Large Scale Cold Stores, An Innovative Design Approach

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ABSTRACT

This paper describes the design details of a dual stage ammonia refrigeration system servicing a large, new commercial cold storage facility in South East Queensland, Australia. Actual key performance data including store temperatures, power consumption and annual energy consumption for the facility are described. Comparisons between a similar existing facility and the new facility both operated by the same owner show a reduction in annual energy consumption per unit volume of cold storage in the new store of more than 30%. The innovative penthouse evaporator concept reduced specific evaporator power consumption to approximately one third of conventional designs operated by the same owner.

Keywords:

Large Cold Stores, New Penthouse Evaporator Design, Fan Speed Control, New Control Concept, Energy Management, Reduced Annual Energy Consumption.

1.0 INTRODUCTION

The construction of the commercial cold storage facility which is the subject of this paper commenced in February, 1999. Commissioning of the refrigeration system was completed September, 1999. The facility is owned and operated by Frigmobile Pty. Ltd. a wholly owned subsidiary of John Swire & Sons Pty. Ltd.

The Swire Group is a major player in the public refrigerated warehouse industry around the world with a total of 50 plants in the U.S.A. and Australia totalling 3,700,000 m³. The group's Australian market share is approximately 35% with nearly 1,200,000 m³ at 16 sites.

The development stage 1 of the cold storage facility has an initial volume of approximately 46,000 m³ plus 7,000 m³ annex. In the third and final stage of the development, the total cold storage volume will be 121,000 m³ with 12,000 m³ annex. Low and high temperature refrigeration capacities are approximately 0.8 MW and 0.1 MW respectively.

The cold store will in the final stage of development comprise three large low temperature storage rooms and one common annex operating at 2°C. The design storage temperature for the low temperature segment is -30°C. There are no blast freezing tunnels.

2.0 BUILDING

The insulated building enclosure is manufactured from expanded polystyrene panels.. Exterior metal sheathing on all walls is included to prevent direct exposure of exterior walls to sunlight. The mechanical and electrical services are in the main located in the cavity between the insulated ceiling panels and the roof. The general building layout is as shown in plan view in fig. 1. The building layout

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reflects the need to maximise utilisation of the land and to accommodate the requirements of the local road planning authorities. The internal height of the cold store varies from approximately 13.5 meters to 14 meters.

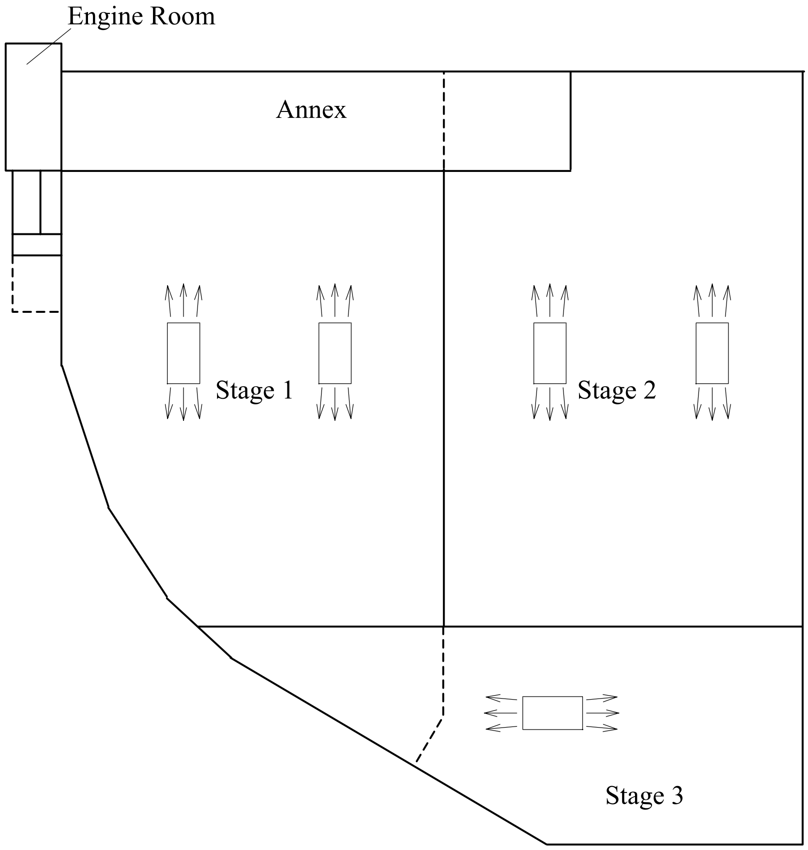


FIGURE 1. Layout of the cold store in plan view.

The exterior structural steel frame of the building is of adequate capacity to minimise the need for structural supports in the form of columns within the refrigerated space. To maximise the number of pallet spaces, the finned air coolers are of the penthouse type also supported by the steel frame.

2.1 Purpose and Location

The purpose of the cold store is medium to long term frozen storage of a variety of frozen foods. The hours of operation are from approximately 0400 hours to approximately 2200 hours daily except week-ends.

The cold store is located in Brisbane, Australia. The building is in relatively close proximity to a creek and the building foundation features relatively high water content and poor load bearing capacity. This necessitated pre-loading of the site with fill for several weeks prior to commencement of construction.

2.2 Future Extension Plans

The future extension plans for the facility are indicated in fig. 1 as stages 2 and 3. The refrigeration system was, as part of stage 1, prepared for the addition of stages 2 and 3. These preparations included the provision for the addition of compressors, air coolers and condensers as required. The main refrigerant pipe lines were selected on the basis of the final duty and in accordance with the maximum permissible pressure drops specified by the owner.

3.0 REFRIGERATION PLANT DESIGN

3.1 Design Objectives

In addition to the usual technical design requirements in terms of storage temperatures, air distribution and noise levels set for the facility by the owner, the overall design objectives for the refrigeration system were in summary:

- Annual energy consumption per unit volume of storage minimum 20% lower than any other similar facility operated by the same owner in the Brisbane area.

- Minimisation of all maintenance costs.

These design objectives as well as the tender evaluation methodology were clearly stipulated by the owner. The determining factor from the owner's point of view was lifecycle costs. In the context of the tender evaluation, lifecycle costs were considered to comprise of the following three main elements:

1. Initial capital cost of the plant.

2. Total annual maintenance costs on an all risk, all included basis.

3. Total annual energy costs.

3.2 Plant Concept

Three general plant concepts were considered by the owner. These were in summary:

- A dual stage liquid recirculation refrigeration plant employing refrigerant ammonia and screw compressors with synthetic refrigeration machine oil non-miscible with ammonia.

- A single stage dry expansion refrigeration plant employing refrigerant ammonia and screw compressors with economizers and ammonia soluble refrigeration machine oil.

- A single stage liquid recirculation refrigeration plant employing refrigerant ammonia and screw compressors with economizers and synthetic refrigeration machine oil non-miscible with ammonia.

The desire on the part of the owner to employ natural refrigerants only eliminated any HCFC and HFC refrigerants as possible options. An ammonia/carbon dioxide dual stage cascade refrigeration system was contemplated, but not seriously considered by the owner.

The design objectives set for the plant eliminated the two latter concepts. The analysis which resulted in the selection of the first concept is not the subject of this paper. However, the anticipated 5 to 10% greater annual energy consumption of the two latter concepts put their respective total lifecycle costs at a significant disadvantage.

3.2.1 Finned Air Coolers

The finned air coolers are of the penthouse type with a unit refrigeration capacity of nominally 150 kW including fouling allowances. This air cooler concept was selected to maximise the available pallet space within the cold store. A cross sectional view of the unit design in end elevation is shown in fig. 2. Only the air return grille and the air discharge ducts are visible from within the refrigerated space.

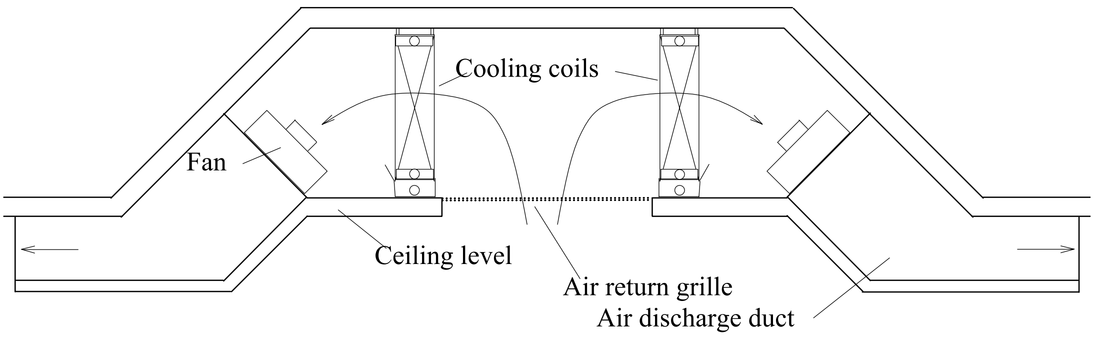


FIGURE 2. Cross sectional view of penthouse evaporator design in end elevation.

During the penthouse unit design, particular emphasis was placed on minimisation of power consumption through reducing the air path pressure drops. Other design considerations were low noise levels, long intervals between defrosts, prevention of slow ice accumulation on coils and in drain pans, adequate air throw within the cold store and minimisation of sealing problems in the unit housing for long service life.

Compared with traditional penthouse unit designs where the air paths feature several relatively sharp angles, this new design largely avoids any 90° turns except for the air return point. The air return grille was theoretically found to contribute significantly to the total air pressure drop. Hence the air intake velocity was reduced by increasing the grille area in order to minimise this problem. Low noise and power consumption levels were achieved by employing variable speed drive fan motors. These measures reduced the peak fan power consumption to 7% of the nominal clean design evaporator refrigeration capacity. In conventional penthouse concepts operated by the same owner, the fan power consumption represents ~22% of the nominal clean design evaporator performance. The new penthouse evaporator concept therefore represents an improvement in terms of power consumption of a factor of approximately three over conventional penthouse designs.

Often the argument is presented that forced draught air flow is thermodynamically more advantageous than induced draught as employed with this new penthouse concept. Forced draught (fans upstream of the cooling coil) gives rise to non-uniform face velocity distribution compared with induced draught where the air is drawn through the cooling coil (coil at fan suction side). The evaporator performance penalty associated with this non-uniform face velocity distribution is approximately 5% [1]. Induced draught air flow was retained in this new concept because the performance improvement derived from uniform face velocity outweighs the penalty associated with adding the fan heat downstream of the cooling coil. At design conditions, the fan heat in this case heats the air stream ~0.2K. This would translate into a theoretical cycle efficiency improvement of ~0.5% if forced draught was employed. A 5% evaporator performance penalty, however, reduces cycle efficiency by ~1%. The fact that induced draught air flow also improves air throw by a factor of three to four compared with forced draught [3] contributed to retaining induced draught as the preferred arrangement.

Long intervals between defrosts of around two days were to some extent achieved by increasing the fin spacing to 12 mm. Air throws ensuring air movement at the cold store walls even at 20% fan speeds became possible through the air discharge duct layout and design as well as through retaining induced draught air flow pattern. The air discharge velocity from the ducts was selected to provide a theoretical air velocity at the cold store walls of 2.0 to 2.5 m/s at peak fan speed. Rapid defrosts were achieved by selecting a light weight air cooler geometry with 321SS tubes and aluminium fins combined with single header circuiting, insulated hot gas supply lines, double bottom drain pans and fast removal of subcooled liquid refrigerant at the initiation of the defrost.

3.2.2 Automatic Control Concept

When fully extended, the cold store complex will comprise three separate low temperature stores. All will operate at identical temperatures. The only high temperature area is the ante room. This relatively low number of refrigerated areas and the lack of room temperature variation between individual stores enabled a relatively simple yet very energy efficient overall automatic control concept to be devised.

At both first and second stage compression levels, the prevailing compressor suction temperatures are determined by the required room temperatures. The suction or evaporating temperatures are therefore not constant, but vary according to the refrigeration load. At high loads, the temperature difference across the air coolers is relatively large and vice versa at low loads. At all operating conditions, the finned air cooler surface areas are hence fully utilised and the refrigerant flow through these is normally not cycled.

The power consumption of the air cooler fans has a compounding effect on plant energy consumption. Firstly, the direct power consumed by the fans must be considered. Secondly, the heat rejection from the fans inside the cold store will indirectly affect the compressor power consumption. The rotational speed of the air cooler fans is therefore in automatic mode controlled in such a way that the overall power consumption of the plant is minimised at all times. The control system monitors the sum of the fan power and the compressor power usage and modulates the fan speed until a minimum is found. The downwards modulation of the fan speed may be overridden either by the operator or by temperature sensors in the store detecting poor temperature uniformity.

The condensing pressure is normally not controlled. The evaporative condenser fans operate continuously, the condenser surface areas are fully utilised at all times and the condensing pressure is a function of plant load and ambient conditions. This control concept requires that the plant is designed in such a way that reliable refrigerant expansion is possible also at exceptionally low condensing pressures. The system ensures lowest possible compression ratios at all times. Only in situations where poor refrigerant feed to the intercooler is detected will the condenser fans be cycled to artificially elevate the condensing pressure and hence the liquid feed pressure.

3.2.3 Engine Room Layout

The engine room layout is shown in fig. 3 below. The layout comprises three levels. On the low level, the compressors and the ammonia pumps are located. On the second level the accumulators and the receiver are located. The condenser, which is not visible on the photo, is situated on the top level. Pipe lines and valves directly above the compressors have been avoided to ensure monorail crane access.

FIGURE 3. Engine room layout.

The layout offers several advantages which may be summarised as follows:

- Very safe and accessible compressor area where the amount of equipment and obstructions on the floor are minimised ensuring good escape routes for personnel in emergencies.

- The static head available for the refrigerant pumps is >4 m ensuring reliable operation at all times with very limited risk of cavitation. The cavitation risk is further minimised by the employment of modulating, electronic refrigerant expansion devices at both temperature levels.

- All manually operated isolation valves for the isolation/cross connection of compressors and vessels are accessible without ladders from the mezzanine floor supporting the accumulators/receiver.

- Good static head for the refrigerant cooled oil coolers ensuring reliable refrigerant feed and hence sufficient oil cooling at all times.

- Provision for liberally sized vertical liquid columns between the condenser liquid outlet and the liquid trap upstream of the receiver. This ensures complete equalisation of any differences in refrigerant pressure drops between condensers operating in parallel and hence full utilisation of all condenser surface areas at all times.

Other features of the engine room include:

- Central oil charging system for the compressors complete with central oil charging pump and permanent oil charging lines.

- Two refrigerant charging points (high pressure and low pressure side) connected permanently to a charging point located outside the engine room at ground level adjacent to the emergency shower.

- Automatic emergency engine room ventilation system with discharge through a water deluge eliminating ammonia vapour prior to discharge to atmosphere.

- All high pressure ammonia pipe lines exposed to the weather are hot dipped galvanised.

- Full size compressor plinths with peripheral kerbs and individual drain points preventing any condensate drip and oil leakage on the engine room floor.

- Chilled glycol for office air conditioning provided via a 25 kW welded cassette plate heat exchanger connected to the second compression stage.

3.2.4 Automatic Monitoring and Alarm System

The plant supervision and control comprises a RS View 32 SCADA system operating with a 400 MHz Pentium Computer with a 15” digital monitor located in the control room adjacent to the compressor room. The SCADA system is interfaced to the Allen Bradley PLC system controlling the plant.

The PLC interfaces to each Compressor Control Unit via a local area communication network. This enables all instrumentation, controls, set points and process variables on the compressor to be controlled and monitored at the SCADA interface. Each compressor is provided with an independent control system. It is possible to operate the plant in local, hard wired mode should the PLC/SCADA system malfunction. Remote monitoring and operation via the telephone line is also possible.

3.2.5 Under Floor Heating

Under floor heating is carried out by means of forced ventilation of air through a grid of PVC pipes located under the cold store. Air at ambient conditions is by means of a central, dual speed axial fan drawn through the PVC pipe grid into a central duct under the cold store and discharged into the engine room where it provides cooling and ventilation.

The subsoil temperatures below the facility are monitored and the temperatures are used as a control signal for the fan speed. At the design subsoil temperature, the supply air temperature to the engine room is 15°C.

3.2.6 Automatic Defrost

Automatic defrost is carried out by means of hot gas. Each penthouse air cooler is provided with one valve station, the two air coolers located in the ante room are serviced by one common valve station. During defrost, hot gas enters the penthouse coils at the top header and flows towards the bottom header opposite the normal direction of refrigerant flow. This ensures fast removal of subcooled refrigerant liquid from the coil and hence rapid defrost. Experience has shown that a defrost can be completed with a hot gas “on” time of only 15 minutes.

For improved reliability and minimum attendance, the penthouse unit drain trays are of a double pan design. During defrost, the pan heating takes place using hot refrigerant gas flowing through a grid of galvanised steel tubes located in the cavity between the inner and the outer pan in direct contact with the surface of the inner pan. The wet part of the pan exposed to the condensate slopes in two directions towards the drain point of the pan and contains no obstructions inhibiting the free flow of water. Following several months of automatic operation there is no evidence of gradual ice accumulation anywhere in the penthouse evaporator units.

3.2.7 Pipe Line Layout for Minimum Pressure Drop

The wet return lines for the low temperature and the high temperature segments were designed for maximum 2K and 1K temperature drops respectively [2]. These pressure drops take into account the accumulator stop valve pressure drops and were calculated from the most remote air cooler to the engine room when the plant is operating at full load and comprises all three development stages.

3.2.8 Condenser Selection and Installation

To minimise condensing pressures and hence power consumption, evaporative condensers were used. To minimise maintenance, units with fibreglass reinforced polyester casings were selected. To ensure maximum reliability, the condensers were provided with standby spray water pumps fully piped and ready to operate.

In the interest of minimum energy consumption, the evaporative condensers were oversized by a factor of 1.25. In the final development stage, the plant will comprise two evaporative condensers. The capacity of each individual condenser is adequate to service the entire plant in the event of emergencies.

4.0 PERFORMANCES

4.1 Temperature Pull-Down

The temperature as a function of time for the initial temperature pull-down period following completion of the facility is shown in fig. 4.

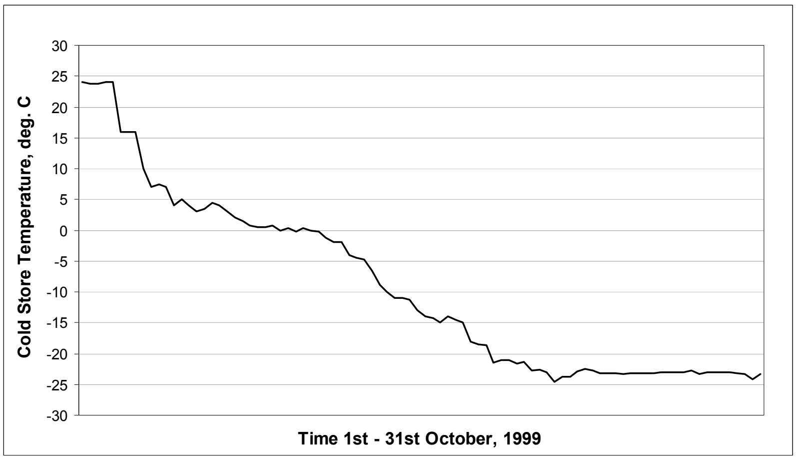


FIGURE 4. Temperature pull-down as a function of time following completion of the facility.

4.2 Normal Operation

The key performance data (store temperatures and total power consumption), for the plant during the third month of commercial operation (December 1999) are shown in fig. 5.

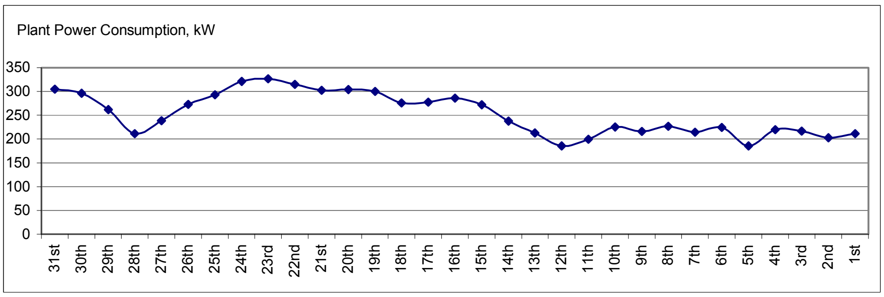
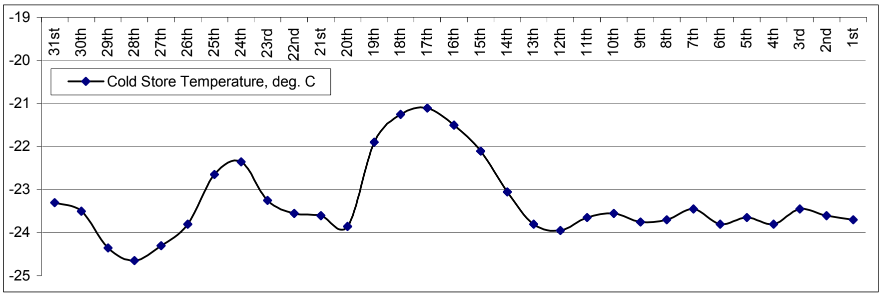


FIGURE 5. Key performance data for the third month of commercial operation (December, 1999).

The month of December 1999 is not entirely representative of normal plant operation. At approximately mid December 1999, the operator commenced loading 2750 tonnes of product entering at -4°C. This explains the temperature and the power increases commencing around 12th of December.

4.3 Energy Consumption



The annual energy consumption of an existing cold store and the new facility are compared in fig. 6 below. Both stores are owned and operated by the same cold store operator. The existing cold store has a ceiling height of between 9.5 and 10 m and the mix between peak low temperature and peak high temperature loads is approximately 0.61 MW and 0.92 MW respectively. The high temperature load is hence relatively greater for the existing cold store than for the new facility. The comparison in fig. 6 is therefore biased somewhat in favour of the existing facility.

The annual specific energy consumption in kWh/m³ of storage volume for the new facility is extrapolated on the basis of the first three months of commercial operation (4th quarter, 1999). During these three months of operation, the energy management features were not fully implemented. In addition, extensive compressor part load operation was experienced because the machines were selected to suit the second development stage. Further reductions in the annual energy consumption of the new facility may therefore be anticipated following implementation of the two additional development stages as well as the energy management features.

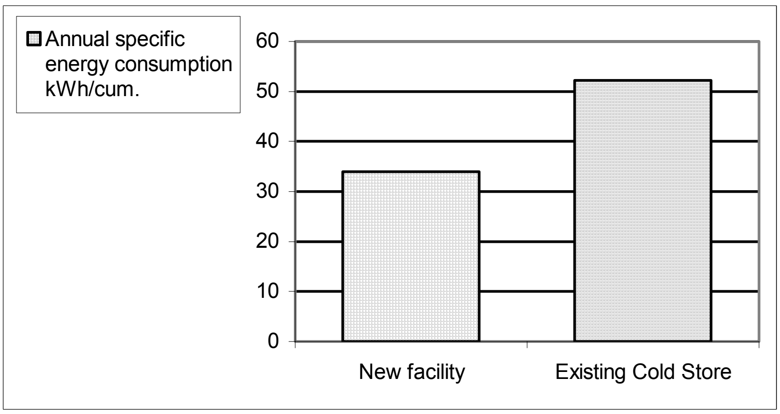


FIGURE 6. Annual specific energy consumption in kWh/m³ of storage volume including annexes for an existing cold store and the new cold store.

The comparison in fig. 6 indicates that the annual specific energy consumption in kWh/m^3 of the new facility is 35% less than for a comparable existing facility operated by the same owner. Part of the energy saving achieved in the new facility is explained by the difference in ceiling heights between the two stores compared. For 14 m versus 10 m ceiling height, the difference in volume/envelope ratio is 21%. With conduction contributing approximately 40% of the design load of the new facility, the ceiling height difference accounts for 8% to 9% of the energy saving achieved.

4.4 Maintenance

During the first three months of commercial operation, the refrigeration plant has operated completely automatically apart from two plant stops. One stop was caused by a power supply failure and required PLC modifications. The other stop was due to a faulty circuit breaker on a second stage compressor.

At the time of writing, the statistical records in terms of plant maintenance are inadequate to conclude whether or not the annual maintenance costs are within the limits projected. The annual labour costs for maintenance currently represent approximately 1.3% of the initial capital costs of the plant. The annual parts and material consumption is unknown at this stage. However, experience from similar installations indicate that material and labour consumption usually are in a ratio of approximately 1 to 1. Annual maintenance costs are therefore on target to be within the projection of <3% of the initial capital costs of the system.

In industrial cold storage applications, maintenance and service calls outside normal scheduled maintenance are often caused by inadequate or faulty automatic defrosts. During the first three months of commercial operation there has been two incidents of gradual accumulation of ice or frost in the finned air coolers located in the dock area. This problem was eliminated by extending the defrost times.

5.0 CONCLUSION

Following the first three months of commercial operation, the new cold storage facility is expected to exceed the design objectives set in terms of energy consumption by a significant margin. The objective was to reduce the annual energy consumption per unit volume of cold storage by minimum 20% compared with a similar cold store operated by the same owner in the Brisbane area. By extrapolation there is a strong indication that the reduction will be >35%. Although 8% to 9% of this saving may be attributed to the ceiling height difference between the facilities compared, the result remains in excess of expectations.

Due to lack of statistical data, it cannot be concluded that the maintenance costs of the new facility will be significantly less than for other equivalent facilities operated by the same owner. However, indications after the first three months of commercial operation are that the annual maintenance costs will be less than the projection of 3% p.a. of initial capital cost of the system made at the commencement of construction.

ACKNOWLEDGEMENTS

The valuable assistance of Mr. Myles Forbes, Chief Engineer of Frigmobile Pty. Ltd. Queensland during the preparation of this paper is acknowledged with thanks.

REFERENCES

1. Wildridge & Sinclair Pty. Ltd.: *Design and Reference Data Manual*. “HE” Type Coil Performance Factors. Section 4.02-F, Page 1 of 2 dated 80.09.19.

2. Aflekt, Kåre. “Trykktap i rør og dimensjonering av rørsystem i kuldeanlegg”. Institutt for Kjøleteknikk, NTH, Trondheim, Norway. (Pressure drops in pipes and sizing of pipes for refrigeration systems).

3. ASHRAE: *ASHRAE Handbook - 1994. Refrigeration, Systems and Applications.* American Society of Heating, Refrigeration and Air-conditioning Engineers, Inc. Chapter 41.

BIBLIOGRAPHY

Jensen, S.S. “Extended Surface Steel Air Coolers for Industrial Refrigeration”. *Journal of the Australian Institute of Refrigeration Air Conditioning and Heating,* August 1989.

Glent & Co. A/S. *Glent Ventilation.* Glent & Co. A/S, Kanalholmen 1, DK-2650 Hvidovre, Denmark, 1977.