

2004

# An Investigation into the Feasibility of the Use of Water as a Refrigerant

Frandon F. Lachner

*University of Wisconsin-Madison*

Gregory F. Nellis

*University of Wisconsin-Madison*

Douglas T. Reindl

*University of Wisconsin-Madison*

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

---

Lachner, Frandon F.; Nellis, Gregory F.; and Reindl, Douglas T., "An Investigation into the Feasibility of the Use of Water as a Refrigerant" (2004). *International Refrigeration and Air Conditioning Conference*. Paper 723.

<http://docs.lib.purdue.edu/iracc/723>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact [epubs@purdue.edu](mailto: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>

# AN INVESTIGATION INTO THE FEASIBILITY OF THE USE OF WATER AS A REFRIGERANT

Brandon F. LACHNER<sup>1</sup>, Gregory F. NELLIS<sup>1</sup>, Douglas T. REINDL<sup>1</sup>

<sup>1</sup>University of Wisconsin, Solar Energy Lab  
Madison, WI, USA  
(608) 263-1592, bflachner@wisc.edu

## ABSTRACT

This paper summarizes the results of a 21CR research project funded by the Air-Conditioning and Refrigeration Technology Institute (ARTI) that investigated the economic feasibility of a water-based vapor compression chiller with a nominal capacity of 1000 tons. Presented are various potential cycle configurations and results of simulations for those cycle configurations. The simulations were performed using component level models developed to accurately size equipment and predict system performance for the most attractive cycle configuration. These component models address issues that are particularly crucial in water as refrigerant cycles, such as compressor discharge superheat and refrigerant side pressure drop. Where possible, these component models were verified through comparison against the current state-of-the-art technology for large chillers. The capital cost and the expected operating costs are determined to quantify the payback time associated with using water as a refrigerant relative to traditional halocarbon refrigerants currently in use.

## 1. INTRODUCTION

The use of water as a refrigerant in a vapor compression chiller does have several attractive attributes. Water is environmentally friendly, low cost, and non-toxic. The thermophysical properties of water are consistent with a vapor compression chiller system that has the potential to achieve a high coefficient of performance (COP). The possibility of using water as both the primary refrigerant and secondary system working fluid offers the potential to implement direct-contact heat exchangers. Direct-contact heat exchangers are both less expensive and can yield improved performance over their indirect heat exchanger counterparts.

Along with the above-mentioned advantages, a water-based vapor compression chiller does have several challenging characteristics. As a point of reference, Table 1 shows that water can compete with R134a, in terms of its theoretical COP; however, Table 1 also highlights some of the key differences between water (R-718) and R-134a operating between conditions (i.e. saturation temperatures) typically encountered in large tonnage chillers. The major differences are related to cycle parameters such as peak temperature, pressure ratio, operating pressures, and volumetric flow rates.

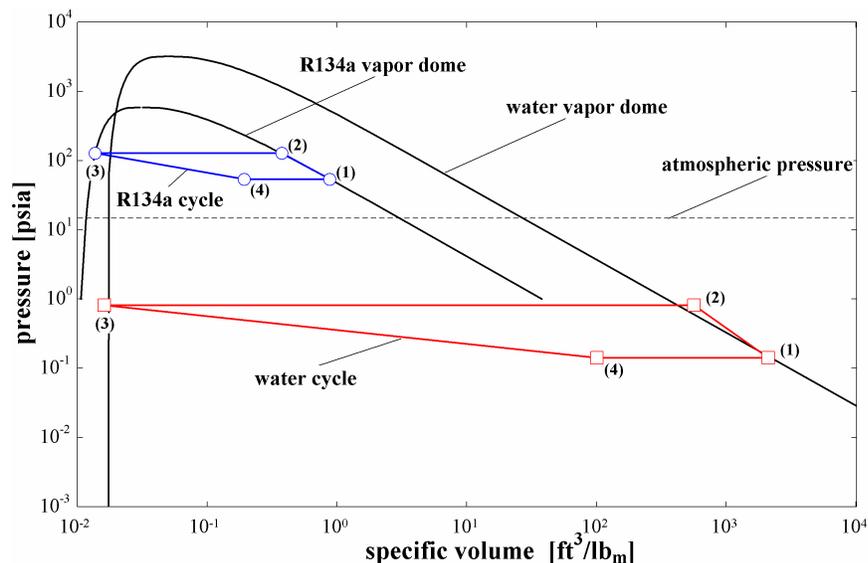
Figure 1 shows a pressure-volume diagram for the ideal R-134a and water vapor compression cycles. This figure illustrates the several problems with a water-based vapor compression cycle that must be addressed. One of the challenges lies in the compression process itself. Water has an extremely high specific volume at normal suction conditions for chiller applications and also requires a large compression ratio to reject heat from the cycle. The superheat loss incurred in the water cycle is nearly two orders of magnitude larger than in the R-134a cycle. On the other hand, Table 1 indicates that throttling loss in a water cycle is relatively low so that the benefit associated with any work extracting expansion device is negligible.

The combination of the large superheat loss and pressure ratio requirements suggest that a multi-stage compression cycle with intercooling may be the most practical and economically-viable alternative. Therefore, various cycle configurations with different methods of intercooling were screened using low-level models. The most attractive cycle configuration is modeled in more detail with component-level models, as described in the second section of the paper. Detailed component models are necessary in order to realistically assess performance and develop relationships between component sizing alternatives and life-cycle costs. A similar analysis of a more conventional R-134a cycle is carried out to provide a meaningful basis of comparison.

**Table 1** Characteristics of water v. R-134a refrigeration cycle

Quantity	Water	R-134a
Theoretical coefficient of performance, COP	8.39	8.47
Carnot Coefficient of Performance, $COP_{Carnot}$		9.88
$COP/COP_{Carnot}$	0.85	0.86
Compressor discharge temperature, $T_{2s}$	313°F	101°F
Throttling loss, $(s_4-s_3)/Q$	$4.7e-6 R^{-1}$	$3.0e-5 R^{-1}$
Superheat loss, $S_{gen,sh}/Q^1$	$2.83e-4 R^{-1}$	$9.0e-6 R^{-1}$
Compressor inlet specific volume, $v_1$	2111 ft <sup>3</sup> /lbm	0.88 ft <sup>3</sup> /lbm
Compressor suction pressure, $p_1$	0.14 psia	54 psia
Compressor saturated suction temperature, $T_{1,sat}$		44°F
Compressor discharge pressure, $p_2$	0.82 psia	129 psia
Compressor saturated discharge temperature, $T_{2,sat}$		95°F
Compression ratio, PR	5.75	2.4
Pressure lift, $p_2-p_1$	0.7 psia	75 psia
Compressor inlet volumetric flow rate	6900 ft <sup>3</sup> /s	45 ft <sup>3</sup> /s
Isentropic enthalpy difference across compressor, $\Delta h_s$	121 Btu/lbm	7.8 Btu/lbm

1.  $S_{gen,sh}$  is the entropy generated by the irreversible de-superheat process



**Figure 1:** Pressure-specific volume diagrams of ideal water and R-134a vapor compression refrigeration cycles.

## 2. CYCLE SELECTION

Multi-stage compression will be required for a water as refrigerant cycle due to the large pressure ratio and high volumetric flow rate requirements. Therefore, the implementation of intercooling may not add significantly to cycle complexity but does provide substantial gains in system COP, as evidenced by Table 1. Therefore, the focus of the cycle selection study was on comparing a non-intercooled cycle with a number of different cycle configurations capable of providing intercooling, including: economized cycle, flash intercooled cycle, direct intercooled cycle, and indirect intercooled cycle. A liquid subcooled cycle was also investigated. These cycle configurations are compared against the non-intercooled, two-stage compression cycle shown in Figure 6. A schematic of some of these cycle configurations are shown in Figures 2 through 5. All of the cycles are modeled using the Engineering Equation Solver (EES) software (Klein and Alvarado, 2004). Component modeling for these screening models

consists of single-value performance metrics including a polytropic compression efficiency<sup>1</sup> (assumed to be 70%) and heat exchanger approach temperatures (assumed to be 0°C for the evaporator and 0.5°C for the condenser). The use of a polytropic rather than isentropic compression efficiency ensures that the overall performance of the compression process was not influenced by the number of compression stages; thereby masking true changes in the system performance related to intercooling between stages.

Figure 6 illustrates the resulting COP for several of the cycle configurations as a function of the number of compression stages. Also shown in Figure 6 is the COP of a non-intercooled R-134a cycle modeled using the same performance metrics. The economized configuration shown in Figure 3 works well in an R-134a cycle but provides very little intercooling in a water cycle. In an economized cycle, the two-phase fluid leaving the expansion valve is separated by phase: the vapor is mixed with the compressor exit stream (superheated steam) in order to provide some cooling while the intermediate pressure/temperature liquid is passed on to the next expansion valve in series. Because the throttling loss in the water cycle is so small (see Table 1), the amount of flash gas produced by each expansion is small and therefore there is very little intercooling realized between stages.

Flash intercooling (Figure 4) uses the direct injection (e.g. via spray) of liquid water into the superheated vapor produced between compression stages. Liquid water from the condenser exit is expanded into the compressor discharge so that the heat of compression is absorbed by the process of vaporizing the throttled liquid. The benefit of this cycle configuration is that the temperature of the gas between stages can be reduced dramatically, in theory to the saturation temperature. The drawback of the flash intercooled cycle is that the mass flow rate passing through subsequent compressor stages increases; however, Figure 6 illustrates that the COP of the flash intercooled cycle is significantly improved relative to the non-intercooled cycle. The degree of improvement diminishes with the number of compression stages but a large increase is seen with even a two-stage compression process.

In an indirect intercooled cycle (Figure 5), the intercooling is provided using an indirect contact heat exchanger that cools the intermediate pressure vapor using the condenser cooling water. The drawback of this cycle configuration is that the temperature of the vapor entering the subsequent compressor stage can only be cooled to the condenser supply water temperature plus some approach temperature. Also, a significant amount of condenser water must be used because it is not being evaporated, as in the flash intercooled cycle, and therefore the heat rejection results only in a sensible energy gain. The benefit of this cycle configuration is that the downstream compressors do not have to process additional refrigerant mass as in the flash intercooled cycle. Figure 6 shows that these two effects approximately cancel so that the COP of the indirect intercooled cycle is nearly the same as that of the flash intercooled cycle. In a water cycle, the use of the indirect intercooler presents some significant practical difficulties including added pressure loss between compression stages as well as additional capital costs for the heat exchangers. Although pressure drop is neglected in this simplified analysis, more detailed models show that the pressure drop is a driving consideration for system design due to the combination of large volumetric flow rates and low absolute cycle pressures and pressure lift.

The direct intercooled and liquid subcooled cycles were found to be impractical for a water cycle and therefore the performance of these cycles are not presented in Figure 6. Any significant amount of standing water (>1") is associated with a hydrostatic pressure rise that is large relative to the overall pressure lift for the cycle. Therefore, the COP penalty associated with standing water eliminates direct and liquid subcooled cycles from consideration.

The baseline configuration selected for this project is the flash intercooled cycle. This cycle is analyzed in more detail by integrating the component-level models, described in Section 3, with the thermodynamic system models in order to allow an economic analysis to be carried out, described in Section 4. Issues that are not captured by this analysis but still important when considering water as a refrigerant are discussed in Section 5.

---

<sup>1</sup> The polytropic efficiency is defined as being equal to the isentropic efficiency at an infinitesimally small pressure ratio. When applied to a finite pressure ratio, a compressor with a given polytropic efficiency will have a lower isentropic efficiency.

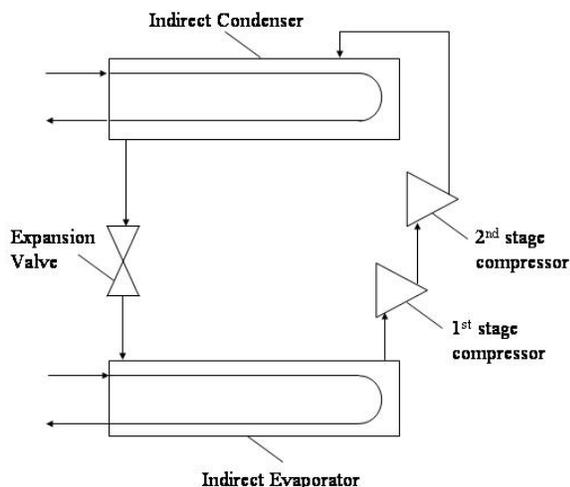


Figure 2. Non-intercooled cycle

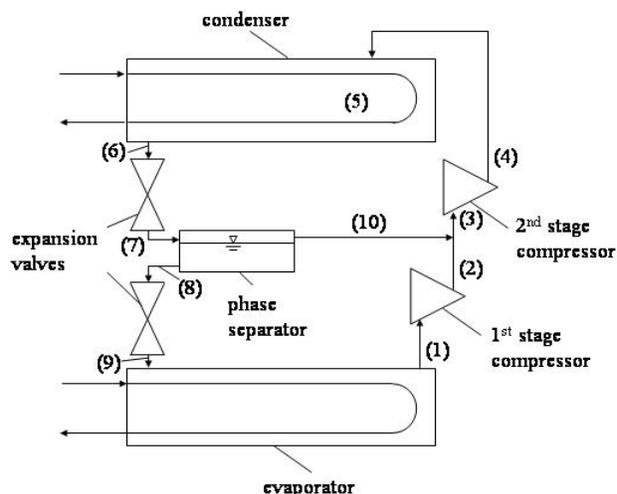


Figure 3. Economized cycle

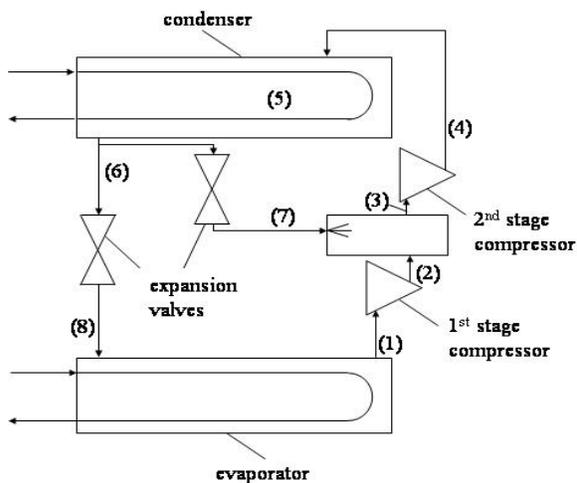


Figure 4. Flash intercooled cycle

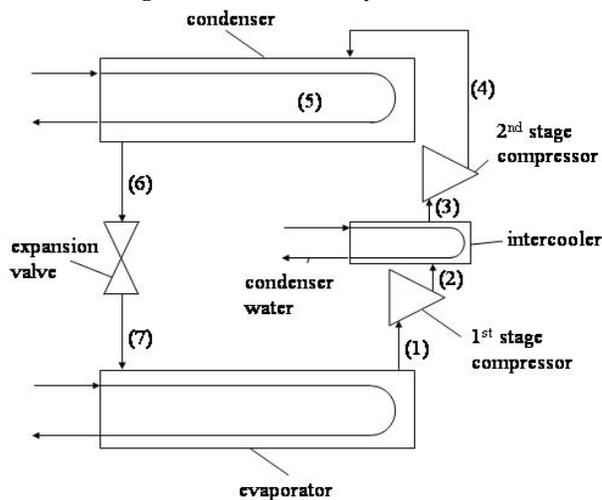


Figure 5. Indirect intercooled cycle

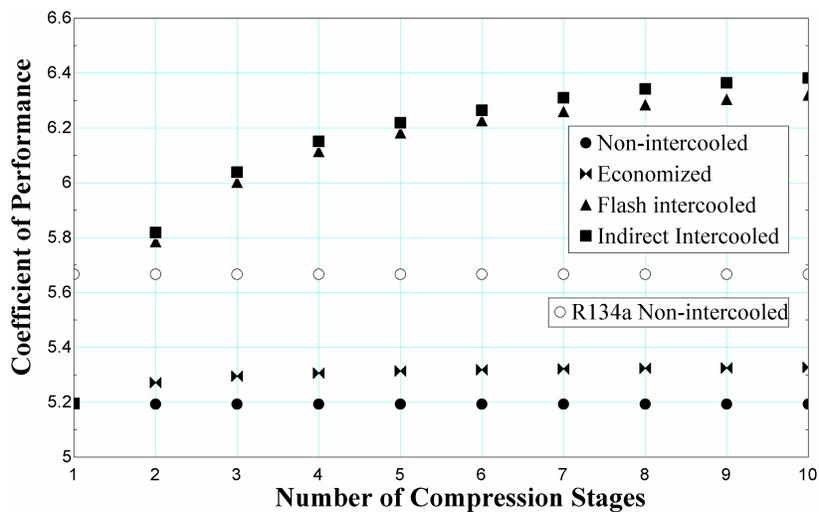


Figure 6: Cycle COP as a function of the number of compression stages for a single stage R-134a cycle and two-stage water cycles that include: non-intercooled, economized, flash intercooled, and indirect intercooled.

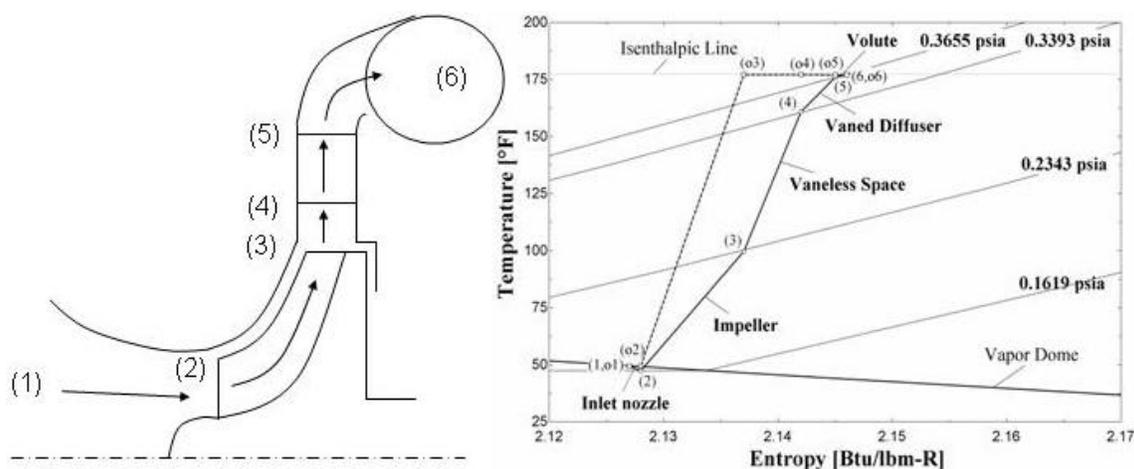
### 3. COMPONENT MODELING

Component models are developed to provide more detailed sizing and performance estimates for the components integrated into a chiller system. A summary of the models are given in the sections that follow and additional details can be found in Lachner (2004).

#### 3.1 Compressor

Wight et al. (2000) carried out a detailed scoping analysis of turbo-compressor technology applied to steam compression in a water as a refrigerant cycle. The authors concluded that a single-stage compressor was not feasible for several reasons including: low efficiency (less than 75% for the radial bladed unit), very large impellers (ca. 20 ft in diameter), and very high tip velocities (ca. 2200 ft/sec). The high tip velocities lead to large stresses and high Mach numbers. These factors combine to yield extremely high capital cost and technically-challenging compressor design/development. A two stage centrifugal compressor was also evaluated with improved results. The two-stage configuration was able to achieve predicted efficiencies of 80% with large impeller diameters but with somewhat lower tip velocities (nominally 1600 ft/sec) implying a more technically achievable compressor. The most promising compressor technology identified by Wight et al., in terms of total geometric size and performance, was a multistage axial compressor configuration consisting of between 6 & 7 stages. The efficiency of this type of compressor was estimated at 82% and the size and tip speeds required were reasonable, 4.5 ft and 1400 ft/sec, respectively. The disadvantage is that axial compressors are sophisticated and expensive devices when compared to centrifugal compressors. Therefore, the use of axial compressors is generally limited to high unit cost devices such as jet engines and stationary gas turbines for power production. In contrast, centrifugal compressors are sufficiently inexpensive to allow their use in very large capacity refrigeration systems today. Given the inherently high cost of axial compression systems, it seems likely that a multi-stage compression process using centrifugal compressors will be more economically attractive and therefore this technology became the focus of the compressor modeling carried out in this project.

The centrifugal compressor is modeled as a series of thermomechanical processes that the refrigerant transitions through corresponding to the inlet nozzle, impeller, vaneless space, vaned diffuser, and collector. Simple relations are used to establish the performance of each sub-component. Taken together, these sub-component models make up the centrifugal compressor model which is capable of predicting the static and stagnation states of the refrigerant as it moves along a mean-line path between the compressor inlet and the exit. The static and stagnation state points are illustrated graphically in Figure 7 on a T-s diagram (for a steam compression process). The compressor model is used to estimate the performance, size, and cost of both water vapor compressors and R-134a compressors.



**Figure 7:** Schematic illustrating the static and stagnation (designated with o) state points within a compressor alongside the corresponding T-s diagram (for a water compression process).

### 3.2 Heat Exchangers

The heat exchanger models rely on establishing a given a set of geometric and other relationships (i.e. aspect ratio, tube pitch, approach temperature difference, refrigerant velocity) that can be used to develop performance predictions which lead to specific heat exchanger sizes. Both the direct and indirect-contact heat exchangers function similarly in this respect. Based on the discussion above regarding the effect of standing water on cycle performance, heat transfer coefficients for water as refrigerant cycles are based on correlations for falling film evaporation and condensation over tube-banks. The R-134a baseline cycle is assumed to use a flooded evaporator. These heat transfer coefficients are all based on correlations for smooth tubes found in Rohsenow et al. (1998). The Petukhov equation (Incropera and DeWitt, 2002) for internal pipe flow is used on the water side. These relationships allow the overall conductance to be calculated and the heat exchanger geometry is iterated until the calculated conductance meets performance requirements. Conductance values were corrected using a multiplier that accounts for the effect of enhanced surfaces in order to match performance data obtained from several chiller manufacturers for an R-134a chiller in the 1000-ton size range.

The direct contact heat exchanger is assumed to be a shell filled with a packing material having a high specific surface area in order to increase the wetted heat transfer area. The direct contact heat exchanger model predicts the volume of packing material that is required to provide adequate heat transfer surface area given the cycle requirements. Heat transfer coefficients are calculated based on hydrodynamic conditions which are heavily dependent on the allowable refrigerant pressure drop (velocity) through the shell.

In the water cycle pressure drop becomes a key issue, almost as important as heat transfer surface area. The flexibility of the heat exchanger models to adjust geometric levers in order to control the pressure drop is critical to optimizing the water cycle. Tube pitch and refrigerant velocity are the main levers for controlling pressure drop. Pressure drop in the R-134a cycle is a negligible penalty, thereby making tube pitch a minor concern.

## 4. ECONOMIC ANALYSIS

The viability of a water-based vapor compression refrigeration system revolves around its life-cycle cost competitiveness with current technology. The life-cycle cost of investing in either an R-134a cycle or a water cycle are evaluated using a multiplier method known as the “ $P_1$ - $P_2$ ” approach (Duffie and Beckman, 1991). The parameter,  $P_1$  is a multiplier that establishes the present value of future energy costs. This essentially modifies the operating cost so that it can be compared directly (i.e. in present value) to the capital cost. The parameter,  $P_2$ , is a multiplier that accounts for all of the economic parameters that pertain to the capital cost of the equipment. This multiplier varies depending parameters such as the proportion of the capital cost that is financed, tax rates, inflation rates, etc. Using these multipliers, the life-cycle cost (LCC) is defined by the following equation in real (today’s) dollars.

$$LCC = P_1 \cdot F + P_2 \cdot E \quad (1)$$

where  $F$  is the yearly operating cost and  $E$  is the capital cost. In order for a water as a refrigerant cycle to be a viable economic alternative to more traditional synthetic refrigerant-based cycles, any capital cost premium that is required for the water refrigeration cycle equipment (e.g., the large centrifugal compressors) must be balanced by an operating cost saving associated with the more efficient water cycle configurations. With this in mind, the maximum allowable first cost is calculated by setting the life-cycle costs of the two cycles, the baseline R-134a cycle and an alternative water cycle, equal to one another in the following way.

$$E_{\text{water}} = \frac{P_1 \cdot F_{\text{R134a}} + P_2 \cdot E_{\text{R134a}} - P_1 \cdot F_{\text{water}}}{P_2} \quad (2)$$

where  $E_{\text{water}}$  is the maximum allowable first cost of a competing water cycle,  $E_{\text{R134a}}$  is the first cost for the R-134a cycle (obtained via a survey of chiller manufacturers), and  $F_{\text{water}}$  and  $F_{\text{R134a}}$  are the yearly operating costs associated with a water and R-134a cycle, respectively. For the four possible permutations of the flash intercooled water cycle considered, Table 2 summarizes the results of a 20-year life cycle analysis. Table 2 shows both the maximum

allowable and the estimated first cost of each water cycle permutation. The permutations correspond to the various combinations of direct and indirect heat exchangers that are possible.

**Table 2** Results of 20-year life-cycle cost analysis

<b>Cycle Configuration</b>	<b>Maximum Allowable First Cost</b>	<b>Predicted Actual First Cost</b>
Indirect Condenser and Evaporator	\$83,700	\$1,098,000
Indirect Condenser / Direct Evaporator	83,900	1,045,900
Direct Condenser / Indirect Evaporator	85,900	1,048,800
Direct Condenser and Evaporator	88,400	1,007,600

Table 2 indicates that there is no permutation of a water cycle that will be economically competitive with an R-134a cycle over a 20-year horizon. The most attractive water cycle takes advantage of the cheaper, higher performance direct contact heat exchangers to accomplish both the evaporation and condensation functions; the benefit of this is significant but insufficient to overcome the very high cost of the centrifugal compressors. Over a longer expected life-cycle, the maximum allowable first cost will approach the actual first cost. However, the greater likelihood that contamination or fouling issues will degrade the performance of the water cycle may eliminate its advantage over the R-134a cycle. Note that Table 2 provides a convenient means of illustrating the results of this study. The greatest uncertainty in the economic analysis lies in predicting the first cost of a production chiller using water as a refrigerant. However, the maximum allowable first cost that can be tolerated depends only on the current first cost of a centrifugal chiller and the performance of the water and R-134a cycles; these are quantities that can be predicted with some degree of accuracy.

## 5. OTHER ISSUES

This project identified and explicitly modeled many of the significant challenges associated with an economically viable water as a refrigerant cycle, including the high volumetric flow/high pressure ratio compression process as well as the large, indirect contact heat exchangers that would be required. This last drawback may be alleviated somewhat by the possible use of direct-contact heat exchangers. However, there are many additional practical obstacles that are less easily quantified but still extremely important.

The use of direct-contact heat exchangers will come with some penalty associated with contamination within the refrigeration cycle. The possibility of the infiltration of air and other non-condensable gases, solids, and entrained liquids is increased. In particular, a direct-contact condenser used in conjunction with a cooling tower will allow air to be absorbed into the water as it circulates through the cooling tower. A significant amount of non-condensable gas in the refrigerant results in a loss of cycle performance and so the non-condensable gases must be purged. The condenser water associated with a 1000-ton chiller system would result in a 1800 cfm volumetric flow rate of air that must be purged from the system, assuming that the water is completely saturated with air. Based on current vacuum pump technology, this flow rate would require nominally 100 hundred horsepower which corresponds to nominally 10% of the input power and would therefore negate any COP benefit associated with the water cycle. In any case, the purging and filtration units required will add to the capital and operating costs of the system and therefore reinforce the conclusion illustrated by Table 2, that the water as a refrigerant cycle is not economically viable.

Another unique aspect of the water-based vapor compression refrigeration cycle is that the effect of any pressure loss in heat exchangers that operate between 0.01 and 0.1 atm are magnified greatly. Tube pitch in indirect contact heat exchangers and packing density in direct contact heat exchangers are critical design parameters and result in larger heat exchangers than might be expected based on the thermal loading alone.

Finally, there is some concern regarding the presence of droplets at the inlet to the compressor. Because the flow is accelerated to match the impeller blade velocity and angle, the static temperature drops. In a water vapor compressor this effect is quite significant and raises the possibility that condensation will occur at the nozzle exit. In order to avoid this possibility, it is necessary to add some superheat to prevent this situation. The penalty associated with superheat is very large for a water cycle. To first order, the penalty on cycle COP due to superheat is linear and

decreases the COP by approximately 0.03 for each °F of superheat. However, there is some evidence from the power generation industry to indicate a supersaturated state may result in which case droplets will not form.

## 6. SUMMARY

This project investigated the economic viability of using water as the refrigerant in a 1000-ton chiller application. The most attractive water cycle configuration was found to be a flash-intercooled, two-stage cycle using centrifugal compressors and direct contact heat exchangers. Component level models were developed that could be used to predict the size and performance of the compressors and heat exchangers in this cycle as well as in a baseline, R-134a refrigeration cycle consistent with chillers in use today. A survey of several chiller manufacturers provided information that was used to validate and refine these component models.

The component models were integrated into cycle models that were subsequently used to investigate the life-cycle costs of both an R-134a and water refrigeration cycle. It was found that the first cost associated with the water as a refrigerant cycle greatly exceeded the savings in operating costs associated with its somewhat higher COP. Therefore, the water refrigeration cycle is not an economically attractive option to today's R-134a refrigeration system. There are a number of other issues, most notably the requirements associated with purging non-condensable gases that accumulate in a direct contact heat exchanger, which will further reduce the economic viability of the water cycle.

## NOMENCLATURE

COP	coefficient of performance		<b>Subscripts</b>	
F	first year operating cost	(US\$)	water	water
E	capital cost (first cost)	(US\$)	R134a	R-134a
LCC	life-cycle costs	(US\$)		
P <sub>1</sub>	operating cost multiplier			
P <sub>2</sub>	capital cost multiplier			

## REFERENCES

- Duffie, J.A., and Beckman, W.A., 1991, "*Solar Engineering of Thermal Processes*", 2<sup>nd</sup> Edition, Wiley and Sons.
- Klein, S. A., and Alvarado, F. L., 2004, *EES – Engineering Equation Solver*, F-Chart Software.
- Incopera, F.P., DeWitt, D.P., 2002, "*Introduction to Heat Transfer*", 4<sup>th</sup> Edition, Wiley and Sons.
- Lachner, B.F., Nellis, G.F., and Reindl, D.T., 2004, "*Use of water vapor as a refrigerant: Impact of cycle modifications on commercial feasibility*", University of Wisconsin – Madison, Contract #: 611-10080.
- Rohsenow, W. M., Hartnett, J. P., and Cho, Y. I., eds., 1998, *Handbook of Heat Transfer*, 3<sup>rd</sup> Edition, McGraw-Hill, New York.
- Wight, S.E., Tsukasa, Y., Le Drew, B.A., D'Orsi, N.C., 2000 "*The efficiency limits of water vapor compressors*", Concepts, ETI, Contract #: 605-10010.

## ACKNOWLEDGEMENT

This work was sponsored by the Air-Conditioning and Refrigeration Technology Institute (ARTI) under ARTI-21-CR Program Contract No. 611-10080. The technical assistance of the project monitor, Steve Syzmurski, as well as Jean-Luc Caillat, John Judge, Lee Burgett, and Thomas Watson on the Project Monitoring Committee is gratefully acknowledged. In particular, the assistance of Tom Watson in obtaining critical technical information is appreciated.