Purdue University Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

High Pressure Ammonia Systems – New Opportunities

Andy Pearson Star Refrigeration

Follow this and additional works at: http://docs.lib.purdue.edu/iracc

Pearson, Andy, "High Pressure Ammonia Systems – New Opportunities" (2010). *International Refrigeration and Air Conditioning Conference*. Paper 1111. http://docs.lib.purdue.edu/iracc/1111

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

High Pressure Ammonia Systems - New Opportunities

Andy PEARSON

Star Refrigeration Ltd, Glasgow, UK apearson@star-ref.co.uk

ABSTRACT

Recent developments in the industrial refrigeration, air conditioning and heat pump sector have included significant increases in the typical operating pressures of systems. With ammonia this offers several opportunities to capitalise on the high critical temperature and high index of compression of the refrigerant in order to deliver heat efficiently in the temperature range $70^{\circ}C - 90^{\circ}C$. It also suggests that high pressure ammonia systems might be suitable for use in the Rankine Cycle to generate power from relatively low grade waste heat.

This paper explores the opportunities presented by the combination of ammonia with high pressure equipment, and gives several case studies and worked examples of the way these concepts are currently being deployed to provide higher efficiency heating and cooling systems.

1. INTRODUCTION

Ammonia has been in use as a low temperature refrigerant for over 150 years but despite its high critical temperature it has not been applied so readily as a heat pump working fluid. Several recent technical and legislative developments have created increased interest in this market, and successful implementation in a range of applications has helped to emphasise the potential of this technology.

1.1 Legislative developments

The move from CFCs to HCFCs produced a wide range of "blended refrigerants" – mixtures of two or more components which were intended to match the properties of the restricted fluids as closely as possible. This included the use of R-404A and R-507 as alternatives to R-502, although the operating pressure at 55°C for the new blends was respectively 10% and 14% higher than the equivalent pressure in a R-502 system. The move from HCFCs to HFCs prompted further blend development, including R-410A – a mixture of the higher pressure components R-32 and R-125. At 55°C saturation temperature the operating pressure of R-410A is 45% higher than that of R-502. The relative pressures are shown in Table 1.

| Refrigerant | R-502 | R-22 | R-404A | R-507 | R-410A |
|----------------------|--------------|--------------|--------------|--------------|--------------|
| Sat Pressure at 55°C | 23.4 bar abs | 21.7 bar abs | 25.7 bar abs | 26.7 bar abs | 34.0 bar abs |
| Relative to R-502 | (-) | -7.7% | +10% | +14% | +43% |

Table 1 - Relative pressures of various CFC replacements

In other respects R-410A is a more suitable alternative to R-502 than the earlier blends: the boiling point at atmospheric pressure is lower, and the system efficiency is better, although the discharge temperature is somewhat higher (albeit not as high temperature nor as efficient as R-22). In some regions, for example Scandinavian countries, HFC use is also quite restricted, and this has led to an increased interest in "natural refrigerants" for applications previously served by halocarbons. At the same time there has been a growing concern about electrical energy use and its effect on climate. Several schemes have been introduced to encourage more efficient operation of equipment, and heat pumps are now classed as a form of "renewable energy", for example in the United States Emergency Economic Stabilization Act (2008). In the United Kingdom the government is set to introduce the

Renewable Heat Incentive in April 2011. In this scheme users are paid a tariff for each kWh of heat provided by their heat pump system. This is likely to create a significant demand for both ground source and air source heat pumps provided the public can be persuaded that the payment commitments in the scheme are sufficiently robust and will survive several changes of government in the life of the scheme.

1.2 Technical developments

Once the difficulties inherent in designing for higher pressures had been addressed and once designers were familiar with the properties of the components of these new blends, it was realised that higher operating pressure delivered some significant benefits. In particular the compressor swept volume is less for a given duty and the evaporating and condensing heat transfer coefficients are typically better than for lower pressure gases. The increasingly difficult demands presented by the legislative programme were therefore prompting designers to accept the challenge of producing higher pressure equipment and maximising the benefits gained. R-410A quickly became widely used, but in high temperature applications like small air-cooled water chillers and split air conditioners. In parallel with this development carbon dioxide (R-744) was becoming more widely used, and required even higher pressure equipment. There is no doubt that product development in the mass market for small R-410A systems provided a boost for R-744 systems, and equipment designed for 120 bar allowable pressure is now common. A further parallel development was in the increased use of high pressure fuel gas compressors, for example in recovering methane from landfill sites and feeding it to generating turbines to produce electricity. The single screw compressor was modified for this purpose in the United States, and the result was a compressor with a relatively large swept volume that could operate at up to 75 bar (1100psig). This machine is also suitable, with minor modifications, for use with R-744 in evaporatively cooled applications, or for use with R-717 in heat pumps.

2. REVIEW OF AMMONIA PROPERTIES

Lorentzen (1988) pointed out that the suitability of a fluid as a refrigerant can be determined by consideration of four key properties: the vapour pressure at the evaporating temperature, the critical pressure, the critical temperature and the molar mass. He concluded that ammonia was "an excellent alternative" to CFCs for refrigeration. When the high latent heat, high index of compression and other factors are considered (Pearson, 2008) the possibilities as a heat pump fluid are also clear – the relevant properties compared with R-134a are shown in Table 2.

| Refrigerant | R-717 | R-134a |
|---|--------------|--------|
| Boiling temperature at atmospheric pressure (°C) | -33.33 | -26.07 |
| Critical temperature (°C) | 133.25 | 101.06 |
| Critical pressure (bar abs) | 113.3 | 40.59 |
| Molar mass (g/mol) | 17 | 102 |
| Specific enthalpy of evaporation at -10°C (kJ/kg) | 1297 | 206 |
| Specific heat capacity of liquid at -10°C (kJ/kg.K) | 4.56 | 1.31 |
| Specific heat capacity of vapour at -10°C (kJ/kg.K) | 2.51 | 0.84 |
| Thermal conductivity of liquid at -10°C (W/m.K) | 540.5 | 98 |
| Viscosity of liquid at -10°C (Pa.s) | 196.8 | 326.3 |
| Density of liquid at -10° C (kg/m ³) | 652.0 | 1325.6 |

Table 2 – Relevant properties comparison

(extracted from Tables 3 and 6 of "Ammonia as a Refrigerant")

The critical temperature indicates that ammonia is well suited for use in high temperature heat pumps. R-134a has the highest critical temperature of the common fluorocarbon options. In comparison, R-22 has a critical temperature of 96°C, and the high pressure refrigerant blends are all substantially lower, typically less than 75°C. Ammonia and R-134a are therefore the only common refrigerants considered for use in heat pumps (with the exception of isobutane in some special applications). The reduced temperature is the ratio of the condensing temperature to the critical temperature. A good rule of thumb is that system efficiency is impaired if the reduced temperature is greater than 0.5, and the limit of operation is a reduced temperature of 0.9 unless an alternative technology such as transcritical operation is used. For ammonia the reduced temperature of 0.5 equates to a condensing temperature of 95°C, whereas for R-134a it is 68°C. The limits of operation, according to the 0.9 rule, are 126°C and 96°C respectively. In both cases raising the reduced temperature from 0.5 to 0.9 reduces the specific enthalpy of

evaporation by about 50%. However this masks a further disadvantage of R-134a in high discharge pressure applications. In a cycle which condenses at 95°C and evaporates at 50°C the amount of flash gas created at the expansion valve is 55.8% of the total mass flow for R-134a, but only 22.8% for R-717. This can be deduced from the shapes of the saturated vapour line and saturated liquid line on a pressure-enthalpy plot: for R-134a the vapour "dome" has a significant overhang to the right, whereas for R-717 the line is vertical. The saturated liquid line is sloped in both cases, but when plotted to the same scale as shown in Figure 1 the difference is apparent. The change in liquid enthalpy from evaporating to condensing condition is a far higher proportion of the specific enthalpy of evaporation for R-134a than it is for R-717. Furthermore, although the operating pressures for R-717 are higher than R-134a the pressure ratio is almost the same and so compressor efficiency in a reciprocating machine is not significantly different, and the volume ratio for a screw compressor is within the normal range.



(pressure scale 0.5 bar abs – 200 bar abs)



The large latent heat of evaporation of R-717 evident in Figure 1a does not translate directly to smaller compressors and gas pipe lines because the reduced mass flow is offset by reduced vapour density, due to the low molecular weight. However it does provide the dual benefits of reduced liquid line size and reduced pressure losses in the vapour lines (or more specifically a reduction in the drop in saturation temperature caused by the pressure loss).

The effect of the high index of compression of R-717 can also be seen in Figure 1a. The lines of constant entropy are relatively flat compared with R-134a, and so the compressor discharge reaches higher temperatures for a given saturated temperature lift. The resultant temperature peak is not however an indication of the temperature that can be reached in the heated fluid - the majority of the heat is recovered from the phase change and therefore is available at a temperature slightly below the condensing temperature for the R-717. The high discharge temperature can either be used to add a small increment to the main heat recovery, or can be used to heat a smaller mass flow to a much higher temperature.

The high thermal conductivity and low viscosity of R-717 provide high coefficients of heat transfer in the evaporator without suffering the penalty of excessive pressure drop. Stoecker (2000) shows that ammonia offers higher heat transfer coefficients than any other common refrigerating fluid, even high pressure carbon dioxide.

The poorer performance of R-134a at these elevated conditions can be partially offset by the use of large additional "internal heat exchangers", such as economisers or suction-liquid heat exchangers. However these need to be sized to transfer a lot of heat - sometimes as much as the evaporator - and they need to be designed for the high discharge pressure, so can be expensive. In comparison, the ammonia heat pump, despite using a somewhat exotic and hazardous working fluid, is likely to be more efficient and cheaper to build, due to the combination of excellent properties.

3. CASE STUDY – FACTORY HEAT RECOVERY FOR PROCESS HEATING

An ammonia heat pump was installed in 2009 in a food factory in England. The system is designed to provide useful cooling of process glycol (30% monopropylene glycol) to 0°C while at the same time heating a closed loop heating system to 60° C. The system comprises four compressors on a central plant, with two of them rejecting heat to the water heating circuit when required. All four machines are also capable of rejecting heat to atmosphere

through a bank of air-cooled condensers. A schematic of the system is shown in Figure 2. Up to 1.25MW of heat can be recovered, with a combined cycle coefficient of performance of 5.46.



Figure 2 - Schematic diagram of combined factory cooling and heating system

The high discharge temperature of the ammonia compressors allows the condensing condition to be held at 59° C, with desuperheating of the discharge gas from 100° C to 59° C providing the additional heating to raise the hot water to 60° C. The heating loop is split into two requirements: clean in place (CIP) and closed loop (product heating).

| | CIP circuit | Closed loop circuit |
|------------------------|-------------|---------------------|
| Water inlet (°C) | 10 | 40 |
| Water outlet (°C) | 60 | 60 |
| Water flow rate (kg/s) | 3.57 | 5.95 |
| Heating Duty (kW) | 745 | 500 |
| % in desuperheater | 10.4% | 10% |
| % in condenser | 75.6% | 75% |
| % in subcooler | 14% | 15%* |

The heating load configuration is summarised in Table 3

Table 3 – summary of heating capacity.

* Note – the subcooler in the closed loop circuit exchanges heat with the water cooled oil cooler circuit, not shown in Figure 2.

The total glycol cooling capacity of the plant is 3,300kW. About one third of this is provided by the heat pump compressors, which are configured to run as the lead cooling requirement. Since a large proportion of the closed loop cooling duty is related to product heating the two duties are coincident and so a significant saving can be made by serving them both from the same refrigerating system. The CIP load is much less frequent but requires a significantly higher instantaneous load. This is reduced by using a hot water buffer system which is charged and discharged during normal operation. This enables the heating requirement to be better matched to the cooling load. Use of a buffer system like this also allows the rate of heat supply to the process to be far higher than the rate of heat recovery from the refrigeration system, provided the total volume supplied does not exceed the buffer capacity and the recovery time between discharges is sufficiently long.

4. CASE STUDY – DISTRICT HEATING

A higher temperature requirement was specified for a district heating system. The nominal design condition has a flow temperature of 90°C and a return temperature of 60°C, with a maximum heating load of 14MW. This is achieved with three two-stage modules connected in series on the heating side. The modules each comprise two single screw compressors, low stage and high stage. The low stage machines are cast iron bodies, rated for 36.9 bar gauge allowable pressure. The high stage machines however are cast steel bodies, rated for 69 bar gauge.

The source of heat in this case is the town harbour – even though the plant is installed in Norway, on the Oslo fjord, there is sufficient heat in the harbour water to justify operation of the heat pump. The water is drawn from a depth of 30m and has a reasonably consistent year-round temperature of 8°C. It is at its coldest in April, due to melt water flowing down from the mountains, but even under these conditions it is not less than 5°C. The refrigeration system is designed to evaporate at 2°C, using spray chillers on each module, with an average cooling capacity of 3,500kW. The average heating capacity of the three modules is 4,700kW, with a power input of 1,490kW, giving a heating coefficient of performance of 3.15. To raise the heating circuit from 60°C to 90°C the modules condense at 72.4°C, 81.3°C and 89.0°C respectively. A flow diagram for a single unit is shown in Figure 3 and a block diagram showing how the three are connected together is shown in Figure 4.



Figure 2 – High Pressure Ammonia Heat Pump Note: temperatures shown are for the second stage of three

Once again the majority of the heat is recovered in the condenser, but in this case because the reduced pressure is higher than in Case Study 1, the amount of high temperature heat available from oil coolers, desuperheaters and the intercooler is greater. This enables a delivery temperature of 90°C to be achieved even though the three units in the system all condense at lower temperatures than this, with condenser water outlet temperatures of 70.6°C, 79.8°C and 87.2°C respectively. The water circuit is quite complex at first sight: the condensers of the three units are piped in series to produce as low a pressure lift on the first unit as possible. The desuperheaters and subcoolers are piped in parallel, partly to reduce the flow rates through these smaller heat exchangers and partly to enable optimum performance to be achieved. Putting the subcoolers in parallel means that all units are able to cool the liquid



ammonia as much as possible – likewise having the desuperheaters in parallel enables maximum temperature benefit to be derived from the compressor discharge superheat.

Figure 3 – Schematic of the interconnecting hot and cold water systems for district heating

On the chiller side of the heat pump the evaporators are piped in parallel. Automatic isolating valves, not shown in Figure 4, prevent flow through chillers when the compressors on that unit are not running.

When the system is required to operate on part load the user can configure the mass flows on the hot and cold water circuits to suit the part load requirement. For example if the part load is two-thirds flow across the full temperature difference of 60° C heated to 90° C then two units can run on full capacity to meet the demand. If the load is the full flow heated from 60° C to 80° C then running all three units at part capacity might be more appropriate.

5. FUTURE POTENTIAL

High pressure compressors of relatively large swept volume have now been demonstrated for heat pumps operating on ammonia. A logical next step is to develop expanders capable of running at similar conditions and adapt the technology for use in a Rankine Cycle power generation system. In these plants heat is extracted from a waste stream at relatively high temperature and used to create high pressure gas from a reservoir of pressurized liquid. The high pressure gas passes through an expander which drives an alternator to generate electricity, and the resulting lower pressure gas is then condensed, rejecting the remaining latent heat of condensation to atmosphere, usually through a cooling tower loop. Previous systems have used R-245fa as the working fluid, but it has a toxicology class B and has a global warming potential over 1,000 relative to CO2 on a 100 year horizon, so it is not an ideal fluid.

Favourable characteristics of a suitable working fluid include a low latent heat, a high index of compression, a high critical temperature and a high saturation pressure for a given temperature. As has been shown earlier, ammonia possesses most of these properties – the major disadvantage in this respect is the latent heat. To avoid expansion into the two-phase region of the pressure-enthalpy map it is necessary to ensure that the discharge is significantly superheated. At elevated temperatures the saturated vapour enthalpy reduces as pressure increases, which significantly increases the amount of superheat required. However the isentropic efficiency of the expansion process raises the expander outlet temperature above the isentropic condition, so somewhat offsets the need for superheat. Previous uses of ammonia in power cycles have focused on mixtures of ammonia and water, for example the Kalina cycle which is widely studied, for example Desideri and Gianni (1997), but the addition of water is somewhat problematic. The use of high pressure ammonia liquid (or possibly even transcritical ammonia) in the evaporator (or "gas heater") would offer the potential for improved efficiency without the difficulties of handling a two-component mixture with a wide temperature glide.

6. CONCLUSIONS

Compressors have now been developed in sufficient size and with sufficient pressure rating to make large scale ammonia heat pumps a viable technology. Efficiency of the heat pump is likely to be better than any fluorocarbon system because of the generally favourable physical properties of ammonia. Such systems can be retrofitted to existing ammonia installations, or can be constructed as stand-alone facilities.

The higher pressures required for carbon dioxide transcritical systems, with discharge pressures up to 120 bar, could also be applied to ammonia, pushing the heat recovery temperatures up to even higher levels than seen to date.

When these compressors are available for ammonia there will also be the prospect of adapting them to work as expanders which would enable electrical generation from waste heat to be developed using ammonia. It would be wrong to refer to this as "organic" Rankine cycle since it uses an inorganic fluid, but the principle is the same, without the environmental impact of fluids with high global warming potential.

REFERENCES

Department of Energy and Climate Change, Renewable Heat Incentive: Consultation on the proposed RHI financial support scheme, London, 2010

Desideri, U. and Gianni, B., Study of possible optimisation criteria for geothermal power plants Energy Conversion and Management, Volume 38, Issues 15-17, October-November 1997, Pages 1681-1691

Lorentzen, G., Ammonia: an excellent alternative, Int. J. Refrig. 11, 248-252, 1988

Pearson, A. (ed), Ammonia as a Refrigerant, IIR, Paris, 2008

Stoecker, WF, "Ammonia Carbon Dioxide Hybrid Systems: Advantages and Disadvantages", International Institute of Ammonia Refrigeration, Nashville, 2000

United States Government, Emergency Economic Stabilization Act of 2008, H.R. 1424, Washington DC, 2008

ACKNOWLEDGEMENT

Thanks are due to the directors of Star Refrigeration for funding this study and authorizing the publication of the results.