Large Commercial Air-conditioning Plant Using Ammonia as a Natural Refrigerant - A Case Study

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INTRODUCTION

To date in Australia the majority of large air conditioning plants use HFC refrigerant based chillers (i.e. R410A,R134a or similar). With increasing world focus on global warming, air conditioning designers need to provide solutions that have true long term sustainability, high efficiency and low global warming potential.
This paper provides a comparison between a modern ammonia refrigerant water chiller system and a high efficiency HFC based chiller system to show that a natural refrigerant alternative is viable in many commercial applications. The use of the Total equivalent warming Index or TEWI is demonstrated to show how direct refrigerant emissions from system leakage can impact on the overall global warming potential of a refrigeration system.

This paper is an update of a previously presented paper by Stefan Jensen at the AIRAH 2009 Natural Refrigerants Update in Brisbane. [1]

The basic safety features that should be incorporated in an ammonia based system as outlined in the recently released Victorian Code of Practice for Ammonia Refrigeration [2] are also addressed.

Ammonia as a refrigerant is the only refrigerant with no direct climate damaging effect. Its ozone depletion (ODP) as well as its global warming potential (GWP) are zero, while its thermodynamic properties are excellent. [3]

There is a growing recognition internationally, that natural refrigerants such as ammonia offer long term sustainability [4]. To date different governments and regulators have taken a range of approaches towards this , from regulations in Denmark that ban large HFC new installations to voluntary agreements in the USA between developers and the EPA to ban CFCs and HFCS, and reworking of regulations in France that reduce barriers to the use of ammonia as a refrigerant . [3] [5]

As previously identified by Stefan Jensen, ammonia is increasingly displacing synthetic CFC's, HCFC's and HFC's in applications, which traditionally were thought not to be suitable for an "industrial" working fluid such as ammonia. [1]

He further points out that extensive research and development focusing on air conditioning chiller charge minimization to levels around 0.05 kg of NH_3 per kW cooling capacity is accelerating this trend. Values of around 0.018 kg of

NH_3 per kW cooling effect in five years are predicted by employment of micro channel heat exchanger technology. This compares with HFC current generation chillers with approximately 0.29kg/kW.

Charge minimization efforts are continually reducing earlier figures. Star refrigeration in the UK have just released the Azane chiller range of air cooled high efficiency water chillers which have a specific refrigerant charge of 0.1kg/kW of duty. These systems use tried and tested technologies that offer robust low maintenance equipment.

Figure 1 Factory Packaged Air Cooled Chiller using ammonia refrigerant

One typical “non-traditional” ammonia application is large-scale air conditioning plant employing reticulated chilled water as secondary refrigerant.

The use of ammonia refrigerant in large-scale air conditioning installations in Australia is very rare. On the basis of a practical case study, this paper updates the previous broad comparison between a conventional air conditioning system and an equivalent system using ammonia as the working fluid.

The conventional system is based around high efficiency multiple liquid chilling packages using HFC refrigerant with Turbocor oil free centrifugal compressors. The alternative is a liquid chilling package using ammonia refrigerant. The Turbocor HFC option has been deliberately chosen as a comparison to demonstrate that a well designed ammonia refrigerant alternative is also capable of providing high efficiency.

The comparison specifically analyzes energy efficiency, capital costs and the challenges facing the system designers when incorporating the two chiller types with the two different working fluids into a commercial building in a safe manner.

Particular emphasis is placed upon addressing the barriers that exist in terms of embracing modern ammonia based liquid chillers in air conditioning applications in Australia today. By referencing the Code of Practice the paper aims to help consolidate the information necessary to successfully use the natural refrigerant technology for selected Australian air conditioning applications.

SUITABLE BUILDING TYPES

Large scale chilled water systems are required in a range of buildings. Of these typically exhibition buildings, sporting complexes, air ports and medical centers would be suitable candidates for ammonia based chilled water systems. For the purposes of this paper it is not proposed to use this solution in CBD high rise locations which often have basement plant rooms. In general any large development which has either a remote energy centre or plant located on an outdoor deck or even roof top would be suitable.

To demonstrate that this has been achieved in other locations already, the following table of existing ammonia chilled water installations is presented.

Table 1. Ammonia Chilled Water Applications

LOCATION	NOTES
Homerton University Hospital London	2x0.5MW capacity
Ice skating rink refurbishment New Brunswick Canada	Including hot water heat reclaim
Energy center London Olympic site & another at the Aquatic center	NA
Radiology Group Medical device cooling in Germany	NA
Copenhagen Airport	NA
Heathrow Terminal 5	4x6.6MW
Hannover Trade fair building	3.5MW

The case study is based on a 6.5MW maximum cooling load and could be any of the above applications designed for sustainability and a green image. The air conditioning concept is reticulated chilled water provided via a central, chilled water plant. The condenser type is water cooled with cooling water provided by a cooling tower. The plant room is situated either on a plant deck or in a remote energy center. The plant concept is shown schematically in figure 2.

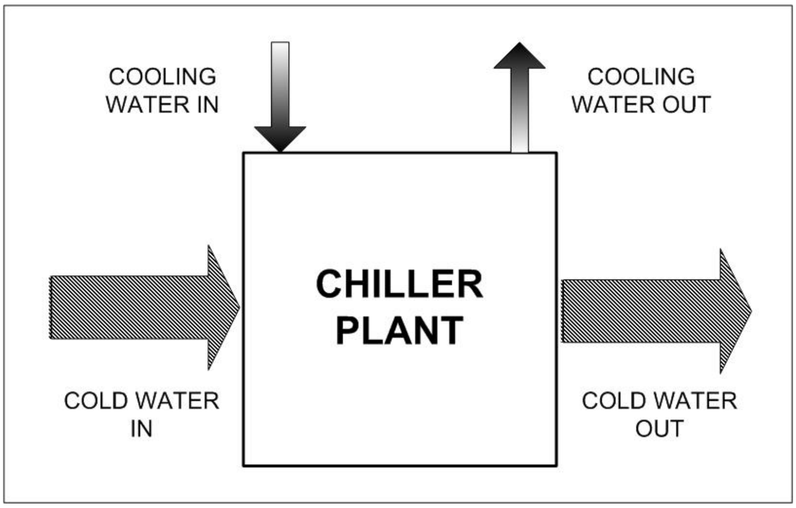

Figure 2. Air Conditioning Plant Concept

The calculated heat load of the building is maximum 6500 kW. The chilled water system is required to provide this peak cooling capacity without diversity.

It has been assumed that the load profile is the same as that proposed in Stefan Jensen's earlier paper [1] . Around two thirds of the operating hours are in the cooling capacity range 30% to 70%. The estimated heat load profile for the building is shown in table 2 and figure 2. The cooling energy requirement is the product of the mean load and the operating hours.

It is evident that good part load efficiency is important due to the extensive periods that the plant operates at less than full capacity. This requires particular attention not only to compressor sequencing, but also to the capacity control of the auxiliary equipment such as chilled water pumps and cooling tower pumps.

Table 2. Building Heat Load Profile

Load [MW]	Load [%]	Hours p.a. [h]	Cooling Energy [MWh]
6.0-6.5	92-100	70	438
5.5-6.0	85-92	190	1093
5.0-5.5	77-85	470	2468
4.5-5.0	69-77	780	3705
4.0-4.5	62-69	1060	4505
3.5-4.0	54-62	1210	4538
3.0-3.5	46-54	1450	4713
2.5-3.0	38-46	1240	3410
2.0-2.5	31-38	890	2003
1.5-2.0	23-31	680	1190
1.0-1.5	15-23	480	600
0.5-1.0	8-15	180	135
0.0-0.5	0-8	60	15
Σ	-	8760	28813

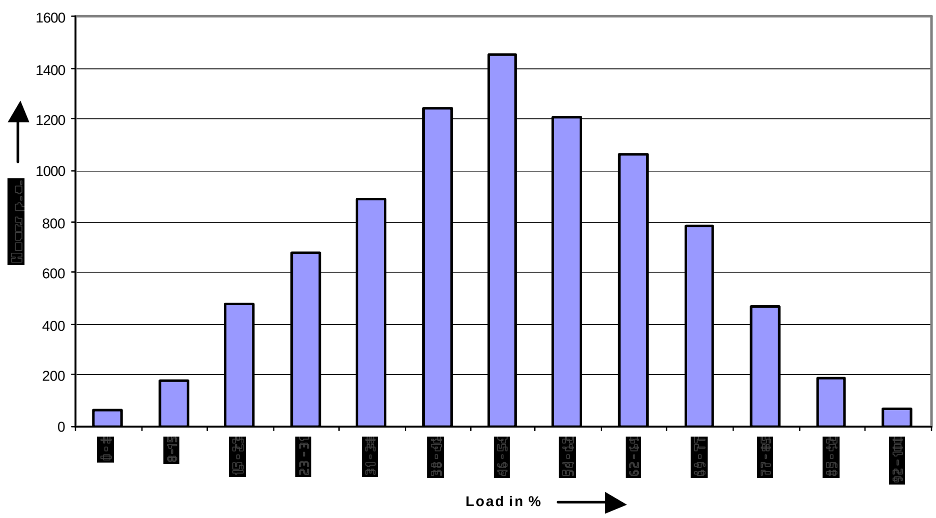

Figure 3. Heat Load Profile

DESCRIPTION OF CONVENTIONAL HFC AIR CONDITIONING PLANT

The conventional air conditioning plant is shown in figure 4. This plant comprises three liquid chilling units each with 5 Turbocor oil free centrifugal compressors screw compressors, shell and tube evaporators with electronic tx valves and water cooled shell and tube condensers. All units are identical and the refrigerant charge per chiller is around 645 kg.

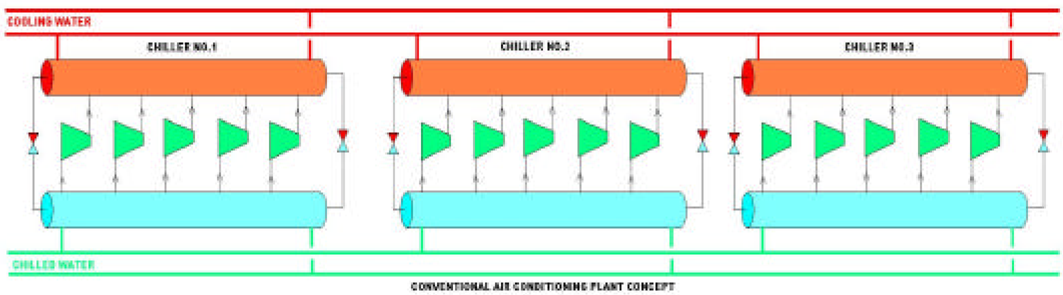

Figure 4. Conventional Air Conditioning Plant Concept

The key full load performance data for each chiller unit are summarized in table 3. The total cooling capacity for the three units is 6480 kW, total compressor shaft power 1125 kW and overall full load coefficient of performance (C.O.P.) is 5.76.

Table 3. Full Load Unit Performance Data

Refrigerant	R134A
Chilled water entering/leaving temperatures, °C	12.0/6.0

Chilled water pressure drop, kPa	47
Cooling capacity per chiller , kW	2160
Compressor shaft power per chiller , kW	375.0
Cooling water entering/leaving temperatures, °C	29.5/35.0
Cooling water pressure drop, kPa	55
Heat rejection, kW	2516
Coefficient of Performance	5.76

These chillers utilize oil free magnetic bearings and proprietary variable speed drives and high centrifugal impellor speeds to provide an optimized C.O.P. To reflect realistic condenser water fouling resistance the entering condenser water temperature has been adjusted.

It is noted that improved COP values will be achieved at reduced condenser water temperatures experienced at lower ambient. This has not been factored into the analysis for either option considered to be consistent.

It should be noted that the HFC R134A has a GWP of 1,300 which means that for every kg released into the atmosphere a CO2 equivalent of 1.3 tonnes is released.

The unit part load performances for constant chilled water leaving temperature of 6.0°C, constant cooling water inlet temperature of 29.5°C and constant water flows are shown in table 4 and figure 4 for the chiller unit in question (unit described in table 2).

Table 4. Part Load Unit Performance

Compressor load percentage	**Coefficient of Performance or C.O.P.**
100	5.76
90	6.0
80	6.2
70	6.3
60	6.4
50	6.3
40	6.5
30	6.7
20	6.9
15	6.9

From the above it is evident that the Turbocor compressor & magnetic bearing variable speed combination C.O.P actually improves at part load as the compressor speed drops. To avoid unnecessary machine wear out the study assumes that the chiller sequencing control would still attempt to minimize the number of chillers running to meet the load.

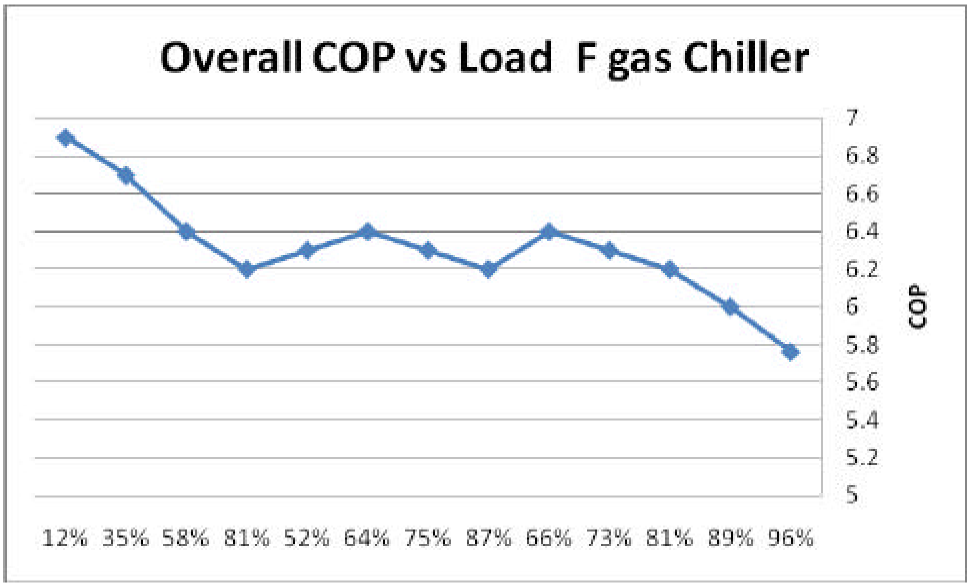

Figure 5. Part Load Unit Performance

The overall liquid chilling plant performances are as estimated in table 5. These are based on the heat load profile detailed in table 2.

The chilled water flow is controlled in response to the cooling load so as to maintain constant water temperature differential across the evaporators. The condenser water flow is not controlled, but is maintained at full flow at all times. The chilled water leaving temperature is constant at 6.0°C. The cooling water inlet temperature is constant at 29.5°C.

Table 5. Overall liquid chilling plant performance

Load [MW]	Hours p.a. [h]	Cooling Energy [MWh]	Compressor shaft power [MW]	Compressor shaft energy [MWh]
6.0-6.5	70	438	1.085	75.9
5.5-6.0	190	1093	0.958	182.0
5.0-5.5	470	2468	0.847	397.9
4.5-5.0	780	3705	0.754	588.0
4.0-4.5	1060	4505	0.664	703.9
3.5-4.0	1210	4538	0.605	731.8
3.0-3.5	1450	4713	0.516	748.0
2.5-3.0	1240	3410	0.430	532.8
2.0-2.5	890	2003	0.357	317.8
1.5-2.0	680	1190	0.282	191.9
1.0-1.5	480	600	0.195	93.75
0.5-1.0	180	135	0.112	20.14
0.0-0.5	60	15	0.036	2.17
Σ	8760	28813	-	4586

The annual average C.O.P. excluding auxiliary equipment may be estimated at 28813/4586=6.28. The average cooling energy provided may be estimated at 28813/8760=3.29 MW or 51% of the peak capacity of the plant. The average compressor shaft power may be estimated as 4586/8760=0.52MW.

The total October 2010 capital cost of the three liquid chilling packages erected within the engine room, charged with refrigerant and oil and including starter panels, but excluding interconnecting chilled water and cooling water piping, cooling tower, electrical sub mains etc. is approximately $1,400,000.

DESCRIPTION OF NH_3 BASED CHILLER PLANT

The liquid chilling system employing refrigerant NH_3 is shown schematically in figure 6. This plant is similar to that in the earlier paper [1].The plant employs three industrial screw compressors. Two of the compressors are of identical size; the third unit has a swept volume, which is approximately ninety percent of each of the two larger machines when it runs at 60HZ. Evaporators and condensers are of the semi-welded cassette type. To provide increased efficiency the refrigerant liquid is sub cooled from a plate heat exchanger connected to the compressors economizer ports. The smaller compressor is fitted with variable speed drive to allow higher efficiency running at part load. All compressor motors are high efficiency type.

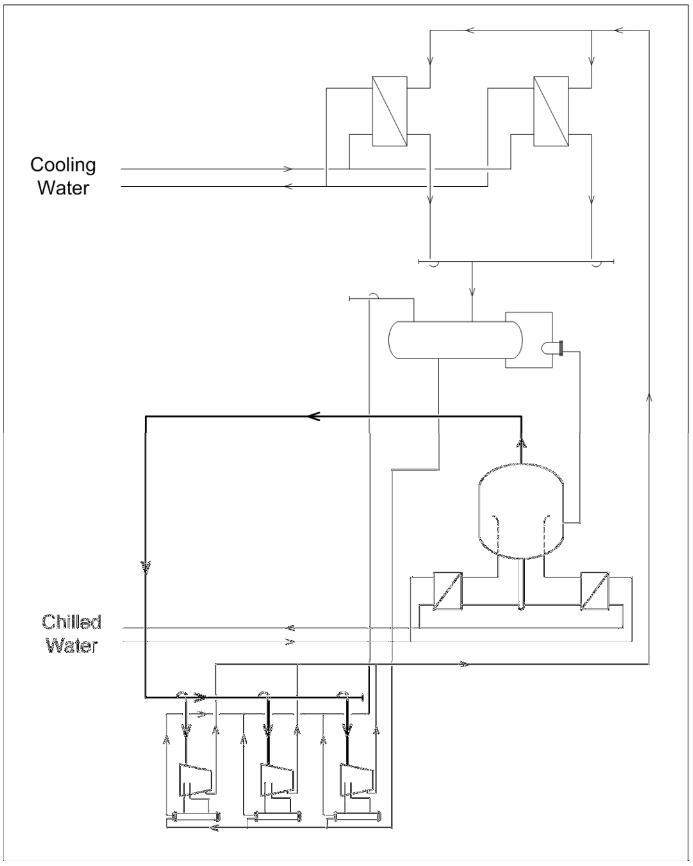

Figure 6. Chiller plant employing NH_3 refrigerant (not all items shown for clarity)

The refrigerant feed is of the gravity flooded type due to the nature of the refrigerant. The expansion device is a high pressure float valve opening at rising liquid level in the receiver. Oil cooling is by means of the thermosyphon principle ensuring maintenance free operation.

The total NH_3 refrigerant charge is estimated at 1300 kg. This represents around 0.2 kg per kW cooling capacity. This is not as low as factory packaged chiller units, which may be around 0.05 to 0.07 kg per kW cooling capacity. However, the latter do also not readily offer the advantage of full utilization of all evaporator and condenser surface areas at part load – a factor that improves plant C.O.P. considerably. This plant represents a robust design using well proven components which deliver high reliability and long life.

Some further reduction in refrigerant charge can be made by using a vertical liquid receiver and plate and shell thermosyphon oil coolers.

The key full load performance data for the chiller plant are summarized in table 6. The total cooling capacity is 6500 kW, total compressor shaft power 1014 kW and overall full load coefficient of performance (C.O.P.) is 6.41.

Table 6. Full Load Plant Performance Data

Refrigerant	R717
Chilled water entering/leaving temperatures, °C	12.0/6.0
Chilled water pressure drop, kPa	28
Cooling capacity, kW	6500
Compressor shaft power, kW	1014.
Evaporating temperature, °C	3.0
Evaporator fouling factor, m^2K/W	0.000035
Suction line temperature drop, K	0.5
Cooling water entering/leaving temperatures, °C	29.5/35.0
Cooling water pressure drop, kPa	10
Condensing temperature, °C	36.6
Condenser fouling factor, m^2K/W	0.00002
Discharge line temperature drop, K	0.2
Heat rejection, kW	7514
Coefficient of Performance	6.41

The overall liquid chilling plant performances are as estimated in table 7 with a graphic representation in figure 7. These are based on the heat load profile detailed in table 2.

The chilled water flow is controlled in response to the cooling load so as to maintain constant water temperature differential across the evaporators. The condenser water flow is not controlled, but is maintained at full flow at all times. The chilled water leaving temperature is constant at 6.0°C. The cooling water inlet temperature is constant at 29.5°C.

Table 7. Overall liquid chilling plant performance

Load [MW]	Hours p.a. [h]	Cooling Energy [MWh]	Compressor shaft power [MW]	Compressor shaft energy [MWh]
6.0-6.5	70	438	0.975	68.25
5.5-6.0	190	1093	0.896	170.1
5.0-5.5	470	2468	0.848	398.35
4.5-5.0	780	3705	0.732	571.27
4.0-4.5	1060	4505	0.689	730.15
3.5-4.0	1210	4538	0.587	710.1
3.0-3.5	1450	4713	0.523	758.7
2.5-3.0	1240	3410	0.459	569.26
2.0-2.5	890	2003	0.357	317.8
1.5-2.0	680	1190	0.278	189.1
1.0-1.5	480	600	0.199	95.4
0.5-1.0	180	135	0.128	23.0
0.0-0.5	60	15	0.066	3.98
Σ	8760	28813		4605

The annual average C.O.P. excluding auxiliary equipment may be estimated at 28813/4605=6.25. The average compressor shaft power may be estimated as 4605/8760=0.53MW.

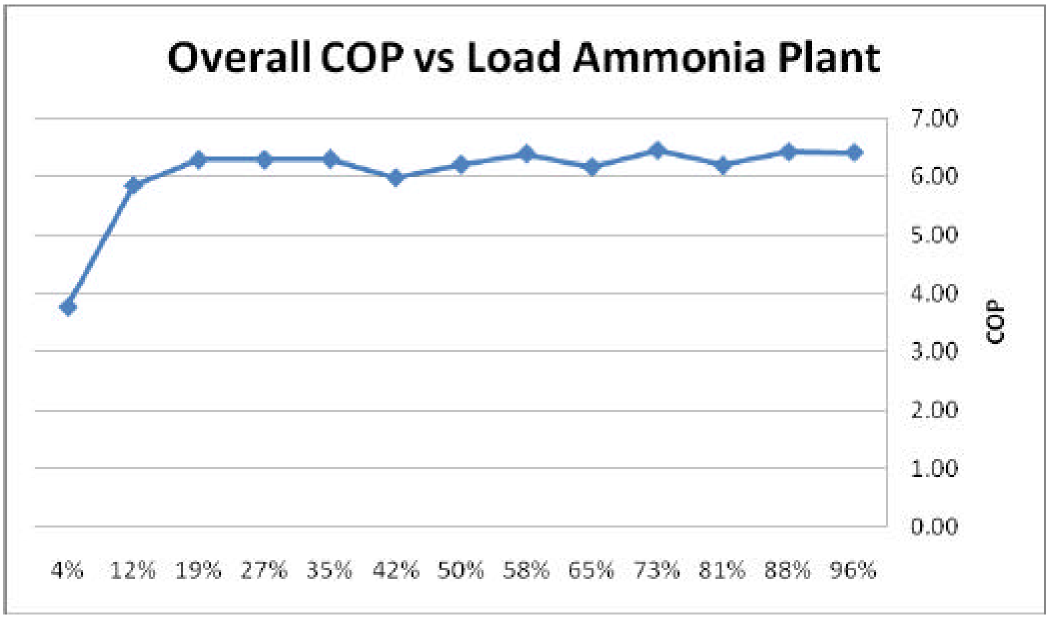

Figure 7. Overall coefficient of performance (COP) as a function of load percentage

The total October 2010 capital cost of the liquid chilling package erected within the engine room, charged with refrigerant and including automatic oil recovery, engine room ventilation system, personal protective equipment, electrical motor starters, controls, site wiring and SCADA system, but excluding interconnecting chilled water and cooling water piping, cooling tower, electrical sub mains etc. is approximately $1,750,000.

PLANT ROOM DESIGN and SAFETY ASPECTS

The preferred plant room location for an ammonia based liquid chiller is either in a ground level energy center or on a plant mezzanine deck which has open space around it well clear of any air-conditioning or ventilation intakes.

The Victorian Code of Practice Ammonia Refrigeration is based on the Australian Standard AS 1677 Part 2 Safety Requirements for Fixed Applications and other associated standards, and [2] gathers together in one document the safety and regulatory compliance requirements for sites with ammonia refrigeration plants. Section 2.3 of the Code outlines recommendations and best practice for plant rooms.

The Code states that due to ammonias relatively high lower explosion limit (LEL), that ammonia concentrations can be monitored and alarmed at levels well below the LEL. Australian and international standards indicate that specially protected electrical items are thus not required in ammonia refrigeration plant rooms provided all electrical circuits are isolated at a safe location should the ammonia concentration detected reach 20% of the LEL which is 30,000PPM. The only exceptions are the exhaust fan/s and emergency lighting systems.

In terms of construction, ammonia plant rooms should be constructed from fire resistant materials such as concrete. This provides effective fire separation in most instances and helps to contain any ammonia leaks. Walls and doors which separate ammonia refrigeration plant rooms from other parts of the building , ceiling spaces, switch rooms , boiler rooms etc should have at least 1 hour fire ratings . (Local building or fire regulations may call for higher ratings)

In summary the code requires:

- Ventilation either static or forced to comply with AS 1677. If mechanical fans are used these must have EXN or explosion proof fan motors and be wired to comply with Zone 2 of AS/NZS 2381.
- Emergency lighting & battery backup. The emergency lighting should be explosion proof as this may be required to operate during a gas leak.
- Ability to isolate all other electrical circuits in the plant room from a safe location.
- Automatic shut down of all electrical circuits at 20% of the LEL of ammonia (30,000PPM)

Other Best Practice recommendations for plant room construction and layout in the code include:

- Construction from fire resistant materials such as concrete to provide effective fire separation.
- A separate sealed electrical control room with 1 hour fire rated walls between the control room and the plant room.

Other Sections of the Code which will directly relevant include:

Part 4 Emergency Planning and Manifest requirements
Part 5 Maintenance and inspection of ammonia refrigeration systems
Part 6 Signage requirements
Part 7 Requirements for Personal Protective Equipment (PPE) that is to be held on site.
Part 8 Type and location of automatic ammonia detection systems and alarm levels.
Part 9 Training for operators and maintenance staff
Part 10 Auditing to demonstrate risk management

Other features which can be included to enhance system safety include:

- Piping relief valves to discharge into water tanks to reduce the amount of accidental release to atmosphere.

- Installing automatically activated water spray scrubbers on plant room exhaust ducting or discharges to partially neutralize the ammonia gas discharge.

- Installing motorized shut off valves to isolate liquid ammonia supply from the receiver.

SYSTEM COMPARISON BETWEEN THE CONVENTIONAL HFC AND THE NH_3 BASED SOLUTIONS

The concept of Total Equivalent Warming Index (TEWI) is a useful way of comparing the combined influences of chiller efficiency, indirect CO2 emissions caused by energy generation, and refrigerant leakage or direct emissions. This method is recommended in a number of International and European Standards [6]. The TEWI comparisons between the conventional and the NH_3 based solutions are provided in table 8.

Table 8. TEWI comparisons between the conventional and the NH_3 based solutions

TOTAL EQUIVALENT WARMING INDEX -TEWI		
	Ammonia	**Turbocor**
DIRECT EFFECT	**NH3**	**R134a**
System Charge kgs	1300	1935
Loss % Per Year	3	8

DIRECT EFFECT (Continued)	**Ammonia**	**Turbocor**
Loss in kg per 10 years	390	1548
GWP	0	1300
CO2 Tonnes 10 Years Lifetime Direct	0	2012
INDIRECT EFFECT		
Total mW hrs compressors	4,605	4,586
TOTAL kW hrs per year	4,605	4,586
Kg OF CO_2 PER MWhr	1220	1220
Tonnes of CO2 in 10 years Indirect	56181	55949
RECOVERY LOSSES		
Recycling factor	1	0.85
Recovery Loss	0	377
TOTAL TEWI in 10 Years tonnes	**56181**	**58339**
Difference %	-4%	0%
Equipment capital cost	$1.75 million	$1.4 million
GWP = Global Warming Potential		
Leakage % estimated for life of plant		
TEWI= Total Equivalent Warming Index		

This comparison shows a number of important points. Firstly the importance of high chiller or compressor efficiency which makes up a significant part of the TEWI through the indirect emissions. This is particularly so in countries such as Australia which has a relatively high CO_2 emission per kWh of power generation. The above comparison shows that a well designed ammonia plant can match current generation high efficiency F gas chiller options.

The influence of refrigerant leakage is shown in the Direct Effect emissions which shows the R134a based chiller as having over 2012 tonnes higher emissions over 10 years . The 8% leakage rates for the R134a chiller comes from a range of sources including IPCC good practice guidelines , NGERS technical guidelines and figures used by the Department of Environment, Water, Heritage & the Arts which range from 2 to 15%.

The ammonia based chiller system does have a higher equipment cost, however international trends show that CO_2 emissions should have costs attributed to them , it is likely at some point in the future that Australia will also adopt some form of carbon tax or equal . When this happens the cost of R134a refrigerant will increase, at a starting point of $15 per tonne as used by NAB [7] this would increase the cost of R134a by another $19.50 per kg to approximately $44 per kg which represents a significant cost exposure over the life of the plant. If CO_{2e} is costed at $26 per tonne as proposed in the governments 2009 white paper , the cost of refrigerant leakage over 10 years would be $56,100.

The ammonia based plant uses industrial design components which typically remain operational for 20 plus years with modest maintenance costs. This should be compared with the R134a based option which has 15 high speed compressors; common sense would suggest that the maintenance costs of this plant option are likely to increase as time goes on.

From the above it can be seen that for applications where minimizing overall CO_2 emissions is a driver, the ammonia based plant is a good long term proposition. The attractiveness of this option is even greater for applications such as sports centers which have significant simultaneous hot water demands. A basic de super heater heat exchanger on the ammonia plant compressor discharge could provide 15 to 20 percent of the refrigeration capacity as heat reclaim which could represent 1000 to 1200 kW of heating with no additional fuel usage or emissions. For a heating demand of 110,000GJ per month this represents 55 tonnes per month less CO_2 emissions from burning natural gas at 51.2 $kgCO_{2e}$.

CONCLUSIONS

Ammonia based chillers are capable of delivering high efficiencies when compared on an equal basis with modern F gas chillers. Using the TEWI calculation to compare options, the effects of direct refrigerant emissions over the life of the equipment can be quantified showing the advantage of using low global warming potential refrigerants. A further benefit of the ammonia chiller option is the heat reclaim which can be used to further reduce water heating fuel usage.

Natural refrigerants can provide truly sustainable long term solutions which significantly reduce future commercial risk against carbon taxes of similar initiatives. For large scale chiller & refrigeration applications ammonia is likely to be the most appropriate natural refrigerant to use, it is inherently safer than hydrocarbons and it’s likely to be cheaper to implement than carbon dioxide. [8]

The increasing adoption of modern ammonia refrigeration technologies in Europe combined with a demonstrated safety record, allow designers to consider using ammonia in applications previously dominated by F gas equipment. This includes airports, sporting facilities, exhibition buildings and facilities generally served by remote energy centers.

We can expect to see further development of low kg/kW charge systems, reduced plant foot prints, automatic oil return and factory packaging to reduce traditional barriers of ammonia plant. System safety is being increased by using better sealing technologies and in the future we anticipate the use of permanent magnet hermetic compressors for use with ammonia. Increased factory packaging will also drive pricing differentials down.

Best practice safety design and regulatory requirements can now be identified by using the Victorian Code of Practice which allows the design team to confidently incorporate the equipment into their projects and reduces further the barriers to uptake.

ACKNOWLEDGMENTS

The author would like to thank Stefan Jensen for his permission to use his previously presented paper to form the basis of this update.

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