Ammonia in traditional HFC territory – Part 1

Stefan S. Jensen, F.AIRAH (B.Sc.Eng. FIEAust, CPEng, M.IIAR) Scantec Refrigeration Technologies Pty Ltd
ssjensen@scantec.com.au

ABSTRACT

During the lead-up to the historic passing of the Carbon Tax Bill in the Australian Senate in November 2011, refrigeration plant users explored ways of minimising the impact of this new legislation. Not only did the legislation assign a price on carbon pollution, thereby increasing electricity costs, it also included a special levy on hydrofluorocarbon (HFC) refrigerants. The latter increased the retail price of HFCs by 300 to 500 per cent as of July 2012.

The actual introduction of the HFC levy in July 2012 generally encouraged a continuation of the efforts of both refrigeration plant users and designers to minimise the impacts of the levy. There are several pathways available to refrigeration and air conditioning plant users of minimising/avoiding the effects of the increasing costs of HFC refrigerants. It has been a common misconception that these pathways are generally characterised by a capital cost penalty and lack of acceptable benefit(s) in terms of return on additional investment.

By way of practical, real-life comparisons, this paper details the end-user business benefits associated with considering ammonia in those medium-size applications, which prior to the advent of the Carbon Tax Bill were reserved for HFC refrigerants.

These real-life comparisons are based on four dual-stage ammonia refrigeration systems in different geographical locations in Australia, ranging from temperate to subtropical environments. The plant designs are characterised by the application of a range of relatively innovative design concepts, including automatic ambient air defrost in frozen storage facilities, automatic oil return and oil distribution to the compressors, office air conditioning by means of ammonia refrigerant, variable- speed drive semi-industrial and industrial reciprocating compressors, employment of secondary refrigerant in chilled storage rooms, automatic venting of ammonia vapours from frozen storage rooms in the event of leaks, floating evaporating and condensing pressures and so on.

The facilities described would traditionally have been reserved for HFC-based refrigeration systems had it not been for the increase in HFC costs. This is commercial reality based on plant capital costs, plant simplicity, the cost of electrical energy and the cost of maintenance.

The paper describes the decision-making process on the part of the four end users that led to a departure from traditional thinking and what the practical and commercial consequences have been of a decision in favour of natural refrigerants and high energy-efficiency plant design. In the case of one particular end user, the paper will compare the annual energy consumption of two facilities that are almost identical in terms of floor area and refrigerated volume, but where the two facilities are serviced by two different types of refrigeration systems. One plant is serviced by a traditional HFC-based refrigeration system, the other by a new generation ammonia-based refrigeration system.

1. INTRODUCTION

The Australian Governments Clean Energy Future (CEF) Plan was first released July 10, 2011 and came into effect July 1, 2012. The objectives of the plan were to reduce pollution and to drive investment in lower polluting industries. Part of the plan was a carbon-equivalent levy applicable to the six Kyoto Protocol gases: carbon dioxide, methane, nitrous oxide, hydrofluorocarbons, perfluorocarbons and sulphur hexafluoride.

This carbon-equivalent levy was a direct function of the global warming potential (GWP) of the fluid. For the calculation of the magnitude of the levy, the Australian Government used the GWP values of Assessment Report 2 (AR2) of the International Panel of Climate Change (IPCC). For the year July 1, 2012 to June 30, 2013, the levy applicable was \$23/tonne of CO_2e. The term CO_2e refers to the equivalent warming impact of a chemical with CO_2 as the base of one (1). Using HFC404A with a GWP of 3,260 (AR2) as an example, the levy payable upon importation was therefore 23 x 3.260 = \$74.98 per kg. This does not include the non-carbon price import levy of \$165/tonne.

The non-carbon price import levy was introduced by the Australian Government prior to the introduction of the carbon-equivalent levy. It did also not include the cost of the refrigerant itself, supply chain margins, labour to charge systems etc. To assist all stakeholders in calculating the levy, the Department of Sustainability, Environment, Water, Population and Communities (DSEWPaC) published a "Calculator for the Import Levy and Equivalent Carbon Price for SGGs and SGG/HCFC blends".

Upon introduction, the levy was fixed for the first three years from 2012 to 2015 at the values shown in Table 1.

Period	Carbon Price/metric tonne
July 1, 2012 – June 30, 2013	\$23.00
July 1, 2013 – June 30, 2014	\$24.15
July 1, 2014 – June 30, 2015	\$25.40

Table 1: Carbon-equivalent levy applicable to HFC refrigerants.

Segment	1		2	
Geographic location	Brisbane	Mackay	Tweed Heads	Sydney
Design refrigeration capacities, low/high temperature, [kW]	133/140	110/164	70.0/72.0	83.3/78.5
Operating conditions, ET/IT/CT, [°C]	-33/-10/35	-33/-9/35	-30/-10/33	-32/-11/33
Compressor shaft power, PL/PH, [kW]	24.3/79.4	20.6/78.7	12.2/46.0	14.2/50.1
Condenser type	Evaporative	Evaporative	Evaporative	Evaporative
Refrigerant feed LT/HT (LR=liquid overfeed; F=Flooded)	LR/LR	LR/LR	F/F	LR/F
Secondary refrigerant (MT=medium temperature)	—	MT	MT	MT
Defrost for freezer segment	Hot gas	Ambient air	Ambient air	Ambient air
Freezer evaporator type	IDC	Alcove	Alcove	Alcove

Table 2: Technical data of the four ammonia refrigeration systems.

As a consequence of the election result on September 7, 2013, the political circumstances have changed. In October 2013 the new government introduced the Ozone Protection and Synthetic Greenhouse Gas (Import Levy) Amendment (Carbon Tax Repeal) Act 2013 to the House of Representatives. This Bill has passed the Lower House, but has yet to pass the Senate.

The momentum for change that was initiated by the introduction of the HFC levy appears to show signs of continuing despite the significant changes to political circumstances. There are broad industry initiatives seeking to prepare all stakeholders for a lower emission future. There is growing realisation on the part of end-users that well-designed natural refrigerant-based refrigeration and air conditioning systems have a range of attractive benefits, are technologically mature and safe. There is also an increasing realisation that due to wide-ranging HFC emission-reduction initiatives in other countries, most of which are supported by Australia, a change towards a future with significantly reduced consumption and emission of HFCs is inevitable.

2. THE FOUR REFRIGERATION PLANTS

This section summarises the fundamental design details of the ammonia refrigeration systems and the equivalent, traditional HFC-based systems that are being compared theoretically and practically. The technical data of the four ammonia refrigeration plants are summarised in Table 2.

The four plants are grouped into two segments, each comprising two systems. The segmentation is based on similarities with respect to compressor types, compressor swept volumes, plant capacities and so on. The two larger plants in the first segment employ industrial reciprocating compressors; the two smaller plants in the second segment are based around semi-industrial compressors and the use of a secondary refrigerant for all medium-temperature services. All plants service refrigerated distribution facilities.

Plan layouts of the distribution facilities are reproduced in Appendix 1. The Brisbane, Mackay and Sydney facilities are new developments. The Tweed Heads facility was an existing cold store, which prior to the upgrade was serviced by an HFC-based refrigeration system.

The Tweed Heads extension coincided with the replacement of the existing HFC plant with the new, dual-stage ammonia plant. Prior to the Tweed Heads upgrade, the chiller/freezer volumes were approximately 1400/1400 m³ respectively. Following the upgrade, these chiller/freezer volumes increased to 1750/4180 m³ respectively. The term IDC in Table 2 refers to an induced-draught cooler. An example of an alcove evaporator is shown in Figure 1.

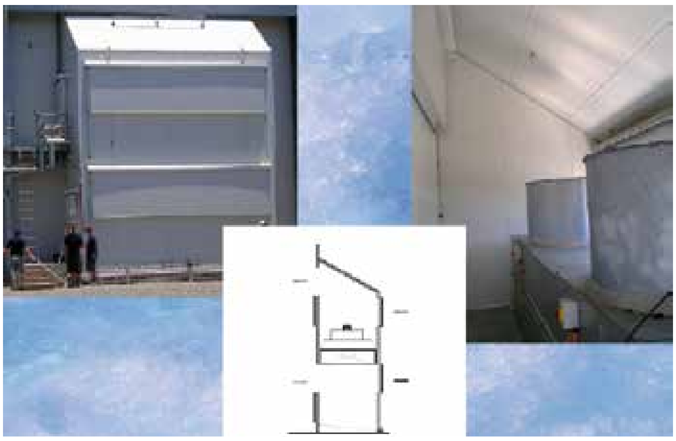

Figure 1: Single-coil alcove unit.

Segment	1		2	
Geographic location	Brisbane	Mackay	Tweed Heads	Sydney
Design refrigeration capacities, low/high temperature, [kW]	133.6/168.0	142.2/185.6	93.2/54.8	87.6/85.2
Operating conditions, ET/IT/CT, [°C]	-33.5/-8.5/46.0	-34.0/-7.0/48.0	-31.0/-7.0/46.0	-32.5/-7.0/48.0
Compressor shaft power, PL/PH, [kW]	127.4/74.6	154.2/104.4	81.8/27.8	81.0/44.2
Condenser type	Air cooled	Air cooled	Air cooled	Air cooled
Refrigerant feed LT/HT	Dry expansion	Dry expansion	Dry expansion	Dry expansion
Defrost for freezer segment	Electric	Electric	Electric	Electric
Freezer evaporator type	IDC	IDC	IDC	IDC

Table 3: Technical data of the four HFC 404A-based refrigeration systems.

Segment	1		2	
Geographic location	Brisbane	Mackay	Tweed Heads	Sydney
HFC, [kW]	482	417	247	294
NH_3, [kW]	273	248	157	153
Difference, [kW]	209	169	90	141

Table 4: Connected electric system power.

The four HFC 404A plants that were being contemplated for the same applications are summarised in Table 3.

The energy performance for the low-temperature element of the Brisbane facility is based on two single-stage HFC 404A economised screw compressors (unit swept volume 250 m³/h) operating in parallel, useful evaporator superheat 7K, evaporating and condensing temperatures as shown, economiser temperature 2.3°C, economiser approach 10K, unit evaporator capacity 66.8kW, unit shaft power 63.7kW, coefficient of performance 66.8/63.7=1.05. Other performances are generated in a similar manner using commercially available compressors and software.

Note that the minor variations in refrigeration capacities between Table 2 and Table 3 are a result of minor building changes between the initial HFC design and the final ammonia design. Some discrepancies are also a result of capacity steps between compressor capacities and the need for redundancy.

The connected electric power for the refrigeration plant is in many jurisdictions also important because it affects the structure of the energy charges. Secondly, the cost of the dedicated transformer for a new facility is often the responsibility of the facility owner.

Table 4 provides a comparison of connected total electric power for the HFC and NH_3 systems that are the topic of this paper. The connected power includes all auxiliary equipment such as fans, pumps and electric defrost heaters.

3. THEORETICAL ENERGY PERFORMANCE COMPARISONS

Ammonia refrigeration systems are in most cases more capital cost-intensive than HFC-based refrigeration systems of equivalent refrigeration capacity. In most cases there are also a number of benefits associated with ammonia compared with HFC-based systems. Quantification of these benefits to the user/owner of the refrigeration plant is what drives the investment decision in favour of ammonia.

The main reasons for the capital cost differences between ammonia and HFC-based systems are a) materials of construction, b) much more stringent refrigerant leakage detection requirements for ammonia systems, c) generally more robust and "industrial" design principles, d) occupational health and safety requirements and e) component costs. If those design requirements that are either compulsory or traditional for ammonia refrigeration systems were to be applied to HFC-based systems, then HFC-based systems could of course potentially deliver significantly improved energy performances, and fugitive emissions could be greatly reduced.

Dual compression-stage liquid-overfeed HFC-based systems with evaporative condensers and featuring ferrous materials throughout do not proliferate in the capacity ranges discussed in this paper. The main reason for this is that the capital costs would most likely be equal to or greater than the cost of an equivalent HFC-based system. The less favourable properties of HFCs compared with ammonia affect component performances. These differences in component performances can be compensated for by increasing the heat exchanger surface areas and refrigerant pipe line dimensions of HFC systems, but this is not common practice – on the contrary; systems are often designed to a price and not a standard.

Refrigerant →	NH_3	HFC 404A
COP	3.65	3.02

Table 5: Comparison of NH_3 and HFC 404A compressor coefficient of performance (COP).

Table 5 details the typical 17 per cent cycle performance difference for a well-known open reciprocating compressor brand with a swept volume of 141.6m³/h. In each case it is the same compressor, the saturated suction and discharge temperatures are identical (-10/35°C) and there is no useful superheat and no subcooling.

In jurisdictions where unit energy costs are comparatively high, the energy consumption over the life of the plant should be one of the most important investment decisions. In this context the emphasis is on the expression "should be" because there are examples where the energy costs are either not considered at all, are considered to be constant whatever the plant type, or are viewed as being relatively unimportant compared with system capital costs. It is, of course, also not in the commercial interest of proponents of low capital-cost systems featuring high energy consumption to disclose lifecycle cost details to the investor. In view of the fact that the energy consumption cost over the life of a refrigeration system is often several times the initial capital outlay, ignoring energy performance comparisons between plant alternatives can be an expensive error.

Evaluation of annual energy performances for systems is often associated with complicated computer models taking into account a large number of variables. For relative energy performance comparisons between system concepts (investment proposals), simpler methodologies can be useful and can exhibit sufficient accuracy for the investment decision. An example in relation to the Brisbane-based plant in Table 2 demonstrates this; for details refer to Appendix 2 showing the detailed calculation for plant "A". A summary of the results of this relatively simple manual calculation using a spreadsheet is provided in Table 6.

These values are based on a unit electrical energy cost of $150/MWh, a power factor of 0.85 and an overall electric motor efficiency of 0.85. Electric motor efficiencies vary greatly depending on size and application from around 0.6 to 0.95 – the 0.85 value used here is an attempt at an average realistic value applicable to the mix of motors employed in the systems analysed. The plant concepts A, B and C are described below.

A) Dual-stage ammonia (NH_3) refrigeration system with reciprocating compressors (rotational speed ≤ 970 rpm), liquid overfeed at both temperature levels, alcove evaporators with automatic ambient air defrost for the freezers, ceiling-mounted induced-draught air coolers for the high-temperature areas, six-pole fan motors for all air coolers, stainless steel/fibreglass evaporative condenser oversized by a factor of 1.23 and variable-frequency fan drives throughout except in the ante rooms.

B) Ammonia refrigeration system with two single-stage economised screw compressors for the freezer rooms, gravity-flooded refrigerant feed for the low-temperature segment, alcove evaporators with automatic ambient-air defrost for the freezers, ceiling-mounted induced-draught air coolers for the high-temperature areas, two reciprocating compressors for the high-temperature areas, glycol reticulation for the high-temperature areas, six-pole fan motors for all air coolers, adiabatically assisted air-cooled condensers at both temperature levels, and variable-frequency fan drives throughout except in the ante rooms.

C) HFC refrigeration system (R404A) comprising four single-stage economised screw compressors (three duty and one standby) two of which service the freezer rooms and one which services the high-temperature areas, dry expansion refrigerant feed throughout, ceiling-mounted induced-draught air coolers, electric defrost for freezers and chiller, common air-cooled dual-circuit condenser, air-cooled oil cooler for the freezer compressors, six-pole fan motors for all air coolers and condensers and variable- frequency fan drives throughout except in the ante rooms.

Plant concept	A	B	C
Energy consumption, [MWh/a]	635	942	1248
Energy consumption, [kWh/m³a]	25.3	37.5	49.6
Annual energy cost	$132,000	$196,000	$259,000

Table 6: Relative energy performance comparison for the Brisbane Plant of Table 2.

The methodologies behind the calculations in Appendix 2 are in summary:

- Peak hours represent ~250 operating days/annum of eight hours each; these hours are taken as being those from 8am to 4pm.
- Shoulder hours represent week days after hours; these hours are taken as being those from 4pm to 8am.
- Minimum hours represent non-peak and non-shoulder hours; these hours are therefore predominantly week-end hours and public holidays.
- Heat loads are calculated at the average conditions matching the hours.
- Compressor speeds and shaft power are calculated to match loads.
- Condenser fan speed is varied linearly with heat rejection.
- Condenser spray pump operates at full capacity constantly.

- Ammonia pumps operate at full load constantly.
- Fan power is reduced by speed reduction cubed.
- Subfloor ventilation fan operates at full capacity constantly.
- Engine room ventilation fan power is varied with ambient temperature.

The energy bill reproduced in Figure 2 verifies that the simplified energy performance modelling is sufficiently accurate for the purposes of arriving at an appropriate investment decision. The energy consumed over 56 billing days is 106.5MWh for the entire site. This value includes light and power for the offices and some office air conditioning.

Empirically for facilities such as this, light and power accounts for around 20–25 per cent of the total electrical energy consumed. The billing period is also for the cooler autumn/winter months, so a 15–20 per cent increase can be anticipated for the summer period from December to end of February.

Account period: 07 Apr 2011 to 02 Jun 2011 - Bill Days 56
Next anticipated reading: 02 Sep 2011 (± 2 business days)

Electricity Tariff - Tariff 20 - General Supply

	Meter Number	Previous Reading	Current Reading	Usage kWh	@Rate c/kWh	Amount $
Peak Use						
Peak	1326765	0	306.2[1]			
	1326765	0	34.6[2]			
	1326765	0	324.7[3]			
				106480	21.7500	23,159.40

[1]Your base usage of 306.2 is multiplied by 160 to calculate Total usage
[2]Your base usage of 34.6 is multiplied by 160 to calculate Total usage
[3]Your base usage of 324.7 is multiplied by 160 to calculate Total usage

Other Charges
Service to Property 24.94

Figure 2: Energy bill for the Brisbane plant.

Based on these assumptions, the energy consumption for the refrigeration plant based on meter readings may be estimated:

(106.5/56 x 182 + 106.5/56 x 183*1.2) x 0.8 = 611MWh

The factor 0.8 represents the 80 per cent of the total consumed by the ammonia plant, and the factor 1.2 represents the approximate increase in plant energy consumption between summer and winter. The corrected annual meter reading of 611MWh is within 5 per cent of the estimate of 635MWh in Table 6, which was actually the estimate upon which the client based the investment decision. An interesting detail in Figure 2 is that the unit cost of electricity is significantly higher than the value used in Table 6.

Applying a unit electricity cost of $0.2175/kWh in Table 6 would increase the annual electricity cost difference between "A" and "C" from $127,000 to around $184,000.

4. QUANTIFIABLE BENEFITS OF NH_3 OTHER THAN ENERGY

Compared with a single-stage HFC 404A system, the example in Table 6 indicates a substantial energy consumption cost advantage associated with a dual-stage ammonia plant. The energy consumption cost advantage will of course escalate with increasing unit electricity costs as demonstrated in the preceding section. Other quantifiable benefits are a) immunity to any environmental legislation and synthetic refrigerant levies, b) low refrigerant leakage rates and c) longer technical plant life.

The immunity to environmental levies has become highly relevant in Australia following the introduction of the carbon-equivalent levy on those synthetic refrigerants with high global warming potential. Mounting scientific evidence that continued unabated release of HFC refrigerant may be responsible for 28–45 per cent of projected global CO_2 emissions (CO_2e basis) by 2050 [1] will most likely lead to further restrictions of their use world-wide.

The historic agreement between presidents Obama and Xi in June 2013 to work together to use the Montreal Protocol as a tool for an HFC phase-down supports this view. Based on a recent verbal survey among industrial refrigeration practitioners within AIRAH, leakage rates in modern ammonia refrigeration systems have been reduced to <1% per cent of the system charge per annum. Compared with HFC systems, this is exceptional. In Australia, the working bank of HFC/HCFC/CFC refrigerants is officially estimated at 40,100 metric tonnes; the annual service consumption rate is estimated at 9 per cent [2]. Applying this annual service consumption rate to plant "C" of Table 6 would give rise to an annual R404A replenishment cost of $8,000–$10,000. The low ammonia system leakage rates are a result of the high safety standards that are applicable, the materials of construction and the skill levels of designers, installers and maintenance staff [2]. The longer technical life of ammonia refrigeration plants compared with equivalent HFC systems is a reflection of the traditionally more industrial approach to component and system design displayed by ammonia practitioners. This in part explains the usually higher capital cost of ammonia plant, but "you get what you pay for" as this paper attempts to show.

Part 2 of the report will be published in July Ecolibrium.

About the Author

Stefan Jensen, F.AIRAH, graduated in 1978 in Denmark with a Bachelor of Science degree in mechanical engineering. His professional career commenced in 1978 with Danfoss Denmark, followed by two years at SABROE Refrigeration A/S as a project engineer. In 1996 Stefan co-founded Scantec Refrigeration Technologies. He now holds the position of managing director. He has authored over 30 technical papers for AIRAH, IIR and IIAR conferences.