COMPARISON OF THE R744 CASCADE WITH THE R404A AND R22 CONVENTIONAL SYSTEM FOR SUPERMARKETS

Alessandro da Silva, alessandro.silva@bitzer.com.br
Bitzer Compressores Ltda , Av. Joao Paulo Ablas 777 – Sao Paulo – 06711-250 – Brazil.

Enio Pedone Bandarra Filho, bandarra@mecanica.ufu.br
Arthur Heleno Pontes Antunes, arthur.h.p.antunes@gmail.com
Faculty of Mechanical Engineering – Federal University of Uberlandia, Av. Joao Naves de Avila, 2121 – Uberlandia-MG-Brazil

Abstract. The present article focuses on the energy efficiency of three different systems used in supermarket applications. The refrigeration systems consist of a cascade system (CO_2/HFC404A) with carbon dioxide for subcritical operation and HFC-404A in the high stage temperature (pump circuit for normal refrigeration and direct expansion for deep-freezing), and also HFC-404A and HCFC-22 with direct expansion systems. The cascade system presents a lower refrigerant charge, 47 kg of both fluids, which represents less than a half of the refrigerant charge of the other systems. An important factor is the total GWP in the case of leakage, where the impact in the atmosphere of the cascade system operating with CO_2 was much less than the direct expansion systems.

Keywords: *Cascade System, CO2, Carbon Dioxide, Supermarket, R22*

1. INTRODUCTION

CO_2 or R744 or Carbon dioxide is a climate-friendly refrigerant because it does not contribute to the depletion of the ozone layer and it has a low direct global warming potential with the reference value of 1. Due to its specific thermodynamic properties, including high operating pressure, low critical temperature and low viscosity, CO_2 offers great scope as a new energy-efficient product. Furthermore, it will encourage development of modern systems that will put the refrigeration industry on a more sustainable footing.

With the focus primarily on supermarket applications, this paper will analyze energy efficiency comparisons carried out between the CO_2 cascade system and the direct expansion conventional system using R404A and R22, and discusses their advantages and disadvantages, along with a comparison of saving costs with carbon dioxide. Relevant issues for application of CO_2 will also be raised in this paper. These energy efficiency comparisons were conducted in the CO_2 technology center that has been operating in Bitzer Brazil since 2008. In this center were installed three refrigerating systems with similar cooling capacities. They have been running, week on week off, so an accurate comparison can be drawn among them.

2. EXPERIMENTAL SYSTEMS

The experimental systems consist of a cascade system with carbon dioxide for subcritical operation in the high stage with HFC-404A (pump circuit for normal refrigeration and direct expansion for deep-freezing), and also HFC-404A and HCFC-22 direct expansion systems. Figure 1 shows the three refrigeration racks. They cool two storage rooms down to 0 to 2°C and a deep-freeze room, -25°C.

There are also two deep-freeze islands working at -25°C that are only connected to the carbon dioxide circuit. The cooling capacity for normal refrigeration is about 20 kW, and about 10 kW in the deep-freeze range. The evaporators of the three refrigerating systems are designed as air coolers and fitted under the ceiling of each cold room. The condensers operate with either air-cooling or water-cooling. All machines and cold rooms are equipped with infrared sensors and a carbon dioxide extraction system. Only one system is in use at any one time to permit energy comparisons.

Figure 1. Refrigeration racks used in the present research.

Table 1 presents the major technical data for each refrigeration rack of both the Medium Temperature (MT) and Low Temperature (LT) systems:

Table 1. Technical data of multicompressor refrigeration systems.

	Subcritical CO_2 Rack (CO_2/R404A)	**R404A Rack**	**R22 Rack**
MT design condition	T_{Cond} = -5ºC (CO_2) T_{Evap} = -10ºC (high stage) T_{Cond} = 40ºC (high stage)	T_{Evap} = -10ºC T_{Cond} = 40ºC	T_{Evap} = -10ºC T_{Cond} = 40ºC
LT design condition	T_{Evap} = -30ºC (CO_2 - DX) T_{Cond} = -5ºC (CO_2)	T_{Evap} = -30ºC T_{Cond} = 40ºC	T_{Evap} = -30ºC T_{Cond} = 40ºC
Compressor models	01 x 2KC-3.2K (CO_2) 01 x 4CC-9.2.Y (R404A)	01 x 4CC-9.2Y (MT) 01 x 4TCS-8.2Y (LT)	01 x 4CC-9.2 (MT) 01 x 4TCS-8.2 (LT)
MT cooling capacity	**21.0 kW**	**21.0 kW**	**19.8 kW**
LT cooling capacity	**9.8 kW**	**10.7 kW**	**9.9 kW**

The cooling capacity of the MT and LT multicompressor refrigeration systems is higher than the required thermal load from cold rooms and islands, as shown in the Table 2.

Table 2. Overview of refrigeration points.

	MT cold room	**MT walk-in cooler**	**LT deep-freeze room**	***LT Islands**
Dimensions	3.5m x 4m x 3.5m	3.5m x 4m x 3.5m	3.5m x 4m x 3.5m	5m total length
Thermal load	**7.5 kW**	**7.5 kW**	**7.5 kW**	**2.5 kW**
Internal temperature	0ºC	+2ºC	-25ºC	-25ºC

* The two LT Islands only run with the CO_2 refrigeration rack.

3. RESULTS

3.1 Compressors:

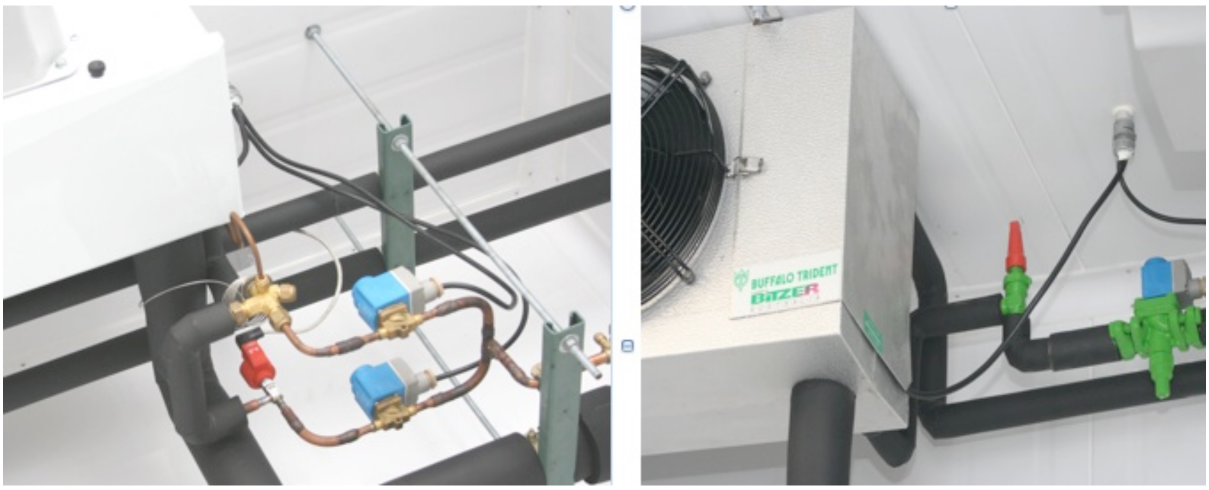
The compressors of each rack also have the operating option of both a frequency inverter and a head capacity control unit, except for compressor model 4TCS-8.2 which is used in connection with the R22 in low temperature with controlled injection cooling (CIC), and also for the CO_2 compressor model 2KC-3.2K which has only one head (2-cylinder). As a result it is not possible for both compressors to operate with a head capacity control unit. The application range is 30Hz to 70Hz for compressors with frequency inverters.

3.2 Condensers:

Each rack has the operating option of both air-cooled and water-cooled condensers (mainly the high stage of the subcritical CO_2 rack). Air-cooled and water-cooled condensers can be used to compare the energy efficiency of the system. The air-cooled condenser fans also have the option of operating with frequency inverters as well as On/Off control pressure switches to control the condensing temperature. The water-cooled condenser types are shell-and-tube and they operate with a water-cooling tower.

3.3 Evaporators:

The air-coolers that use R404A and R22, which are fitted in the cold rooms, are direct expansion (DX) type and use thermostatic expansion valves (TEV) and electronic expansion valves (EEV). The CO_2 air-coolers are used for both MT cold rooms and LT deep-freeze room. The air-cooler used in the LT deep-freeze room is DX and uses only an electronic expansion valve. The other two CO_2 air-coolers, for medium temperature, run with liquid recirculation and only use manual expansion valves to control the refrigerant flow.



(a) **(b)**

Figure 2. **(a)** Thermostatic and electronic expansion valves used in air-coolers with R404A and R22; **(b)** CO_2 manual expansion valve used in evaporators with liquid recirculation.

All relevant information comes together in a central monitoring unit that can also be controlled via LAN and the Internet. Medium temperature (MT) evaporator defrosting is achieved by off cycle (air) while low temperature (LT) evaporator defrosting is achieved with the electric heater, mainly for the LT CO_2 evaporators (Islands and LT deep-freeze room).

4. SYSTEM DESIGN

4.1 R404A and R22 Refrigeration Racks

Both refrigeration racks work with two semi hermetic piston compressors (Octagon model 4CC-9.2 for MT and 4TCS-8.2 for LT) in parallel applications. Each rack has a common discharge, but the suction header is split for MT and LT suction lines, while the discharge is collected in a common header and directed into a single oil separator. The oil return pipe enters into an oil receiver, which pushes the oil to the oil regulators which are fitted on the compressor sumps. The discharge line goes to the condenser and then goes to a vertical liquid receiver, where a header liquid line distributes the liquid to the evaporators. The operating conditions are –30ºC for low temperature (LT), -10ºC for medium temperature (MT) and 40ºC for condensation temperature.

4.2 CO_2/R404A Subcritical Rack

Figure 3 shows a schematic diagram of the cascade CO_2/R404A system. According to this figure, the MT CO_2 evaporators run with liquid recirculation at –5ºC, while the LT CO_2 evaporators run with direct expansion at –30ºC of evaporating temperature, through the vapour compressor cycle using a semi hermetic reciprocating compressor.

In the R404A/CO_2 cascade system, the CO_2 and the R404A are in two separate circuits. These two circuits come into thermal contact in the interstage heat exchanger where they exchange heat with each other without mixing the two refrigerants. The interstage heat exchanger serves as a condenser for the CO_2 system and as an evaporator for the R404A system. CO_2 is used as pumped liquid for normal refrigeration and direct expansion for deep-freezing.

The design of the CO_2 rack has some unusual features, which are required to maintain compressor temperatures at the recommended level. It was found that the performance of the CO_2 compressor suffered in very low operating temperatures, which if left unchecked, would result in a high concentration of refrigerant in the oil within the compressor sump, causing premature compressor failure. Superheating degrees of 20K to 30K at the CO_2 compressor suction were required to maintain acceptable sump temperatures in the CO_2 rack.

To prevent this, an additional heat exchanger was added between the CO_2 suction line and the R404A high stage liquid line, which maintained the CO_2 suction gas temperature at the compressor at between –10ºC to 0ºC. Experience has shown that maintaining sufficient heat in the compressor is sometimes a problem. Because the return vapour to the compressors is dense, CO_2 has a much larger capacity to absorb heat out of the compressor castings than other gases. This can result in the compressor being chilled to a point where the compressor discharge line, and the compressor crankcase are covered in frost and ice and this will almost certainly mean that the oil is being diluted by refrigerant. Any refrigerant dilution will have an adverse effect on the life expectance of the compressors running gear. It is best to keep the compressor sump temperature at least at body temperature, and the discharge should always be hot.

Controlled return vapour superheating needs to be provided by some means such as liquid-suction heat exchangers, utilising the liquid from R404A high stage thus providing "free" subcooling of the high stage liquid. Stress is on controlled superheating of the return vapour, as uncontrolled superheating of the return vapour will cause the usual system problems. But some form of control must be installed to limit the compressor return vapour temperature, either a bypass system or multiple heat exchangers staged to provide accurate control of the vapour inlet temperature. Discharge vapour temperature can be used or suction return temperature to control the heat exchanger operation. Low return vapour superheating will give rise to oil and lubrication problems, while high superheat levels will cause motor overheating and subsequent failures, as well as high discharge temperatures.

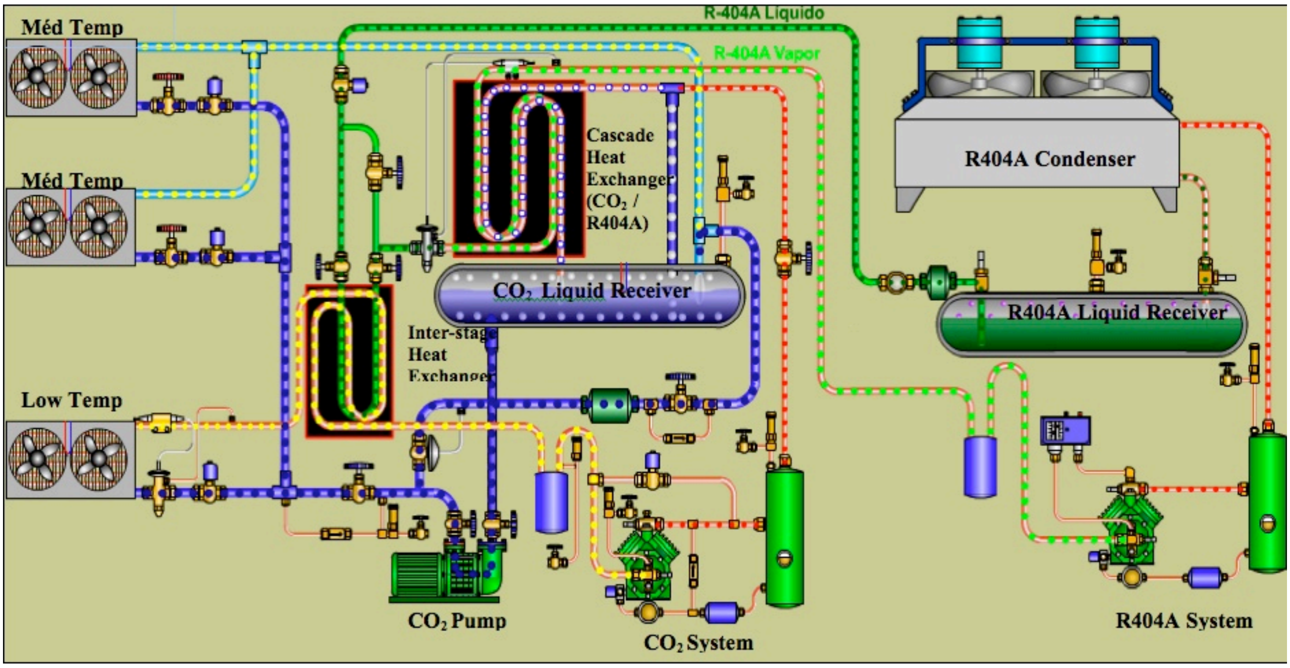


Figure 3. Schematic diagram of the cascade CO_2/R404A system.

5. PIPE WORK COMPARISON

The cascade system design can also take advantage of a high degree of liquid subcooling, which results in substantial reductions in pipe line diameters and a reduced refrigeration charge, compared to conventional refrigerants such as R404A and R22. Table 3 shows this comparison.

Table 3. Suction and liquid line size comparison used in the two cold rooms for MT using CO_2, R404A and R22.

Refrigerant		**CO_2**	**R404A**	**R22**
Suction line	Cooling capacity (kW)	10	10	10
(wet return line for CO_2, dry return line	ΔT (K)	0.67	0.47	0.55
for R404A and R22)	Velocity (m/s)	6.64	10.36	9.02
	Diameter (mm)	12.7	28.58	28.58
Liquid line	Velocity (m/s)	1.36	0.84	0.57
(-5ºC for CO_2)	Diameter (mm)	9.52	15.88	12.7

Cold room nº 1 and nº 2 for MT (T_{Evap}= -10ºC; T_{Cond} = 40ºC); L_{eqv} = 20 (m);

Table 4: Suction and liquid line size comparison used in the two deep-freeze room for LT using CO_2, R404A and R22.

Refrigerant		**CO_2**	**R404A**	**R22**
Suction line	Cooling capacity (kW)	10	10	10
(dry return line for CO_2, R404A and	ΔT (K)	0.35	0.53	0.39
R22)	Velocity (m/s)	8.35	11.42	10.28
	Diameter (mm)	15.88	34.93	28.58
Liquid line	Velocity (m/s)	0.85	0.97	0.68
(-5ºC for CO_2)	Diameter (mm)	9.52	15.88	12.7

Deep-freeze room LT (T_{Evap}= -30ºC; T_{Cond}= 40ºC); L_{eqv} = 20 (m);

In the Table 5 is found the pipe work comparison in kg/m, only about the suction and liquid lines used each evaporator:

Table 5. Total pipe work used in two cold rooms for MT as well as in the deep-freeze room for LT.

	Pipe work real length (m) – only suction line (SL) and liquid line (LL) each evaporator					
Diameter (mm)	9.52	12.7	15.88	28.58	34.93	
R22 Cold room 01		15 LL		15 SL		
R22 Cold room 02		11 LL		11 SL		
R22 Deep-freeze room		15 LL		15 SL		
R404A Cold room 01			15 LL	15 SL		
R404A Cold room 02			11 LL	11 SL		
R404A Deep-freeze room			15 LL		15 SL	
CO_2 Cold room 01	15 LL		15 SL			
CO_2 Cold room 02	11 LL		11 SL			
CO_2 Deep-freeze room	15 LL		15 SL			
ASTM B-280 – kg/m	0.186	0.294	0.424	0.971	1.314	**TOTAL (kg)**
R22		41 LL		41 SL		**51.86**
R404A			41 LL	26 SL	15 SL	**62.34**
CO_2	41 LL		41 SL			**25.01**

As a general guide, pipeline sizes can be reduced to approx 1/5 of the line sizes currently used with R404A and R22 for the same system capacity, mainly the suction pipe work lines.

Due to the purchase price of CO_2 being considerably less expensive than what is currently used commercially, such as R404A and R22, the total cost of the refrigerant charge can be significantly reduced. Table 6 shows the total refrigerant charge used each rack.

Table 6. Total refrigerant charge used in each rack.

Refrigeration rack	CO_2/R404A (subcritical rack)	R404A rack	R22 rack
Total refrigerant charge	CO_2 – **32 kg** R404A – **15 kg**	R404A – **125 kg**	R22 – **115 kg**

While the medium temperature system through the liquid re-circulation system does not generally provide significant reductions in energy costs, substantial savings can be achieved through a reduced refrigerant charge and a real reduction in the actual cost of the refrigerant.

6. EQUIPMENT COST

The cost of the three racks, the six air-cooled evaporators and the condensers, were all tracked so a comparison could be drawn. The rack system costs were calculated separately. This has been done because the contractor supplies the interconnecting pipe work, as well as the pipe insulation, between the various items.

The two racks that make up the cascade system using CO_2 on low temperature and R404A on high temperature stage were found to be 18.5% (based on 2008) more expensive than single stage racks using R22 and R404A based on the same cooling capacity. This higher cost was largely due to the additional safety equipment that the CO_2 system required under the Brazilian occupational health and safety codes, and the fact that a reasonable amount of the components were specially built and had to be air freighted from Australia. As CO_2 gains in popularity and more CO_2 equipment becomes available this additional cost will be reduced.

The main factors at work here are the large reduction in the size of the pipe work and insulation respectively. In addition, the CO_2 evaporators were physically smaller and less expensive due to the increased specific cooling capacity of the refrigerant. It was found that the both R404A and R22 evaporators need approximately 20% more surface area to achieve the same performance as the CO_2 evaporators (based on the same temperature difference between evaporating temp and room temperature).

The refrigerant in the three systems also has an influence on the total cost. According to the Table 6 the cascade system has 32 kg of CO_2 as well as an additional 15 kg of R404A, (32+15= 47kg). The other two racks using R404A and R22, they have 125 kg and 115 kg, respectively.

In Brazil, the refrigerant HFC-404A has an average cost (at the time of purchase) of \$22 (twenty two dollars) per kg, the HCFC-22 costs \$ 7 (seven dollars) per kg, while CO_2 has a cost of \$ 1.40 (one dollar and forty cents) per kg. The CO_2/R404A cascade system has an advantage over the R404A system of the \$ 2375.20 and for the R22 system of \$ 430.20.

The direct global warming potential (GWP) of the three systems, due to direct emissions in the event of a total loss of the entire refrigerant charge, is also of great importance. Since the CO_2 is used as the base unit for measuring GWP, this comparison is relatively simple. One kg of R404A has a GWP of 3260 units; one kg of R22 has a GWP of 1500 units, whereas one kg CO_2 is equal to 1 unit. Therefore, a CO_2/R404A cascade system has 48932 units, R404A system has 407500 units and the R22 system has 172500 units, according to each refrigerant charge in the system. As can be noted, the difference between the CO_2/R404A cascade system and R404A system is of the order of 358468 units and for R22 is 123568 units.

7. POWER CONSUMPTION

Each of the three refrigeration systems are fitted with watt nods, which are able to capture the total power consumption of the entire system. Power is recorded at 15min intervals for the plants in operation, and includes all aspects of the system. Compressor motors and sump heaters, fan motors defrost heaters, evaporator fans, and so on. The energy efficiency comparisons are an average over one year where the condensing temperature was maintained of the order of 38ºC. As the trial is ongoing, and a full year has not elapsed, some assumptions have been made.

Table 7. Total power usage data.

Power Consumption per year CO_2-system [kWh] (compressor with frequency inverter ; LT evaporator with EEV)	103,234
Power Consumption per year R404A-system [kWh] (compressor with frequency inverter ; evaporators with EEV's)	126,295
Power Consumption per year R22-system [kWh] (compressor with frequency inverter ; evaporators with EEV's)	117,435
Difference in porcentage [%] - CO_2 vs. R404A ; CO_2 vs. R22	22.33 (R404A); 13.75 (R22)

Power Consumption per year CO_2-system [kWh] (compressor with frequency inverter ; LT evaporator with EEV)	103,234
Power Consumption per year R404A-system [kWh] (compressor with frequency inverter ; evaporators with TEV's)	128,701
Power Consumption per year R22-system [kWh] (compressor with frequency inverter ; evaporators with TEV's)	119,212
Difference in porcentage [%] – CO_2 vs. R404A ; CO_2 vs. R22	24.67 (R404A); 15.47 (R22)

It is most likely that with a CO_2-system, a good proportion of the energy savings can be attributed to the subcooling of the high stage liquid, by the low stage suction gas.

According to the Table 7, CO_2 is most efficient 22.33% on R404A system, and 13.75% on R22 system in connection with frequency inverter in the compressor motors and electronic expansion valves (EEVs). However, when both the R404A and R22 systems use thermostatic expansion valves (TEVs), CO_2 becomes even more efficient, in which it represents 24.67% on R404A system and 15.47% on R22 system. Electronic expansion valves save more energy costs because it is more reliable and precise in its way to control the refrigerant mass flow through the evaporator, as it receives all the information regarding the temperature and pressure in the evaporator outlet in order to control the opening and closing of the valve according to the superheating. It uses a PID control algorithm that guarantees stabilization of the temperature, as well as controlling the evaporator superheating, and defrost routines in real time.

8. CONCLUSION

This comparison has proven that superior performance, and a more environmentally friendly process can be applied to reduce the effects of direct and indirect global warming, while achieving long term cost reduction for the plant operator.

Clearly, there are numerous advantages, which will ensure that carbon dioxide cascade systems have a place in future refrigeration systems. Many advantages of CO_2 systems with R404A and R22 can be listed, such as, (i) Reduction of the electric energy consumption (in this case it varies between 13 to 24%); (ii) Low compression ratio and increased useful life of the CO_2 compressor; (iii) High CO_2 density and high pressure in the low pressure stage; (iv) Reduction of CO_2 piping diameter sizes; (v) Reduction of CO_2 refrigerant charge; (vi) Low price of CO_2 purchase; (vii) High enthalpy and high degree of liquid subcooling and higher cooling capacity; (viii) Low GWP and less carbon taxes (CO_2); (ix) Small volumetric displacement and smaller sized CO_2 compressors; (x) Smaller refrigeration rack and compact installation and smaller compressor numbers; (xi) Smaller and efficient evaporator coils; (xii) Reduced installation and maintenance costs.

Given the rapidly changing cost of refrigerants and the expected reduction in the cost of CO_2 compatible components, plus the enormous variation in the cost of power around the world, it is not possible to provide the exact payback period that is required to offset the more expensive cascade system. But it is safe to mention that the larger the plant the more attractive CO_2 becomes.

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