Simulation of an Ammonia–Water Heat Pump Water Heater with Combustion Products-Driven Evaporator

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Abstract

The objective of this work is to simulate a single effect (SE) ammonia-water heat pump for domestic water heating, with innovative aspects for cycle simulation and eventual practical implementation. The following practical difficulties are addressed in the simulation: 1. seasonal temperature variations change the operating conditions of the distillation column, calling for insightful design to maintain a suitable refrigerant concentration in all seasons, and particularly in winter; 2. The evaporator activated by outdoor air suffers from immoderate heat transfer requirements, and these demands are considerably reduced if the activation is done by products of combustion; 3. Pumps have head requirements that can be assuaged by judicious selection and inlet solution subcooling. The variables that need to be controlled if the same column is to be used all year round are specified. As configured with the innovations mentioned, the cycle simulation yields a coefficient of performance within the expected range for a single effect, but it harbors the promise of a much smaller evaporator, of small overall height, and of a distillation column capable of operating effectively all year round with the same feed point.

Nomenclature

A	Heat transfer area, m ² .
cp	Combustion products.
COP	Coefficient of performance.
CPAE	Combustion products assisted evaporator.
f	Fuel to air mass ratio.
g	Gaseous combustion products.
GAX	Generator absorber heat exchange.
h	Specific enthalpy, kJ kg ⁻¹ .
H_a	Absolute pressure equivalent head, m.
H_f	Head loss in feed pipe to pump, m.
H_s	Head at suction point, m.
H_{vp}	Vapor pressure equivalent head, m.
HX	Heat exchanger.
$\dot{m}$	Mass flow rate, kg s ⁻¹ .
NTU	Number of transfer units.
NPSHa	Net positive suction head available, m.
NPSHr	Net positive suction head required, m.
p	Pressure, bar.
$\dot{Q}$	Heat rate, kW.
SE	Single effect.
Su	Summer.
T	Temperature, C.
SCA	Solution-cooled absorber.
SHX	Solution heat exchanger.

U	Overall heat transfer coefficient, $\text{kw m}^{-2} \text{K}^{-1}$.
UA	Overall heat transfer coefficient times heat transfer area, kW K^{-1} .
W_i	Winter.
x	Ammonia liquid mass fraction.
y	Ammonia vapor mass fraction, or mole fraction in combustion products.
Greek letters	
ε	Effectiveness.
Subscripts	
a	Air.
B	Boiler.
comp	Component (Absorber, boiler, condenser or evaporator).
F	Fuel (natural gas).
hot	Hot fluid in heat exchanger.
i	Inlet.
l	Liquid.
n	Plate number.
o	Outlet.
R	Reflux.
sat	Saturation.
v	Vapor.

18

19 1. Introduction

20 The literature on ammonia-water based cooling machines is vast [1] but the number of actual realizations
 21 comparatively small . The interest, translated into patents and applications of many creative minds arises
 22 due to direct conversion of thermal energy into heating or cooling, employing insightful management of
 23 chemical potentials to achieve a thermal end. The lack of massive applications can perhaps be traced to
 24 four chief reasons: the cost of small capacity units, the numerous codes that regulate the ammonia
 25 inventory included in residences and buildings, the sparsity of servicing knowledge, and the relatively low
 26 ratio of electrical to thermal energy prices.

27 The ideal has been, for over a century now, to increase the coefficient of performance of the single
 28 effect (SE) cycle, typically around 0.5. Whereas the initial credit for this trend goes to Altenkirch with the
 29 early invention of the generator-absorber heat exchange cycle [2], other engineers and practitioners have
 30 persisted, in face of considerable difficulty, to make progress in an uncertain field. Because recording
 31 the contributions of all these pioneers is an impossible task and outside the scope of this paper, we
 32 mention just a few. The crucial work of Phillips [3] was to establish a comparative sequence of cycles:
 33 whereas the single effect could reach COPs of 0.5, the solution-cooled absorber promised 0.6, and the
 34 generator-absorber heat-exchange promised about 0.8 to 1. Alefeld and Radermacher [4] established a
 35 system of developing multi-stage cycles virtually covering all possibilities of heat transformation.
 36 Grossman and Zaltash [5] developed insightful, modular software to enable the simulation of many
 37 advanced cycles. Studies of and large physical realizations are the trademarks of D. Erickson [6] in
 38 multiple applications of food, chemical and power industries. Ziegler [7] has contributed much to cycle
 39 characterization, outlining the advantages and drawbacks of many cycles via insightful methods. The
 40 contributions of Coronas and his group [8] to ammonia-water cycles as well as constant innovation on
 41 working fluids are part of the process whereby new technology is developed.

42 Recently, ammonia-water cycles to heat domestic water have been researched. In this regard,
 43 compactness is of paramount importance, and adsorption of ammonia in carbon is being considered as a
 44 strategy to reduce complexity and to improve COP, currently about 1 [9]. Regarding absorption, a unit is
 45 studied theoretically and also experimentally in three configurations: a single effect (SE) , solution-cooled

absorber (SCA), and a GAX configuration [10]. Predicted performance for the SE, SCA, is a remarkable COP of 1.6, verified experimentally [10]. The GAX configuration does not reach the expected performance, with only a small improvement over the SE, SCA cycle. Significantly for the present paper, the evaporator is shown to limit the performance of the cycle. Another entry, with similar COP (1.56) for water heating, is that of [11], although the cycle is not clearly specified.

The appeal of water-heating is clear: a unit would operate continuously in the heating mode, thus avoiding any design compromises for cooling. Whereas the most advanced gas furnaces may reach efficiencies of 0.95, a heat-pump water heater can reach (with the SE configuration) a COP of 1.5, significantly better than the 0.95 of the furnace. There is, however, one obstacle: as the ambient temperature varies between summer (Su) and winter (Wi), the cycle must serve the heating needs with good COPs, and with the same components. It is hence of importance to map out how the different seasonal requirements might impact the parameters defining each cycle component. The intended envelope of ambient conditions and heated water is shown in Table 1.

In this paper, we endeavor to ascertain how to make the SE as simple as possible, while preserving the COP and temperature operating envelope. A SE can obtain a good COP, but in addition is the simplest of cycles, with means, at least to a first order, the one requiring less material and specialized construction. This should lead to a lower first cost in conjunction with reasonable COPs, which would show a definitive improvement over furnaces, even in a historical period when gas prices are low. Matters of some innovation concerning this venerable cycle are presented in what follows.

2. The cycle

After a number of iterations concerning what cycle to adopt, the SE effect cycle dedicated to water heating was modeled according to the schematic of Fig 1. The state points corresponding to the solution (1 to 9) are placed in the schematic P-T-X of Fig 2. The solution out of the boiler (pt. 1) is cooled in the solution heat exchanger SHX (2a), expanded through a valve to 2b, and enriched in ammonia in the absorber (3a), which rejects heat to the domestic water. After the absorber, the solution is subcooled in Sub 2 by domestic water, (3asub) (the subcooling depending on the pump type and particular design) and pumped (3b) via the SHX to the distillation column, pt. 4. The solution entering the generator at pt. 4 is often referred to as “feed” in the literature for distillation. The ammonia refrigerant is separated from the solution and leaves as pt 5 in the boiler, with heat input from combustion products of natural gas. The solution poor in ammonia leaves the boiler at pt. 1, to recommence the cycle. In the condenser, domestic water receives the heat of condensation, whereby saturated liquid results (pt 10), which is then subcooled by domestic water in Sub 1 (pt 6), below the condenser. The subcooling is sufficient as to obviate the use of a precooler, which in normal SE cycles serves to cool the refrigerant with vapor from the evaporator previous to expansion in order to increase the evaporator capacity. Saturated liquid is returned to the distillation column as a stream commonly referred to as reflux R (pt 10). The rest of the liquid is subcooled, as already mentioned, in Sub 1 (pt. 6) and expanded (some vapor is generated by the expansion) to pt. 7. The remaining liquid is evaporated via heat addition, pt. 8. As ammonia evaporates, the water concentration of the liquid remaining in the evaporator increases. The vapor pressure of water is much smaller than that of ammonia, and it is inevitable for the conditions assumed in this work that some liquid will remain unevaporated. To allow such liquid to stay in the evaporator would result in excessive liquid accumulation, which could reabsorb some ammonia during transients. Liquid which due to large water concentration is not evaporated, is returned to the absorber as pt. 9. The source for evaporator heat input can be ambient air, or, as an innovative idea floated by the present work, from combustion products. Clearly, the vapor at pt. 7 and the liquid at pt. 9 imply thermodynamic losses, and are calculated as such in the model presented in an upcoming section. In the case of vapor generated by expansion at pt 7, this vapor does not contribute to the evaporator load. In the case of unevaporated

liquid (pt 9), the same observation can be made. As mentioned before, the subcoolers Sub1 and Sub2, the condenser and the absorber reject heat to the domestic water.

Regarding state points 1cp to 4cp, they correspond to combustion products. The boiler is heated by the combustion products of natural gas, and their temperature is reduced to that of state 2pc. At this point, the combustion products could enter an air preheater. The possibility contemplated in the Fig. 1 is different: the combustion products activate the evaporator, with a further reduction in temperature. Hence, two cases are considered in this work. One is the conventional air-source evaporator, and the other is a combustion products assisted evaporator, or CPAE.

The simulation of the single effect cycle called for a number of assumptions regarding the physical possibilities of each component from a standpoint of heat transfer performance.. Regarding each individual component, configurations were assumed that could be readily translated into practical realizations. A standard boiler vessel, with an adiabatic purification column on top and heat input from combustion products was assumed. The solution pool in the boiler vessel would receive the heat input across a finned wall, and the ensuing vapors would travel upwards in the column, with the liquid from the feed (pt 4, fig1) and from the reflux (pt 10, Fig 1) flowing downwards, washing the vapors in counterflow, and coming to equilibrium in each one of the column plates. The solution pool in ammonia would leave the boiler vessel as pt. 1. The condenser would be contained in a single shell, with an interior coil, cooled by domestic water. The vapors condensed in the counterflow coil (even at low water concentrations, a counterflow coil is deemed better for our purposes) would be collected as saturated liquid in an intermediate tray. From the tray, saturated liquid would flow for reflux duty in the column. Tray overflow would collect at the bottom of the condenser, with incoming domestic water subcooling the solution via an additional coil. The evaporator as simulated for this work could be an air-refrigerant crossflow configuration, whereby the refrigerant in tubes is evaporated by heat input from the air, outside finned tubes. As will be seen below, this air-refrigerant evaporator would call for large air flows and heat transfer area.

In the CPAE, combustion products produce the required evaporation while flowing counterflow to the vapor. The small flow rate of gases would call for a finned tube inside a shell, with the refrigerant flowing inside the tube. The length of this shell/tube arrangement is not known, but the overall UA for this evaporator is projected in this work to be much smaller than that for the air-refrigerant one. In terms of fan power, the CPAE is projected to be much more feasible.

The absorber is assumed of the conventional type, with absorption occurring as solution flows downwards over a surface cooled by domestic water flowing upwards. Typically, a shell enclosing a water-cooled coil with solution flowing downwards as a falling film can be used for the absorber. The SHX is of particular importance for the single effect cycle. A counterflow configuration with high effectiveness is assumed. Typically, plate heat exchangers can accomplish this duty.

3. The numerical model

The cycle model was formulated in EES [12]. The main constraint is to maintain the same vapor concentration (0.998) throughout both Su and Wi seasons (Table 1). Capacities are calculated for 1 kg·s⁻¹ of refrigerant generated at pt 5, Fig1. At the absorber and condenser exit, and at all stages of the column, thermodynamic equilibrium is assumed. Temperature approaches for the air-cooled evaporator, and water-cooled absorber and condenser are larger than 3 C. For the boiler, approaches in the order 300 C are reasonable. The SHX has reasonable effectiveness (0.8) for nearly equal stream heat capacities. The activating fuel is natural gas, with a heat of combustion of 50,000 kJ/kg. For the CPAE, the low Wi pressures were maintained to maximize temperature differences, and to effect a fair comparison to the

air-source evaporator. However, much remains to be gained by letting the CPAE pressure increase, which is not part of the reported work. The model is presented briefly, but the simulation of the distillation column and of the combustion process are covered in some detail, due to their centrality for our topic.

3.1 Generic model

After a comparison of property values between those made available by EES [12], and those by Conde [13, 14], which yielded generally satisfactory results, the simulation of the cycle was undertaken in EES. For each component, the generic steady-state mass and heat balances are:

$$\dot{m}_{l,i} + \dot{m}_{v,i} = \dot{m}_{l,o} + \dot{m}_{v,o} \quad (1)$$

Ammonia balance:

$$\dot{m}_{l,i} \cdot x_i + \dot{m}_{v,i} \cdot y_i = \dot{m}_{l,o} \cdot x_o + \dot{m}_{v,o} \cdot y_o \quad (2)$$

Energy Balance:

$$\dot{m}_{l,i} \cdot h_i + \dot{m}_{v,i} \cdot h_i + \dot{Q}_{comp} = \dot{m}_{l,o} \cdot h_o + \dot{m}_{v,o} \cdot h_o \quad (3)$$

In Eq.3, $\dot{Q}_{comp}$ stands for the heat rate, positive if flowing into the component. The specific enthalpies h depend on three variables, typically pressure, temperature and concentration. Other sets, incorporating quality, are possible. Thermodynamic equilibrium is assumed at the exit of each component, except for state point 3_{asub}. All results, except where otherwise noted, apply to $\dot{m}_5 = 1 \frac{kg}{s}$; $y_5 = 0.998$.

Two extreme conditions are simulated, namely Su and Wi. In Su, domestic water inlet temperatures are high. In Wi, the temperatures are much lower, and so are the high-side pressures. The solution feed temperatures change seasonally, and the simulation pays special heed to balancing the purification column in both seasons. In addition, when an air-source evaporator is modeled, the winter air temperatures force a decrease in the low-side pressure. Yet, one of the strengths of ammonia-water technology is that the desired water delivery temperature can still be maintained at Wi conditions. The CAEP was simulated only for the design Wi conditions, for its feasibility for Su is easily ascertained conceptually if water delivery temperatures do not vary seasonally, as intended.

3.2 Distillation operation

The key variables in distillation are the reflux, number of theoretical plates, and the resultant heat input to obtain, in our case, a vapor exit concentration (y_5) of 0.998. The high ammonia concentration is necessary to limit evaporation losses when evaporating temperatures are low. The inlet conditions (pt. 4, Fig 1) change from Su to Wi, and hence it is important to determine if a Wi design will work in the Su, and whether the reflux or injection point can stay the same or must vary.

With reference to Fig 3, the column has two rectification stages (I is the top one), a mixing one, where stream at pt. 4 enters, and a stripping one, with heat input from hot gases at the bottom. A generic heat and mass balance that applies to each adiabatic plate within the column reads:

$$\dot{m}_{l,n} + \dot{m}_{v,n} + \dot{m}_{l,4} = \dot{m}_{l,n-1} + \dot{m}_{v,n+1} \quad (4)$$

In Eqs.4, 5 and 9, $\dot{m}_{l,4}$ is zero except in the mixing plate. The conservation of ammonia calls for

$$\dot{m}_{l,n} \cdot x_{l,n} + \dot{m}_{v,n} \cdot y_n + \dot{m}_{l,4} \cdot x_4 = \dot{m}_{l,n-1} \cdot x_{n-1} + \dot{m}_{v,n+1} \cdot y_{n+1} \quad (5)$$

172 At the column pressure, the vapor and liquid at any stage are in equilibrium, namely:

173
$$y_n = f(x_n, p, T_n) \quad (6)$$

174 The liquid and vapor enthalpies at saturated conditions are given by EES, so that:

175
$$h_{l,n} = f(T_n, p, Q = 0) \quad (7)$$

176
$$h_{v,n} = f(T_n, p, Q = 1) \quad (8)$$

177 The energy balance for each plate reads:

178
$$\dot{m}_{l,n} \cdot h_{l,n} + \dot{m}_{v,n} \cdot h_{v,n} + \dot{m}_{l,4} \cdot h_{l,4} + \dot{Q}_B = \dot{m}_{l,n-1} \cdot h_{l,n-1} + \dot{m}_{v,n+1} \cdot h_{v,n+1} \quad (9)$$

179 In Eq. 9, $\dot{Q}_B$ is null except at the boiler, which is plate no. V. In the case (infrequent) that pt.4 were
180 superheated, an adiabatic energy balance, with mass balances, can be used to calculate the flashed
181 vapor and ensuing temperature, but in most cases pt. 4 is slightly subcooled.

182 The heat rate required to condense the reflux defines its mass flow rate, relevant for the balances of plate
183 I. The reflux heat rate is adopted as the independent variable for the deliberations that follow. As the
184 domestic water temperature varies from Su to Wi, so does the condenser minimum temperature and
185 pressure. The column pressures for the Wi-Su range (1118 kPa to 1600 kPa) were calculated. To each
186 pressure, there corresponds a solution feed temperature. The feed temperature was varied linearly
187 between the approximate minimum (Wi, 110 C) and the approximate maximum (Su, 137 C). So,
188 corresponding to each column pressure there is a solution inlet temperature.

189 To each set of conditions, there corresponds a minimum reflux rate. Such rate is found numerically, by
190 iterating on the reflux heat rate for a fixed vapor mass flow rate and concentration leaving the column at
191 pt. 5. ($y_5=0.998$, $\dot{m}_5=1 \text{ kg s}^{-1}$). For a given pressure and feed temperature, the reflux heat is reduced
192 from a starting value in a decreasing sequence. Initially we use 10 kW decrements, then several 5 kW
193 ones, and then positive or negative intervals of 1 kW. The search direction is changed when the
194 condition $TV \leq T_4$ is observed. The process stops when $TV > T_4$ in the 1 kW interval domain. So, the
195 reflux rate is known to 1 kW. As the boiler exit temperature decreases, so does the boiler heat input,
196 which is also observed as a feedback variable. Large boiler inputs (>3000 kW) are of no interest, and
197 generally align with large reflux duties.

198 The results of these exercises can be appraised via Figs 4 and 5. Figure 4 shows that the reflux
199 decreases towards the low feed temperatures of the Wi season, which imply a smaller ammonia vapor
200 pressure. The boiler heat input increases towards Wi, also reflecting a cooler solution entering the boiler.
201 This behavior is reasonable, if consideration is given to the higher vapor pressures of ammonia in the
202 summer, arising from higher solution temperatures in that season. The oscillations in the boiler duty
203 curve could be probably eliminated by adopting smaller intervals for searching the reflux rate, but such
204 fineness was deemed unwarranted.

205 The condenser temperature increases as Su conditions are approached, reflecting higher domestic water
206 temperatures, Fig 5. The boiler exit temperature (T_1) also increases as Su conditions are approached,
207 reflecting that the solution increases all its operating temperatures as the evaporator pressure increases
208 with the warmer season.

209 The distillation column seasonal simulations then establish that the reflux rate (pt. 10 in Fig 1) must vary
210 seasonally if the injection point, inlet and exit concentrations to the boiler are to be maintained
211 substantially constant all year around.

4. The combustion products and the evaporator

Evaporator performance prediction is not a simple problem, particularly in Wi. At low pressures, ammonia tends to evaporate readily, leaving behind a solution of progressively increasing water concentration. Assuming that the absorber is capable of maintaining constant evaporator pressure, the liquid concentration can be computed as a function of the vapor mass flow rate, Fig 6.

As the liquid concentration becomes ammonia-poor, the saturation temperature increases, Fig 7. Whereas the increase may be tolerable in Su, the low source air Wi temperatures renders the avoidance of liquid refrigerant a matter of considerable difficulty. The air is not hot enough to sustain water evaporation. The typical outcome is evaporators of large conductance (UAs), and still a liquid stream at pt.10. The latter represents a loss, arguably smaller the higher the evaporating temperature. Yet, reaching those higher temperatures comes at a high expense: large UAs and considerable large air flow rates. This implies large parasitic power required by large fans to move the necessary volumes of air.

Dividing the temperature vs. vapor mass flow rate as to approach linearity for each increment of vapor, and calculating the corresponding UA results in a large value. To attempt a normalization, the UA for the evaporator is divided by its capacity, and the resulting value discussed as this section develops. It is anticipated here that the normalized value is large.

For a heater, another heat source could be used to advantage in reducing both the UA and the unevaporated loss. The process schematic is shown in Fig 8, whereby combustion products activate the evaporator after activating the boiler and previous to preheating air. Assuming ideal gas behavior for the air and combustion products, and a HHV for the fuel of $50,000 \text{ kJ kg}^{-1}$, it turns out that it is possible to serve the boiler, evaporator and air preheater with combustion products. The results of combustion calculations are shown in Table 2 [15]. It is noteworthy that the Wi acid dew point, estimated for this application at 100 C (SO_2 by volume of 0.0012 [16]), is not reached in the evaporator or preheater, if used. Nevertheless, some portions of the evaporator might need special materials to avoid the spectrum of corrosion.

The outcome of this novel evaporator concept is remarkable. In Wi, ambient air activation results (for 1 kg s^{-1} of 0.998 ammonia rich vapor in the distillation column) in the UAs and air flows of Table 3. By contrast, the combustion products assisted evaporator (CPAE) yields a much smaller UA, and the same small fan for forced-draft combustion could activate the evaporator.

What is most remarkable regarding the CPAE is a much smaller gas flow rate (from 250 to $2.7 \text{ kg}^{-1}\text{s}$), possible due to the large temperature differences available to the combustion gases. In Wi, the air is cold. Its function as an effective heating source requires a temperature drop. Circulation of large amounts of air across the evaporator (250 kg s^{-1}), is necessary to keep the temperature drop within acceptable limits. Naturally, the amount of air could be reduced by allowing the peak evaporator temperature to drop. The consequence of such action would be to increase the unevaporated refrigerant loss.

5. The SHX and cycle performance

The mass fraction difference (x_1-x_3 , Figs 1 and 2) is an important variable. Its importance ensues from the fact that the smaller it is, the larger the flow rate of solution will become for a given amount of refrigerant to be absorbed. The general trend over the years has been to increase the concentration spread, in an effort to reduce the circulation losses associated with large solution flow rates. Increasing the concentration spread generally entails higher peak boiler temperatures and additional heat exchanger area. In the present case, a small concentration difference was adopted, since the availability of

effective plate heat exchangers may lead to loss reduction. So, the SHX is anticipated to be a large one, but of achievable effectiveness.

For $x_1=0.2$ and $x_3=0.25$, our calculations based on the cycle temperatures given below, demand an SHX effectiveness of 0.83, with an NTU of 4.7, using the conventional method [17]. Even though the heat capacities of both participating streams are nearly equal, the calculated effectiveness is plausible. Generally, heat capacity ratios close to 1 do not reach high effectiveness unless the NTUs are large.

The Wi and Su performance calculated with the enhancements noted are summarized in Tables 4 and 5. The cycles as calculated could perform well as water heaters. In Wi, when conditions are more stringent, the cycle could deliver 8.8 kg s^{-1} of water, heating it up from 1.67 C to 75 C. This value is normalized for our standard vapor exit conditions, namely:

$$\dot{m}_5 = 1 \text{ kg s}^{-1} \quad y_5 = 0.998 \quad (10)$$

The Su cycle could deliver an even higher flow, in the order of 10.4 kg s^{-1} of domestic water. The projected COP ranges from 1.45 to 1.47, which is, for a SE cycle, a reasonably good value. The reader probably notices in Tables 4 and 5 that the pump inlet solution is subcooled, 2 C in Wi, 2.5 C in Su. The reasons for and magnitude of subcooling will be explained in the next section. To maintain conditions 10, the solution flow rates must vary seasonally.

In addition to the seasonal variations of capacities, Tables 4 and 5 point at two control aspects of definitive importance. To discuss them we invoke, in addition to the said Tables, Fig 4. It is clear that to maintain the same number of stages and injection point of the purification column as operating conditions change with the seasons, two magnitudes must be controlled: the reflux flow (pt 10 in Fig 1), and the solution flow rate (pts 1 and 3b in Fig 1).

The reflux flow control is clearly the most demanding. A modulating flow control valve responding to condenser pressure or temperature deserves consideration, although other controlling variables (i.e. feed temperature (pt 4 in Fig 1) may be more suited. For the solution loop, a variable speed pump in conjunction with a variable expansion valve are recommended, although a variable pump bypass could do as well. The controlling variable could simply be the feed temperature, (pt 4 in Fig 1). Finally, the fuel and air flow rate should be controlled in unison, with a possible control variable being the condenser pressure. Simulation of these control strategies, a difficult subject, is beyond the scope of this work. It should be however mentioned that the possibilities afforded by a single microprocessor controlling all necessary variables certainly deserves consideration.

6. Pumps and subcooling

Pumps and ammonia-water are another difficult topic. The large pressure rise coupled with low flows, and the always present cavitation danger make pump selection a challenge. The specific speeds of pumps for a 15 kW heater are low, which leads to the selection of a positive displacement (PD) pump. In order to gain some knowledge, commercially available pumps for the 15 kW application were surveyed.

The first conclusion of the survey is that all the PD pumps have a $NPSH_r$, which must be met for trouble-free operation. It is customary in pump practice to reduce all the pressures to geometric heads in m of the fluid pumped. This is done by dividing the pressure by the product of the gravitational constant and the fluid density. The available head at the pump inlet $NPSH_a$, must exceed the $NPSH_r$ as given by [18]:

$$NPSH_a = H_a + H_s - H_{vp} - H_f > NPSH_r \quad (11)$$

The geometric meaning of the different heads that make up $NPSH_a$ are given in the nomenclature and illustrated with the aid of Fig 9. H_a is the equivalent head from the absorber pressure. H_s is the static head above the pump inlet, which in the present application should be kept to a minimum for space reasons, although it might be worthwhile to place the pump on the floor with all components on top of the heater in order to maximize H_s . H_f is the friction loss in the feed line, and its minimization calls for short, large diameter lines. Finally, H_{vp} is the equivalent head as given by the solution vapor pressure at the pump inlet.

If H_a and H_{vp} values are close in value, meeting $NPSH_r$ must come from physically elevating the absorber a length H_s above the pump inlet. In addition, H_f must be kept to a minimum by using a large diameter feedpipe. The only other alternative to large gravitational heads is to explore subcooling the solution with the domestic water. A plot of gained head vs subcooling is shown in Fig 10. Large subcoolings imply a more compact design.

Subcooling denotes lack of thermodynamic equilibrium. Subcooling may happen in ammonia-water as a consequence of poor mass transfer in the absorber. If the solution is cooled, but it does not absorb enough refrigerant to reach equilibrium at the vapor pressure of the absorber, then it is subcooled. Whereas subcooling is more frequent in LiBr- H_2O machines than in ammonia ones, a bad absorber design makes subcooling a real possibility. The net outcome is that the circulating solution does not absorb as much refrigerant as projected. The capacity of the cycle then decreases, with deleterious COP effects. In our simulations a subcooling of 0 C at the absorber exit is assumed, and deemed possible with a good design.

If the solution is instead cooled downstream of the absorber (with no additional absorption) in a dedicated small HX, then the effect of subcooling on performance is quite different than subcooling in the absorber. The effect on COP depends on how the heat obtained from cooling the solution is used, and on whether the SHX area is increased to keep approximately the same inlet temperature into the column or not. If the solution is subcooled slightly to ensure pump reliability, and the SHX is slightly upgraded to keep the same column conditions, then the effect on COP is (or should be) null. Even if the SHX is not upgraded, no large COP changes are anticipated.

It was found from the survey that several commercial pumps could meet the needs of a 15 kW heater, with some subcooling (Table 6). Inlet redesign may reduce the required subcooling. Based on expected reliability and service life, two types were found more adequate for the application. Table 6 simply outlines the potential of two pump types, and it is not a recommendation of commercial products for cycle development.

7. Conclusions

As noted above, the cycle could deliver hot water at 75 C all year round. To enable the design, both the reflux and the pumping flow rate must be controlled for proper operation of the distillation column. Gas flow should also vary seasonally. The controlling variable could be the condenser pressure or temperature. Perhaps the ambient temperature could serve as well. Clearly, much of the controls experience will have to be gained experimentally, for transient simulations of ammonia systems are challenging.

Adopting the CPAE could make a significant difference regarding the feasibility of the water heater in cold environments. The heater would call for much smaller fans than the air activated one, or simply for just one small fan to activate combustion and evaporator gas flow. This could entail substantial first and operational cost savings, if one considers that the acid dew point may still be avoided. The Su dewpoint

needs to be investigated further. Creative evaporator designs would be necessary, for the gas path should be much longer than for air activated units. The possibilities of CPAE to increase low-side pressures in all seasons deserve further consideration.

Elimination of the precoolers by subcooling the refrigerant in the condenser is another simplification possible for water heaters. This practice would decrease the number of shells and combine the condenser and subcooler into one unit, albeit with provisions to separate the saturated reflux flow, such as a perforated tray. The SE modeled here calls for an effective counterflow SHX, which is within the possibilities of plate HXs.

Compactness of the unit may be enhanced by subcooling the solution as a provision to meet the *NPSH_r*. Whereas PD pumps do not have large *NPSH_r*s, which depend on the inlet design as well, saturated ammonia solutions are difficult to pump, and subcooling by heating water is a way to keep the pump close to the absorber and to ensure reliable starts of a sealed system.

The SE cycle for a water heater could reach cycle COPs of 1.47 to 1.45. Parasitic power (i.e. fan and pump power) should be kept to a minimum. Recovering pump work with a set of small turbines may be a way to reduce parasitic power, but the concept needs further research. During start-up and shut-down, and during transients, losses tend to occur. Their magnitudes are hard to ascertain by modeling, but it stands to reason that a simplified SE with CPAE, devoid of a large evaporator and of other strategies for internal heat recovery, would stand a better chance to retain high seasonal performance.

All widely accepted technologies are enabled by fulfilling an objective with some clear advantage over the competing ones. Sometimes, lower operational costs are the enabling factor, sometimes low capital costs, more often than not is a favorable combination of both. Whereas it is hard to state what will enable ammonia-water heat pumps, it is easy to benchmark some variables. In going from a COP of 0.9 to 1.4 (gas heater to SE, say), gas costs would be reduced to 64% of those of the heater. At a cost of water heating of \$4500 yr⁻¹ (~100 gal day⁻¹ of hot water) [21], the buyer of the ammonia water heater could see a reduction to \$2880 yr⁻¹. Then, to recover the additional cost of a water heater over that of a gas heater, in the order of \$1600 yr⁻¹ would be available. So, for a 3 yr. time horizon, the heat pump could entail an incremental cost of \$4800. It is hoped that at least some of the ideas advanced in this work are part of an enabling set for heat pump water heaters.

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416 and the absence of a precooler.

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Figure 4. SE cycle schematic with state points. Notice the subcooler in the condenser and pump inlet, and the absence of a precooler.

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Figure 10. About 1 C of subcooling reduces 1 m of absorber elevation.

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Table 1. Seasonal temperatures for water heater, C		
Season	Domestic Water	Air
Su	18.3	29.4
Wi	1.7	-6.7
Delivery	75	N/A.

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Table 2 . Results of combustion calculation (Pt 5 above the dew point)				
State Point, Fig 8	Temperature, C (K)	Enthalpy, kJ kg ⁻¹	Composition (mol fraction)	Load ¹ , kW
1	10 (283)	-15.04	yO2=0.21 yN2=0.79	
2	25 (298)	50000	yCH4=1	
3	997 (1270)	1131	yCO2=0.041 yH2O=0.081 yO2=0.12 yN2=0.758	
4	420 (693)	433	"	1925
5	137 (410)	119	"	864
6	117 (273)	98	"	54
0	-10 (263)	-35.03	yO2=0.21 yN2=0.79	
1. $\dot{m}_g = 2.755 \frac{kg}{s}$ $\dot{m}_F = 0.063 \frac{kg}{s}$ $f = 0.02346$				

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Table 3. Comparison of evaporators, Wi		
Variable	Air	CPAE
$\dot{Q}_E$, kW	780	866
UA , kW K ⁻¹	442	3.2
Assumed U , W m ⁻² K ⁻¹	300	300
$A/\dot{Q}_E$, m ² kW ⁻¹	0.53	0.012
$\dot{m}_{hot}$ kg s ⁻¹	250	2.755

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Table 4. Wi cycle with CPAE and pump subcooling, 2 C					
Point	x	mfr, kg s ⁻¹	T, C	P, bar	h, kJ kg ⁻¹
1	0.2	10.16	120.6	11.18	383.3
2a	0.2	10.16	78.05	11.18	197.5
2b	0.2	10.16	78.19	11.18	197.5
3a	0.25	10.84	68.07	2.8	127
3asub	0.25	10.84	66.07	2.8	118.4
3b	0.25	10.84	66.24	11.18	119.8
4	0.25	10.84	108	11.18	302.5
5	0.998	1	48.94	11.18	1352
6	0.998	0.679	5	11.18	21.99
7l	0.9979	0.641	-10.87	2.8	-51.22
7v	1	0.038	-10.87	2.8	1256
8l	0.5955	0.0024	10	2.8	-184.6
8v	0.9994	0.6386	10	2.8	1306
9	0.5955	0.0024	10	2.8	-184.6
10	0.998	0.321	28.62	11.18	133.9
Other items					
Item		Value		Comment	
Pump work, kW		14.74			
Relative heat balance error		0.004		Error divided by total heat input.	
COPth		1.45			
COPpower		1.42		Includes pump work, @3.14 thermal equivalent	

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Table 5 .Su cycle with CPAE and pump subcooling, 2.5 C					
Point	x	mfr, kg s ⁻¹	T , C	P, bar	h, kJ kg ⁻¹
1	0.2	8.961	144.3	16	491.3
2a	0.2	8.961	112.9	16	361.1
2b	0.2	8.961	113	16	361.1
3a	0.25	9.56	110.6	9	314.2
3asub	0.25	9.56	108.1	9	303
3b	0.25	9.56	108.3	16	304.2
4	0.25	9.56	135	16	426.2
5	0.998	1	57.79	16	1349
6	0.998	0.599	22	16	102
7l	0.9981	0.5979	21.59	9	99.75
7v	1	0.001123	21.59	9	1287
8l	0.8857	0.009831	25	9	34.3
8v	0.9999	0.588	25	9	1297
9	0.9999	0.009831	25	9	34.3
10	0.998	0.401	57.79	16	195.3
Other items					
Item		Value		Comment	
Pump work, kW		11.45			
Relative heat balance error		0.015		Error divided by total heat input.	
COPth		1.47			
COPpower		1.44		Includes pump work, @3.14 thermal equivalent	

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Table 6. Two Potentially feasible pump types for a 15 kW heater operating as per Table 4		
Type	Diaphragm [19]	Piston [20]
Flow, l m ⁻¹	4	Up to 11.4
Pressure rise, bar	8.41	7 to 70
rpm	600	466 (at rated flow 4 l min ⁻¹)
Power, W	108	65
NPSHr, m (Deemed possible)	2.3	2.5
Inlet subcooling, C	2.3	1 to 5
Inlet temperature, C	68	68
Material	Possibly non-metallic	SS, consult manufacturer.

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