International Sorption Heat Pump Conference
June 22-24, 2005; Denver, CO, USA
ISHPC-090-2005

A THERMODYNAMIC INVESTIGATION OF THE NH_3-H_2O WORKING PAIR FOR ENHANCED ENERGY EFFICIENCY OF AIR-CONDITIONING EQUIPMENT VIA WET COMPRESSION

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ABSTRACT

The efficiency of the vapor compression refrigeration cycle can be improved upon by adding an absorbent to the refrigerant at a certain concentration. The unevaporated refrigerant-absorbent mixture can be handled via a solution circuit that bypasses the compressor via a pump and an internal heat exchanger. However, the added cost of these components and problems with flow/pressure balancing have prevented widespread adoption of this technology. This has prompted research into the conceptually simpler means of routing the solution through the compressor. The present work takes a look at cycle thermodynamics with the resulting two-phase compression for ammonia-water. The state-point analysis suggests the promise of an improvement in cycle COP of about 14% over the dry pure-ammonia vapor compression case, under typical air-conditioning water temperatures. The improvement comes from a combination of discharge temperature reduction and mixture thermodynamics, with the latter providing over two-thirds of the enhancement via pressure ratio reduction and temperature glide in the phase-change components. The challenge to further establishing the feasibility of the high-efficiency, all-natural-refrigerants chiller lies in the design of the wet compressor.

Keywords: Chiller, Vapor Compression, Absorption, Ammonia-Water, Wet Compression, Pressure Ratio, Temperature Glide.

INTRODUCTION

In the U.S., commercial and residential buildings account for about 39% of the total energy consumption [1]. A significant portion of this – as much as 48% - is attributable to air-conditioning (cooling and heating) equipment. Interest in the energy efficiency of such equipment has grown in recent years, building up to the ratification of the Kyoto Protocol by 141 nations on February 16, 2005 [2]. This is because it has been established that over 96% of the global warming impact of this sector comes from the release of carbon dioxide into the atmosphere as a result of fossil fuel combustion required for power generation [3]. Improving the efficiency of AC equipment results in less power consumption, fossil-fuel combustion and hence greenhouse gas (CO_2) emission.

An earlier, more widely accepted treaty ratified by the U.S. as well was the Montreal Protocol, which lays down schedules for the various nations of the world to phase out chlorofluorocarbon (CFC) refrigerants [4]. With regulatory pressure more recently turning on hydrochlorofluorocarbons (HCFCs), and in Europe, hydrofluorocarbons (HFCs) as well, there is a renewed interest in adopting natural refrigerants such as hydrocarbons and ammonia [5]. These are environment-friendly, with zero ozone depletion and negligible direct global warming potential, and in many instances, with excellent thermodynamic properties yielding high COPs and hence low indirect global potential comparable to if not better than their fluorocarbon counterparts. What remains in the way of their proliferation in AC equipment is concern over the safety of such refrigerants. This relates to the flammability and toxicity of these refrigerants [6].

The combined drivers of high equipment efficiency and low ozone-depletion & direct global warming potential of the refrigerant therein form the impetus of this work. These drivers have placed ammonia in a unique position because of its superior refrigerating properties and environment-friendliness. This is one of the oldest refrigerants still in use, widely prevalent in industrial refrigeration and process cooling applications for well over a century. In addition to high cycle efficiencies, the relatively low specific volume of ammonia offers compressors whose CFM/ton requirements can be an order of magnitude lower than those of its fluorocarbon counterparts. Finally, owing to its relatively high heat of vaporization, ammonia is a low-charge refrigerant. Compounding this with the fact that ammonia is 10 times cheaper than an HCFC like R22 [7] suggests the possibility of very low installed equipment costs.

Ammonia's weakness lies in its safety classification; it has a B2 classification per ASHRAE Standard 34 [6], which indicates a toxic, moderately flammable refrigerant. This has led to poor acceptance in space-conditioning applications, particularly in the U.S.. Materials compatibility is a *design*-related issue that has affected acceptance; copper cannot be used in ammonia systems [8] as the two react to form a deep-blue complex ion dissolving the copper. This restricts the materials of construction to steel, potentially resulting in more expensive componentry, especially when using stainless steel. Recent advancements in heat exchanger technology, however, could suppress this concern, with all-stainless-steel, compact fusion-brazed plate heat exchangers now available in the market [9].

Most ammonia-based refrigeration applications employ the standard vapor compression cycle. Variations on the vapor compression cycle for efficiency enhancement have been investigated by numerous researchers, both theoretically and in the lab [10]. One such variation is the compression-resorption hybrid cycle, which incorporates elements of the absorption cycle in the vapor compression cycle [11]. Thus, refrigerant-absorbent pairs typically suited for absorption are used here as working fluids. These tend to be environmentally benign, often consisting of natural refrigerants such as ammonia and water. Large efficiency gains have been reported with these fluids in various hybrid cycle configurations, but these are mostly for high-lift applications such as industrial heat-pumping [10, 11]. Few of these focus on the gains possible at conventional air-conditioning temperatures.

An attempt was made previously via cycle thermodynamics to illustrate the advantages of the basic hybrid cycle with a separate solution circuit [12]. Three working fluid pairs, consisting of an HCFC (R22), an HFC (R134a) and a natural refrigerant (ammonia) with a corresponding suitable absorbent each were considered. Efficiency enhancements in the order of 4-7% relative to the pure refrigerant case were projected. The basis for the improved efficiency was threefold: compressor head pressure reduction, temperature glide in the condenser and evaporator and an incremental heat of solution/mixing added to the normal heat of vaporization in the evaporator. However, problems were identified that may be encountered during actual operation of such systems. These include pump-compressor synchronization / flow balancing [13]. Also, the results of this study hinged on the ability to achieve uniform distribution of even small amounts of liquid solution in the condenser, and hence instantaneous vapor-liquid equilibrium. Last but not least, the impact of an extra solution pump, heat exchanger and vapor-liquid separator on cost and size could not be ignored.

The encouraging results obtained above, particularly for the natural-refrigerant based system, coupled with reliability and cost considerations for commercially manufactured product prompted the next phase of this work, presented in this paper. The pump, heat exchanger and separator were eliminated from the ammonia-water hybrid cycle, and the two-phase solution exiting the evaporator/generator was introduced directly in the suction port of a compressor capable of handling wet compression. Such compressors have been designed and tested in laboratories [11], such as the modified air compressor developed by Infante-Ferreira and his team at Delft University [14]. In any case, screw compressors typically handle as much as 11% liquid (mostly oil) by mass through the compression process. Such two-phase mixtures turn out not to be very removed from those suggested by this analysis, strengthening the case for this technology.

SYSTEM DESCRIPTION

A schematic of the wet compression variation of the hybrid cycle is shown in Fig. 1. As pointed out, at a system level, it is a substantially simpler cycle than the hybrid cycle with a solution circuit. The complexity, however, is not entirely eliminated but pushed down to the component level, specifically in the compressor design. For example, rotor lubrication of such a compressor could be accomplished with the process fluid (here, ammonia-water solution). The rotors themselves may then require lower-torque profiles, larger clearances and possibly specially coated surfaces [14].

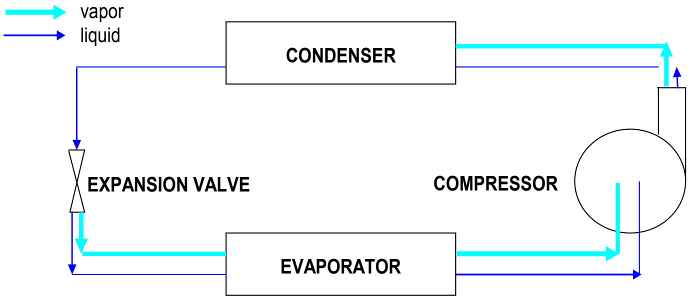


Figure 1: Vapor Compression Cycle with Binary Working Fluid and Wet Compression

As with the hybrid cycle with a solution circuit, the refrigerant-absorbent mixture leaves the evaporator as a two-phase mixture. The liquid phase consists of varying concentrations of ammonia in aqueous solution, depending on the operating conditions, while the vapor phase consists predominantly of ammonia, which is more volatile than water. Unlike other refrigerant-absorbent pairs, however, vapor-liquid equilibrium properties of the non-ideal mixture dictate that traces of water also pass into the vapor phase, which could affect compressor design and performance.

In contrast to the solution circuit variation, the compression of the two-phase, two-species mixture occurs with the liquid and vapor phases in constant contact and well-mixed with each other, i.e. not removed from equilibrium. This has a twofold advantage: the two phases are in constant thermal contact with each other, resulting in lower discharge temperatures, and they undergo constant species transport, resulting in near-equilibrium discharge concentrations. A fall-out of this near-equilibrium heat and mass transfer through the compression process is reduced entropy generation, relative to the post-compression mixing of the two phases, out of thermal and species equilibrium with each other, in the case of a separate solution circuit. This second-law efficiency gain is over and above the first-law gains of reduced pressure ratio and increased refrigerating effect observed in the latter [12].

Assumptions

The following assumptions were made for the analysis of the wet-compression variation of the hybrid cycle:

1. As with the solution-circuit variation, the condenser/absorber and evaporator/generator "UA"s are each held constant at baseline values established for the pure (single-component) refrigerant case. This was done to "lock in" the component hardware and eliminate its effect on system performance. A further justification for this would be the fixed heat capacities (mass flow rate times specific heat) held for the water side, also allowing a comparison strictly between refrigerant-absorbent pair capabilities. It must be kept in mind that the performance characteristics (solution-side heat/mass transfer coefficients) of

these components are unlikely to be as good as those of the same hardware when operating purely as a condenser or evaporator, resulting in larger / more expensive surface areas.

2. The performance of the compressor is also “frozen”, for the above reason. As before, the compressor isentropic efficiency is fixed at 90%. To a limited extent, a more realistic, 75% scenario is also considered. The expansion process through the throttle valve is adiabatic and isenthalpic.

3. Constant vapor-liquid equilibrium through the compression process, as mentioned above. In actuality, there might be some vapor-liquid separation, depending on the entering quality and loading of the compressor. From a practical standpoint, the potentially high local pressures in the rotors due to the incompressibility of the liquid solution must be borne in mind during compressor design.

4. Finally, when both components are present in the vapor phase (e.g. in the case of ammonia-water), the mixture is assumed to be ideal, i.e. the sum of the contributions of the individual enthalpies gives the enthalpy of the vapor mixture.

The cooling tower water always enters the unit at 85°F, while chilled water supply is maintained at 44°F.

Non-Linear Temperature Glide

For refrigerant blends with a more linear temperature glide (e.g. R407C), an overall log-mean-temperature-difference (LMTD) can be calculated given the entering and leaving temperatures on both sides of the heat exchanger. The ammonia-water working pair, however, exhibits a highly non-linear glide during phase change. This is because of the relatively close volatilities of ammonia and water and their strong mutual affinity yielding a non-ideal solution [15]. This impacts the calculation of the LMTD, and hence the fixed “UA”, in the present analysis. Simply using the entering and leaving temperatures on the refrigerant side would violate the definition of LMTD, which assumes a linear temperature vs. heat transferred distribution. An iterative procedure is developed to discretize this distribution into linear intervals, apply the LMTD definition over each interval and integrate over the entire heat exchange (Fig. 2). The resulting overall temperature difference is then the real LMTD.

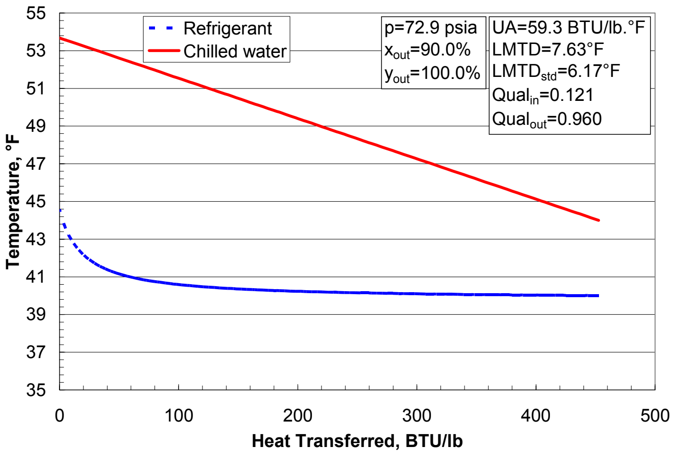


Figure 2(a): Typical (Computed) True Temperature Distributions in Two-Component Evaporator: Non-Linear Glide of Ammonia-Water

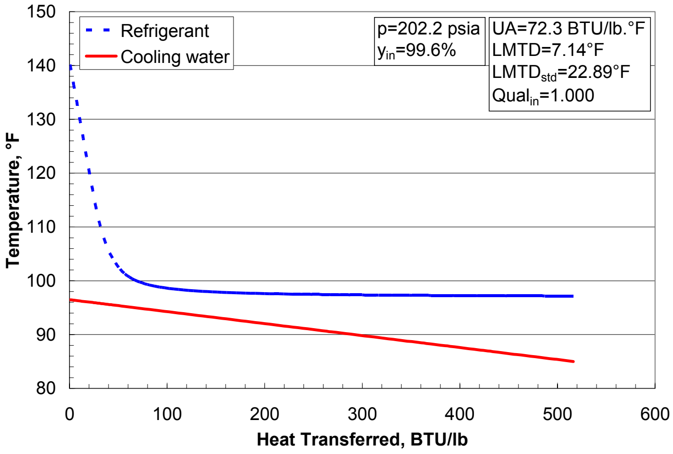
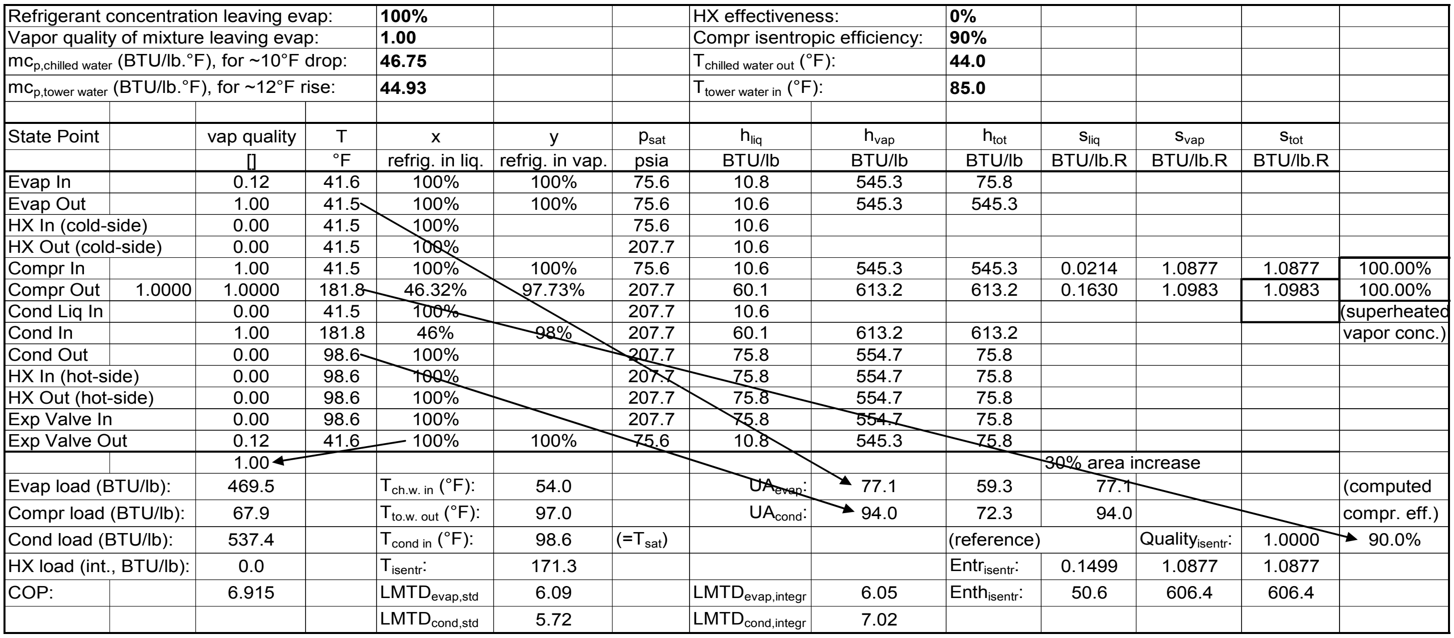


Figure 2(b): Typical (Computed) True Temperature Distributions in Two-Component Condenser: Non-Linear Glide of Ammonia-Water

Computational Procedure

The cycle computations were carried out as a spreadsheet state-point analysis with iterative solution. A snapshot of this is provided in Fig. 3. The formulae in the spreadsheet were derived from mass, species and energy balances around the different components, flow splits and junctions, as well as from thermodynamic (equilibrium) properties where applicable.



Refrigerant concentration leaving evap:				**100%**			HX effectiveness:		**0%**				
Vapor quality of mixture leaving evap:				**1.00**			Compr isentropic efficiency:		**90%**				
$mc_{p,\text{chilled water}}$ (BTU/lb.°F), for ~10°F drop:				**46.75**			$T_{\text{chilled water out}}$ (°F):		**44.0**				
$mc_{p,\text{tower water}}$ (BTU/lb.°F), for ~12°F rise:				**44.93**			$T_{\text{tower water in}}$ (°F):		**85.0**				
State Point		vap quality	T	x	y	p_{sat}	h_{liq}	h_{vap}	h_{tot}	s_{liq}	s_{vap}	s_{tot}	
		[]	°F	refrig. in liq.	refrig. in vap.	psia	BTU/lb	BTU/lb	BTU/lb	BTU/lb.R	BTU/lb.R	BTU/lb.R	
Evap In		0.12	41.6	100%	100%	75.6	10.8	545.3	75.8				
Evap Out		1.00	41.5	100%	100%	75.6	10.6	545.3	545.3				
HX In (cold-side)		0.00	41.5	100%		75.6	10.6						
HX Out (cold-side)		0.00	41.5	100%		207.7	10.6						
Compr In		1.00	41.5	100%	100%	75.6	10.6	545.3	545.3	0.0214	1.0877	1.0877	100.00%
Compr Out	1.0000	1.0000	181.8	46.32%	97.73%	207.7	60.1	613.2	613.2	0.1630	1.0983	1.0983	100.00%
Cond Liq In		0.00	41.5	100%		207.7	10.6						(superheated
Cond In		1.00	181.8	46%	98%	207.7	60.1	613.2	613.2				vapor conc.)
Cond Out		0.00	98.6	100%		207.7	75.8	554.7	75.8				
HX In (hot-side)		0.00	98.6	100%		207.7	75.8	554.7	75.8				
HX Out (hot-side)		0.00	98.6	100%		207.7	75.8	554.7	75.8				
Exp Valve In		0.00	98.6	100%		207.7	75.8	554.7	75.8				
Exp Valve Out		0.12	41.6	100%	100%	75.6	10.8	545.3	75.8				
		1.00								30% area increase			
Evap load (BTU/lb):		469.5		$T_{\text{ch.w. in}}$ (°F):	54.0		UA_{evap}:	77.1	59.3	77.1			(computed
Compr load (BTU/lb):		67.9		$T_{\text{to.w. out}}$ (°F):	97.0		UA_{cond}:	94.0	72.3	94.0			compr. eff.)
Cond load (BTU/lb):		537.4		$T_{\text{cond in}}$ (°F):	98.6	(=T_{sat})			(reference)		Quality_{isentr}:	1.0000	90.0%
HX load (int., BTU/lb):		0.0		T_{isentr}:	171.3				Entr_{isentr}:	0.1499	1.0877	1.0877	
COP:		6.915		$\text{LMTD}_{evap,std}$	6.09		$\text{LMTD}_{evap,integr}$	6.05	Enth_{isentr}:	50.6	606.4	606.4	
				$\text{LMTD}_{cond,std}$	5.72		$\text{LMTD}_{cond,integr}$	7.02					

Figure 3: Fixed Hardware (Glide-Advantage) Analysis; Zero Liquid Enthalpy & Entropy at 32°F (0°C)

As with the solution-circuit analysis, the two input variables of the analysis pertained to the exit refrigerant condition from the evaporator: vapor quality, and solution concentration of the liquid phase (i.e. concentration of refrigerant in the refrigerant-absorbent liquid mixture). The direct output variables were the missing information at any given state point (i.e. temperature, pressure, concentration, enthalpy, and in the case of the compressor, entropy). The indirect output variables, estimated from the calculated state points, were the component loads, coefficient of performance, and the leaving and entering tower- and chilled-water temperatures, respectively.

RESULTS & DISCUSSION

Pure Ammonia

To establish a benchmark, calculations were first performed for the pure ammonia case with fixed low- and high-side saturation temperatures. This amounted to variable hardware (UA's), as different extents of wet compression, given by the varying vapor quality exiting the evaporator, were considered. The effect on cycle efficiency is shown in Fig. 4.

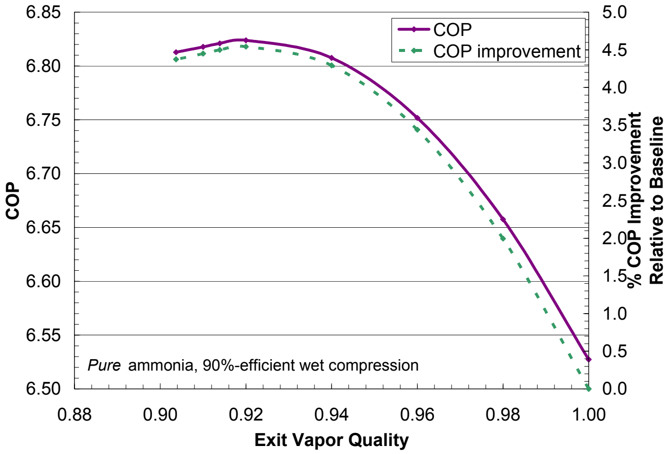


Figure 4: COP Variation & Improvement with Evaporator Exit Vapor Quality for Pure Ammonia (Wet-Compression Simple Cycle)

It is evident from this plot that there exists an optimum "wetness" yielding the highest boost in COP, 4.5% over the baseline case (refrigerant leaving the evaporator at 100% quality, 0°F superheat). This occurs at about 0.92 quality. As the quality decreases from unity to this value, the compressor vapor discharge temperature is brought down toward saturation, and the cycle progressively resembles the Carnot cycle lying entirely within the saturation dome. The efficiency benefit with this adiabatic cooling, however, increases at a decreasing rate, governed by the relation:

$$COP = \frac{1}{\dfrac{Q_{cond}}{Q_{evap}} - 1} = \frac{1}{\dfrac{T_{sat,cond}.\Delta s_{cond} + \displaystyle\int_{T_{sat,cond}}^{T_{discharge}} T.ds}{T_{sat,evap}.\Delta s_{evap}} - 1}$$

The decline in benefit below the 0.92 quality is due the fact that the loss of refrigerating effect with reduced quality increases in greater proportion than the drop in heat rejection. Below a quality of 0.90, the refrigerant leaves the compressor saturated.

Interestingly, the COP of the wet-compression cycle with a fluorocarbon such as R123 actually *drops* with decreasing suction quality. This is due to the (gentler) slope of T-s or p-h curve in the superheated region.

Ammonia-Water

Next, water was added to the ammonia refrigerant in varying concentrations and, as with the solution-circuit -based cycle analysis [12], calculations were performed assuming unchanging hardware performance (UA's). Consistent with the previous analyses, a 90% compressor isentropic efficiency was assumed.

The effect of the varying vapor qualities and liquid concentrations reveal optimum values for the latter (Fig. 5), yielding absolute COP enhancements as high as 14.6%. (The absolute enhancement references back to the same baseline case as for the pure ammonia case.)

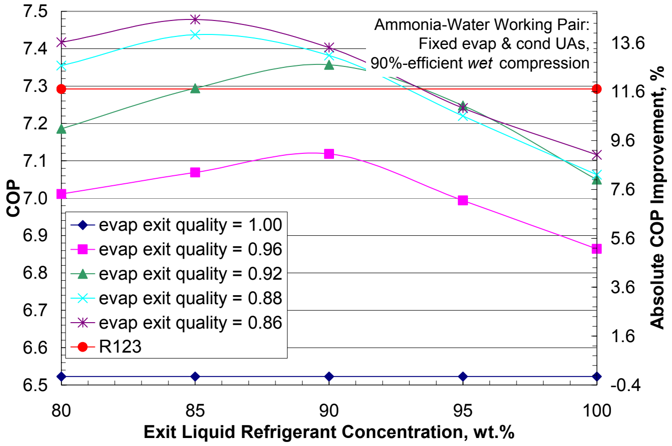


Figure 5: COP Variation & Improvement with Evaporator Exit Vapor Quality & Liquid Refrigerant Concentration (Wet-Compression Hybrid Cycle)

For a fair comparison with the solution-circuit –based cycle, and to separate out the benefits of wet compression, the relative enhancements are shown in Fig. 6. (The relative enhancement references back to the performance obtained for pure refrigerant at a given quality.)

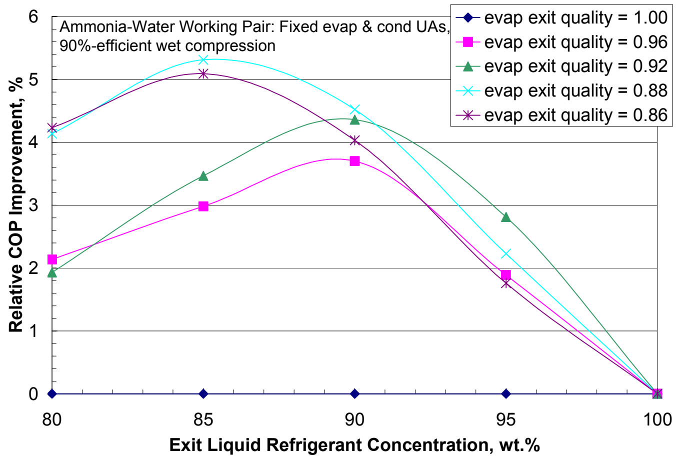


Figure 6: COP Improvement Variation with Evaporator Exit Vapor Quality & Liquid Refrigerant Concentration (Relative to Pure Refrigerant at Given Quality)

The cycle performance enhancement seen here is of a similar order as that with the solution circuit. The ~1% improvement here is partly the result of the continuous direct contact, and hence species equilibrium, between the vapor and liquid through the compression process, as opposed to the instantaneous adiabatic equilibrium assumed at the pre-condenser vapor-liquid junction in the solution circuit case. Some of the performance difference also comes from the more accurate present treatment of LMTD in the two-species phase-change components. The results point to an optimum liquid concentration of 85% and an optimum vapor quality of 0.88 leaving the evaporator. As before, the reduced pressure ratio is the biggest factor in the efficiency boost (Fig. 7).

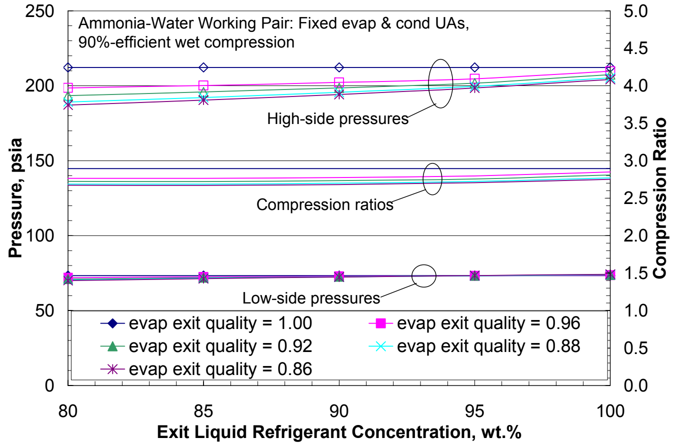


Figure 7: Pressure and Compression Ratio Variation with Evaporator Exit Vapor Quality & Liquid Refrigerant Concentration

An alternative view of both absolute and relative performance improvement is shown in Fig. 8. For the range considered here, the effect of decreasing quality on the improvement is different from the solution-circuit case, even for pure ammonia. This can again be attributed to the different handling of the vapor-liquid mixture entering the condenser (approach and entry into the saturation dome), as well as the presence of an internal heat exchanger in the solution circuit. The extra inflection in the presence of water comes from the condensation of a non-ideal mixture.

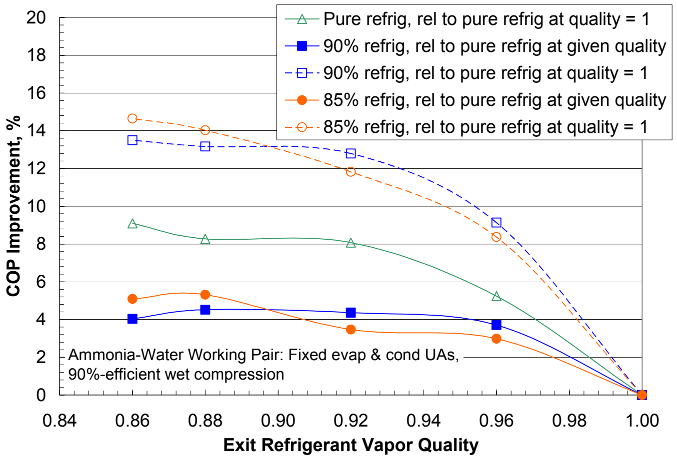


Figure 8: COP Improvement Variation with Evaporator Exit Vapor Quality for Optimum Refrigerant Concentrations

To confirm that the different mixing models at the condenser entrance impact performance only outside the saturation dome, the analysis was extended down to lower qualities for pure ammonia. As shown in Fig. 9, behavior is not unlike the solution-circuit case past the hump-kink due to dome entry (0.86-0.92 quality), i.e. the COP does rise monotonically with increasing "wetness" (up to a point), due to progressive unloading. This would, however, be at the expense of higher "UA/ton" or first cost, as one might expect.

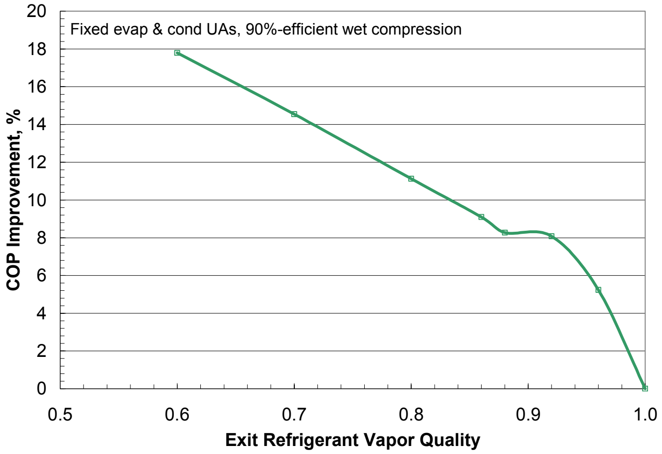


Figure 9: COP Improvement Variation with Evaporator Exit Vapor Quality for Pure Ammonia

The "UA/ton" is the capability of a heat (and mass) exchanger for a given cooling capacity and provides a measure of its specific first cost. The reduction in requirement of this capability with the addition of water is evident from Fig. 10 for both the evaporator and the condenser. Also illustrated is the underutilization of a given piece of hardware in part-load. Thus, the relative increase in the "UA/ton" requirement for the evaporator from the baseline to the best-case scenario is 12.7%.

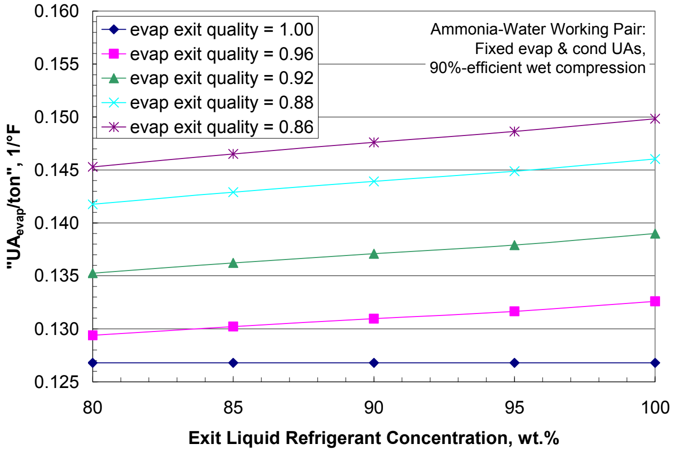


Figure 10(a): Evaporator "UA/Ton" Variation with Exit Vapor Quality & Liquid Refrigerant Concentration

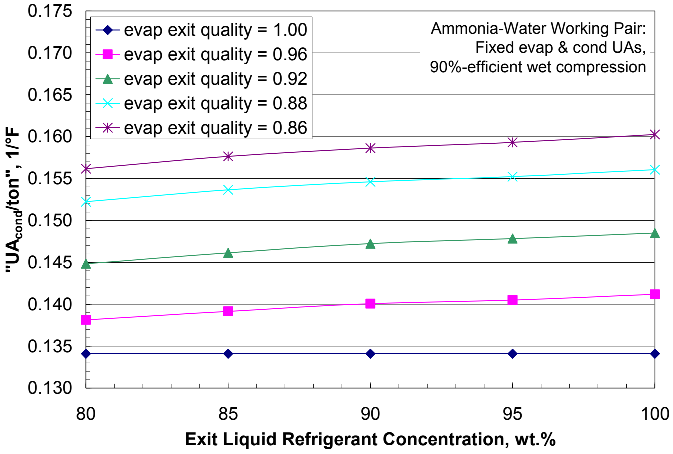


Figure 10(b): Condenser "UA/Ton" Variation with Evaporator Exit Vapor Quality & Liquid Refrigerant Concentration

Comparison to Fluorocarbons

Holding the "UA/ton" constant for both components takes out the effect of varying load on efficiency. In a per unit lb/hr analysis such as this, the "UA/ton" translates to simply "1/LMTD" in the case of the evaporator and the inverse of a representative driving temperature difference/approach in the case of the condenser. The final stage of this analysis involved extending the assumption of fixed "UA" to one of fixed "UA/ton" for the two components, to see the impact of load normalization. This was done in the context of comparison with ideal-cycle (40/100 °F sat.) performance of R123, for a more representative compressor efficiency of 75%. 90%-efficiency results have been restricted to the ideal cycle analysis, to show the impact of this parameter. Limited comparison with R134a performance is also provided. The comparisons are presented in Table 1.

Table 1: Relative Cycle Performance

40°F/100°F SATURATED EXIT

90% Compressor Isentropic Efficiency

Refrigerant	COP	% Improv.
R123	6.692	0
R717	6.527	-2.5
R134a	6.248	-6.6

75% Compressor Isentropic Efficiency

Refrigerant	COP	% Improv.
R123	5.577	0
R717	5.439	-2.5
R134a	5.206	-6.7

With wet compression (Qual$_{\text{evap exit}}$=0.92):

R717	5.687	2.0

FIXED UA/ton (=1/LMTD)

With wet compression (Qual$_{\text{evap exit}}$=0.92):

R717	5.741	2.9

With wet compression (Qual$_{\text{evap exit}}$=0.88) and water (x$_{\text{evap exit}}$=0.85):

R717	6.031	8.1

Based on the ideal cycle comparisons, compressor efficiency has minimal effect on relative deviation from the COP of R123. Also, it takes wet compression for ammonia to thermodynamically outperform R123.

Based on the analysis assuming fixed hardware for a given cooling capacity (fixed "LMTD"s), the ammonia wet compression cycle is 3% more efficient than the standard R123 ideal cycle. Introducing water in the former, in the right proportions (nearly 2% overall circulation rate) gives a boost of up to 8% over R123 efficiency.

Property Variation and Enhancement Mechanisms

A variety of sources exist for the thermodynamic properties of ammonia. However, these are not always in agreement, and their usage can influence analysis results. The ideal cycle analysis without and with wet compression was carried out using the properties of Haar & Gallagher [16], Van Wylen et al. [17] and Phillips Engineering [18]. The last was curve-fit from the data of Gillespie et al. [19] and also the source of the ammonia-water mixture properties, collapsing to pure ammonia properties in the absence of water. Table 2 shows the comparisons.

Table 2: Relative Cycle Performance Improvement Depending on Ammonia(-Water) Properties Source; 40°F/100°F Saturated Exit (Unless Otherwise Noted), 90% Compressor Isentropic Efficiency

A. PURE AMMONIA, SATURATED VAPOR AT EVAP EXIT (DRY COMPRESSION)

Quality	Data Source	COP
1.00	Haar & Gallagher (Trane routines)	6.527
	Van Wylen & Sonntag (computer)	6.528
	Phillips Engineering (mixture rout.)	6.523

B. PURE AMMONIA, SATURATED VAPOR-LIQUID AT EVAP EXIT (WET COMPRESSION)

Quality	Data Source	COP	
0.88	Haar & Gallagher (Trane routines)	6.793	
	Van Wylen & Sonntag (computer)	6.799	
	Phillips Engineering (mixture rout.)	6.718	
	Phillips Engineering (mixture rout.)	7.062	with fixed "UA"

C. AMMONIA-WATER*, SATURATED VAPOR-LIQUID AT EVAP EXIT (WET COMPRESSION)

Quality	Data Source	COP	
0.88	Phillips Engineering (mixture rout.)	5.913	
0.88	Phillips Engineering (mixture rout.)	7.438	with fixed "UA" (glide advantage)

* 85 wt.% ammonia in liquid component of mixture exiting evap.

RELATIVE IMPROVEMENTS

Data Source	Wet improve- -ment (%)	Mix. + Glide improve- -ment (%)	Mixture improve- -ment (%)	Glide improve- -ment (%)	Overall improve- -ment (%)
Haar & Gallagher (Trane routines)	4.1	9.5	-13.0		14.0
Van Wylen & Sonntag (computer)	4.2	9.4	-13.0		13.9
Phillips Engineering (mixture rout.)	3.0	10.7	-12.0	25.8	14.0

For the ideal cycle, the differences in results are statistically insignificant, although the Phillips properties lead to more conservative results, especially away from dry compression. Using a given set of properties, the fixed-UA analysis, by allowing the saturation temperatures to float, yields higher COPs than the ideal cycle analysis. This effect is accentuated substantially in the case of the ammonia-water mixture, owing to the added advantage of temperature glide and counterflow heat exchange in the condenser and the evaporator.

A breakdown of the different effects contributing to the COP enhancement has been attempted for the three property sources. The overlap between these effects must be recognized, preventing a linear superposition of the corresponding increments in COP from yielding the overall enhancement. Regardless of the property source for pure ammonia (Phillips was the sole source adopted for mixture properties), the overall improvement going to a wet compression cycle that includes small amounts of water was ~14%, at the optimal concentration and quality. 20-30% of this is the consequence of the second-law advantages of wet compression.

RISK ASSESSMENT

Apart from compressor modifications to handle liquid ammonia solution, which might be the biggest challenge of this work, the two-phase components could show lower refrigerant-side heat transfer coefficients (therefore "U" values) for mixture condensation / absorption and evaporation / desorption relative to pure-refrigerant phase-change. This is because of the additional mass transfer resistance due to the presence of the second species (water). The potential shortfall in "U" would have to be compensated for with larger components (and hence more cost), if the "UA" is to remain the same.

Counterflow heat exchange in both phase-change components of the proposed system is essential to reap the full benefit of the hybrid cycle. This can be accomplished with a plate heat exchanger for the evaporator (brazed-aluminum or welded-steel), but might prove more challenging for an air-cooled condenser, the standard coil-and-fin configuration being *crossflow*. If the condenser must be crossflow, the COP enhancement is ~3% relative to the pure-ammonia cycle with the same UAs and suction quality (88%).

Another point of concern is the assumption of constant compressor efficiency. How is the efficiency actually impacted by the presence of liquid? How does it vary with pressure ratio (specifically, over the 2.67-2.89 range) as the absorbent concentration varies? If the efficiency is affected adversely by these two conditions, a further reduction in benefit could be expected.

FUTURE WORK

As one might expect, in addition to validation of the analysis, there are a number of practical questions pertaining to wet compression that need to be addressed with a proof-of-concept system. Although ammonia screw compressors are widely used for commercial & industrial refrigeration applications, can a standard mono- or twin-screw ammonia compressor handle 12% by mass of ammonia-rich liquid solution (85 wt.% NH_3) without losing lubrication via the dilution or emulsification of the oil? How will oil separation from this solution be accomplished after discharge from the compressor? Will the solution be adequate for rotor lubrication, if the bearing lubrication is effected separately with conventional oil? This would be a means to obviate oil separation. Issues like these form the impetus for future hardware-oriented work. Finally, current discharge superheat - based refrigerant flow controls would not work with wet compression; alternative algorithms would have to be developed.

At the very least, the proof-of-concept testing could demonstrate whether the COP improvement is masked by UA and/or compressor efficiency changes with absorbent concentration and/or vapor quality. The behavior of these two parameters with the two main independent variables can then be mapped.

CONCLUSIONS

The vapor compression - resorption hybrid cycle, where the solution circuit is combined with the vapor stream emerging from the evaporator and sent through the compressor resulting in wet compression, has been analyzed thermodynamically. The analysis was carried out for ammonia-water at ARI-type water-to-water air-conditioning conditions and indicated a maximum total enhancement of about 14% over the COP of the standard (dry) pure-ammonia vapor compression cycle. The contribution of the non-ideal mixing of ammonia and water through reduced pressure ratio, negative heat of solution (increasing the heat of vaporization in the evaporator) and temperature glide in the condenser and evaporator is over two-thirds of this, i.e. 10% enhancement, with the remaining 4% coming from 2nd-law gains such as the lowering of the discharge temperature (and hence enthalpy, entropy) due to the presence of liquid in the compressor.

Analytical comparison with COPs of other fluorocarbon-based systems with comparable UA/ton operating at water-cooled conditions shows that an ammonia system can exceed the efficiency of an R123 system by almost 3% with wet compression and 8% with wet compression of a mixture with absorbent water.

The present analysis is based on proven property data and cycle analysis methods. The results are sufficiently accurate to assess the COP benefits of the vapor compression - resorption hybrid cycle with wet compression and compare it to other vapor compression cycles using ammonia and alternative refrigerants. However, hardware challenges remain, the most prominent of these being the design of a compressor capable of handling significant amounts of liquid without compromising its integrity (localized high pressures) and reliability (diluted lubrication).

A proof-of-concept test is the logical next step, to verify the presented thermodynamics and provide a starting point for hardware design, including the heat exchangers, towards a commercializable air-conditioning technology that is sustainable.

ACKNOWLEDGEMENTS

The author wishes to thank Paul Glamm and Bill Plzak of Trane, for insightful discussions that shaped some of this work, as well as Carlos Infante Ferreira of Delft University of Technology, for dialogue that shed light on practical implementation.

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