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# Energy Performance of Low Charge, Central Type, Dual Stage $NH_3$ Refrigeration Systems in Practice

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## Abstract

*Development of low charge $NH_3$ refrigeration systems is taking place throughout the world for various applications, predominantly for refrigerated storage and packaged solutions to chill liquids and/or condition refrigerated spaces. These developments are initiated by a global phase-down of high global warming (GWP) refrigerants of the hydrofluorocarbon (HFC) type. This phase-down is a direct result of the "top-down" emissions reduction agreement resulting from the Conference of the Parties (COP 21) in Paris, December 12, 2015.*

*Ammonia refrigerant offers improved vapor compression cycle efficiencies in comparison with most other refrigerants. The energy performance improvements associated with the application of ammonia refrigerant in combination with other energy efficiency engineering techniques such as state-of-the-art superheat/quality (SH/X) expansion control, advanced low charge evaporator technology, extensive integration of variable frequency drives, two-stage compression, low oil carry-over compressors, low friction loss pipe lines, and genuine central plant concept have not been the subject of widespread investigations and reporting.*

*Where compliance costs are directly proportional with $NH_3$ inventory, favoring multiplexing as opposed to central plants is tempting for stakeholders. This, of course, risks sacrificing energy performance in return for reduction in $NH_3$ inventory. For signatories to the COP 21 agreement, this is neither in the national interest nor is it in the commercial interest of plant owners if compliance cost increases outweigh energy cost reductions.*

*This paper describes the energy performances of several refrigerated distribution centers with storage volumes of approximately 10,000 to 50,000 m³ (353,000 to 1,766,000 ft³). The performance evaluations are based on the electrical energy consumption as measured by the electrical energy providers over representative periods of time. All systems are serviced by central, state-of-the-art low charge, dual stage $NH_3$ refrigeration systems. In the case of one plant the contribution of the photovoltaic panels to the energy requirement of the facility as a whole is shown on a month-by-month basis.*

*An energy performance comparison is also made between two refrigerated distribution centers with a volume of approximately 10,000 m³ (353,000 ft³), but serviced by two different types of ammonia refrigeration systems. In one case the plant is a single-stage economized dual screw compressor based system with gravity flooded refrigerant feed. In the other case, the plant is a central, low charge $NH_3$ dual-stage system with speed controlled semi industrial reciprocating compressors. Other features of the two facilities include general warehouse designs that are more or less identical. The energy performance comparisons are again based on the electrical energy consumption as recorded by the electrical energy provider over one calendar year.*

## Introduction

In the wake of the pending global phase-down of HFCs due to their contribution to global warming, users and owners of refrigeration systems are faced with decisions that at times appear difficult. These decisions relate to whether or not users continue to employ HFC-based refrigeration systems, switch to low GWP synthetic refrigerants or consider future-proof natural refrigerants such as $NH_3$, $CO_2$, hydrocarbons, water, or air in their new and/or expanded systems. In this decision-making process, one very important factor is often either overlooked or underestimated. This factor is energy performance—particularly the energy performance of low charge $NH_3$ systems.

Discussing the energy performance of systems that proponents of refrigeration systems that use synthetic refrigerants market is not in their commercial interest so they rarely do. Their marketing focus is often attractive capital costs, refrigerant "safety," availability of service/maintenance resources, and simplicity. Synthetic refrigerant proponents often seek to marginalize the pending HFC phase-down by referring to several factors: the anticipated relatively long time frame of the HFC phase-down; the future availability of alternative synthetic low GWP refrigerants; the capital cost penalties associated with a switch to natural refrigerants; and the allegedly expensive, frequent, and specialized service/maintenance requirements associated with refrigerants such as ammonia or $NH_3$.

Promoters of natural refrigerant-based systems, however, tend to undersell the excellent energy performances of natural refrigerant-based refrigeration systems—particularly low charge $NH_3$ systems. This is understandable because low charge $NH_3$ refrigeration systems are not yet as common as liquid overfeed or gravity-flooded systems and documented annual energy performances for low charge $NH_3$ plants—particularly the modern versions—are relatively scarce.

  

The decision between HFCs and low GWP synthetics or natural refrigerants is often made difficult by the quality, the independency (or lack thereof) of, and the sources of the decision-making material presented to users. Claims of improvements in energy performances of 40–70% associated with low charge $NH_3$ refrigeration plants as compared with industry standard HFC-based systems are often dismissed as exaggerated, biased, and therefore irrelevant. The confusion on the part of end users when faced with conflicting technical information is understandable and decisions in favor of low-capital cost solutions is perhaps not surprising.

As this paper will show, the claims of 40–70% improvement in energy performance are not exaggerated. In fact low charge $NH_3$ systems can, if designed correctly, present an attractive business case in favor of straight replacement of existing outdated HFC-based systems with new, modern low charge $NH_3$ plants. Modern low charge $NH_3$ refrigeration plants can also provide significant energy savings compared with conventional liquid overfeed $NH_3$ systems with screw compressors.

## The Refrigeration Plants

The following sections summarize the refrigeration plants that are the main subjects of this paper.

### *Perth*

This 43,000 m³ (1,519,000 ft³) refrigerated distribution facility is situated in Perth, Western Australia. The facility comprises a 16°C (61°F) room, a 4°C (39°F) cool room, a -25°C (-13°F) freezer, and a 4°C (39°F) annex. The refrigeration plant is a dual-stage low charge $NH_3$ system with four identical speed-controlled reciprocating compressors, evaporative condenser, internally surface-enhanced evaporators suitable for dry expansion refrigerant feed, and refrigerant injection controlled by superheat.

All interconnecting refrigerant pipe lines are carbon steel. A plan layout of the facility is shown in Figure 1.

*Tamworth*

This 10,000 m³ (353,000 ft³) refrigerated distribution facility is located in Tamworth, New South Wales. The facility comprises a 4°C (39°F) cool room, a -25°C (-13°F) freezer, and a 4°C (39°F) annex. The refrigeration plant is a dual-stage low charge $NH_3$ system with four speed-controlled semi-industrial reciprocating compressors, evaporative condenser, internally surface-enhanced evaporators suitable for dry expansion refrigerant feed, and refrigerant injection controlled by superheat. Figure 2 shows the floor plan.

*Lismore*

This 10,000 m³ (353,000 ft³) refrigerated distribution facility is located in Lismore, New South Wales. The facility comprises a 4°C (39°F) cool room, a -25°C (-13°F) freezer, and a 4°C (39°F) annex. The refrigeration plant is a single-stage $NH_3$ system with two fixed-speed industrial screw compressors with common economizer and evaporative condenser. The medium temperature evaporators are arranged for dry expansion refrigerant feed; the freezer is fitted with evaporators arranged for gravity-flooded feed and hot gas defrost. The plan layout is similar to that shown in Figure 2.

*Melbourne*

This 43,000 m³ (1,519,000 ft³) refrigerated distribution facility is situated in Melbourne, Victoria. The facility comprises a 4°C (39°F) cool room, a -25°C (-13°F) freezer, and a 4°C (39°F) annex. The refrigeration plant is a dual stage, low charge, central $NH_3$ system with four identical speed-controlled reciprocating compressors, oversized evaporative condenser, internally surface-enhanced evaporators with longer circuits and suitable for dry expansion refrigerant feed, and refrigerant injection controlled by a combination of superheat and quality signal. All interconnecting refrigerant pipe lines are 304 stainless steel. The system was from the outset fitted with a desiccant drier in the freezer. This decision was based on the good result obtained in Perth when a desiccant drier was retrofitted at that facility.

**Townsville**

This 31,000 m³ (1,095,000 ft³) refrigerated distribution facility is situated in Townsville, Queensland. The facility comprises a -25°C (-13°F) freezer, a 4°C (39°F) cool room, a 16°C (61°F) flour room, and a 4°C (39°F) annex. The refrigeration plant is a dual-stage low charge $NH_3$ system with four identical speed-controlled reciprocating compressors, oversized evaporative condenser, internally surface-enhanced evaporators with longer circuits and suitable for dry expansion refrigerant feed, and refrigerant injection controlled by a combination of superheat and quality signal. All interconnecting refrigerant pipe lines are 304 stainless steel. The system is fitted with a desiccant drier in the freezer.

## Recorded Energy Consumption Details

Table 1 shows the measured annual energy consumption details for the five facilities. The value specific energy consumption (SEC) is derived by dividing the annual energy consumption of the refrigerated facility measured in kWh per year (kWh/yr) by the total refrigerated volume measured in m³ (ft³). The unit for SEC is hence kWh/m³*yr (kWh/ft³*yr).

| **Plant** | **Total Annual Energy Consumption (kWh)** | **Record Period** | **Refrigerated Volume ($m^3/ft^3$)** | **Specific Energy Consumption (SEC) ($kWh/m^3*yr/kWh/ft^3*yr$)** |
|---|---|---|---|---|
| Perth | (700,072 + 219,440)/ 9*12 = 1,226,016 | 1.7.14 to 31.3.15 | 43,289/ 1,528,737 | 28.3/0.801 |
| Tamworth | 409,597 | 1.7.14 to 30.6.15 | 9,474/ 334,571 | 43.2/1.22 |
| Lismore | 1,135,027 | 1.7.14 to 30.6.15 | 10,748/ 379,562 | 105.6/2.99 |
| Melbourne | 1,098,390 | 1.1.16 to 30.6.16 | 42,619/ 1,505,076 | 25.8/0.731 |
| Townsville | 406,781/7*12 = 697,339 | 1.2.2016 to 31.8.2016 | 31,344/ 1,106,903 | 22.2/0.630 |

**Table 1. Recorded Energy Consumption**

In each case except Townsville, these figures are for the entire facility and typically include other services such as information technology (IT), general light and power, services for refrigerated trucks, forklift charging, and office air conditioning. Detailed records of the energy consumption of the $NH_3$ systems in isolation only exist for the Townsville plant. In this facility the supervisory control and data acquisition (SCADA) system is fitted with hardware and software to facilitate the separate measurement of the $NH_3$ plant energy consumption. In part this is to evaluate the feasibility and economic viability of providing office air conditioning services via the central low charge $NH_3$ system as opposed to fitting individual, HFC-based, air-cooled split air conditioning systems.

Some evidence in Table 1 shows that the energy consumption of auxiliary services such as office air conditioning, IT, general light and power, etc. is sufficiently significant to warrant separate recording. The Townsville facility is located in a subtropical area, has 26% less refrigerated volume than the Melbourne facility, yet features 14% lower SEC. Both facilities are fitted with desiccant driers. Based on the difference in refrigerated volume between Melbourne and Townsville, the SEC for Townsville should have been 27.7 kWh/m³*yr (0.78 kWh/ft³*yr) using the SEC for Melbourne as the base for extrapolation and possibly even a little higher due to the approximately 3°K (5.4°F) higher wet bulb temperature in Townsville. The difference between 22.2 and 27.7 kWh/m³*yr (~20%) may therefore be taken as being representative of the energy consumption of the auxiliary services.

On more recent installations that are not included in this paper, part of the regeneration heat for the desiccant drier is being recovered from the $NH_3$ plant via a desuperheater, which then also provides heat for the subfloor heating below the freezer via a water/ethylene glycol heat exchanger. Figure 3 shows an example of such an installation. This recent installation also employs a secondary refrigerant for the medium temperature segment (food processing area) and horizontal accumulators. The latter two features reduce the specific refrigerant charge to approximately 0.65 kg/kW (5 lbs/TR).

In the case of Perth, the 700 MWh represents the electrical energy supplied from the grid and the 219 MWh is the energy supplied from the photovoltaic (PV) panels. The sum of 919 MWh represents the energy consumption of the entire facility for the nine-month period shown. The total amount of electrical energy supplied from the grid to the Perth facility is 915.6 MWh for the period July 1, 2014, to June 30, 2015. The contribution of the PV panels is only known for this period. Therefore Table 1 only shows the nine-month period where supplies from the grid and the PV panels overlap. Figure 4 shows the monthly contribution from the PV panels. The annual electrical energy consumption of 1,226 MWh for the Perth facility is simply estimated by extrapolation as shown in Table 1. If an assumed 20% of the annual electrical energy consumption is allocated to services other than the $NH_3$ plant, the annual specific energy consumption (SEC) of the $NH_3$ system becomes 22.7 kWh/m³*yr (0.643 kWh/ft³*yr).

At the conclusion of the energy consumption recording period for Lismore, the attention of the plant owner was drawn to the fact that the set point for the condensing pressure control was higher than necessary. Following reduction of the condensing pressure set point to enable floating condensing pressure, the monthly average electricity account fell from approximately $22,000 to approximately $12,000 according to the plant owner. (These values are in Australian dollars; the conversion is approximately A$1.0 = US$0.7). The electrical energy consumption recording period after the condensing pressure set point adjustment was too short to establish the exact impact on SEC. For the Tamworth facility the average monthly electricity account ranges from A$6,000 to A$8,000. The condensing pressure set point adjustment at Lismore is estimated to have reduced SEC by 20% to 40% to around 65–85 kWh/m³*yr (1.84–2.41 kWh/ft³*yr). Figure 5 compares the electrical energy consumption for the Tamworth and Lismore facilities.

The Perth and Tamworth energy performance results are excellent for facilities of these sizes. Other facilities of similar volume and function consume around double based on the correlation shown in Figure 6, which originates from a study carried out by the California Energy Commission in 2008 covering 67 public and 96 private refrigerated warehouses. The graph shows specific energy consumption as a function of warehouse volume.

The significant difference in energy consumption between Tamworth and Lismore is most likely mainly attributable to the selection of the type of compressor, the plant configuration, and the fixed-speed compressor drives. Table 2 details modeled annual energy consumption values of various compressor configurations and two different load patterns (Lorentzen 1981). The advantage of reciprocating compressors compared with screw compressors in terms of energy performance is clear.

| Compressor combination → | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|
| Screw compressor capacity (kW)<br><br>Reciprocating compressor capacity (kW) | 1x(500)<br><br>- | 1x(500)<br><br>- | 1x(452)<br><br>1x(48) | 1x(452)<br><br>1x(48) | 1x(500) | 1x(452)<br><br>1x(48) | -<br><br>3x(151)<br><br>1x(47) |
| Energy consumption, pattern I, MWh/yr | 1,845 | 1,812 | 740 | 725 | 898 | 665 | 675 |
| Energy consumption, pattern II, MWh/yr | 1,890 | 1,825 | 1,440 | 1,370 | 1,250 | 1,150 | 1,075 |

**Table 2. Annual energy consumption for various compressor configurations and load patterns**

The various compressor configurations and load patterns are as follows.

Compressor combinations:

- 1: Single-stage screw compressor,
- 2: Single-stage screw compressor with economizer,
- 3: Single-stage screw and dual-stage reciprocating compressor,
- 4: Single-stage screw compressor with economizer and dual-stage reciprocating compressor,
- 5: Dual-stage screw compressor,
- 6: Dual-stage screw and dual-stage reciprocating compressor, and
- 7: Dual-stage reciprocating compressors.

Load patterns:

- I: Combination of plate freezers and freezer stores, load variation 10–100%; and
- II: Combination of blast freezers and freezer stores, load variation 40–100%.

The maximum refrigeration capacity at -40°C evaporating temperature is 500 kW in all cases.

  

The modeling results in Table 2 do not reflect the presence of medium-temperature refrigeration loads and the use of variable-frequency compressor drives. As such they do not fully explain the energy consumption differences between Tamworth and Lismore. However, the results in Table 2 do illustrate the importance of compressor part load efficiency with respect to the delivery of superior system energy performance. The comparison of typical compressor part load efficiencies in Table 3 further illustrates the importance of considering this element during system design (Grasso Comsel Compressor Software version v3.20.02). All values are coefficients of performance (COP) calculated as refrigeration capacity divided by compressor shaft power. The operating condition is -10°C saturated evaporating temperature, 35°C saturated condensing temperature, 0°C superheat, 0°C subcooling, refrigerant $NH_3$. The reciprocating compressor is a Grasso V600 with a refrigeration capacity at 100% (1,500 rpm) of 315.7 kW, corresponding shaft power consumption 83.1 kW. The screw compressor is a Grasso HR2655S without economizer with a refrigeration capacity at 100% (2,940 rpm) of 294.5 kW, corresponding shaft power 82.9 kW.

| Load (%) ↓ | Fixed speed | | Variable speed | |
|---|---|---|---|---|
| | Reciprocating | Screw | Reciprocating | Screw |
| 100 | 3.80 | 3.55 | 3.80 | 3.55 |
| 87 | 3.74 | 3.42 | 3.83 | 3.47 |
| 75 | 3.66 | 3.31 | 3.84 | 3.36 |
| 62 | 3.55 | 3.12 | 3.84 | 3.18 |
| 50 | 3.40 | 2.82 | 3.82 | 2.98 |
| 37 | 3.16 | 2.36 | 3.80 | 2.70 |
| 25 | - | 1.79 | 3.64 | 2.44 |

**Table 3. Comparison between part load efficiencies (COPs) for screw and reciprocating compressors**

  

## Low Charge $NH_3$ versus Industry Standard HFC

The owner of the Perth facility operates a second distribution facility in the same suburb around two km (1.2 mi) from the warehouse serviced by the low charge $NH_3$ plant.

The second facility is referred to as the Cocos Dr. warehouse. The Cocos Dr. warehouse is serviced by industry standard, individual air-cooled HFC-based, single-stage condensing units with electric defrost in the low-temperature areas. Tables 4 and 5 show the design refrigeration loads for the Perth and Cocos Dr. distribution centers. Clearly the sum of the design refrigeration loads for the two distribution centers are similar.

| Estimated heat loads kW (TR) | LT | HT |
|---|---|---|
| Refrigerant temperature, oC (°F) | -31 (-23.8) | -3 (26.6) |
| Flour room, 16oC (60.8°F), LxWxH = 40.5x5.9x10.0 m (133x19.4x32.8 ft) | n.a. | 11.1 (3.16) |
| Chiller, 4oC (39.2°F), LxWxH = 40.5x22.5x10.0 m (133x73.8x32.8 ft) | n.a. | 51.1 (14.6) |
| Freezer, -25oC (-13°F), LxWxH = 55.5x40.5x10.0 m (182x133x32.8 ft) | 173.8 (49.5) | n.a. |
| Annex, 4.0oC (39.2°F), LxWxH = 71.6x13.0x10.0 m (235x42.7x32.8 ft) | n.a. | 166.2 (47.4) |
| Total ~43,000 m³ (1,518,500 ft³) | 173.8 (49.5) | 228.4 (65.1) |

Table 4. Estimated design heat loads for Perth warehouse serviced by low charge $NH_3$ system

| Estimated heat loads kW (TR) | LT | HT |
|---|---|---|
| Refrigerant temperature, oC (°F) | -31 (-23.8) | -3 (26.6) |
| Freezer 1, -25oC (-13°F), LxWxH = 35.5x24.0x9.0 m (116x78.7x29.5 ft) | 88.8 (25.3) | n.a. |
| Freezer 2, -25oC (-13°F), LxWxH = 30.0x29.5x9.0 m (98.4x96.8x29.5 ft) | 94.8 (27.0) | n.a. |
| Chiller 1, 4oC (39.2°F), LxWxH = 35.5x7.5x9.0 m (116x24.6x29.5 ft) | n.a. | 46.1 (13.1) |
| Chiller corridor, 4oC (39.2°F) | n.a. | 6.0 (1.7) |
| Chiller 2, 4oC (39.2°F), LxWxH = 14.5x8.3x9.0 m (47.6x27.2x29.5 ft) | n.a. | 17.7 (5.0) |
| Dock, 4°C (39.2°F), LxWxH = 20.0x19.0x4.5 m (65.6x62.3x14.8 ft) | n.a. | 41.0 (11.7) |
| Annex, 4.0oC (39.2°F), LxWxH = 37.5x6.0x4.5 m (123x19.7x14.8 ft) | n.a. | 82.3 (23.4) |
| Total ~ 22,000 m³ (776,923 ft³) | 183.6 (52.3) | 193.1 (55.0) |

**Table 5. Estimated design heat loads for Cocos Dr. warehouse serviced by HFC systems**

The financial records of the operator of the Cocos Dr. warehouse indicate monthly electrical energy supply costs of around A$42,000 on average. The electricity account for the Perth warehouse for the period April 1–30, 2015, was A$13,751.57, including 10% Goods and Services Tax (GST). This was for a total supply of 81,264 kWh. Based on Figure 4, this level of monthly electrical energy consumption is not unusual. Given that this is the same electrical energy provider for both the Perth and the Cocos Dr. warehouses, it may be concluded that the energy performance improvement of the Perth warehouse serviced by a low charge $NH_3$ system could represent a reduction of approximately (1-13,752/42,000) *100, or 67% compared with Cocos Dr. This significant difference in energy consumption between HFC and $NH_3$ may appear extraordinary, but it is not when comparisons are made between other facilities operated by the same owner. A 1,385 m² (14,908 ft²) facility situated at Kunda Park in Southeast Queensland, Australia, and serviced by HFC-based air-cooled systems with electric defrost consumes around 1,265 MWh annually. An 1,130 m² (12,163 ft²) facility serviced by a dual-stage, liquid overfeed $NH_3$ system situated at Somersby north of Sydney in New South Wales consumes 546 MWh annually (Jensen 2013).

## Factors Affecting Low Charge $NH_3$ System Energy Performance

A low charge $NH_3$ refrigeration plant does not necessarily feature superior energy performance compared with other $NH_3$ based systems. As the comparison between Tamworth and Lismore shows, using $NH_3$ as the refrigerant is no guarantee of above-average energy performance either. Several factors individually contribute to the improvement of energy efficiency. Table 6 summarizes the author's order of significance for nine factors. The percentage improvements shown cannot be interpreted as cumulative. Each factor is to be considered as one individual change all other things being equal.

| Item | Energy Conservation Factor | Percentage Impact (%) |
|---|---|---|
| 1 | Selection of compressor type | 15–25 |
| 2 | Evaporator fan speed control | 15–25 |
| 3 | Evaporator design | 5–25 |
| 4 | Compressor capacity control | 10–20 |
| 5 | Quality of match between compressor turn-down ratios and heat load variations | 0–15 |
| 6 | Condenser size, condenser fan speed control, and condenser efficiency | 5–10 |
| 7 | Liquid injection control into the evaporators | 5–10 |
| 8 | Elimination of liquid within suction lines | 2–15 |
| 9 | Use of low-friction-loss 304SS schedule 10 refrigerant pipe lines in lieu of carbon steel | 1–2 |

Table 6. Factors impacting upon energy performance of low charge $NH_3$ systems

*Item 1. Compressor type*

A refrigerated distribution facility comprising 46,000 m³ (1,624,475 ft³) frozen storage plus a 7,000 m³ (247,203 ft³) annex (Jensen 2000) recorded a specific energy consumption of 35 kWh/m³*yr (0.991 kWh/ft³*yr). This facility was serviced by a dual-stage, liquid overfeed system comprising three identical fixed-speed drive screw compressors, one booster, one second-stage compressor, and one dual duty standby machine. The penthouse evaporators were fitted with variable frequency drive fans. A 23,000 m³ (812,237 ft³) refrigerated storage facility in the same geographic location with a slightly different mix between low- and medium-temperature services recorded a specific energy consumption of 27 kWh/m³*yr (0.765 kWh/ft³*yr) (Jensen 2013). The latter facility was serviced by a dual-stage, liquid overfeed system with four fixed-speed drive reciprocating compressors. The percentage shown in Table 6, item 1, refers to the comparison between these two practical systems, but a similar energy performance improvement estimate may be derived from Table 2.

*Item 2. Evaporator fan speed control*

A refrigerated distribution facility comprising 46,000 m³ (1,624,475 ft³) frozen storage plus a 7,000 m³ (247,203 ft³) annex (Jensen 2000) with variable-speed penthouse fans recorded a 35% lower specific energy consumption than a similar neighboring facility with the same owner and fixed-speed fans fitted to the penthouse evaporators. Around 8–9% of the 35% energy performance difference was attributable to warehouse design. This forms the basis for the percentage in item 2, Table 6.

*Item 3. Evaporator design*

Many practical examples of dry expansion feed air coolers for $NH_3$ failing to meet performance expectations exist (Jensen 2006 and Jensen 2011). There are several reasons for this. The most important are summarized below:

- Incorrect evaporator circuiting causing inadequate turbulence and stratified flow;
- Nonuniform refrigerant distribution within the air cooler;
- Presence of water in the refrigerant causing a refrigerant bubble point rise toward the conclusion of the evaporation process, which in turn provides a false superheat control signal;
- Air cooler core tube material with inadequate thermal conductivity again causing lack of turbulence and stratified flow;
- Mismatch between the operating envelope provided by the air cooler manufacturer and the operating envelope required by the system;
- Oil fouling on the internal tube surfaces of the air coolers;
- Inadequate condensate removal during hot gas defrost due to inappropriate condensate drainage provisions;
- Inappropriate selection of expansion valve for the application; and
- Suboptimal control methodology applied to the refrigerant injection and the control of the hot gas defrost procedure.

New air cooler technologies are available that address the problem of inadequate exposure of the internal tube surfaces to the boiling refrigerant. These are based on internal tube surface enhancement that causes a capillary effect or the insertion of turbulators. New liquid distribution technologies have also been made available to enlarge the operating envelope (Nelson 2013; Jensen 2015a and Jensen 2015b). The main issue for the refrigeration plant designer to understand here is that reliance on air cooler suppliers to provide heat exchangers that deliver the specified thermal performances will not necessarily guarantee a successful outcome. The system designer must look at all heat exchanger designs critically with a view to addressing all of the previously summarized issues.

*Item 4. Compressor capacity control*

Table 6 refers to the retrofitting of variable frequency drives to an existing refrigerated warehouse in Sydney, Australia. This measure reduced annual energy consumption by >15% (New South Wales Office of Environment and Heritage 2012). The plant is a dual-stage liquid overfeed system with screw compressors servicing a mixture of low-temperature, medium-temperature, and blast-freezing rooms, with a total area around 30,000 m² (323,000 ft²).

*Item 5. Compressor turn-down ratio and heat load variation*

Extensive part load operation of compressors is a common problem in many industrial refrigeration systems. The percentage impact referenced in item 5, Table 6, is a function of the severity of the problem; Table 2 describes the magnitude of the potential efficiency loss.

*Item 6. Condenser size, control, and efficiency*

Evaporative condensers may be designed and selected such that the energy consumed by the condenser (the sum of fan and pump energy) is less than 1% of the design heat rejection, but ratios of 2–3% are no rarity in practice. Furthermore, oversizing the condenser such that it reduces the saturated condensing temperature by 1°K (1.8°F) improves the coefficient of performance of a typical second stage compressor by 2.6%. The percentage improvement in item 6, Table 6, is readily within reach with this simple measure.

*Item 7. Liquid injection control into the evaporators*

Quality-based control of the liquid injection into the evaporators is superior to conventional superheat-based control (Jensen 2015b). Practice has shown that entering temperature differences (ETD) between air and refrigerant of around 2.5°K (4.5°F) are possible without excessive control instability. In this context minimizing the possibility of liquid hold-up in the evaporator is important. The percentage range nominated in item 7, Table 6, is derived by estimating the impact on energy efficiency of raised plant suction pressure that reduced ETD gives rise to.

*Item 8. Elimination of liquid in suction lines*

As yet no experimental basis exists for the claimed 2–15% impact. This value range is a result of many practical observations of the energy performances of conventional liquid overfeed systems versus central type, low charge, dry expansion $NH_3$ plants of a design as described herein.

*Item 9. Low friction piping*

The claimed value range of 1–2% is based on actual line pressure drop measurements at the Melbourne plant. The measurements were based on the SCADA system and the pressure transmitters fitted at the evaporators (for superheat-based injection control) and the pressure transmitter fitted in the central engine room for provision of the compressor capacity control signal. The pressure drops measured were minimal and not particularly accurate due to the accuracy of the instrumentation. This is reflected in the value range in Table 6.

## Conclusion

Ammonia refrigeration systems with reduced refrigerant inventory (low charge $NH_3$ systems) have been presented as potentially highly attractive alternatives to both industry standard HFC-based systems and conventional liquid overfeed and/or gravity-flooded $NH_3$ systems. Appropriately designed low charge $NH_3$ systems demonstrate measured specific energy consumption values in kWh/m³*yr (kWh/ft³*yr) that are up to 67% lower than industry standard HFC-based air-cooled, single-stage systems with electric defrost and up to 50% lower than gravity-flooded, single-stage screw compressor based systems employing $NH_3$ refrigerant. The energy performances of low charge $NH_3$ systems are sufficiently attractive to warrant straight replacement of existing industry standard HFC-based systems with new $NH_3$ systems provided plant owners can accept rates of return of 20% and prevailing unit electricity prices are ≥A$200/MWh. Added benefits of low charge $NH_3$ systems are the exceptionally low refrigerant inventories in the air coolers located within the refrigerated space. Complete loss of the operating charge from one of three air coolers within a refrigerated warehouse will under normal circumstances not give rise to an ammonia concentration within the warehouse of more than 200 ppm (complete mixing) and usually less. $NH_3$ concentrations that are 20–25 times greater and exposure times of 0.5–2 hours are required to pose significant risks to human health.

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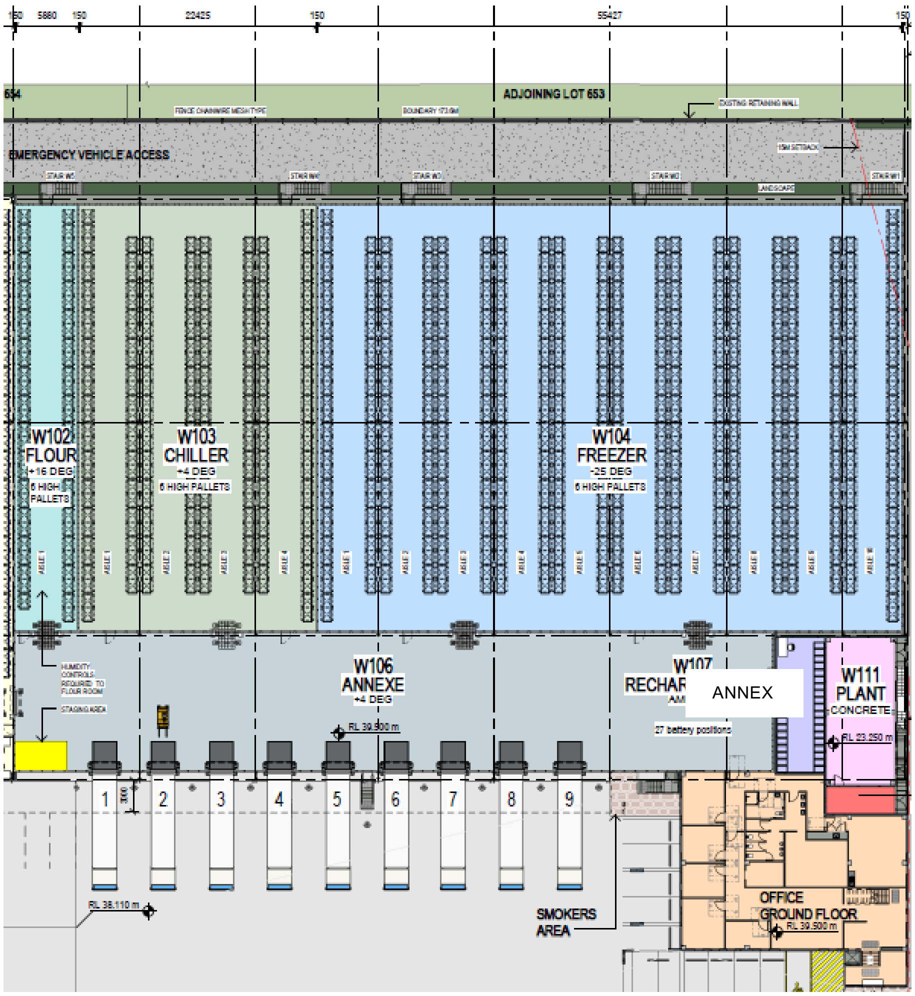


Figure 1.Perth facility in plan view

  

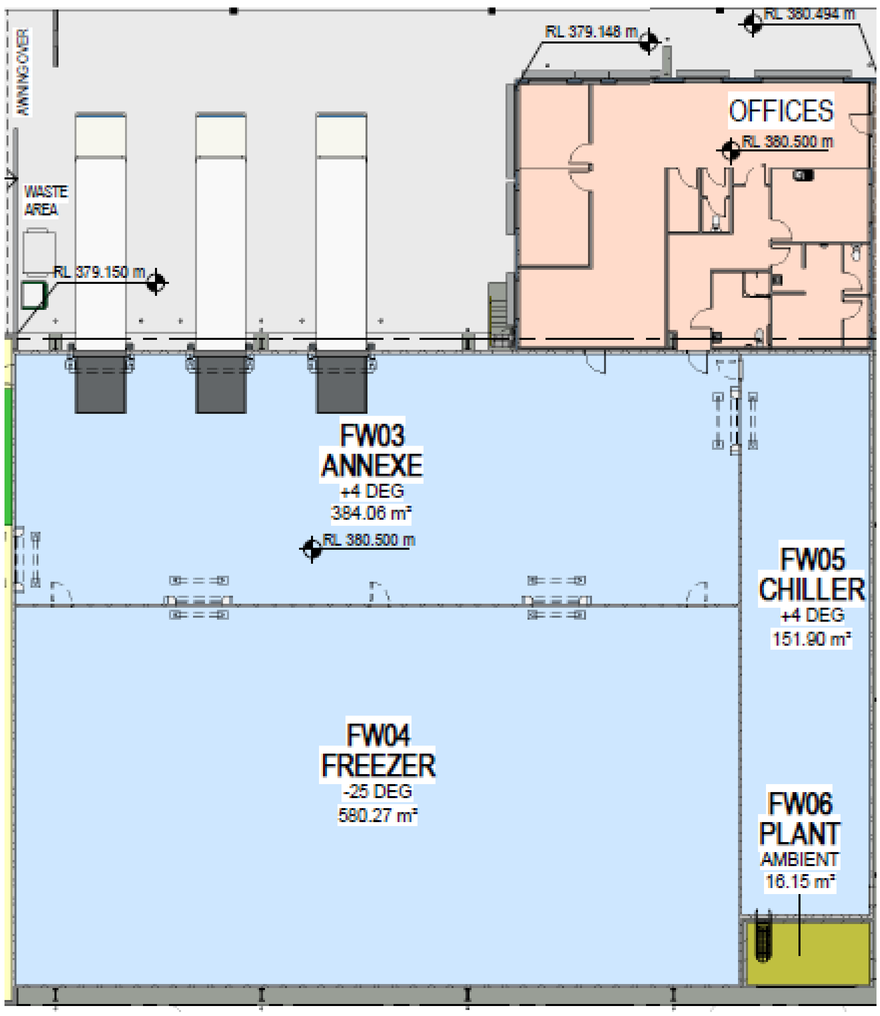


Figure 2. Tamworth facility in plan view

Figure 3. Low charge $NH_3$ system with secondary refrigerant and energy recovery from the second stage compressor discharge line for the desiccant drier

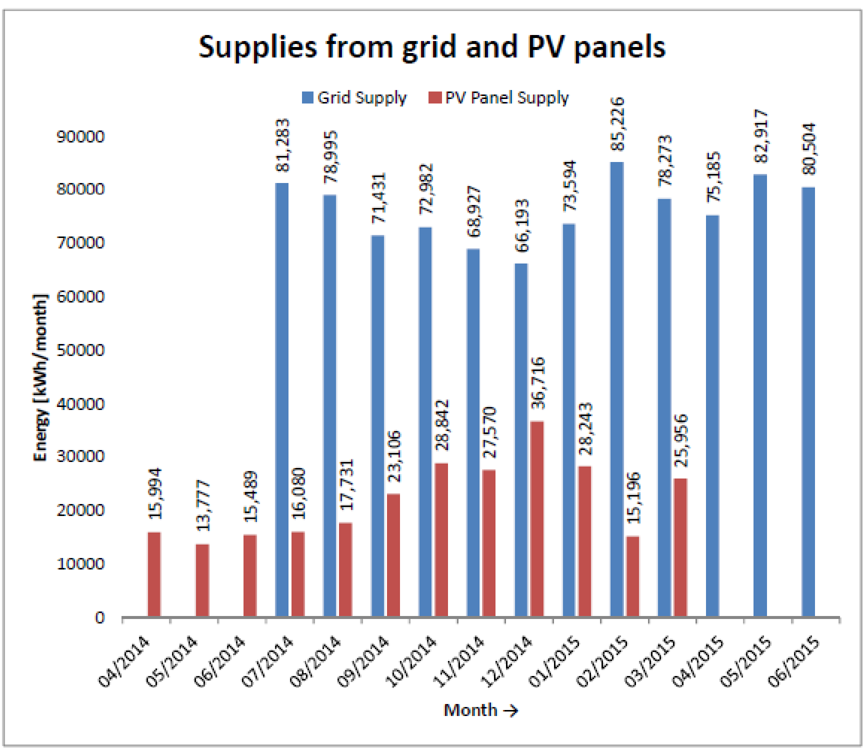


Figure 4. PV panel contribution for Perth facility

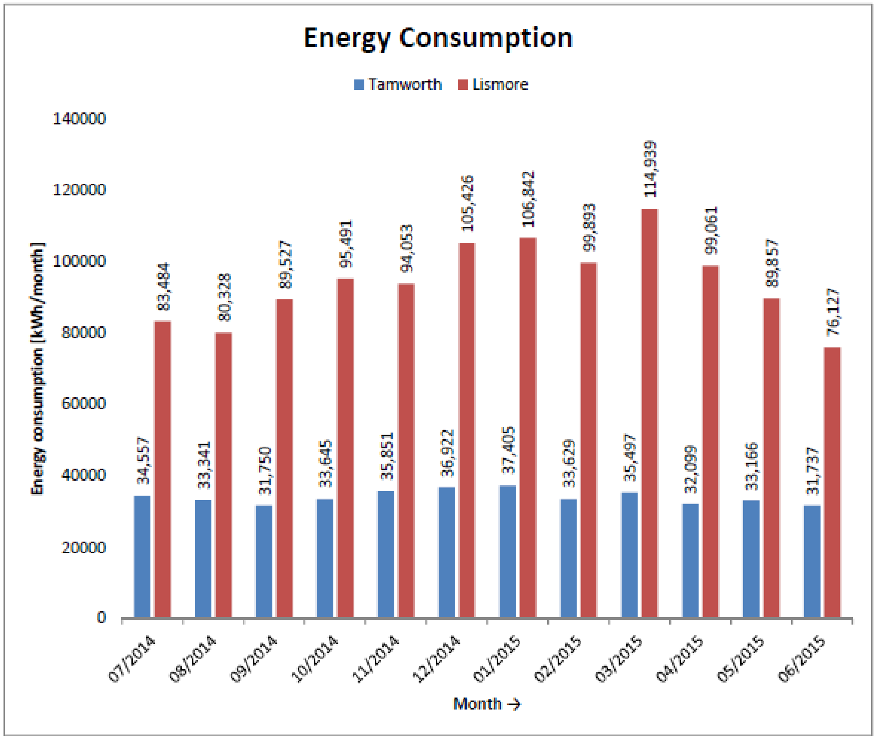


Figure 5. Energy consumption comparison for Tamworth and Lismore facilities

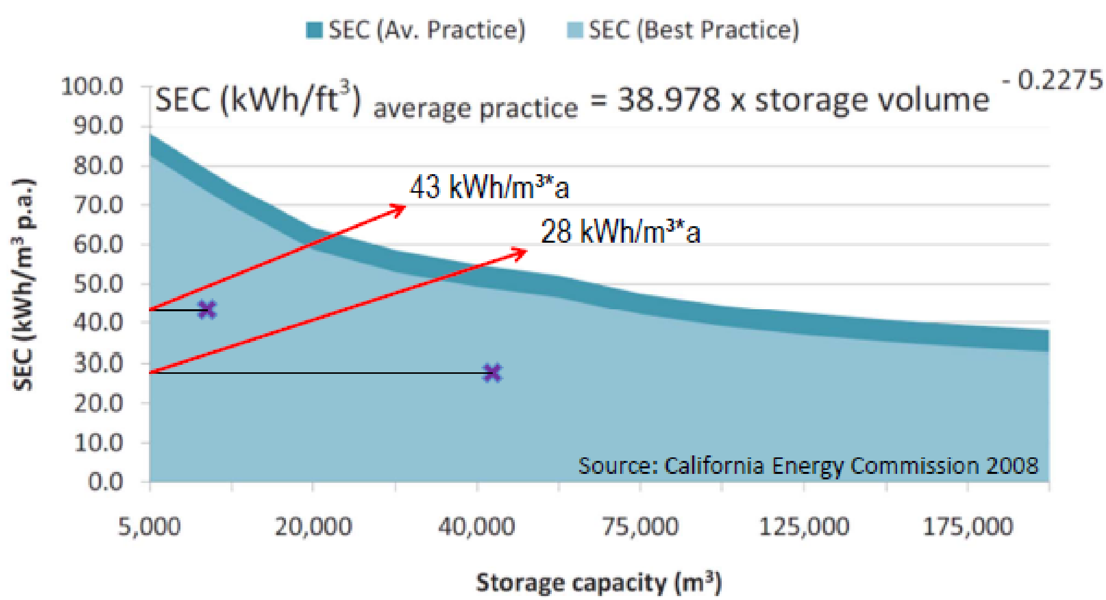


Figure 6. Specific energy consumption (SEC)
Source: California Energy Commission (2008).