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Energy performances of low-charge NH_3 systems in practice

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ABSTRACT

The paper describes the measured energy performances of two medium-sized refrigerated distribution centres with respective storage volumes of approximately 10,000m³ and 40,000m³. The performance evaluations are based on the electrical energy consumption as measured by the electrical energy provider over one calendar year. Both systems are serviced by state-of-the-art low-charge, dual-stage NH_3 refrigeration systems with VFD-controlled reciprocating compressors; evaporative condensers and hot gas defrost.

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 centres, each with a volume of approximately 10,000m³, but serviced by two different types of ammonia refrigeration systems. In one case the plant is a single-stage economised dual-screw compressor-based system with gravity-flooded refrigerant feed. In the other case the plant is a low-charge NH_3 dual-stage system with speed-controlled semi-industrial reciprocating compressors. Other features of the two facilities including general warehouse design 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.

All facilities have a common owner. The differences in the modes of operation and the products stored are therefore minimal between the plants compared. There are minor differences in climatic conditions between the three sites.

INTRODUCTION

In the wake of the global phase-down of hydrofluorocarbon refrigerants (HFCs) due to their contribution to global warming, users and owners of refrigeration systems are faced with decisions, which at times appear difficult. These decisions relate to whether or not users continue to employ HFC-based refrigeration systems, switch to low global warming (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. The factor referred to is energy performance – particularly the energy performance of low-charge NH_3 systems.

It is not in the commercial interest of proponents of refrigeration systems employing synthetic refrigerants to discuss the energy performance of the systems they market, so they rarely do. Their marketing focus is often attractive capital costs, refrigerant "safety", availability of service/maintenance resources and simplicity. The HFC phase-down is often marginalised by synthetic refrigerant proponents by referring to 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 specialised service/maintenance requirements associated with refrigerants such as NH_3.

Promoters of natural refrigerant-based systems on the other hand, have had a tendency 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 as yet not as common as liquid overfeed or gravity flooded systems and measured annual energy performances for low-charge NH_3 plants – particularly the modern versions – are relatively scarce.

The decision referred to earlier – HFCs versus low-GWP synthetics or natural refrigerants – is often made difficult by the quality, the independency (or lack thereof) 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 being exaggerated, biased and therefore irrelevant. The confusion on the part of end-users when faced with large amounts of conflicting technical information is understandable, and decisions in favour of low-capital-cost solutions perhaps not so surprising.

As this paper will show, the claims of 40–70% improvement in energy performance referred to above are not exaggerated. In fact, low-charge NH_3 systems can, if designed correctly, present an attractive business case in favour of straight replacement of existing out-dated 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 three refrigeration plants that are the main subjects of this paper are in summary:

1. A 43,000m^3 refrigerated distribution facility situated in Perth, Western Australia. The facility comprises a 16°C room, a 4°C chiller, a -25°C freezer and a 4°C 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.
2. A 10,000m^3 refrigerated distribution facility located in Tamworth, New South Wales. The facility comprises a 4°C chiller, a -25°C freezer and a 4°C 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 plan view.
3. A 10,000m^3 refrigerated distribution facility located in Lismore, New South Wales. The facility comprises a 4°C chiller, a -25°C freezer and a 4°C annex. The refrigeration plant is a single-stage NH_3 system with two fixed-speed industrial screw compressors with common economiser 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.

A fourth system similar to plant number one (Perth) was commissioned in Melbourne in September 2015. The main differences between the Melbourne and the Perth facilities are:

a) quality-based injection control in lieu of superheat based,

b) schedule 10 304SS NH_3 pipe lines in lieu of carbon steel,

c) larger evaporative condenser,

d) longer circuits and hence higher vapour velocities in the evaporators, and

e) desiccant drier in the freezer (later retrofitted to the Perth facility).

At the time of writing the energy consumption records for the Melbourne facility were limited to a few months. Evaluation of the effects of actions a) to e) on energy performance is therefore considered premature.

RECORDED ENERGY CONSUMPTION DETAILS

The measured annual energy consumption details for the three facilities are shown in Table 1.

In each case these are for the entire facility and typically include other services such as IT, general light and power, services for refrigerated trucks, forklift charging and office air conditioning. Unfortunately there are no detailed records of the energy consumption of the NH_3 systems in isolation. On more recent installations the SCADA systems are being fitted with hardware and software to facilitate the 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. Again on more recent installations, 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.

In the case of Perth, the 700MWh represent the electrical energy supplied from the grid and the 219MWh the energy supplied from the photovoltaic panels. The sum of 919MWh 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 915.6MWh for the period July 1, 2014 to June 30, 2015.

The contribution of the PV panels is only known from 1.7.2014 to 31.3.2015 inclusive. Therefore, only the nine-month period where supplies from the grid and the PV panels overlap is shown in Table 1. The monthly contribution from the PV Panels is shown in Figure 3.

Figure 1. Perth facility in plan view

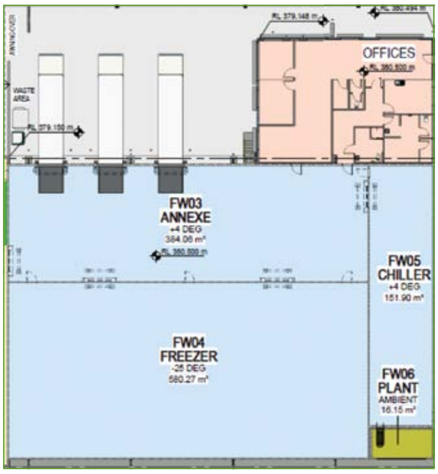

Figure 2. Tamworth facility in plan view

Plant	Total annual energy consumption [kWh]	Record period	Refrigerated volume [m³]	Specific energy consumption (SEC) [kWh/m³*a]
Perth	(700,072+219,440)/9*12 = 1,226,016	1.7.14 to 31.3.15	43,289	28.3
Tamworth	409,597	1.7.14 to 30.6.15	9,474	43.2
Lismore	1,135,027	1.7.14 to 30.6.15	10,748	105.6

Table 1. Recorded energy consumption

The annual electrical energy consumption of 1,226MWh for the Perth facility is simply estimated by extrapolation as shown in Table 1. If an assumed 20% of the annual electrical energy consumption are allocated to services other than the NH_3 plant, the annual specific energy consumption (SEC) of the NH_3 system becomes 22.7kWh/m³*a.

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 reduced from around AU$22,000 to around AU$12,000 according to the plant owner. 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 AU$6,000 to AU$8,000. It is estimated that the condensing pressure set-point adjustment at Lismore reduced SEC by 20 – 40% to around 65 – 85 kWh/m³*a. The comparison between the electrical energy consumption for the Tamworth and Lismore facilities is provided in graph form in Figure 4.

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 5. This is based on 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 modelled 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 combinations:

1: Single-stage screw compressor

2: Single-stage screw compressor with economiser

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

7: Dual-stage reciprocating compressors

Load patterns:

I: Combination of plate freezers and freezer stores, load variation 10–100%

II: Combination of blast freezers and freezer stores, load variation 40–100%

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

The modelling 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

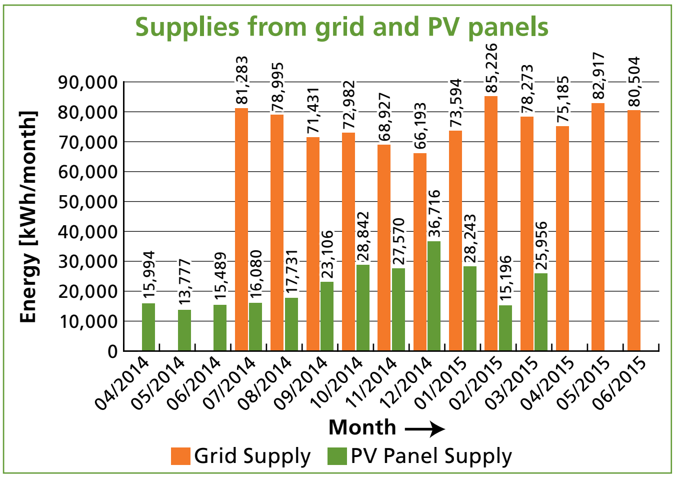

Figure 3. PV panel contribution for Perth facility.

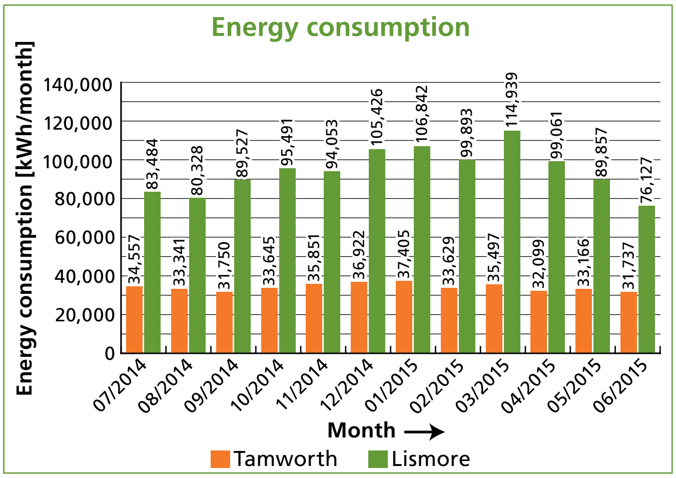

Figure 4. Energy consumption comparison for Tamworth and Lismore facilities.

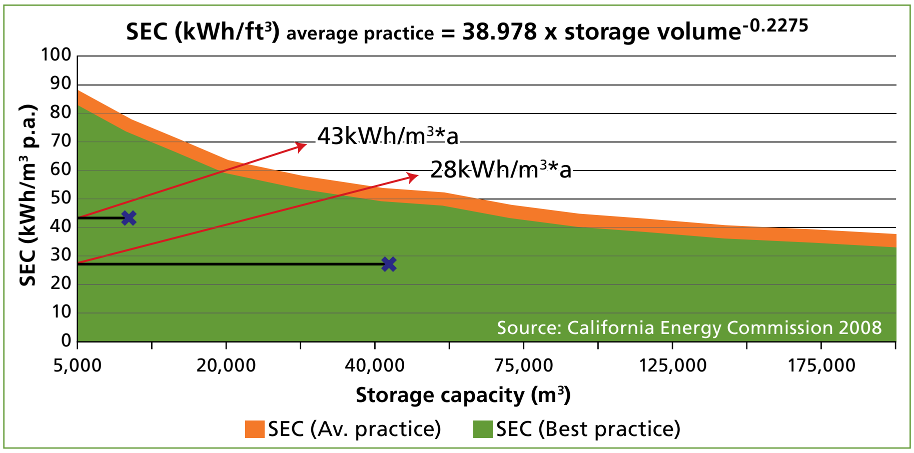

Figure 5. Specific energy consumption (SEC)

Lismore. The results reproduced in Table 2 do illustrate the importance of compressor part-load efficiency with respect to the delivery of superior energy performance.

The comparison of typical compressor part-load efficiencies in Table 3 further illustrates the importance of considering this element during system design (Comsel 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 315.7kW, corresponding shaft power consumption 83.1kW. The screw compressor is a Grasso HR2655S without economiser with a refrigeration capacity at 100% (2,940 rpm) of 294.5kW, corresponding shaft power 82.9kW.

LOW-CHARGE NH_3 VERSUS INDUSTRY-STANDARD HFC

The owner of the Perth facility operates a second distribution facility in the same suburb around 2km 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 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 centres. It is evident that the design refrigeration loads for the two distribution centres are similar.

The financial records of the operator of the Cocos Dr. warehouse indicate monthly electrical energy supply costs of around AU$42,000 per month on average. The electricity account for the Perth warehouse for the period April 1–30, 2015 was AU$13,751.57 incl. 10% GST. This was for a total supply of 81,264kWh. Based on Figure 3 this level of monthly electrical energy consumption is not unusual. It is the same electrical energy provider for both the Perth and the Cocos Dr. warehouses. From this it may be concluded that the energy performance improvement of the Perth warehouse serviced by a low-charge NH_3 system represents around (1–13,752/42,000)*100≈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 1,385m² facility situated at Kunda Park in Southeast Queensland and serviced by HFC-based air-cooled systems with electric defrost consumes around 1,265MWh annually. A 1,130m² facility serviced by a dual-stage, liquid overfeed NH_3 system situated at Somersby north of Sydney in New South Wales consumes 546MWh annually (Jensen, 2013).

Compressor combination	1	2	3	4	5	6	7
Screw compressor capacity, [kW]	1x500	1x500	1x452	1x452	1x500	1x452	—
Reciprocating compressor capacity, [kW]	—	—	1x48	1x48	1x48	3x151	1x47
Energy consumption, pattern I, [MWh/a]	1845	1812	740	725	898	665	675
Energy consumption, pattern II, [MWh/a]	1890	1825	1440	1370	1250	1150	1075

Table 2. Modelled annual energy consumption for various compressor configurations and load patterns

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 for screw and reciprocating compressors

Estimated heat loads:	LT	HT
Refrigerant temperature, °C	-31	-3
Flour Room, 16°C, L x W x H = 40.5 x 5.9 x 10.0m		11.1
Chiller, 4°C, L x W x H = 40.5 x 22.5 x 10.0m		51.1
Freezer, -25°C, L x W x H = 55.5 x 40.5 x 10.0m	173.8	
Annex, 4°C, L x W x H = 71.6 x 13.0 x 10.0m		166.2
Total (~43,000m³)	173.8	228.4

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

Estimated heat loads:	LT	HT
Refrigerant temperature, °C	-31	-3
Freezer 1, -25°C, L x W x H = 35.5 x 24.0 x 9.0m	88.8	
Freezer 2, -25°C, L x W x H = 30.0 x 29.5 x 9.0m	94.8	
Chiller 1, 4°C, L x W x H = 35.5 x 7.5 x 9.0m		46.1
Chiller Corridor, 4°C		6.0
Chiller 2, 4°C, L x W x H = 14.5 x 8.3 x 9.0m		17.7
Dock, 4°C, L x W x H = 20.0 x 19.0 x 4.5m		41.0
Annex, 4°C, L x W x H = 37.5 x 6.0 x 4.5m		82.3
Total (~22,000m³)	**183.6**	**193.1**

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

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, employment of NH_3 as the refrigerant is no guarantee of above-average energy performance, either. There are several factors that individually contribute towards the improvement of the energy efficiency. In order of significance these may be summarised as shown in Table 6. The percentage improvements shown cannot be interpreted as cumulative. Each factor is to be considered as one individual change, all other things being equal.

A refrigerated distribution facility comprising 46,000m³ frozen storage plus a 7,000m³ annex (Jensen, 2000) recorded a specific energy consumption of 35kWh/m³*a. 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 23,000m³ 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 27kWh/m³*a (Jensen, 2013). The latter facility was serviced by a dual-stage, liquid-overfeed system with four fixed-speed-drive reciprocating compressors. The percentage nominated 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.

A refrigerated distribution facility comprising 46,000m³ frozen storage plus a 7,000m³ annex (Jensen, 2000) with variable-speed penthouse fans recorded a 35% lower specific energy consumption than a similar neighbouring 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 nominated in item 2, Table 6.

In relation to item 3, Table 6 there are many practical examples of dry-expansion-feed air coolers for NH_3 failing to meet performance expectations (Jensen, 2011). There are several reasons for this. The most important are summarised below:

- Incorrect evaporator circuiting causing inadequate turbulence and stratified flow,
- Non-uniform liquid distribution within the air cooler,
- Presence of water in the refrigerant causing a refrigerant bubble point rise towards 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,

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

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

- Inadequate condensate removal during hot gas defrost due to inappropriate condensate drainage provisions,
- Inappropriate selection of expansion valve for the application
- Sub-optimal control methodology applied to the refrigerant injection and the control of the hot gas defrost procedure.

There are new air cooler technologies 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, which causes a capillary effect. New liquid distribution technologies have also been made available to enlarge the operating envelope (Nelson, 2013; Jensen 2015). 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 issues summarised above.

Item 4, 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% (NSW 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, total area around 30,000m^2.

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; the magnitude of the potential efficiency loss is as described in Table 2.

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. Secondly, oversizing the condenser such that it reduces the saturated condensing temperature by 1K, improves the coefficient of performance of a typical second-stage compressor by 2.6%. The percentage improvement nominated in item 6, Table 6 is readily within reach with this simple measure.

Quality-based control of the liquid injection into the evaporators is superior to conventional superheat-based control (Jensen, 2015). Practice has shown that entering temperature differences (ETD) between air and refrigerant of around 2.5K are possible without excessive control instability. In this context it is important to ensure that the possibility of liquid hold-up in the evaporator is minimised. 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.

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 also 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/m^3*a that are up to 67% lower than industry-standard HFC-based air-cooled, single-stage systems 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 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 ≥AUD200/MWh.

An added benefit is 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 200ppm (complete mixing) and usually less. It requires NH_3 concentrations that are 20–25 times greater and exposure times of 0.5 to two hours to pose significant risks to human health.

ABOUT THE AUTHOR

Stefan Jensen, F.AIRAH, is the managing director of Scantec Refrigeration Technologies. A member of AIRAH's national board of directors, he is widely recognised as a pioneer and an expert in energy-efficient systems and natural refrigerants.

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