Application of CO_2 (R744) Refrigerant in Industrial Cold Storage Refrigeration Plant

Michael Bellstedt, Ph.D.Eng, MIEAust, MAIRAH, MSAIRAC; Frank Elefsen Ph.D. Eng. MAIRAH Ass. Member IIR; Stefan S. Jensen, B. Sc. Eng. MIE Aust, MAIRAH

ABSTRACT

This paper describes four single stage CO_2 (R744) vapour compression refrigeration systems servicing two medium sized freezer stores located in a food processing facility situated in Brisbane, Australia. The process which lead to the selection of carbon dioxide as a refrigerant as well as the selection of condensing medium are detailed. In addition, the general plant design concept, plant control, automatic defrost method, practical issues relating to the installation and practical difficulties encountered during commissioning are described in detail. An assessment of the long term reliability of the R744 systems is made and future applications of R744 as a refrigerant are discussed.

Keywords:

CO_2 (R744) refrigerant, single stage vapour compression system, freezer stores, food processing facility, practical installation issues, automatic defrost, plant control.

1.0 GENERAL BACKGROUND – WHY CO_2?

Prior to 1930, the refrigerants most commonly in use were the natural refrigerants Ammonia, Carbon Dioxide and Sulphur Dioxide as well as hydrocarbons such as Ethane and Propane (natural = occurring in nature). Also in use at the time were the flammable chemical refrigerants Methyl Chloride and Ethyl Chloride.

Due to the level of technology available at the time, both with regard to mechanical design and control systems, the relative toxicity, flammability and high pressures associated with these refrigerants posed great problems to design engineers and operators, and accidents and injuries were common.

During the 1930s the so-called “safety refrigerants” were introduced, primarily Chlorofluorocarbons (CFC’s) such as R11, R12 and R13, and Hydrochlorofluorocarbons (HCFC’s) such as R22 and R502. These refrigerants displaced all the old refrigerants in virtually all applications, other than sectors of industrial refrigeration, where Ammonia and some Hydrocarbons remained in use due to their superior characteristics.

Then the disastrous effect of the CFCs and HCFC’s on the ozone layer was discovered during the 1970s, leading to the phase-out of all CFCs by the 1990s, followed by the phase-out of HCFC’s in the early 2000’s.

The chemical alternatives introduced by the chemical industry for the CFC and HCFC refrigerants are various Hydrofluorocarbons (HFCs) and their blends, such as R134a, R507, R404a, R407c and R410a. While the HFC refrigerants do not directly damage the ozone layer, they are synthetic greenhouse gases (SGGs) with high Global Warming Potential, higher than some of the CFC’s and HCFC’s they replace, and significantly higher than the original natural refrigerants, i.e. Ammonia (R717), Carbon Dioxide (R744), Sulphur Dioxide (R764) and hydrocarbons such as Propane (R290), refer to Table 1. Therefore, emission of these refrigerant to the atmosphere contributes greatly and disproportionately to the global warming phenomenon, which has been widely reported in the media.

Michael Bellstedt is Director of Minus40 Pty. Ltd. Sydney, Frank Elefsen is Managing Director of Danish Technological Institute Pty. Ltd. Brisbane and Stefan S. Jensen is Managing Director of **scantec** refrigeration technologies pty. ltd. Brisbane, Queensland, Australia.

Refrigerant	GWP (100 years, relative to CO2)	ODP (relative to R11)	Atmospheric life (years)
R11	3,800	1	50
R12	8,100	1	102
R13b1	5,400	10	65
R22	1,500	0.055	13.3
R23	12,000	0	260
R134a	1,300	0	13.6
R717 – Ammonia	0	0	-
R744 – CO_2	1	0	-
R290 - Propane	3	0	-

Table 1. GWP and ODP for various refrigerants [1] and [2]

In addition, the HFCs have a number of drawbacks that complicate their application and render the systems less reliable and more costly to maintain than comparable CFC or HCFC systems. These drawbacks include high cost of both refrigerant and special oils, the hygroscopic nature of these special oils, difficulties associated with application to flooded systems, reduced efficiency (COP) compared to CFC or HFCF systems, high discharge pressures and decreased chemical stability.

The justification for the return to the original natural refrigerants is therefore great. These refrigerants are cheap, do not damage the environment, and are technically excellent in their range of application. Modern design techniques and equipment have dramatically reduced the risks associated with the use of these fluids, and it only remains to demonstrate to the industry that these refrigerants can be safely and cost effectively applied in order to achieve an extensive reduction in the use of fluorocarbons.

The natural refrigerants that merit extensive re-evaluation and re-introduction are Ammonia (R717), Carbon Dioxide (R744), Hydrocarbons and Water. Ammonia remains in widespread use, and the application thereof to areas such as commercial refrigeration and air conditioning, particularly as alternative to the still widely used R22, is growing.

The application of CO_2 is unlikely to encounter many barriers, as modern equipment can effectively cope with the high pressures, and the refrigerant as such is much less hazardous than even a fluorocarbon as it remains safe and benign even in the event of a fire.

The major drawbacks of CO_2 are the low critical temperature (31.1 °C) and the high operating pressures. These properties render CO_2 less useful as a refrigerant in a single stage system, as this would require inefficient trans-critical operation. Therefore CO_2 is best used as the low stage refrigerant in a cascade system, such that the condensing temperature of CO_2 remains sufficiently low to limit the pressures to manageable magnitudes. Under these conditions CO_2 refrigeration systems exhibit superb efficiency and very compact plant design is possible for a given refrigeration capacity. As a rule, compressor and suction line dimensions reduce to $1/6^{th}$ or $1/8^{th}$ of the comparable R717 or Fluorocarbon equipment size, thereby saving equipment cost.

The facility described herein is the first modern-day Carbon Dioxide refrigeration system in Australia and as such represents both a breakthrough and a landmark in Australian refrigeration history.

2.0 GENERAL DESCRIPTION OF THE APPLICATION/FACILITY

Snap Fresh, a wholly owned Qantas subsidiary, was established to centrally manufacture high quality meals from a greenfield site in Southeast Queensland. The Snap Fresh facility is the first of its kind in Australia with the flexibility to satisfy airline demands for frequent menu changes. The facility has been constructed to worlds best practice.

The facility consists of a number of temperature controlled areas comprising loading bays, dry waste room, raw materials chillers, raw materials cold store, preparation areas, processing rooms, packaging areas, finished product cold store and offices.

The temperatures specified during design ranged from <-30°C in the in-line spiral freezer through to 25°C in the offices and some thermal processing areas. The major objectives during the design development, with regard to the refrigeration and air conditioning systems were:

- High energy efficiency at all times, even during weekend turn-down.
- Highest degree of environmental compatibility and elimination of fluorocarbons from the facility
- Highest possible level of redundancy and reliability
- Maintenance of room temperature and pressurisation to ensure highest levels of hygiene.
- Ease of future expansion
- Acceptable capital cost

3.0 GENERAL DESCRIPTION OF OVERALL PLANT DESIGN CONCEPT

An heat load analysis was conducted to determine the load distribution and duration. This analysis revealed that there was a great disparity between the peak daytime design loads and weekend design loads. Furthermore several loads were strongly process dependent and seasonal. Hence the requirement for a system capable of operating efficiently at part load and at >90% turndown on weekends became clear.

Environmental considerations dictated the use of a central ammonia plant located in the plant room to service all the above loads including office air conditioning. In order to achieve a system with efficient turndown even at the above conditions, a system with 2 identical variable compression ratio screw compressors (one high stage and one low stage) and 3 high stage reciprocating compressors was selected.

Chilled water (+1.5°C) and propylene glycol / water (-8°C) reticulation systems were installed within the ceiling space to provide for the high and medium temperature loads in the facility. The gravity flooded welded cassette plate heat exchangers for these loops were located within the ceiling space and employed innovative cyclonic separator designs for liquid/vapour separation to ensure minimum mass and minimum refrigerant charges.

An initial design concept involved the use of a 2-stage ammonia plant with the low stage serving both the spiral freezer and the 2 cold stores. This idea was rejected and the two-stage ammonia plant limited to the spiral freezer for the following reasons:

- The 2 cold stores are located at opposite ends of a large building and use of ammonia (either pumped or dry expansion) would have implied extensive presence of ammonia pipe lines in the ceiling space,

- ♦ In the interests of operator safety, the consulting engineer wished to avoid the use of ammonia inside storage rooms, where the units could possibly be damaged by fork lift trucks. Adequate space for penthouse units was not available.

- ♦ The low stage compressor would operate at low capacity (<30%) and hence low efficiency, for more than 82% of the time

Therefore in the interests of achieving highest possible plant efficiency, it was decided to select dedicated CO_2 (R744) condensing units with direct expansion fan coil units arranged as two separate refrigeration circuits per cold store to achieve adequate redundancy (i.e. a total of four (4) such circuits). Although these units could have been supplied as air cooled or liquid cooled units using R404a, this would have compromised the objective of achieving a facility that is largely fluorocarbon free. Since cold glycol/water at -8°C is available in the ceiling space at all times to serve the various other refrigerated areas, the concept of using a CO_2 system with glycol as condensing medium was generated.

4.0 R744 SYSTEM DESIGN CONCEPT

Design of CO_2 systems is generally straightforward and for the most part like HCFC and HFC systems, but the thermo physical and thermodynamic properties of CO_2 have to be taken into account.

Special attention should be paid to the slope of the saturation curve for CO_2 compared to other refrigerants. Figure 1 shows a comparison between four different refrigerants, and it is evident that the slope of the CO_2 graph is significantly steeper than for the other refrigerants.

This means that CO_2 is more sensitive to heat input in liquid lines and pump inlets than other traditional refrigerants used. An increase in liquid temperature and a small liquid line pressure drop can easily cause flash gas in liquid lines and inlets to circulation pumps when using CO_2.

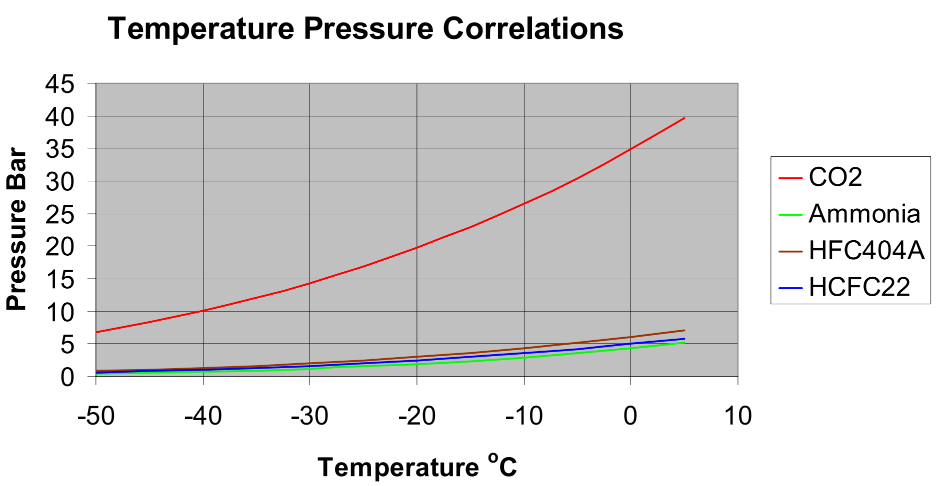

Fig. 1. Temperature – pressure relationships for different refrigerants.

A typical layout of an ammonia/CO_2 cascade system is shown in Figure 2. The CO_2 system is shown as a dry expansion system on the low temperature side. Using CO_2 in compressor dry expansion systems gives the following benefits compared with traditional ammonia or HFC systems:

- No ammonia in occupied areas
- Minimum ammonia or HFC charge
- High efficiency of low-stage CO_2 compressor (lower pressure ratio for CO_2 results in higher compressor efficiency compared with other refrigerants)
- In the case of in-line freezers, higher production capacity due to lower process temperatures
- Reduced compressor swept volumes compared to systems employing ammonia and HFC’s
- Small dimensions of liquid and suction lines

- Use of copper tubes instead of welded steel tubes

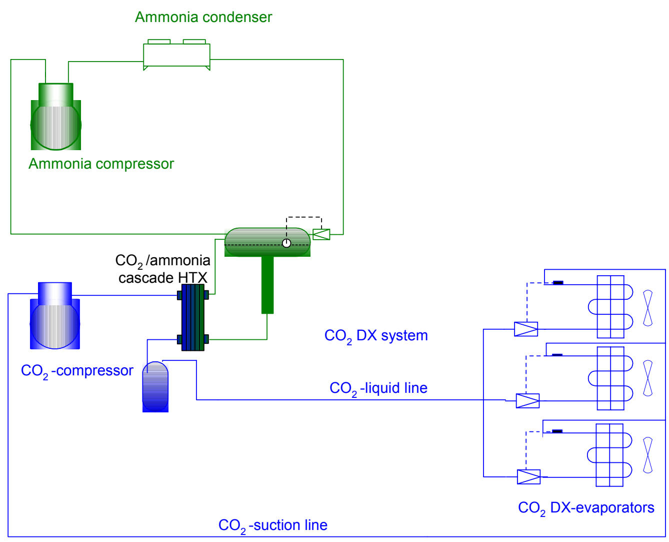

Fig. 2. CO_2 compressor system with dry expansion evaporators

In Figure 3, the calculated COP (Coefficient of Performance) is shown at different compressor suction pressures (shown as saturation temperatures) for CO_2 and ammonia in a two-stage refrigeration plant. The calculations are based on manufacturers' data for industrial reciprocating compressors. CO_2 has a lower pressure ratio compared to ammonia and the resulting improvements in isentropic and volumetric efficiencies are indicated in Figure 3 in the form of improved COP values.

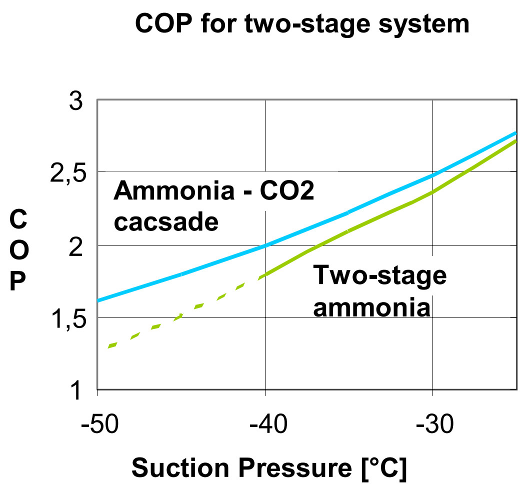

Fig. 3. COP comparison between ammonia two- stage plant and CO_2/ammonia cascade plant at various compressor suction presssures (shown as saturation temperatures)

The calculated performance of CO_2 in a cascade system with ammonia as shown in Figure 3 is better than the equivalent two-stage ammonia plant (open intercooler) despite the effective temperature loss (3K in the example shown) in the cascade heat exchanger. The high-stage ammonia compressor working conditions in the two cases are -8°C/30°C and -10°C/30°C respectively in the CO_2/ammonia cascade and the two-stage ammonia comparison.

In large installations, the CO_2 system can be designed with liquid overfeed in lieu of dry expansion feed in order to improve the efficiency of the total system. When using this design the advantages of a traditional liquid overfeed system can be achieved, but the CO_2 charge will be larger and the risk of high system pressures during power failure and other system faults should be taken into account.

The CO_2 systems meet the cooling demand in two cold stores at a room temperature of -20°C. To ensure reasonable redundancy, each cold store is equipped with two identical CO_2 units each with a cooling capacity of 14 kW. The system layout of each of the four CO_2 units is shown in Figure 4.

The design specifications of each unit are:

Cooling capacity	:	14 kW
Evaporating temperature	:	-28°C
Condensing temperature	:	0°C

The finned air cooler is of the Lu-Ve Contardo dry expansion type with electric defrost. The compressor is a Dorin TCS 340H with a swept volume of 7 m^3/h. The condenser is a Swep type B26x60 brazed plate heat exchanger designed for high pressure operation. All components on the high pressure side are rated for 40 bar working pressure. On the low-pressure side of the system all components are rated for 35 bar working pressure.

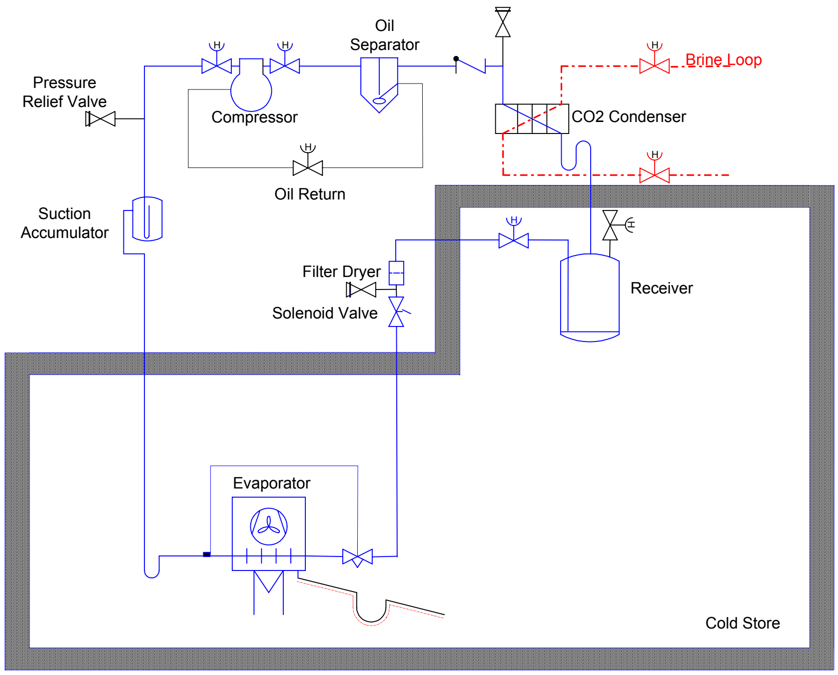

Fig. 4. CO_2 systems at Snap Fresh

Pressure relief valves are mounted where there is a risk of isolating a section of the CO_2 system either manually or automatically. To minimise the risk of CO_2 leakage through the relief valves in the event of a power or brine supply failure, the receiver is located in the cold store. This will prevent the pressure in the CO_2 system from exceeding a saturation pressure corresponding to the room temperature level.

5.0 DEFROST METHOD AND PLANT CONTROL CONCEPT

The operation of the system is controlled directly by the room temperature. When the room temperature sensor is below the temperature set point, the solenoid valve in the liquid line closes and the system cycles off on the pump-out pressure switch. When the room temperature sensor is above the temperature set point the solenoid valve in the liquid line opens and the evaporator commences filling. When the suction pressure reaches the set point of the pump-out switch, the compressor starts and the cooling of the room will continue until the room temperature reaches the temperature set point.

The two systems in each room are defrosted staggered (one coil at a time) 4 times per day controlled by a timer function in the central PLC. The defrost is carried out by means of electric heaters in the coil and the tray. The defrost signal initiates the pump down sequence described above and the defrost heaters are switched on for a period of 20 minutes. If, during defrost, the suction pressure rises and reaches the set point of the pump-out pressure switch, the compressor starts and operates until it is stopped again by the pump-out pressure switch. This ensures the evaporator pressure is maintained below the set point of the relief valves during defrost.

6.0 PRACTICAL ISSUES RELATING TO THE R744 INSTALLATION

If CO_2 in liquid form is expanded into a space with a pressure less than approximately 5.5 bar, then the liquid will solidify and form what is commonly known as dry ice. At saturated operating pressures below approximately 5.5 bar within the refrigeration system, the same will occur i.e. the refrigerant will solidify. Evaporating temperatures below approximately -55°C are hence not possible. The working fluid will solidify and the plant will stop.

These properties are unusual when comparing with conventional refrigerants and they give rise to the following special practical design considerations:

- How is the refrigerant prevented from escaping through the safety relief valves during shorter system standstills caused by power failure, loss of condensing medium or system faults?

- Where are the safety relief valves from the system to be positioned in order to ensure that no dry ice is formed in the safety valve outlet, should the relief valve lift?

- Where are the refrigeration system components suitable for design pressures of 45 bar and test pressures of 67 bar to be sourced?

In modern times the application of CO_2 is relatively novel. The availability of suitable standard system components is therefore relatively poor. With increasing system design pressures, the availability of standard refrigeration components gradually becomes very scarce. The designer is therefore forced to design and manufacture custom engineered pressure vessels and source other components from suppliers of hydraulic, petrochemical, general industrial and gas liquefaction equipment.

To minimise the problems illustrated in fig. 1 (liquid line pressure drop) and for various other practical reasons including service access, the R744 condensing units were mounted in the ceiling cavity of the facility directly adjacent to the freezer stores, refer fig. 5.

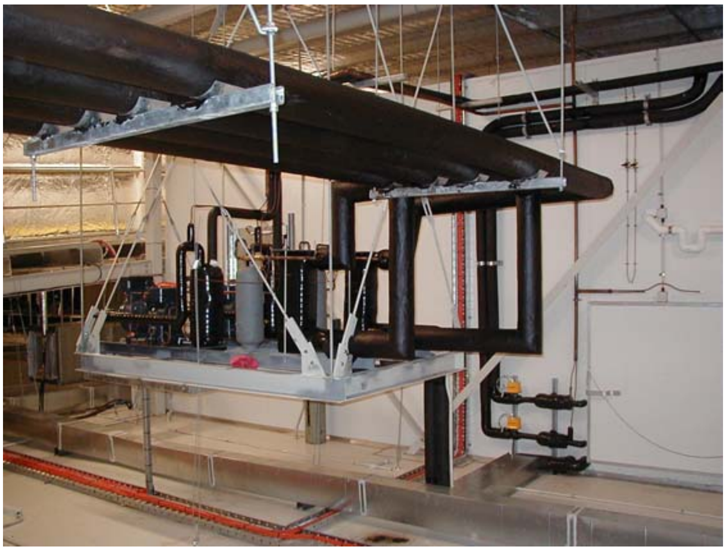

Fig. 5. R744 (CO_2) Condensing Unit installed in ceiling cavity

The two electrically actuated liquid line shut-off valves are visible in the bottom right hand corner. These feature a three second opening/closing time to prevent liquid hammer at the expansion device located at the air cooler. The two liquid receivers are located behind the sliding door to the right of the shut-off valves. The air coolers are installed in the cold store as shown in fig. 6. Due to limited space in the ceiling cavity above the cold store, it was not possible to install the refrigerant pipe lines in any other location than exposed within the refrigerated space.

Fig. 6. Finned air cooler suitable for R744 (CO_2) refrigerant and installed in the cold store

The temperature sensor used for room temperature control and monitoring is visible upstream of the induced draught air cooler. The air coolers are standard copper/aluminium air coolers with modified circuiting for R744 operation. In addition, they are factory pressure tested at elevated test pressure to suit the system design pressure on the evaporator side.

7.0 CHARGING AND COMMISSIONING – PRACTICAL ASPECTS

Refrigeration systems using R744 must be thoroughly evacuated like other HFC/HCFC refrigeration systems. Standard filter/driers for R744 and suitable for the design operating pressures required here are available. However, the system operating pressures are generally very close to the limits of the pressure range being offered by reputable suppliers.

Charging the R744 system like a normal HFC/HCFC system is not readily possible. With HFC/HCFC and other systems it is common practice to charge liquid into the liquid line upon completion of the evacuation i.e. when the plant is still under vacuum. This is not possible with an R744 system because dry ice would form internally at the charging point when the system pressure is less than ~5.5 bar.

It is therefore necessary to increase the system pressure to a level >5.5 bar using CO_2 gas before it is possible to charge liquid into the liquid line upstream of the expansion valve. Liquid line sight glasses are not commercially available for the pressures in question. Determining the correct system charge by observing vapour bubbles or lack thereof in the sight glass is hence not possible. The correct system charge must therefore be determined by weighing. Observing the frost line on the receiver during normal operation is also helpful – the normal temperature of the liquid in the receiver is below 0°C.

In practice it was found that an isolation valve in the equalizing line between the compressor discharge line and the top of the receiver was helpful in preventing excessive compressor discharge pressures – particularly during charging. Care was taken during the installation that the condenser was situated at a higher level than the receiver to ensure liquid drainage by gravity. Occasionally during charging the compressor discharge pressure was elevated – presumably caused by a relatively warm receiver and hence liquid accumulating in the condenser. In these situations, opening and/or closing the equalizing line was a helpful remedy.

Loss of condenser coolant, plant stoppages, power failures and other faults all have the potential to cause elevated pressures within the R744 plant. During plant standstill and without coolant circulating in the condenser, the system pressure will rapidly increase beyond the safety valve setting and the R744 charge will be lost gradually. To counter this problem, the system must either be fitted with an expansion vessel of adequate capacity to store the charge in gas form at reasonable pressure or the receiver must be kept cool at all times. In this case the receiver is kept cool by means of the cold store featuring substantial thermal buffer capacity.

8.0 PERFORMANCE TRENDS, PRACTICAL LONG TERM OPERATION

Fig. 7 shows a graph of a typical system operating cycle incorporating temperature control at the set point, automatic electric defrost and temperature pull-down following the defrost. As shown on the graph, the compressor cycles on/off during defrost in response to a suction pressure signal.

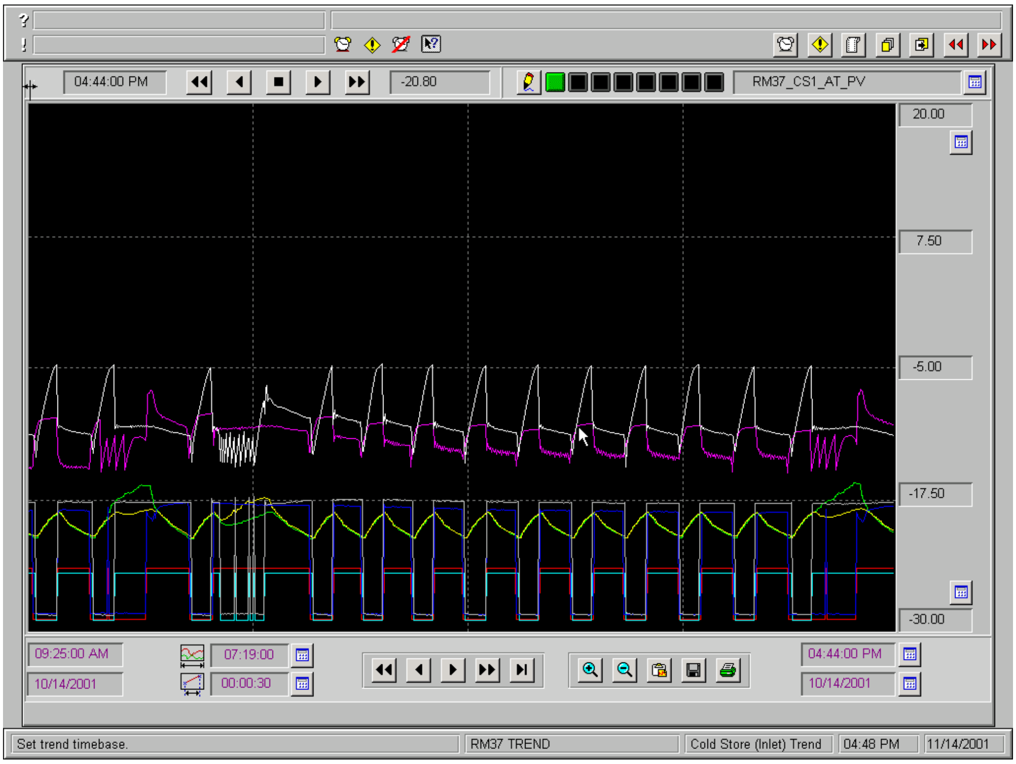

Fig. 7. Graph describing typical R744 system operating cycle

9.0 MISCELLANEOUS PRACTICAL DIFFICULTIES ENCOUNTERED

The systems are all fitted with an oil separator downstream of the compressor. As in any other similar refrigeration system, the oil separated in the separator must be returned to the compressor. The device usually applied is a high pressure float valve fitted to the oil separator. The high pressure float valve opens when the oil level in the separator rises and then passes oil to the compressor sump.

Due to the system design pressure of 45 bar, it was not readily possible to obtain a suitable high pressure float valve for the oil return line. Initial trials with a normal stop valve fitted at the oil outlet from the separator were not successful. It was impossible to determine a correct valve setting and constant manual adjustment was necessary to maintain an oil level in the compressor at all times. The fact that the pressure differential across the valve changed constantly during temperature pull-down in the cold store made correct valve setting more difficult to determine.

The normal stop valve was then replaced with a needle valve. The flow coefficient of the needle valve was determined on the basis of an assumed oil carry-over from the compressor of approximately 300 ppm. The difficulty here is of course taking into account the gas expansion which occurs in the oil return valve. Refrigerant contained in the oil leaving the separator will expand in the needle valve and this expansion increases the necessary flow coefficient of the valve. In addition, no data in relation to actual oil carry-over from the compressor were available from the compressor manufacturer, so this second valve was also found to be unsuitable due to inadequate capacity causing gradual loss of oil from the compressor.

A larger needle valve was then fitted. By this time the cold store temperatures had been reduced to the design temperature of -20°C and an appropriate valve setting was determined quite easily and has been found to be satisfactory since.

The thermostatic expansion valves used were manufactured specifically for the application by Danfoss, Nordborg, Denmark. They are of the conventional mechanical type with external pressure equalisation. Initially, the intention was to employ electronic expansion valves. These were, however, substituted against mechanical valves for greater system simplicity. During commissioning when the cold store temperature was gradually reduced over a 12 day period, the mechanical expansion valves required frequent superheat adjustment. This was to be expected and would have been less labour intensive had electronic expansion valves been employed.

Approximately 3 to 4 weeks after handover of the plant, a compressor failed. The fault appeared to be of electrical nature. The current drawn by the individual motor windings varied by a factor of approximately three causing overheating of the motor and subsequent thermistor protection fault. The compressor was replaced and the faulty compressor repaired by the manufacturer under warranty.

10.0 ASSESSMENT OF R744 SYSTEM RELIABILITY

Making allowances for the relative novelty of this installation, the long term reliability of R744 systems of a similar design concept is considered acceptable and at the same level as conventional refrigeration plants employing HFC/HCFC refrigerants.

During commissioning and during the first three months of operation, the main difficulties encountered with these installations were one compressor failure (electrical), oil differential pressure switch failures, three system leaks, one high pressure safety switch failure and sealing problems with the motorised liquid line shut-off valves. The majority of faults related to novel applications of components at the limit of the application ranges or possibly outside. Had standard R744 components for standard R744 systems been available, the number of faults would have been less.

11.0 FUTURE APPLICATIONS

CO_2 refrigerant is anticipated to find economic and efficient application in the following areas:

- Medium and large Cold Storage systems, as alternative to R22 and R717

- Supermarket freezer display cabinets, as alternative to R404a

- Low temperature blast, spiral, belt and tunnel freezers, as alternative to R717

Large end-users in the food industry, like Nestlé and Unilever, have already declared their intention to use CO_2 in future low temperature applications due to both economic and environmental reasons. Some very large applications (up to 16MW of cooling capacity) are currently being planned and built in Europe and the USA. CO_2 is accepted as an economically viable solution for low temperature applications and the number of new installations will grow rapidly in the near future.

In the supermarket sector the use of CO_2 in freezer cabinets and cold rooms is standard in new supermarket applications in Denmark today. This technology has developed very rapidly in the last two years into an economically competitive solution accepted by both supermarkets and installers.

12.0 CONCLUSION

Considering the fact that these R744 refrigeration plants are the first in modern times to service a commercial cold storage application, the practical problems experienced during commissioning and commercial operation have been less than anticipated at the outset. As previously indicated, identifying suitable system components during the design phase was difficult. This was, however, a direct result of the required system design pressure of 45 bar, which in turn was dictated by the design temperature for the glycol loop servicing the remainder of the food processing facility.

These first Australian R744 installations open up vast new possibilities in terms of providing safer, more energy efficient and more environmentally friendly refrigeration plants. In situations where system leaks from low temperature ammonia refrigeration systems may cause a risk to occupants and/or products, CO_2 or R744 presents an alternative which is acceptable from a technical as well as a commercial point of view. The owner of the food processing facility being serviced by these novel R744 systems is to be congratulated for the initiative and vision displayed when taking the decision in favour of an environmentally friendly natural refrigerant as opposed to conventional HFC/HCFC fluids.

ACKNOWLEDGEMENTS

The valuable assistance of Mr. Jan Bijsterbosch, General Manager and Peter Franklin, Chief Engineer both of Snap Fresh Pty. Ltd. Queensland during the preparation of this paper is acknowledged with thanks.

REFERENCES

1. IPCC Working Group report, 2001

2. IPCC Working Group report 1995