Proceedings of the
Renewable and Advanced Energy Systems for the 21st Century
April 11-15, 1999, Lahaina, Maui, Hawaii

SLA-2

SECOND LAW STUDY OF THE EINSTEIN REFRIGERATION CYCLE

Sam V. Shelton, Ph.D.
Assoc. Professor
G.W.W. School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Georgia 30332-0405
sam.shelton@me.gatech.edu

Andrew Delano, Ph.D.
Design Engineer
Hewlett-Packard, Inc.
3400 East Harmony Road, MS 69
Fort Collins, Colorado 80528-9599
adelano@fc.hp.com

Laura A. Schaefer
Ph.D. Student
G.W.W. School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Georgia 30332-0405
gt4867g@prism.gatech.edu

Keywords: Second Law, Absorption, Einstein Cycle, Coefficient of Performance

ABSTRACT

After formulating the theory of relativity, Albert Einstein spent several years developing absorption refrigeration cycles. In 1930, he obtained a U.S. patent for a unique single pressure absorption cycle. The single pressure throughout the cycle eliminates the need for the solution pump found in conventional absorption cycles. The Einstein cycle utilizes butane as a refrigerant, ammonia as a pressure equalizing fluid, and water as an absorbing fluid. This cycle is dramatically different in both concept and detail than the better known ammonia-water-hydrogen cycle.

Recent studies have shown that the cycle's COP is 0.17, which is relatively low compared to two-pressure cycles. This limits the cycle to refrigeration applications where simplicity, compactness, silent operation, and low cost are the important characteristics. Improved efficiency would open up other potential applications.

In this study, a comprehensive second law analysis of the cycle was carried out on each component and process to determine the thermodynamic source of the low efficiency. The results show that the reversible COP for the cycle is 0.58, and that the component with the largest irreversibility is the generator. The entropic average temperatures for the heat flows into and out of the cycle are 353 K for the generator, 266 K for the evaporator, and 315 K for the absorber/condenser. The COP degradations from the ideal due to irreversibilities are 0.12 for the evaporator, 0.11 for the absorber/condenser, and 0.17 for the generator. The generator irreversibility is due to the inherent temperature difference in the internal heat exchange. The results show that there is a large potential for increasing the cycle's efficiency through design changes to raise the low generator temperature and to reduce the large generator irreversibilities.

NOMENCLATURE

Symbols

COP	Coefficient of Performance
h	Enthalpy (kJ)
m	Mass (kg)
Q	Heat Transfer (kJ)
S	Entropy (kJ/K)
T	Temperature (°C, K)
x	Liquid Concentration
y	Vapor Concentration

Subscripts

#	State Point as Identified in Figure 1
a	Ammonia
b	Butane
f	Liquid
g	Vapor

gi	Generator, Internal
i	Component i
il	Internal Liquid
iv	Internal Vapor
rev	Reversible
s	Entropic

Superscripts

•	Rate
–	Average

INTRODUCTION

Current refrigeration cycles used in air conditioning, refrigeration, and heat pump equipment are two-pressure cycles where a temperature difference between a condenser and evaporator is established by a pressure difference. A compressor establishes this pressure difference in vapor-compression cycles, or a solution pump in absorption cycles. These two-pressure cycles require mechanical devices with moving parts and access to shaft or electrical power to drive the compressor or pump. This compressor or pump adds significantly to the system costs, reduces reliability, generates noise, and limits portability. In addition, these refrigeration systems must be connected to a fuel burning power generating facility which converts only 25 to 35 percent of this fuel to power. The one exception to this is the Platen and Munters cycle using hydrogen gas to establish a lower refrigerant partial pressure in the evaporator, while maintaining a higher refrigerant pressure in the condenser. Because of its low efficiency and limited temperature lift, it has very limited commercial application.

Over 65 years ago, Albert Einstein and Leo Szilard obtained a U.S. patent for a pump-less absorption refrigeration cycle (Einstein, 1930). This cycle operates with an evaporator, a combined condenser/absorber, and a generator at a single uniform pressure. It utilizes butane as a refrigerant, ammonia as a pressure equalizing fluid, and water as an absorbing fluid. In the evaporator, the partial pressure on the entering liquid butane is reduced by ammonia vapor, allowing it to evaporate at a lower temperature. In the condenser/absorber, the partial pressure on the vapor butane coming from the evaporator is increased when the ammonia vapor is absorbed by liquid water, thus allowing the butane to condense at a higher temperature. Liquid butane and liquid ammonia-water are immiscible and naturally separate. The ammonia is then separated from the water in a generator by the application of heat. Fluid flow in the cycle is accomplished with a heat driven bubble pump and gravity.

In this study, a first and second law thermodynamic model of the Einstein refrigeration cycle was used to investigate the cycle's performance characteristics. Two regenerative heat exchangers not included in the Einstein patent were added to

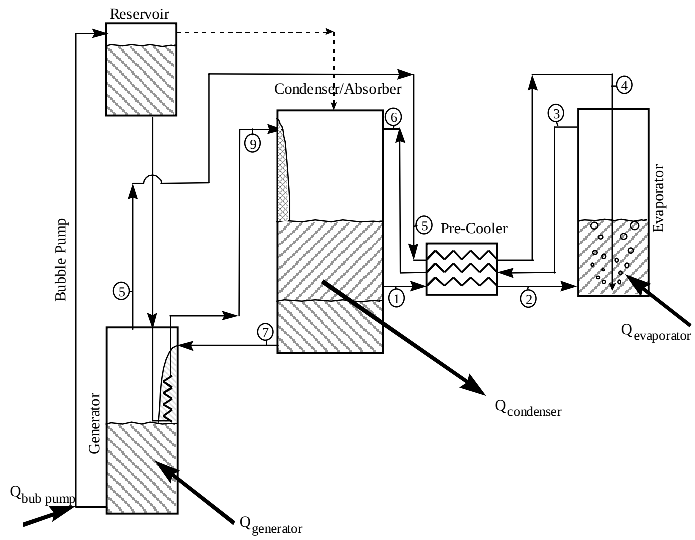

Figure 1 Einstein Refrigeration Cycle Schematic

improve performance. These are a generator internal regenerative heat exchanger and an evaporator pre-cooler. The ammonia-butane and ammonia-water fluid mixture properties were modeled with a cubic equation of state and fitted to the limited available experimental data using mixing rules.

As described below, the cycle COP was calculated through both first and second law analyses. The entropic average temperatures, the ideal reversible COP, the entropy generation in each process, and the irreversible COP degradations from the reversible COP for each component are analyzed.

CYCLE DESCRIPTION

Figure 1 shows a schematic of the Einstein refrigeration cycle. In the Einstein cycle, ammonia acts as an inert gas to lower the partial pressure over the refrigerant, butane. Water later provides separation by absorbing the ammonia.

Starting in the evaporator, liquid butane arrives from the condenser/absorber. In the evaporator, the partial pressure above the butane is reduced by ammonia vapor flowing from the generator. With its partial pressure reduced, the butane evaporates near the saturation temperature of its partial pressure, cooling itself and the ammonia, and providing external refrigeration. The ammonia-butane vapor mixture leaves the evaporator and enters the pre-cooler where it cools the hot vapor ammonia that is counter-flowing from the generator. The now superheated ammonia-butane mixture flows out of the pre-cooler into the condenser/absorber, which is continuously cooled by an environmental heat sink. Meanwhile, liquid water from the generator is sprayed into the condenser/absorber. With its affinity for ammonia vapor, this water absorbs the vapor ammonia from the ammonia-butane mixture. The absorption of the ammonia vapor increases the

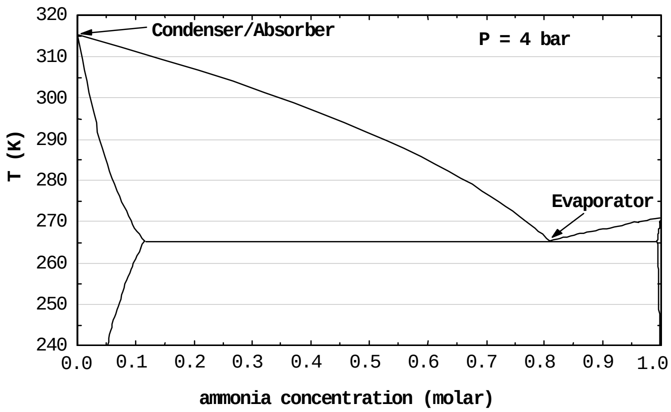

Figure 2 T-x-x-y Diagram for Ammonia-Butane

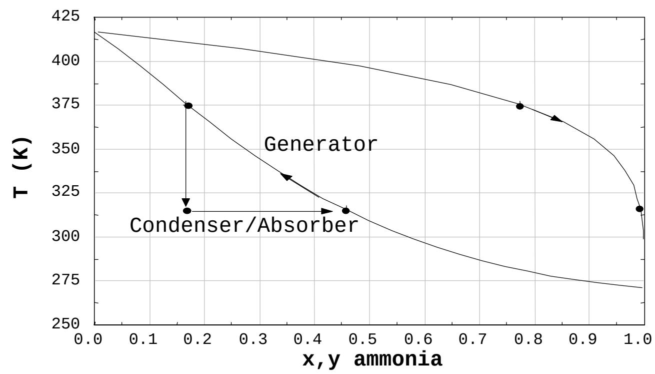

Figure 3 T-x-y Diagram for Ammonia-Water, P=4 bar

partial pressure on the butane vapor to nearly the total pressure, allowing it now to condense at butane's saturation temperature for the total pressure. Note that this is higher than butane's saturation temperature at the partial pressure in the evaporator. The butane and the ammonia water separate due to their respective density differences and the fact that ammonia-water is immiscible with butane at the condenser/absorber's temperature and pressure. Since liquid butane is less dense than liquid ammonia-water, it is the top liquid and is siphoned back to the evaporator. Meanwhile, the ammonia-water mixture leaves from the bottom of the condenser/absorber and enters the solution heat exchanger. Here, the mixture is pre-heated by the internal regenerative heat exchanger before being heated by an external heat source.

Inside the generator, heat is applied to the strong ammonia-water solution, driving off ammonia vapor where it rises and is carried to the evaporator. The remaining weak ammonia-water solution is pumped up to a reservoir via a bubble pump. In the reservoir, any residual ammonia vapor from the bubble pump is sent to the condenser/absorber. This vapor flow into the condenser is small (Delano, 1998), and has been neglected in this analysis of the cycle. The weak ammonia water solution falls to the generator internal regenerative heat exchanger where it gives up its heat to the strong ammonia-water solution coming from the condenser. Finally, this water is sprayed into the condenser/absorber.

While the overall pressure of the cycle is constant, there are slight pressure variations within the cycle necessary for fluid motion. These are due to height variations and are not large enough to significantly affect property evaluation.

PROPERTY MODEL

Accurate working property models are available for the ammonia-water mixtures needed in this study, but none were found for the required ammonia-butane. Therefore, for consistency, the vapor-liquid equilibrium thermodynamic properties of both the ammonia-butane and ammonia-water mixtures were modeled with the Patel-Teja cubic equation of state (Patel, 1980) in conjunction with the Panagiotopoulos and Reid mixing rules (Panagiotopoulos and Reid, 1986). These mixing rules allow the equation of state to be fitted to experimental mixture data. For the ammonia-butane mixture, the equation of state was fitted to the experimental data of Wilding et al. (1996). For the ammonia-water mixture, the equation of state was fitted to the experimental data of Gillespie et al. (1987).

Typical property model results for ammonia-butane are shown on a temperature-concentration plot in Figure 2. The pressure in this case is taken at four bars, which is the total operating pressure at which the cycle was analyzed. Note that the property model shows an azeotrope as well as the appearance of a liquid-liquid equilibrium. This liquid-liquid separation disappears at lower pressures, which agrees with the experimental data.

Figure 3 shows the ammonia-water property model result on a temperature-concentration diagram. Again, the total pressure is taken at 4 bars, and the results agree very well with the Gillespie et al. (1987) experimental data. A more detailed description of the agreement between the property models and the experimental data is given in Delano (1998).

SECOND LAW ANALYSIS

The first and second laws can be combined to provide a relationship for the reversible COP of a three-temperature reservoir heat pump in terms of only the constant temperature reservoir temperatures. Due to irreversibilities such as fluid mixing and heat transfer across a finite temperature difference, the COP_{rev} is degraded to the actual COP. Using the concept of direct second law analysis (Alefeld, 1990), it is possible to calculate the amount by which each process in a system of interest degrades the system's reversible COP.

This approach is different in detail to carrying out an exergy or availability analysis where the loss of work due to irreversible entropy generation is calculated. Since no work is required or produced by the Einstein cycle, work is not a commodity of interest. The effect on the COP is of interest. This COP is the ratio of the "desired output," which is the refrigeration heat absorbed, to the "input paid for," which is the driving high temperature generator heat input. This direct second law approach directly calculates the degradation of this COP due to irreversible entropy generation in the various cycle processes.

For the Einstein refrigerator, direct second law analysis begins by applying the first and second law to a control volume encompassing the entire system, with only heat crossing the boundary, as shown in Figure 1.

$$\dot{Q}_{evaporator} + \dot{Q}_{generator,overall} - \dot{Q}_{condenser} = 0 \qquad (1)$$

$$\frac{\dot{Q}_{evaporator}}{\overline{T}_{evaporator,s}} + \frac{\dot{Q}_{generator,overall}}{\overline{T}_{generator,s}} - \frac{\dot{Q}_{condenser}}{\overline{T}_{condenser,s}} = -\sum_k \dot{S}_k \qquad (2)$$

In the previous equation, $\dot{S}_k$ represents the entropy generation of each process, *k*, occurring within the system, and $\dot{Q}_{generator,overall}$ is the heat added to the overall generator subsystem. The temperatures $\overline{T}_{evaporator,s}$, $\overline{T}_{condenser,s}$, and $\overline{T}_{generator,s}$ are defined to be the entropic average temperatures at which heat flows across the defined control volumes (Herold et al., 1996). The condenser/absorber and the evaporator exchange heat with their respective heat sinks and source at constant temperatures. Therefore, their entropic average temperatures are merely equal to $T_{evaporator}$ and $T_{condenser}$, which are fixed at 315 K and 266 K. However, the generator fluid receives its heat at a varying temperature, starting at the internal heat exchanger exit at T_{7i} to the maximum generator temperature, T_8, as shown in Figure 7b. Calculation of the exact entropic average temperature for this desorbing process is complex, and was therefore estimated as the numerical average of the entering and leaving temperatures. The error introduced by this calculation is minimal for this small temperature range, as demonstrated by Alefeld (1993).

Now, equation 2 is multiplied by $T_{condenser}$ and subtracted from equation 1. After some rearrangement, the following equation is produced:

$$\frac{\dot{Q}_{evap}}{\dot{Q}_{gen,ov}} = \left[\frac{\left(\dfrac{\overline{T}_{gen,s} - T_{cond}}{\overline{T}_{gen,s}}\right)}{\left(\dfrac{T_{cond} - T_{evap}}{T_{evap}}\right)}\right] - \left(\frac{T_{evap} \cdot T_{cond}}{T_{cond} - T_{evap}}\right) \cdot \left(\frac{\sum_k \dot{S}_k}{\dot{Q}_{gen,ov}}\right) \qquad (3)$$

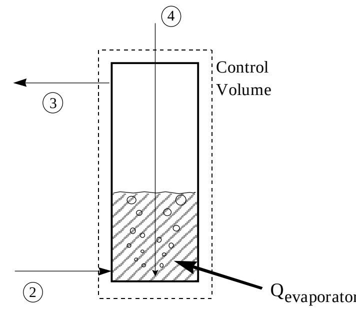

Figure 4 The Evaporator

The first term in square brackets on the right side in equation 3 is the COP_{rev} for a thermally-driven refrigerator operating between three temperature reservoirs. Equation 3 may thus be rewritten as follows:

$$COP = COP_{rev} - \sum_k COP_{degradation,k} \qquad (4)$$

where

$$COP_{degradation,k} = \left(\frac{T_{evaporator} \cdot T_{condenser}}{T_{condenser} - T_{evaporator}}\right) \cdot \left(\frac{\dot{S}_k}{\dot{Q}_{gen,overall}}\right) \qquad (5)$$

Equation 5 conveniently shows how much the entropy generation of each process, *k*, inside the control volume of Figure 1 degrades the reversible COP to the actual irreversible COP. This allows for an absolute evaluation of the degradation impact on the cycle COP for each irreversible process.

THERMODYNAMIC MODEL

To create a thermodynamic model of the Einstein cycle, the overall cycle was divided into 5 separate control volumes: the evaporator, the pre-cooler, the condenser/absorber, the generator, and the internal generator. Next, the mass conservation and first and second law equations were written for each control volume. Each control volume was assumed to operate under steady state conditions with no fluid friction.

Figure 4 shows the control volume for the evaporator. Conserving mass for both the ammonia and the butane found in the evaporator yields the following equations, with the subscripts referring to the Figure 4 state points:

$$y_{a,3} \cdot \dot{m}_3 = x_{a,2} \cdot \dot{m}_2 + \dot{m}_4 \qquad (6)$$

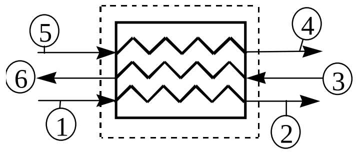

Figure 5 The Pre-Cooler

$$y_{b,3}\cdot\dot{m}_3 = x_{b,2}\cdot\dot{m}_2 \tag{7}$$

Conservation of energy for this same evaporator control volume in Figure 4 yields:

$$\dot{Q}_{evaporator} = \dot{m}_3\cdot h_3 - \dot{m}_1\cdot h_2 - \dot{m}_4\cdot h_4 \tag{8}$$

Furthermore, the second law entropy generation for the evaporator is:

$$\dot{S}_{evaporator} = \dot{m}_3\cdot s_3 - \dot{m}_1\cdot s_2 - \dot{m}_4\cdot s_4 - \frac{\dot{Q}_{evaporator}}{T_{evaporator}} \tag{9}$$

where $T_{evaporator}$ is constant.

The pre-cooler (Figure 5) is insulated so that the only heat transfer occurs between the entering streams and exiting streams. Since the conservation of mass will be satisfied between the evaporator and condenser/absorber, the only equation necessary for the pre-cooler is the conservation of energy.

$$\dot{m}_3\cdot h_3 + \dot{m}_1\cdot h_1 + \dot{m}_4\cdot h_5 = \dot{m}_3\cdot h_6 + \dot{m}_1\cdot h_2 + \dot{m}_4\cdot h_4 \tag{10}$$

The entropy generated by the pre-cooler due to heat transfer across any finite temperature difference is:

$$\dot{S}_{precool} = \dot{m}_3\cdot s_6 + \dot{m}_1\cdot s_2 + \dot{m}_4\cdot s_4 - \dot{m}_3\cdot s_3 - \dot{m}_1\cdot s_1 - \dot{m}_4\cdot s_5 \tag{11}$$

In this study, the effect of non-ideal heat exchange is accommodated simply with the use of pinch points. Since it is a three stream heat exchanger, there are two pinch points in the pre-cooler. One is between the vapor mixture entering at

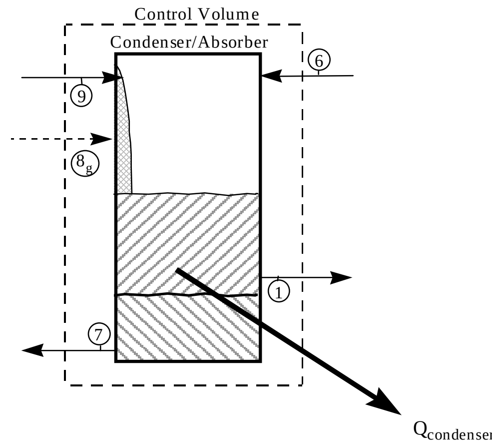

Figure 6 The Condenser/Absorber

state point 3 and the liquid leaving at state point 2. A second exists between the vapor mixture leaving at state point 6 and the liquid entering at state point 5.

$$T_2 = T_3 \tag{12}$$

$$T_6 = T_5 \tag{13}$$

In the Einstein refrigeration cycle, the condenser and absorber are combined into a single component where both processes occur simultaneously as shown in Figure 6. Since $\dot{m}_3 = \dot{m}_6$, conservation of mass for the condenser/absorber control volume in Figure 6 yields the following equations:

$$\dot{m}_1 + \dot{m}_7 = \dot{m}_9 + \dot{m}_3 + \dot{m}_{8g} \tag{14}$$

$$x_{i,1}\cdot\dot{m}_1 + x_{i,7}\cdot\dot{m}_7 = x_{i,9}\cdot\dot{m}_9 + y_{i,3}\cdot\dot{m}_3 + y_{i,8g}\cdot\dot{m}_{8g} \tag{15}$$

Conserving energy for this same control volume yields the heat transfer from the condenser:

$$\dot{Q}_{cond} = \dot{m}_1\cdot h_1 + \dot{m}_7\cdot h_7 - \dot{m}_9\cdot h_9 - \dot{m}_3\cdot h_6 - \dot{m}_{8g}\cdot h_{8g} \tag{16}$$

The second law entropy generation for the control volume in Figure 6 is:

$$\dot{S}_{cond} = \dot{m}_1\cdot s_1 + \dot{m}_7\cdot s_7 - \dot{m}_9\cdot s_9 - \dot{m}_3\cdot s_6 - \dot{m}_{8g}\cdot s_{8g} - \frac{\dot{Q}_{cond}}{T_{cond}} \tag{17}$$

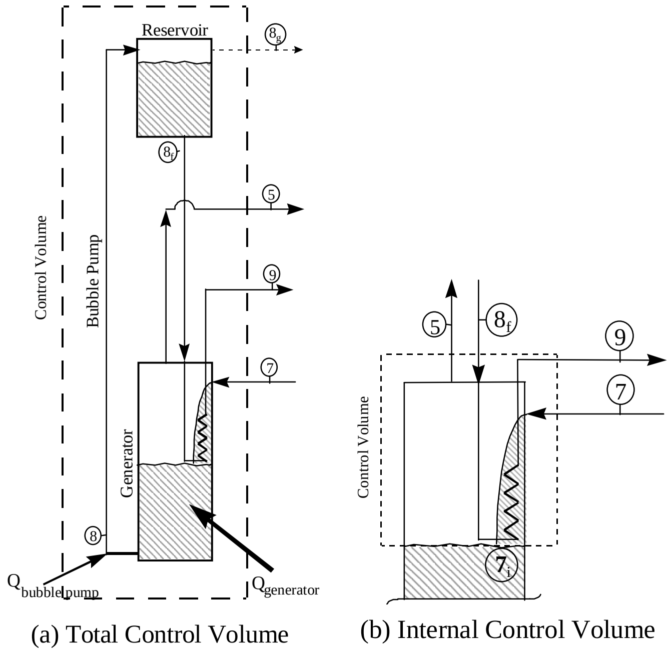

Figure 7 The Generator

In the generator, shown in Figure 7a, ammonia rich water arriving from the condenser is heated. Conserving mass and energy flow for the generator's control volume yields:

$$\dot{m}_7 = \dot{m}_4 + \dot{m}_{8g} + \dot{m}_9 \tag{18}$$

$$x_{a,7} \cdot \dot{m}_7 = x_{a,9} \cdot \dot{m}_9 + \dot{m}_4 + y_{a,8g} \cdot \dot{m}_{8g} \tag{19}$$

$$\dot{Q}_{gen} + \dot{Q}_{bubpump} = \dot{m}_5 \cdot h_5 + \dot{m}_9 \cdot h_9 + \dot{m}_{8g} \cdot h_{8g} - \dot{m}_7 \cdot h_7 \tag{20}$$

The entropy generated by the generator is:

$$\dot{S}_{gen} = \dot{m}_5 \cdot s_5 + \dot{m}_9 \cdot s_9 + \dot{m}_{8g} \cdot s_{8g} - \dot{m}_7 \cdot s_7 - \left(\frac{\dot{Q}_{gen} + \dot{Q}_{bpmp}}{\overline{T}_{gen,s}} \right) \tag{21}$$

To account for the generator's internal heat exchanger, another control volume is necessary (Figure 7b). Conservation of mass and energy yield the following equations:

$$\dot{m}_5 + \dot{m}_{gil} = \dot{m}_7 + \dot{m}_{giv} \tag{22}$$

$$\dot{m}_5 + x_{a,gi} \cdot \dot{m}_{gil} = y_{a,gi} \cdot \dot{m}_{giv} + x_{a,7} \cdot \dot{m}_7 \tag{23}$$

$$\dot{m}_5 \cdot h_5 + \dot{m}_{gil} \cdot h_{gil} + \dot{m}_9 \cdot h_9 = \dot{m}_7 \cdot h_7 + \dot{m}_9 \cdot h_8 + \dot{m}_{giv} \cdot h_{giv} \tag{24}$$

The subscript $(\ )_{giv}$ denotes the vapor entering the bottom of the control volume at the i interface and the subscript $(\ )_{gil}$ denotes the liquid flowing down the wall out of the bottom of the control volume interface at i.

The entropy generated by the generator's internal heat exchanger is accounted for with equation 21, which includes the internal heat exchanger. The internal heat exchanger in the generator also requires a pinch point:

$$T_7 = T_9 \tag{25}$$

To circulate the working fluids without a mechanical pump, the Einstein cycle relies on a bubble pump. The bubble pump is a heated tube communicating with the generator and the higher reservoir. The liquid in the generator initially fills the lower part of the tube. Heat is applied at the bottom of the tube at a rate sufficient to evaporate some of the liquid in the tube. The resulting vapor bubbles rise in the tube. The bulk density of the fluid in the tube is reduced relative to the liquid in the generator, thereby creating an overall buoyancy lift. This buoyancy lift causes liquid and vapor to flow upward in the tube carrying fluid from the generator to the reservoir.

A model of the bubble pump was created using the conservation of mass, momentum and energy, assuming that the bubble pump operated in the slug flow regime. The results of this model showed the energy requirement of the bubble pump to be about ten percent relative to that of the generator. Since the bubble pump has a relatively small effect, details of the analysis are omitted here, but can be found in Delano (1998).

CYCLE OPERATING CONDITIONS

Einstein's patent (Einstein, 1930), specified ammonia, water, and butane as the working fluids. The operating conditions for the cycle were chosen in this study to be at a system pressure of 4 bar, a condenser/absorber temperature of 315 K, an evaporator temperature of 266 K, and a maximum generator temperature, T_8, of 375 K. These were chosen to be consistent with a refrigerator using ambient air as a heat sink and producing ice-making conditions, as discussed below.

Table 1 Base Case Results for Pressure = 4 bars

Parameter	Value
COP_{ideal}	0.57
COP	0.17
$\dot{Q}_{genrator}$	7.09 kW
$\dot{Q}_{bubblepump}$	0.76 kW
$\dot{Q}_{condenser/absorber}$	-9.18 kW
$\dot{Q}_{evaporator}$	1.33 kW
$\dot{m}_1$	3.8 g/s
$\dot{m}_3$	8.5 g/s
$\dot{m}_4$	4.6 g/s
$\dot{m}_7$	14.7 g/s
$\dot{m}_9$	9.6 g/s
$\dot{m}_{bp}$	10 g/s
$T_{evaporator}$	266 K
$T_{condenser/absorber}$	315 K
$T_{generator, max}$	375 K
$\overline{T}_{generator,s}$	352.9 K
$T_{generator,internal}$	325.4 K
T4	278.1 K

The temperature for the condenser/absorber was chosen to be 43ºC, or 315 K, so that heat could be rejected to the ambient air. Next, the behavior of the ammonia-butane mixture, shown in Figure 2, was studied to select a system pressure and evaporator temperature. As seen in Figure 2, pure butane at a pressure of 4 bar condenses at 315 K. The addition of ammonia allows the mixture to boil as low as 266 K. Therefore, the system pressure was chosen to be 4 bar and the evaporator temperature 266 K. This -7ºC evaporator allows the production of ice.

To select a maximum generator outlet temperature, the behavior of the ammonia-water mixture at the system pressure, as shown in Figure 3, was studied. To generate ammonia vapor, the nearly 50/50 ammonia-water mixture flowing from the condenser/absorber at 315 K is heated, driving off mostly ammonia vapor, but also some unwanted water vapor. Heating the ammonia-water to 375 K reduces the mass concentration of ammonia in the liquid from about 0.5 to under 0.2, and does not generate a significant amount of water vapor. Lower temperatures would reduce the amount of the desorbed ammonia vapor, and higher temperatures would boil unwanted water vapor. Therefore, the maximum generator temperature was selected to be 375 K.

Figure 2 shows that at a fixed system pressure, the characteristics of the ammonia-butane mixture constrains the evaporator and condenser/absorber temperatures to a minimum

Table 2 Degrading of Reversible COP

Process	COP degradation
Pre-cooler	0.007
Condenser/Absorber	0.106
Evaporator	0.119
Generator/Bubble Pump	0.173
Total	**0.405**

and a maximum respectively. These extreme temperatures were used to produce the maximum temperature lift.

For the case studied here, infinitely large heat exchangers were assumed by making the pinch points for both heat exchangers zero. Finally, all mass flow rates were normalized to the mass flow rate into the bubble pump, since this flow rate can be controlled by heat input to the bubble pump. With specification of ammonia-water-butane fluids, the evaporator, condenser/absorber, and generator temperatures, zero heat exchanger pinch points, and a bubble pump mass flow rate of 10 g/s, the refrigeration cycle model is fully developed.

RESULTS

To simultaneously solve the large set of nonlinear equations in both the refrigeration cycle thermodynamic model and the Patel-Teja cubic equation of state property model, the *Engineering Equation Solver* software was used (Klein and Alvarado, 1997).

Table 1 provides the important results of this base case. Note that the scaling parameter is the bubble pump mass flow rate, taken arbitrarily in this case.

Utilizing equation 5, Table 2 shows the irreversible degradations calculated for the various processes.

For the ideal process, the reversible COP using equation 3 is 0.57. Including the degradations of all the processes results in an actual COP of 0.17. The generator contributes the largest irreversible degradation. This is due to heat transfer across the inherent temperature difference in the internal heat exchanger. In this regenerative heat exchanger, the liquid entering the generator from the condenser/absorber at 315 K is heated to approximately 325 K. This fluid is heated by the hot fluid side starting at the peak generator temperature of 375 K and decreasing down to 315 K at the exiting zero pinch point. This produces a large temperature difference of 55 K at one end of the heat exchanger, decreasing to zero at the other end.

The mixing of ammonia and butane in the evaporator causes its degradation to be relatively high as well, at 0.12. Likewise, the mixing of water and ammonia occurring in the condenser/absorber causes the third largest degradation to the COP. The pre-cooler contributes relatively minor degradations.

DISCUSSION

A question arises as to the value of adding the generator internal regenerative heat exchanger, which is the single irreversible process occurring in the generator/bubble pump, and the largest irreversibility in the cycle. It was inserted into the cycle in order to reduce the generator external heat requirement and to increase the COP. If it is removed, the generator irreversibility will be reduced. This appears to be a contradiction.

However, the entropic average temperature for the generator heat input will also be reduced with the removal of the generator internal heat exchanger, thereby reducing the COP_{rev}. Specifically, without the internal heat exchanger, the external generator heat input will start at T_7 (315 K), ending at the maximum generator temperature of 375 K, for an average of 345 K, rather than the 352.9 K with the internal heat exchanger. In addition, the exiting fluid temperature leaving the generator and entering the condenser will rise from 315 K to 375 K. This will increase the condenser irreversibility since it will have to reject this additional high quality thermal energy to the 315 K sink. The effect of removing the generator heat exchanger will be to reduce the COP_{rev}, add to the condenser irreversibility, and decrease the generator irreversibility. The net effect will cause a reduction in the resulting COP.

The internal regenerative heat exchanger has an inherent temperature difference at one end due to a large difference in the thermal masses of the two streams. This regenerative heat exchanger is not present in the original Einstein cycle, but was added to improve the COP. The fact that this added component causes a large irreversibility is not contradictory with its intent to improve the overall cycle efficiency. Its addition raises the entropic average temperature for the generator heat input and therefore the reversible COP, from which the irreversibilities degrade performance. It also reduces the temperature of the steam flowing to the absorber, thereby reducing the irreversibility in that component.

The evaporator irreversibility is due to mixing the ammonia vapor with the butane to reduce the partial pressure of the butane, thereby reducing its boiling temperature. This is a fundamental characteristic of the process. However, using other fluids might reduce the magnitude of the irreversibility.

The combined condenser/absorber irreversibility is due solely to the mixing of the liquid water flowing from the generator and the ammonia coming from the evaporator. This mixing process is necessary to absorb and remove the ammonia vapor from the butane-ammonia mixture to increase the partial pressure of the butane, thereby increasing the condensing temperature. This allows the thermal energy absorbed in the low temperature evaporator to be rejected at a higher temperature, thereby pumping thermal energy up a temperature hill.

The pre-cooler heat exchanger between the condenser/ absorber and the evaporator also contributed a small irreversibility of 0.01. This is due to the imbalance of the thermal masses, but the irreversibility is essentially negligible. This heat exchanger was added to improve the cycle COP, which it does by changing the stream temperatures to make the condenser/absorber and evaporator reject and receive heat at constant temperatures. This is similar to the generator internal heat exchanger effect discussed above.

CONCLUSIONS

The Einstein cycle is a three temperature thermal heat pump with no work input or output. It receives high temperature driving heat which is used to pump heat from a low refrigeration temperature to an ambient rejection temperature. The refrigeration application analyzed in this study uses butane-ammonia-water working fluids with the driving heat input temperature varying from 325 K to 375 K. The entropic average temperature for this driving heat input is 342 K. This is a relatively low driving heat source temperature. The refrigeration temperature from which heat is pumped was selected to be 266 K and the single heat rejection temperature was selected to be 315 K. The COP of this cycle is low relative to two-pressure absorption cycles, which require a liquid solution pump. A second law analysis gives insight into the fundamental reasons for the low COP.

The reversible COP was found to be at 0.57. This means that even if the cycle could be made reversible, it still could not reach the COP of advanced two-pressure absorption cycles. This is due primarily to the low generator heat input entropic average temperature of 342 K.

This reversible COP is degraded by three primary irreversibilities: 1) the generator internal regenerative heat exchanger, 2) the evaporator mixing, and 3) the absorber mixing. They degrade the reversible COP by 0.17, 0.12, and 0.11 respectively, down to 0.17.

The cycle demonstrates a creative approach to achieving refrigeration with no work requirement, though at a relative low COP. The second law analysis shows the source of this low performance to be primarily due to a low generator temperature. This should be investigated to determine how the cycle and working fluids might be changed to raise the generator heat input temperature and various achieve temperature lifts.

These developments could make the cycle competitive with current commercial technology in various applications such as residential heat pump space cooling and heating. The cycle can achieve first law heating efficiencies over 100 percent, and reduce summer peak air conditioning loads on electrical power plants. Other applications are possible where low cost and reliability are important, such as remote installations and developing countries without an electrical infrastructure. Silent operation is also a benefit.

REFERENCES

Alefeld, G., 1990, “What Are Thermodynamic Losses and How to Measure Them,” *A Future For Energy*, J. Necco, ed., World Energy Symposium, Firenze, Italy.

Alefeld, G., and Radermacher, R., 1993, *Heat Conversion Systems*, CRC Press, Boca Raton, FL, p. 64.

Delano, A., 1998, “Design Analysis of the Einstein Refrigeration Cycle,” Ph.D. Thesis, Georgia Institute of Technology, Atlanta, Georgia.

Einstein, A., and Szilard, L., 1930, “Refrigeration” (Appl: 16 Dec. 1927; Priority: Germany, 16 Dec. 1926) Pat. No. 1,781,541 (United States).

Gillespie, P.C., Wilding, W.V. and Wilson,G.M., 1987, “Vapor-liquid equilibrium measurements on the ammonia-water system from 313 K to 589 K,” *Experimental Results From the Design Institute for Physical Property Data*, C. Black, ed., AIChE Symposium Series, Vol. 83, No. 256.

Herold, K., Radermacher, R., and Klein, S. A., 1996, *Absorption Chillers and Heat Pumps*, CRC Press, Boca Raton, FL, p. 13.

Klein, S.A., and Alvarado, F.L., 1997, *Engineering Equation Solver*, Version 4.4, F-Chart Software, Middleton, Wisconsin.

Palmer, S.C. and Shelton, S.V., 1996, “Effect of Evaporator Design Pressure on Dual-Pressure Absorption Heat Pump Performance” *Proceedings of the International Ab-Sorption Heat Pump Conference*, R. Radermacher, ed., Vol. 1, pp. 231-236.

Patel, N.C., 1980, “The Calculation of Thermodynamic Properties and Phase Equilibria Using a New Cubic Equation of State,” Ph.D. Thesis, Loughborough University of Technology, Loughborough, Leicestershire, United Kingdom.

Pangiotopoulos, A.Z., and Reid, R.C., 1986, “New Mixing Rule for Cubic Equations of State for Highly Polar, Asymmetric Systems,” *Equations of State: Theories and Applications*, K.C. Chao and R. Robinson, ed., American Chemical Society, Washington, D.C., pp. 571-585.

Wilding, W.V., Giles, N.F., and Wilson, L.C., 1996, “Phase Equilibrium Measurements on Nine Binary Mixtures,” *Journal of Chemical Engineering Data*, Vol. 41, pp. 1239-1251.