2 is thermodynamically efficient for lower ambient temperatures, non-toxic, non-flammable, inexpensive and easily accessible. It is further environmentally benign, with zero ozone depletion potential (ODP) and a GWP equal to 1 CO_2-eqv. If it is taken form a source which would otherwise emit it to the atmosphere, one could argue for that its GWP upon leakage from a refrigerant system is 0 CO_2-eqv., as it adds no effect on the climate compared to the alternative treatment. However, at high ambient temperatures, well-designed and maintained HCFC/HFC systems require slightly less energy compared to the current CO_2 systems. In order to phase out HCFC/HFC systems with negative environmental impact also in hot climates, a new generation of advanced CO_2 systems that could offer comparable energy performance is required. An adaptation and a further development of the currently available CO_2 systems are required to increase the energy efficiency and cost-effectiveness for applications in the Indian market.

An ejector improves the performance of a CO_2 system by up to about 40 %, depending on the ambient temperature [3]. Several system plants are operated in Europe, and also, some system plants are operated in Africa. One installation in Spain obtained 10 % higher energy performance than a standard synthetic system, and 35 % better than the previous solution [11]. Changing refrigerant from synthetic to CO_2, 20-27 % improvement was obtained in small convenient shops in Japan and Indonesia [12, 13], despite that the latter lies close to equator. The performance of a refrigeration system in climates as warm as that in India, considering temperatures of 45 °C, is however rarely considered, despite the large market for cooling equipment in Asia.

The subject of the research was a feasibility study of integrated supermarket installation adapted for climatic conditions in India (ambient temperatures up to 45 °C). An energy performance analysis was carried out for example profiles of Low-Temperature (LT), Medium-Temperature (MT), and Air Conditioning (AC) thermal load, for two alternative systems: a standard, separate, HFC-based installation vs. a novel, integrated, CO_2-based vapour compression pack.

2. METHOD

A refrigeration system with three temperature levels, low (LT), medium (MT) and air-conditioning (AC) were modelled in the modelling environment Modelica (3.2.1) using the Dymola (Dynamic Modelling Library, version 2015, Dassault systems) and TIL libraries (TIL 3.4, TLK-Thermo GmbH, Braunschweig, Germany). Details on the temperature levels in the systems are shown in Table 1. The performance of two working fluids were compared at 45 °C ambient. In one system, CO_2 was used as the only working fluid, to investigate its viability in such high temperature conditions. In the other, R410A was the only working fluid used, because this is commonly applied in refrigerant applications both in India [8, 9] and worldwide, yet having a GWP of 2088 [8] and, being an HFC. To make the comparison as fair as possible, the same evaporation temperatures were used for both fluids, and these and the corresponding pressures are shown in Table 1. All components were also made as similar to those in the CO_2 system as possible, using the same compressor efficiency (0.7) and obtaining the same degree of superheat (7 K) and cooling on the high pressure side (48 °C, 3 K above ambient temperature).

Table 1: Temperature and pressure levels for cooling in the modelled system

Temperature level	Abbreviation	Load	Evaporation temperature [°C]	Evaporation pressure [bar]	
				R744	R410A
Low temperature	LT	3.00 kW	-28.6	15.0	2.85
Medium temperature	MT	10.8 kW	-3.2	32.0	7.20
Air-conditioning	AC	20.0 kW	+12.6	48.0	11.7

A system sketch for the planned CO_2 system is shown in Figure 1. This is an integrated system with five pressure levels in the system. Besides the three mentioned in Table 1, the high pressure was 130 bar and that before the AUX compressor was 56 bar. The separator tank was 80 L, the smaller one did not serve any purpose in the modelling environment and was thus neglected. The system has two ejectors and internal heat exchange. However, running the system revealed that the system could not achieve the desired pressure levels with the MT ejector. This ejector increased the AC pressure by sending its mass flow to this pressure level, and the flow through the AC ejector had to be so large to remove this incoming mass that it required a very high mass flow from the high-pressure side. Because of this, it withdrew too much mass from the high pressure side and the high pressure level sank, reducing its driving force and requiring even higher mass stream to maintain the AC pressure. Thus, in the final system studied, there was only one ejector, the AC ejector, and because it was discovered that the internal heat exchangers and the gas cooler after the LT compressor actually degraded the results, these were also removed.

For the solution using R410A, three separate installations using internal heat exchange were assumed. This might appear to make comparison to CO_2 unfair, as there were no internal heat exchangers in this system, but the aim was to compare a possible, future CO_2 system to normally applied cycles today, and these would probably use internal heat exchangers to improve performance, whereas a new CO_2 system would not include them if these components reduces the performance. Thus, the comparison was more relevant and realistic when assuming that all systems are made to be as good as they can be with a given refrigerant at a non-excessive cost. This involves both internal heat exchange and compression stages. For high pressure ratios, more stages of compression should be applied. 2-4 are normally the optimum ratios, and designing for a high pressure of 29.32 bar (saturated at about 48°C), the low temperature (LT) system was modelled with two compression stages, whereas the MT and AC systems only had one compression stage.

Table 2: Evaporation temperature (T_{evap}) and design pressures in the condenser (p_{high}), evaporator (p_{low}) and eventually between the two compressors (p_{middle}) for the R410A installations at the low temperature (LT), medium temperature (MT) and air-conditioning (AC) temperature levels

	LT	MT	AC
T_{evap} [°C]	-28.6	-3.2	12.6
p_{low}	2.85	7.20	11.7
p_{middle}	9.49	-	-
p_{high}	29.32	29.32	29.32

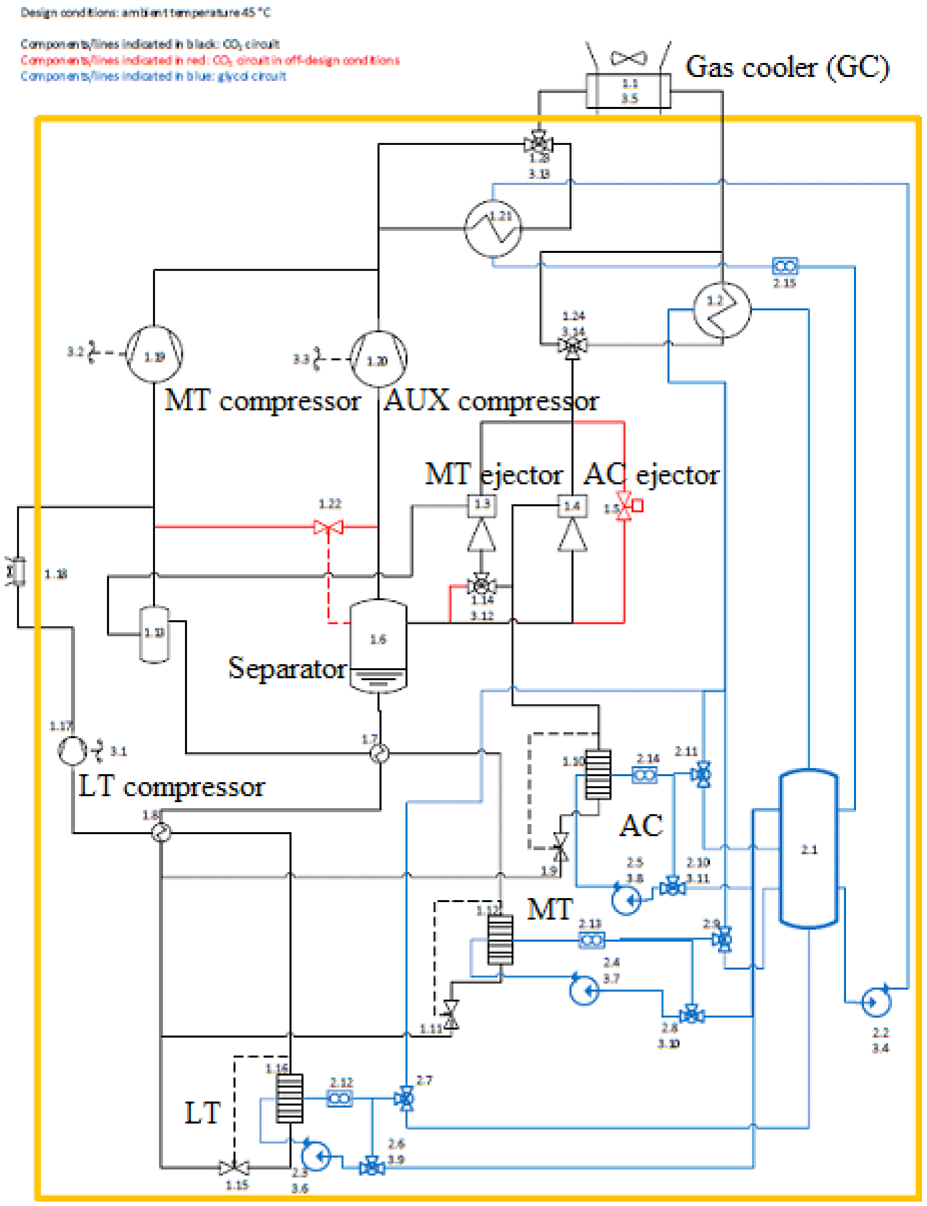

Figure 1. The modelled system with three evaporation levels: low (LT), medium (MT) and air conditioning (AC) temperatures. Two ejectors, three compressors and a gas cooler, and a glycol tank for distribution of cold (in blue) are shown.

2.2. Ejector model

The ejectors were modelled using the flow correlation of Lucas, which fitted experimental data with plus/minus 10 % [14]. This function was built in in the TIL library. To the best of our knowledge, no simple and precise model exists, and the realization of a modelled ejector must still be based on making a design that performs as the desired, thus the obtained values for optimal area will not be exact. Ejectors would really be made with a constant flow area, but as it was not known in advance which size this should be, it was decided to allow this area to be varied, and adjust it to achieve an appropriate value. In contrast to optimization, this meant that a too low area would make the driving pressure (the high side pressure) and the suction pressure (in this study: the AC pressure) too high, as the flow through it is constrained. This would make the AC load too low and the compressor work per mass higher. A too high value would make the high pressure (and potentially the AC pressure) too low and send an excessive mass flow to the pressure level of 56 bar. This means that an excessive mass flow would have to be compressed by the AC compressor and the energy consumption would increase, as much mass would be looping without serving any purpose. The AC load would most likely not be as designed and the gas cooler would not be able to remove all the extra energy added by the excessive mass flow in the compressor. Thus, the ejector area is not simply an optimal value, but an important value for the system to operate as desired.

2.3. System control

PI-regulators were used to steer the systems. In the CO_2 system, three PI-regulators controlled the three valves to obtain seven degrees of superheat of from the three evaporators. The frequencies of the all the compressors were controlled by each their PI-regulator to keep the pressure before them at the desired value. The intermediate pressure in the LT system using R410A was thus controlled by the second compressor to be 9.49 bar, based on the recommendation of Formel in [15]. All evaporation pressures were controlled in this way except for the AC pressure level in the CO_2 system. Originally, this was controlled by the opening in the AC ejector, whereas the sum of ejector areas was adjusted to maintain the high side pressure at 130 bar. However, after removal of the MT ejector, the AC ejector alone was controlled so that the high pressure should be 130 bar, as a high pressure is essential for good performance in such systems. The result of this was that the AC pressure in the CO_2 system, and the condensers in the R410A systems, the pressure was to a large extend controlled by the system design: the sizes of the heat exchangers, mass flow and incoming enthalpy. Therefore, proper system design to achieve desired heat transfer and pressure was a careful, iterative process, in which parameters were adjusted until satisfactory operation conditions were achieved.

2.4 Glycol

The system cooled down three glycol circuits entering with the characteristics in Table 3, and rejected heat to a glycol circuit (entering at 60 °C) and then to ambient air at 45 °C. These glycol boundaries should transfer heat a supermarket, and thus the system boundaries for simulation were set at the glycol circuit. Each of them were modelled as a glycol source and sink, connected by a flow through the heat exchangers. The fluid was modelled as TILMedia_Glysantin_50, which represents a 50 % inhibitor - monoethylene glycol mixture.

Table 3: Conditions of the incoming glycol in the heat exchangers at the low temperature (LT), medium temperature (MT) and air-conditioning (AC) temperature levels

Evaporator	Mass flow [kgs^{-1}]	Inlet temperature [°C]
LT	0.09	-10
MT	0.6	+5
AC	0.6	+25

The numerical simulation spent the first simulation time to adjust the control mechanisms to one another, and thus required some time for stabilization. The results were obtained after that all values had come to rest at a stable value. Simulations were run until a steady state was achieved. Oscillations and variations were always present in the start of all simulations before they reached their stable operation point close to the designed conditions.

3. RESULTS AND DISCUSSION

All systems operated satisfactory at the desired conditions, thus, it should be possible to use CO_2 ejector systems for refrigeration purposes in climates with 45 °C ambient temperature. All pressures were very close to their desired values, deviating by no more than 0.04 bar or at most 0.14 % of the set point value. The obtained values for pressures in the CO_2 simulation are given in Table 2. The AC-pressure was the only pressure that was not directly controlled, but determined by the system design. However, the value was very close to the desired value.

Table 4. Pressures obtained in the CO_2 simulation

Pressure level	Obtained value [bar]
LT	15.00
MT	32.00
AC	48.01
High	56.00
GC	130.00

Table 5: Compressor power required for the three compressors in the CO_2 system, defined in Figure 1

Compressor	Power demand [kW]
LT	0.781
MT	7.432
AUX	10.803
All	19.017

The ejector area became 1.30e-5 m^2, and the mass flow in the gas cooler was only 0.34 kg s^{-1}. The required power for all three compressors are shown in Table 3, and the resulting refrigeration loads in Table 4. The results for the R410A systems are shown in Table 5, and for these systems too, the conditions were very closely achieved, with deviations from design values less than 1 %. From these tables, it is seen that total power required was about 19.0 kW for the CO_2 system, which is 53 % higher than for the R410A systems, which required only 12.4 kW. As there are three different evaporation levels involved, it makes no sense to speak of COP, in the normal way, but still, the ratio of achieved cooling to required compressor power is interesting for comparison of these two systems and shown in Table 8, but this value is not comparable to any other results than those using the same evaporation temperatures, ambient temperature and cooling loads as in this study.

Table 6: Achieved cooling loads at the low temperature (LT), medium temperature (MT) and air-conditioning (AC) temperature levels for the CO_2 system

Pressure level	Cooling supplied [kW]
LT	3.005
MT	10.793
AC	19.966

Table 7: Achieved conditions and performance of the three R410A systems supplying cooling at low temperature (LT), medium temperature (MT) and air-condtioning (AC) temperature levels: Work required for compression in the first, and eventually second, compressor, cooling supplied (Q_{evap}), Coefficient oif performance, temperature after condenser, (T_{high}) and the pressure levels in the condenser (p_{high}), evaporator (p_{low}) and eventually between the two compressors (p_{middle})

Variable/pressure level	LT	MT	MT	AC
Work1 [kW]	1.322	4.207	4.489	5.339
Work2 [kW]	1.278	-	-	-
Q_{evap} [kW]	3.007	10.09	10.80	19.98
COP	1.16	2.40	2.40	3.74
T_{high}	48.00	48.00	47.98	48.06
p_{high}	29.31	29.32	29.31	29.36
p_{low}	2.85	7.20	7.20	11.7
p_{middle}	9.49	-	-	-

Table 8: Overall performance results for the CO_2 and R410A refrigeration solutions: total work input required and ratio of total cooling supplied divided by total work input.

	CO_2	R410A
Work [kW]	19.017	12.428
Ratio of total cooling to total work input	1.76	2.72

The effect of the low pressure drop of CO_2 is a great advantage in many systems, but as no distances for supplying the cold were assumed, this effect was not included, thus, the CO_2 system would probably have been somewhat better than the result here show, however, it would still be about 50 % worse than R410A. Another large opportunity for improvement which was not exploited here is that when using ejectors, flooded evaporators is a possibility, which would enhance the performance quite drastically. Yet, even with a 30 % improvement in energy performance, the CO_2 system would still be slightly worse (7%) than the R410A system.

Even though CO_2 performed worse than R410A, the coming legislation on GWP for refrigerants might make installations with GWPs higher than 150 very costly as they must be replaced after a short time. Vendors cannot sell systems for such refrigerants in Europe, and might stop producing these units. CO_2 taxes are also believed to make the choice of natural refrigerants a more profitable solution, in addition to sparing the environment. Whether CO_2 is a better option in climates with 45 °C is not clear however, as higher power consumption means higher indirect CO_2 emissions. As it is better in colder climates, one must consider which temperature range is the most normal in the area the units are placed. If the ambient temperature lies in ranges where CO_2 outperforms other refrigerants large parts of the year, but is 50 % worse during two months a year, then this might still be a profitable investment. Otherwise, one should look to the use of other natural refrigerants such as ammonia, ammonia-water, cascades, propane or other natural refrigerants, as these will be the sustainable solutions in the long-term.

4. CONCLUSION

A refrigeration system using CO_2 only can be applied with reasonably good performance in climates as hot as 45 °C. The highest pressure was 130 bar, and the use of a PI-controller to determine a suitable ejector area was successful. The system would not be applicable without ejectors. The optimal ejector configuration for the studied steady state case was only one ejector with an area of 1.30e-5 m^2. It was not as energy efficient as the existing R410A systems might be, but one should move away from synthetic refrigerants, harmful to the environment, rather than fining new synthetic fluids that might have even other unknown effects. Long transfer lines for the cold, possibility of flooded evaporators, and an evaporative condenser, future CO_2 taxes, new legislation and market conditions might make CO_2 more profitable (total cost of ownership) than found in this study. Still, as it is far from performing better at 45 °C ambient temperature, other natural refrigerants and the performance over the whole year should be considered before choosing a solution based on HFCs.

NOMENCLATURE

p	pressure	(bar)	Subscripts	
Q	cooling load	(kW)	evap	in evaporator
T	temperature	(°C)	high	in condenser
			low	lowest
			middle	intermediate

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