Large-Scale Plate Freezers for Meat Products – Use of Non-Toxic Secondary Refrigerants

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NRTB Secondary Refrigerant Seminars, February 2005

Executive Summary

Approximately two decades ago, large-scale plate freezers were introduced to the red meat industries in Australia and New Zealand for the freezing of meat products in corrugated cardboard cartons.

The development of these large-scale plate freezers was partly funded by the Australian Government as a more energy efficient alternative to the then commonly used automatic air blast freezing tunnels.

Plate sizes of approximately 5,000x3,000 mm are common. Plate freezer holding capacities are up to 2,500 cartons (more commonly 600 to 1,500 cartons) with a unit mass of 27 kg. Clamp times are ~21 hours, evaporating temperatures -38 to -42°C.

The plate freezers operate with ammonia liquid overfeed rates ranging from about 15 to 1 for the latest models up to about 100 to 1 for the older versions. Ammonia charges are around 3,000 to 3,500 kg per 750 cartons holding capacity – some newer models are about 20% less.

Recent accidental discharges of ammonia refrigerant from large-scale plate freezer installations have highlighted that there is a significant risk associated with these installations not only to human health, but also to products, capital equipment and business operations.

The paper presents the impacts on a typical, industrial, large-scale plate freezer installation by substituting the volatile, primary ammonia refrigerant against a non-toxic, environmentally friendly secondary refrigerant. The impacts are analyzed in the following key areas:

- Freezing performance
- Energy consumption
- Occupational Health and Safety
- Suitability for retro-fitting to existing installations employing volatile, toxic fluids

The paper also describes practical line pressure drop measurements from the liquid inlet of an individual plate of a plate freezer to a point immediately upstream of the freezer valve station when employing the volatile refrigerant ammonia. The intermediate line pressure at plate freezer outlet (upstream of the flexible hose) is also presented.

The paper further describes the operating experiences associated with an existing plate freezer installation, which was designed for the use of a secondary refrigerant.

The paper will show that the energy consumption penalty associated with the additional heat exchange between the primary and the secondary refrigerant loops in a plate freezer installation employing a secondary refrigerant are minimal (<5%) when compared with an installation using NH_3 direct. This is due to the fact that the significant wet return line losses in a plate freezer installation employing NH_3 are substantially eliminated.

Conventional Plate Freezing using refrigerant NH_3

Overview

The conventional plate freezing facility for a meat processing plant producing a certain daily quantity of frozen product is shown in Figure 1 in plan view. The automatic chilling tunnel shown alongside the plate freezers is for the processing of chilled product. The chilling equipment is not relevant to this presentation.

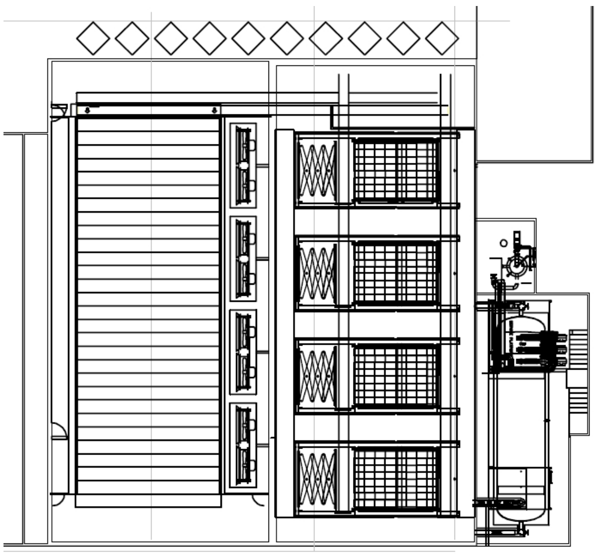

Fig.1 Plan view of conventional meat carton plate freezing plant

The contact freezers shown above may be of the manual load/unload, semi-automatic load/unload or fully automatic load/unload type. The loading mechanism shown above is of the scissor type at constant level.

In fig. 1 is also shown the very large refrigerant accumulator required for large-scale contact freezer installations. The great majority of contact freezers in the meat industry employ refrigerant ammonia.

In large-scale contact freezer installations using refrigerant ammonia, the approximate ammonia system charge is around 3,000 to 3,500 kg per 750 meat cartons. Installations for the freezing of 5,000 cartons per day will typically hold 20 to 25 metric tons of ammonia. The necessary ammonia accumulator size is 40 to 45 m^3 as a result of the need to hold 110% to 120% of the total ammonia (NH_3) charge.

Large quantities of ammonia refrigerant in contact freezer installations are of a serious safety concern mainly to legislators. This safety concern is aggravated by the operating principle of a contact freezer with extensive employment of flexible refrigerant connections, which are subjected to movement at low temperatures on a daily basis during normal production periods.

More importantly, recent uncontrolled releases of ammonia from contact freezer installations in the Australian meat industry have attracted the attention of Authorities charged with the task of enforcing the Occupational Health and Safety legislation in place in all Australian States and Territories.

Occupational Health and Safety legislation is now such that uncontrolled releases of toxic substances including ammonia will transform the site of the incident into a forensic site [1]. The consequence is that the operator(s) in charge are only authorized to take actions to terminate the release. Further actions including recommencement of meat plant operations are in the hands of the Authorities.

Theoretical Analysis of the Freezing Process in a Plate Freezer

The theoretical determination of the necessary temperature of the freezing medium to achieve the required process outcome in a plate freezer is calculated in table 1 for a typical product.

The initial determination of the required temperature of the freezing medium is made using simplified freezing time formulae [2][3].

Freezing concept	Contact Freezer
Thermal conductivity of the product below the freezing point (average temperature ~-11°C), W/mK	1.345
Product density above the freezing point, kg/m³	1000
Product density below the freezing point, kg/m³	950
Product specific heat above the freezing point, kJ/kgK	3.40

Product specific heat below the freezing point, kJ/kgK	1.78
Latent heat of fusion, kJ/kg	254
C+, MJ/m³K	3.40
C-, MJ/m³K	1.69
Freezing point, °C	-2
ΔH (0/-10), MJ/m³	274
Initial product temperature, °C	30
Final product core temperature, °C	-12
Final product mass average temperature, °C	-19
Thermal resistance of carton material, m²K/W	$4.8*10^{-2}$
Thermal resistance of 0.3 mm thick poly bag, m²K/W	$1.76*10^{-3}$
Thermal resistance of air gaps within carton, m²K/W	0.004 (~0.10 mm avg.)
Thermal resistance from exterior of plate surface to refrigerant (δ_{Al} = 0.006 m, λ_{Al} = 210 W/mK), m²K/W	$2.86*10^{-5}$
Oil fouling resistance within contact freezer (oil film thickness 0.05 mm, λ_{OIL} = 0.12W/mK), m²K/W	$4.17*10^{-4}$
Surface film coefficient within plate freezer (d_{HYDR}=0.039 m, cavity length 4 m, 48 cavities, 24 passes, n_{CIRC}=50 to 1, mass flow density = 90.9 kg/sm^2, $\Delta p_R \cong$ 0.3K), W/m²K	821
Total thermal resistances freezing medium/product, m²K/W	0.0554
Heat transfer coefficients freezing medium/product, W/m²K	18.04
Product shape	Infinite slab
Thickness, m	0.165
Required temperature of freezing medium, °C	-39.0
Freezing time, hours	21.0

Table 1. Required freezing medium temperature in a contact freezer with volatile refrigerant.

The thermal carton resistance of 0.048 m^{2}K/W has not been estimated in detail, but it is a relatively optimistic assessment compared with estimates found in technical reports in relation to freezing trials of meat in cartons [4].

Table 1 clearly shows that contact freezer performance for meat packed in corrugated cardboard cartons is severely inhibited not only by the carton material, but also by variations in contact caused by air gaps, differences in carton heights or curled up plastic wrapping material in the top of the box.

The result of the theoretical estimation of freezing medium temperatures shown in table 1 is in good agreement with practice. Many contact freezer installations in the meat industry are designed around -38°C to -40°C evaporating temperature in the freezers with saturated compressor suction temperatures around -42 to -43°C achieved in practical operation.

Practical Performance Measurements in an Existing Plate Freezing Plant

For large overfeed rates in low temperature plants with significant differences in levels, ammonia is not an ideal refrigerant. This is predominantly due to the pressure/temperature relationship of the fluid whereby small pressure changes give rise to significant changes in saturation temperature.

The various pressure drops in the refrigerant circuit (through the plates, hoses, valves and wet return risers) are indeed calculable and are often found to be significant. This is for example the case where, due to building constraints, positioning the accumulator at a lower level than the plate freezer(s) is difficult or impossible.

To verify the impact of these system refrigerant pressure drops on freezer performance and indeed energy consumption, the system pressure drops in a typical existing plate freezer installation as shown in fig. 2 were measured.

Fig. 2. Typical Plate Freezer Installation

The installation shown comprises four plate freezers each with a capacity of around 550 cartons. Using entering/leaving product temperatures of 35/-18°C respectively, the calculated average refrigeration capacity per plate freezer is approximately 52 kW for a clamp time of 21 hours. The installation is shown schematically in fig. 3.

To measure the various refrigerant pressures in the refrigerant circuit, pressure transmitters were positioned as shown in fig. 4 and connected to the existing computerized control and monitoring system for the refrigeration plant. This enabled trending of the pressures throughout the batch freezing process.

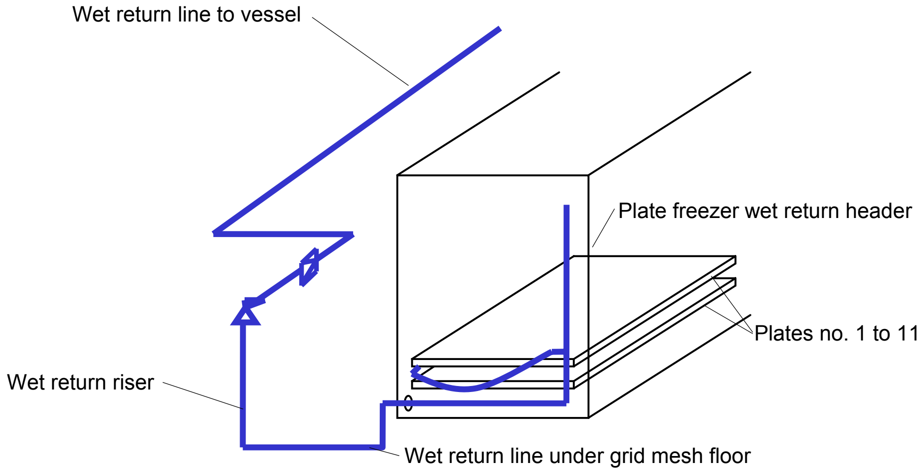

Fig. 3. Schematic diagram of plate freezer installation

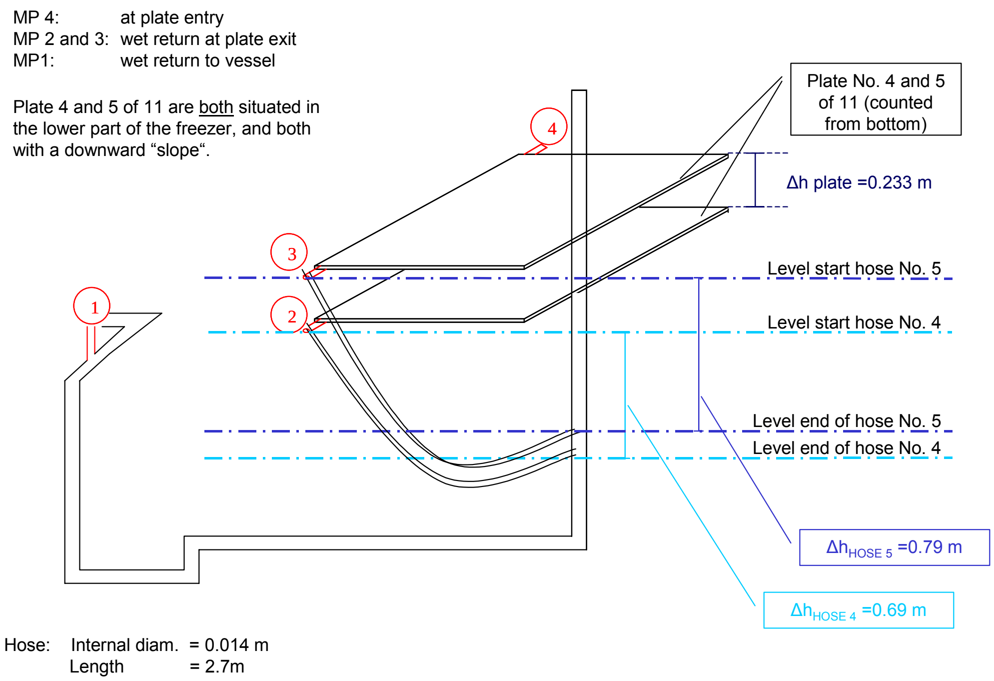

Fig. 4. Location of pressure transmitters in the refrigerant circuit

The pressure transmitters in the wet return connections were positioned as shown in fig. 5 and fig. 6.

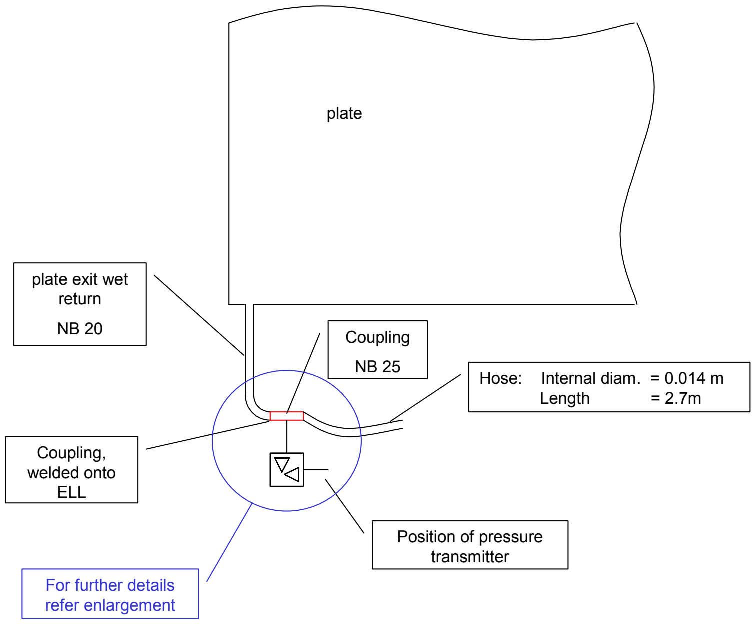

Fig. 5. Plan view of freezer plate

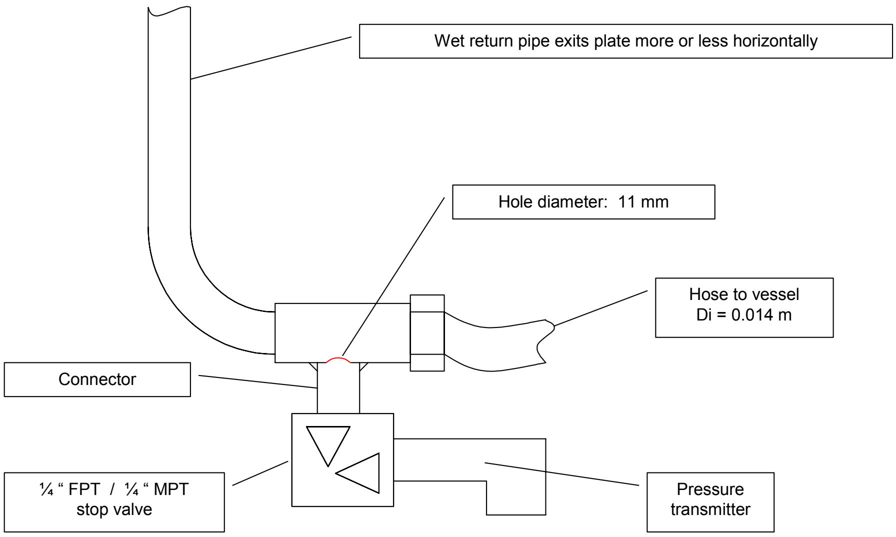

Fig. 6. Enlargement of plate exit detail

The results of the measurements are shown in fig. 7. It is evident that there is a significant pressure (temperature) drop from plate entry to plate exit of close to 6K. The pressure (temperature) drop from the plate exit to the wet return line valve station of around 3 to 3.5K is also substantial.

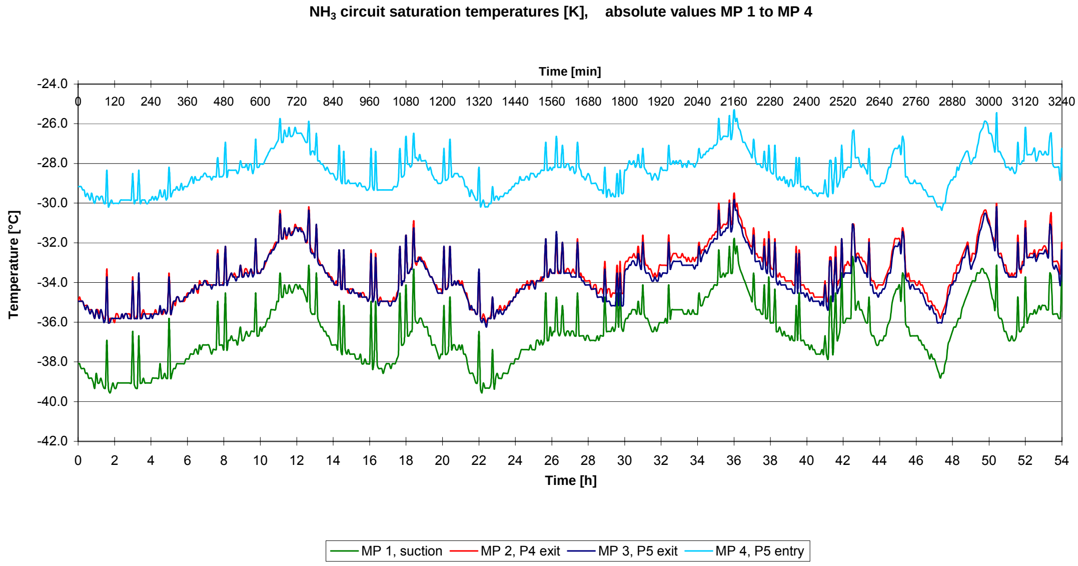

Fig 7. Saturation temperature trends for plate freezer

Another significant finding associated with the pressure measurements was that there was limited to no refrigerant evaporation occurring in the plates. This was due to the extraordinarily high overfeed rates of >100 to 1 in this particular plant.

Essentially, the ammonia refrigerant was therefore used as a volatile secondary refrigerant. This is an operating principle, which has been widely used in many applications for several decades, but it was presumably not the intent to operate this plant this way.

The difficulty in this situation is to determine at which point in the refrigerant circuitry the refrigerant vapour will form and what impact this has on the circuit pressure drop.

At commencement of the freezing process where the load is relatively high, some vapour may form in the plates. Towards the conclusion of the freezing process where refrigeration loads are low, very limited vapour will form. This is a situation, which has been experienced by many plate freezer operators. The refrigerant relocates from the accumulator to the plate freezers and often a low accumulator refrigerant level is experienced.

It is evident from the pressure measurements that there are unavoidable pressure drops in the plate freezer refrigerant circuit. These pressure drops are sufficiently significant to jeopardize the energy efficiency of the freezing process.

Using a total wet return line/dry suction line loss of 3.5K as per the measurement results, the specific energy consumption in kWh per metric ton associated with contact freezing of meat cartons may be approximated as shown in table 2. The calculation is for a dual compression stage ammonia refrigeration system and is not related to the plant measured.

Freezer type →	Contact freezer
Conduction, kW	7.2
Infiltration, kW	4.9
Lighting, kW	2.5
Total product quantity processed, kg	135,000
C_+, kJ/kgK	3.4
Freezing point, °C	-2.0
C_-, kJ/kgK	1.78
Latent heat of fusion, kJ/kg	254
Initial product temperature, °C	30
Final product core temperature, °C	-12
Final product mass average temperature, °C	-19
Total product heat removal per batch, MJ	53,063
Cycle time, hours	21.0
Average product heat removal, kW (kJ/s)	701.9
Total refrigeration plant capacity required, kW	716.5
Suction line loss, K	3.5
Required saturated compressor suction temperatures, °C	-42.5
Booster shaft power, kW	199.6
Compressor shaft power, kW	230.2
Condensing temperature, °C	33.0
Condenser heat rejection, kW	1144
Condenser fans, kW	19.1
Condenser spray water pumps, kW	2.3
Refrigerant pump(s), kW	17.4
Evaporator fans, kW	0
Total system shaft power, kW	471.1
Overall average system efficiency	0.85
Total system energy input per cycle, kWh	11639
System energy input/t, kWh/t	86.2

Table 2. Energy consumption of typical contact freezing installation

Alternative Plate Freezing Plant with Non-Toxic Secondary Refrigerant

Substitution of the toxic, volatile ammonia refrigerant in plate freezers against an environmentally friendly, non-toxic refrigerant appears to be a potentially attractive proposition to operators. In existing plants, this would be conditional upon mechanical modifications to the freezers being kept to a minimum and negligible penalties in terms of freezing performance and energy consumption.

The operating principle of such a system is shown in fig. 8. A shell and tube heat exchanger with marine type water boxes has been used. This is in order to be able to remove the contaminants (oil, debris and miscellaneous chemical compounds) returning from the plate freezers following the conversion to a secondary refrigerant.

In a new installation, welded cassette plate heat exchangers may be used for further reduction in ammonia charge and plant dimensions.

As an example, the shell and tube heat exchanger has been sized for a refrigeration capacity to suit the plant measured (refer fig.2); i.e. ~250 kW. This includes an allowance of 20% or around 40 kW for the situation when a plate freezer is put on line with a batch of warm product (pull-down).

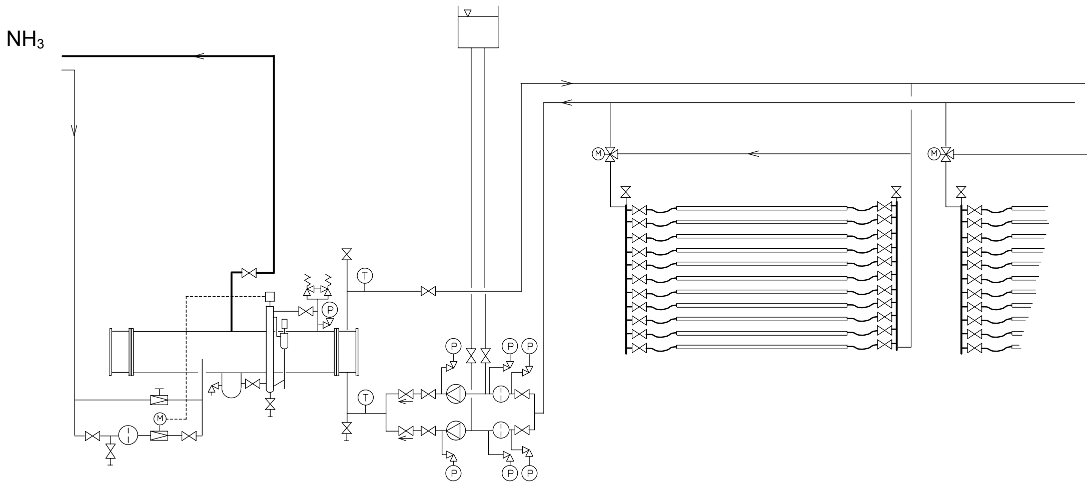

Fig. 8. Typical plate freezer installation employing secondary refrigerant

Each plate is manufactured from a number of aluminium extrusions welded together. The cross section of one refrigerant pass is shown in fig. 9. Each pass comprises three galleries and each gallery has the dimensions shown.

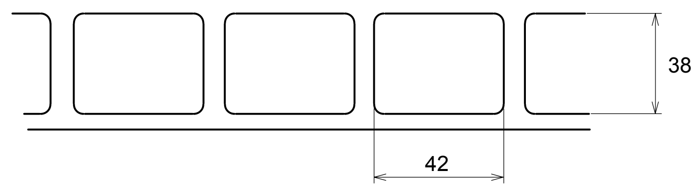

Fig. 9. Plate cross section

The total number of passes in each plate is 21. To ensure the removal of air from the passes it is important to maintain a minimum fluid velocity. Using this minimum fluid velocity yields a flow per pass of 0.0015 m^3/s. With four plate freezers and 11 plates per freezer, the total fluid flow becomes 0.068 m^3/s.

The shell and tube heat exchanger design details are provided in table 3.

Heat exchanger type	Shell & Tube
Shell Outside Diameter, mm	1219
Tube outside diameter, m	0.0603
Tube inside diameter, m	0.05248
Net tube length, m	6.0
No. of active tubes	200
No. of tube passes	10
Tube side flow, m^3/s	0.068
Tube side fluid	$Temper_{-40}$
Tube side fluid mean temperature, °C	-35
Tube side fluid density, kg/m^3	1226
Tube side fluid specific heat, kJ/kgK	2.851
Tube side fluid thermal conductivity, W/mK	0.405
Tube side fluid dynamic viscosity, kg/ms	0.03462
Tube side fluid leaving temperature, °C	-35.0
Thermal conductivity of tube material, W/mK	46
Inside fouling resistance, m^2K/W	0.00005
Shell side fluid	NH_3
Shell side fluid evaporating temperature, °C	-42.0
Pool boiling factor B	2.68
Capacity, kW	254
Tube side fluid in, °C	-33.9
α_{INSIDE}, W/m^2K	368
Tube side fluid pressure drop, kPa	123
$\alpha_{OUTSIDE}$, W/m^2K	289
k_o, W/m^2K	148.6

Table 3. Shell and tube heat exchanger for plate freezer installation

The secondary refrigerant flow and the shell and tube heat exchanger design focus on minimization of plate freezer modifications. Modifications to the plate circuiting as such have been avoided, but replacement of the existing refrigerant hoses with hoses with an internal diameter of minimum 25 mm will be required.

The refrigerant side pressure drop in each plate after the conversion to secondary refrigerant is estimated at 23 kPa. The combined pressure drop in the 2.6 m long inlet and outlet hoses is estimated at a total of ~90 to 100 kPa if smooth hoses with an internal diameter of 25 mm are fitted.

Based on the estimated system pressure drops, the absorbed power of each of the two secondary refrigerant pumps is approximately 15 to 16 kW. The power absorbed by the existing ammonia pumps represents around half of this value. The impact on unit freezing cost of the additional energy consumption of the refrigerant pumps is <5%.

The reduction in primary refrigerant charge by conversion to secondary refrigerant is estimated at a factor of around 5 to 6. For the plant in question, this would mean a reduction in NH_3 charge from around 22 m^3 to around 4 m^3. More importantly, all NH_3 would be eliminated from the plate freezer area and confined to the engine room.

Table 4 shows the relative difference in plate freezer performances when using ammonia as the primary refrigerant and when employing $Temper_{-40}$ in the freezers where the measurements were conducted. The refrigerant temperature used for the ammonia plant is an average value derived from fig. 7.

Freezing concept	Contact Freezer	Contact Freezer
Refrigerant	NH_3	$Temper_{-40}$
Thermal conductivity of the product below the freezing point, W/mK	1.4	1.4
Product density above the freezing point, kg/m³	1000	1000
Product density below the freezing point, kg/m³	950	950
Product specific heat above the freezing point, kJ/kgK	3.40	3.40
Product specific heat below the freezing point, kJ/kgK	1.78	1.78
Latent heat of fusion, kJ/kg	254	254
C+, MJ/m³K	3.40	3.40
C-, MJ/m³K	1.69	1.69
Freezing point, °C	-2	-2
ΔH (0/-10), MJ/m³	274	274
Initial product temperature, °C	30	30
Final product core temperature, °C	-10	-10
Final product mass average temperature, °C	-17	-17
Thermal resistance of carton material, m²K/W	$4.8*10^{-2}$	$4.8*10^{-2}$
Thermal resistance of 0.3 mm thick poly bag, m²K/W	$1.76*10^{-3}$	$1.76*10^{-3}$
Thermal resistance of air gaps within carton, m²K/W	0.004 (~0.10 mm avg.)	0.004 (~0.10 mm avg.)
Thermal resistance from exterior of plate surface to refrigerant (δ_{Al} = 0.006 m, λ_{Al} = 210 W/mK), m²K/W	$2.86*10^{-5}$	$2.86*10^{-5}$
Oil fouling resistance within contact freezer (oil film thickness 0.05 mm, λ_{OIL} = 0.12W/mK), m²K/W	$4.17*10^{-4}$	0
Surface film coeff. within plate freezer, W/m²K	683	126

Total thermal resistances freezing medium/product, m²K/W	0.05567	0.06173
Heat transfer coefficients freezing medium/product, W/m²K	17.96	16.20
Product shape	Infinite slab	Infinite slab
Thickness, m	0.165	0.165
Average temperature of freezing medium, °C	-33.5	-34.5
Freezing time, hours	23.5	24.4

In relative terms, there will be more or less no difference in freezing performance between the two freezing methods.

This conclusion is supported by the freezing results achieved in a plate freezer installation using secondary refrigerants. The installation referred to has been in commercial operation for around 6 years and is shown in fig. 10, fig. 11 and fig. 12 below.

Fig. 10. Plate freezer employing secondary refrigerant – general view

In this particular installation two different secondary refrigerants are used. In one plate freezer a mixture of ethylene glycol and water is used. The other installation was in December 2000 converted from ethylene glycol to Freezium$_{-55}$. The latter conversion was an attempt to reduce freezing time.

The plant generally operates with secondary refrigerant supply temperatures around –30°C to -32°C, the compressor saturated evaporating temperature is around -38° to -40°C.

The refrigerant supply and return hoses are manufactured from rubber. During six years of commercial operation no hoses have been replaced, but it has been necessary to replace some hose clamps as a result of minor refrigerant leaks developing.

Fig. 11. Plate freezer hose and clamp detail

Hose repairs are carried out whilst the plant is operating. Temporarily the refrigerant pump is stopped, the hose is tied off, removed from the plate connector and then re-attached with a new clamp.

Fig. 12. Plate freezer engine room

In fig. 13 is shown the result of a random freezing trial conducted in May 2001. The product was loaded on Friday 4 May 2001 and remained in the plate freezers over the long week-end. The final product core temperature is not shown in the graph, but equilibrated at -37.6°C after a clamp time of around 40 hours.

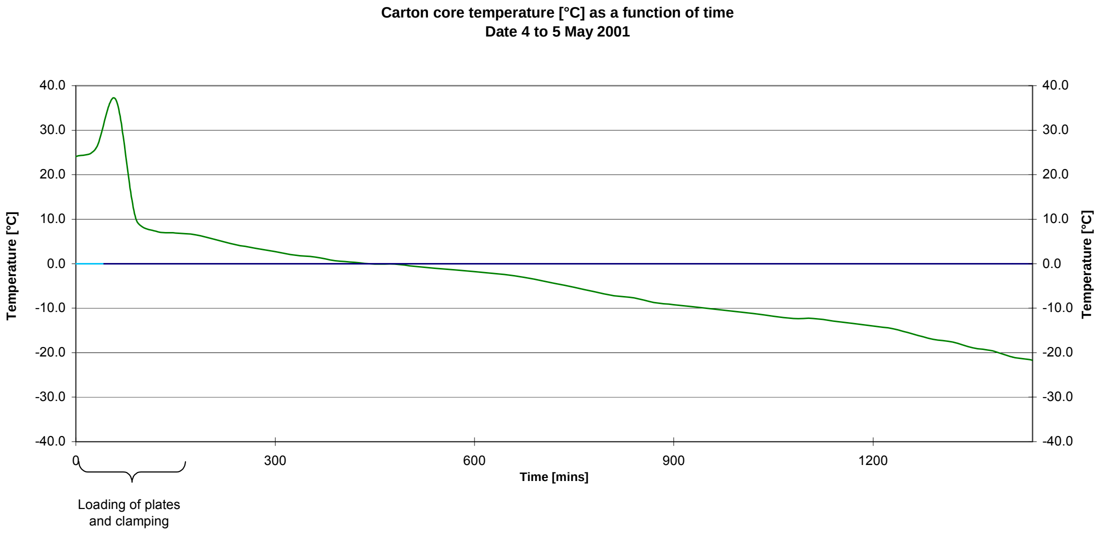

Fig. 13. Result of random freezing trial in plate freezer employing Freezium

Conclusion

The use of modern secondary refrigerants with low viscosity and low environmental impact in large scale plate freezers in the Australian and New Zealand meat industries is presented as a viable alternative to the toxic, volatile primary refrigerant ammonia.

Direct substitution of ammonia against a secondary refrigerant such as Temper is possible with no alterations to plate circuiting. Enlargement of the refrigerant hoses is necessary to maintain reasonable circuit pressure drops.

The freezing performance of the plate freezer after conversion remains more or less unchanged under the operating conditions described in the paper. Prolongation of freezing time remains below an estimated 5%.

The energy consumption penalty associated with substituting the volatile primary refrigerant NH_3 against Temper is in the case described estimated at less than 5%. This penalty is predominantly due to the increased refrigerant pumping power.

The system efficiency penalty associated with the additional heat exchange between the primary and the secondary refrigerants is to a large extent compensated for by substantial reductions in primary refrigerant system pressure drops.

Maintaining existing plate circuiting when substituting NH_3 against Temper is not ideal from a system efficiency point of view. This approach was adopted for simplicity and to make the introduction of the secondary fluid more attractive for retrofitting to existing plants. The penalties are relatively high system fluid flows and hence elevated pumping power.

Plate freezer installations, which are optimized at the outset in terms of design for secondary refrigerants, will feature higher system efficiencies than the plant described in this paper.

Such systems would feature plates with circuiting optimized for less secondary fluid flow. Provided the risk of fouling is negligible, such systems may also be fitted with heat exchangers more efficient and compact than a cleanable shell and tube heat exchanger.

The very substantial reduction in primary refrigerant charge of a factor of 5 to 6 in the case presented and the elimination of any toxic refrigerant from the plate freezers significantly improve operator safety. Even greater reductions in primary refrigerant charge are possible with welded cassette plate type heat exchangers and the use of CO_2 as the primary refrigerant.

Plate freezing systems, which from the outset are designed for secondary refrigerant use, will feature greater flexibility with respect to piping and relative levels within the plant.

Although modern secondary refrigerants are relatively expensive, they are no more costly than the primary refrigerant they are substituting.

The vessels for storage of secondary refrigerant have a significantly lower pressure rating than the large refrigerant accumulator(s) required for plants employing NH_3. The costs of the secondary refrigerant vessels will therefore be less. The need for very large NH_3 accumulators rated for pressures from 1.2 to 1.4 MPa is eliminated.

New installations employing secondary refrigerants are flexible with respect to selection of primary refrigerant. The plant shown in figs. 10 to 12 employs an HFC as the primary refrigerant and semi-hermetic light industrial screw compressors.

Another highly suitable primary low temperature refrigerant is CO_2. Use of CO_2 would enable factory design and assembly of highly space efficient refrigeration packages capable of providing chilled secondary refrigerant whilst condensing against the -10°C NH_3 loop found in most meat plants via a CO_2/NH_3 cascade heat exchanger.

In the case of smaller installations, the units may employ multiple, light industrial semi-hermetic compressors. An added benefit of this concept would be a further significant reduction of the NH_3 charge and hence further reduction of the occupational health and safety risks.

References:

[1] Occupational Health and Safety Act 2001 (NSW).

[2] "Prediction of freezing and thawing times for multi-dimensional shapes by simple formulae, Part 1: regular shapes." D.J. Cleland, A.C. Cleland and R.L. Earle. Massey University, Palmerston North, New Zealand.

[3] Prediction of freezing and thawing times for foods to different final product centre temperatures. D.J. Cleland, Department of Biotechnology, Massey University, Palmerston North, New Zealand.

[4] "Energibesparende Indfrysning" (*Energy Conserving Freezing*). Denmark, 1983. Working group comprising Dirks & Co. (consulting engineers), A/S Thomas Ths. Sabroe & Co. and Vejle Fryse- og Kølehus (commercial cold store).

Acknowledgements:

The valuable assistance of John Perkins, Chief Engineer of Kilcoy Pastoral Company, Kilcoy and Steve Johnstone, Plant Maintenance Officer of Lago Cold Stores Pty. Ltd., Hemmant, Queensland during the preparation of this paper is acknowledged with thanks.