15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

Ammonia-water absorption cycle: heat transformer and hybrid reversible design to upgrade excess heat and produce power

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Abstract

Decarbonization of the industrial sector is a major challenge to reduce global greenhouse gas emissions. 36% of the industrial energy consumption is lost in the form of excess heat, with about 25% being between 40 and 100°C; therefore, the implementation of efficient thermodynamic cycles to recover energy from this heat is appealing. Ammonia-water absorption heat transformers (AHT) can upgrade industrial excess heat (~ 80°C) to higher temperature (> 100°C) suitable to run industrial processes. The valuable output of such cycle design is the heat released by the absorber. This work presents the experimental insights of a 10 kW NH_3-H_2O AHT prototype. Notably, complex dependencies of the components and the challenging operating conditions limit the performance. Another significant perspective of industrial excess heat recovery is power production. Thus, the present study introduces an innovative hybrid reversible design allowing for either upgrading excess heat temperature level or generate electricity. The energy outputs of this versatile cycle might serve internal industrial applications or integrate external power supply grid.

Keywords: Absorption heat transformer; Prototype; Hybrid reversible cycle; Industrial excess heat; Power production

1. Introduction

For the past century, the use of fossil fuels to meet the global energy demand has been largely predominant [1]. As of today, the excessive consumption of these toxic fuels and the irreversible environmental alterations to sustain constant growth led to cross seven out of the nine planetary boundaries [2]. The IPCC reported the scientific consensus about the responsibility of human activities on global warming and biodiversity destruction [3]. Thus, decarbonizing and changing energy consumption behavior is crucial. Most industries require energy in form of heat and about 36% of the heat used to run industrial processes is lost [4]. 25% of these excess heat losses are low-temperature heat, meaning between 40 and 100°C. From these numbers arises a major stake in upgrading excess heat.

NH_3-H_2O absorption systems are well established technologies to produce cold and heat at temperatures < 70°C in heat pump mode. These low-temperature absorption heat pumps are known as Type I heat pumps, which use two heat inputs at lower and higher temperatures and generate heat at an intermediate temperature level [5]. While the literature in regards to NH_3-H_2O absorption heat pumps is extensive, much less is known about thermodynamic cycles to upgrade heat at high-temperatures (> 100°C), such as Absorption Heat Transformers (AHT) [6]. AHT are Type II heat pumps using an intermediate temperature heat source and generating 2 heat sources at low- and high-temperatures; thus, operating reversely than type I absorption heat pump [5]. The diversity of working fluid available can be restricted by considering environmental impacts and operating condition range. For example, fluid categorized as PFAS can be ruled out [7]. NH_3-H_2O absorption cycles are good candidates to develop lower-impact thermodynamic cycles, with no vacuum-leak issues and wider range of operational temperatures and pressures compared to traditional LiBr-H_2O systems [8]. Despite the abondance of numerical modelling studies presenting various AHT architectures, experimental work

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testing high-temperature AHT is still lacking. Moreover, the versatility of energy systems is essential to guarantee resilience. Therefore, hybrid reversible cycles producing either high-temperature heat or electricity are a promising solution and should be further assessed in future research.

The present work characterizes the performance of a 10 kW NH_3-H_2O AHT prototype recently tested in our laboratory generating heat at temperature > 100°C and discusses the technological locks encountered for the development of hybrid reversible cycles.

2. Absorption heat transformer

2.1. Experimental tests

The studied NH_3-H_2O AHT prototype was built based on the modelling work of Collignon *et al.* [9]. The architecture comprises 4 heat exchangers (HX) being the main components of the thermodynamic cycle (generator, condenser, evaporator and absorber), 3 internal HX (Solution HX SHX1 and SHX2, Refrigerant HX RHX) and 2 pumps for circulating the condensed refrigerant and the NH3 poor solution. RHX allows for preheating the liquid refrigerant entering the evaporator, as well as precooling the refrigerant vapor entering the condenser. The architecture corresponds to a single-effect single-stage Type II heat pump, which means that a medium temperature heat input implies higher- and lower-temperature heat outputs [5]. Thus, the HX were sized to produce 10 kW of heat > 100°C at the absorber and reject low-temperature heat at about 20°C at the condenser from excess heat input at 80°C at the generator and evaporator. The NH_3 desorption and condensation take place on the low-pressure level (about 8 bar), while the refrigerant evaporation and absorption occur at about 25 bar. The rich solution exiting the absorber is then expanded through the expansion valve upstream the generator. Fig. 1 presents a simplified schematic of the aforementioned prototype.

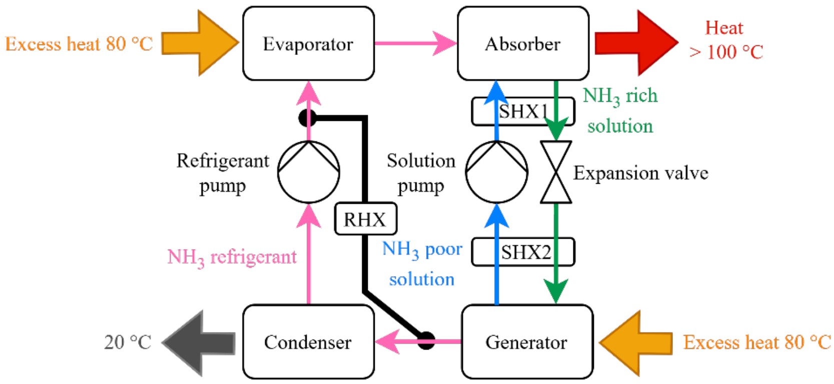

Figure 1. Simplified schematic of the 10 kW absorption heat transformer prototype.

The experimental rig was instrumented with type T thermocouples (± 0.1°C) measuring the fluid temperature leaving and entering each component and 4 pressure gauges (± 0.05%) reading the pressure level at the main HX. 3 mass flow meters (± 0.05%) measured the rich solution, poor solution and refrigerant mass flow rates, as well as their densities, allowing for calculating the NH_3 mass fractions of the fluids using REFPROP [10] for each data point collected every 5 sec. The inlet temperatures of the external heat transfer fluid (HTF) of the 4 main HX were set by the operator at the start of the experiments. The testing conditions to validate the operational proof of concept of the 10 kW prototype upgrading excess heat to temperature > 100°C are listed in Table 1 below.

Table 1. Prototype testing conditions to validate the proof of concept.

Operational parameter	Setting
Excess heat temperature (°C)	80
Absorber HTF inlet temperature (°C)	90
Absorber HTF mass flow rate (kg/h)	600
Condenser HTF inlet temperature (°C)	15

Usual indicators for heat pumps were used to evaluate the performances of the prototype:

- Thermal coefficient of performance (COP_{th}): ratio of the heat output at the absorber $\dot{Q}_a$ and the sum of the excess heat sources.

$$COP_{th} = \frac{|\dot{Q}_a|}{\Sigma|\dot{Q}_{excess\ heat\ source}|} \tag{1}$$

- Electrical coefficient of performance (COP_{elec}): ratio of the heat output at the absorber and the sum of the pumps power consumption.

$$COP_{elec} = \frac{|\dot{Q}_a|}{\Sigma W_{pump}} \tag{2}$$

- Gross temperature lift (GTL): difference between the heat temperature output at the absorber $T_{a,o}$ and the input excess heat temperature.

$$GTL = T_{a,o} - T_{excess\ heat\ source} \tag{3}$$

Varying the absorber HTF inlet temperature between 90 and 100°C and the condenser HTF inlet temperature between 15 and 30°C allowed for evaluating the performances of the prototype at different operating conditions.

2.2. Results and discussion

Fig 2 presents the measured performances at different absorber HTF inlet temperatures ($T_{a,i}$) and condenser HTF inlet temperatures ($T_{c,i}$). At the operational settings listed in Table 1, the prototype reached a GTL ≥ 20°C meaning heat at temperatures > 100°C, a COP_{elec} ≥ 25, a COP_{th} ≥ 0.40 and produced heat up to 10 kW. Thus, the proof of concept of this prototype is validated. However, deeper analysis would enlighten the parameters responsible for the wide distributions of the heat produced and resulting COP_{elec} at identical operating conditions. Moreover, further investigations are required to understand the behavior of the machine, such as the hydrodynamics imposed by the design. The system energy balance (Fig 2-e) is consistent along the different test. The mean 1 kW offset might be lost from the lack of insulation of the prototype, which should be considered for building further prototypes.

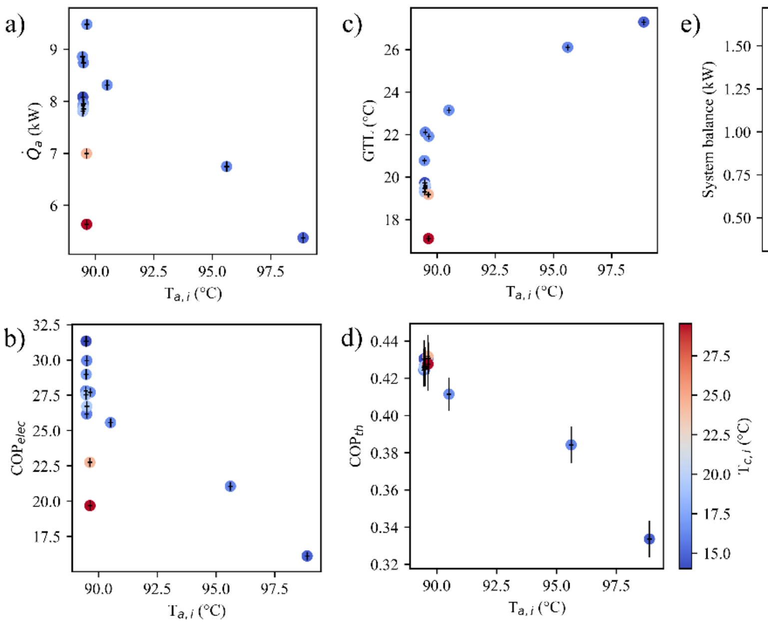

Figure 2. Distribution of the performances a) absorber heat output, b) COP_{elec}, c) GTL, d) COP_{th} and e) the energy balance of the system for the tests at different $T_{a,i}$. The color of the points correspond to $T_{c,i}$. The error bars were calculated applying the Root-Sum-Square method using the measurement uncertainties.

Globally, the performances decreased with increasing $T_{a,i}$. The GTL increased with higher $T_{a,i}$, but looking at the temperature difference (ΔT) between the absorber HTF outlet and inlet, ΔT is around 12, 10 and 7°C for $T_{a,i}$ of 90, 95 and 100°C respectively. Thus, ΔT dropping confirm the performance degrading for $T_{a,i} > 90$°C. The data trends of these experimental results are similar to the reported data of Garone *et al.* [11] on their NH_3-H_2O AHT prototype upgrading excess heat at 60°C with absorber temperature varying from 70 to 80°C.

Looking at the varying parameter $T_{c,i}$ for a $T_{a,i}$ of 90°C, the heat produced at the absorber (Fig 2-a), the COP_{elec} (Fig 2-b) and COP_{th} (Fig 2-d) peaked for a $T_{c,i}$ of 15°C and higher $T_{c,i}$ affected the performances. For example, the heat produced at the absorber dropped < 6 kW for a $T_{c,i}$ of 30°C. The GTL also fell to a low point when $T_{c,i}$ increases from 15 to 30°C (Fig 2-c). Interestingly, the thermal COP did not reflect the heat production and GTL drops at higher $T_{c,i}$.

These primary results highlighted limitations and improvement perspectives for NH_3-H_2O AHT. The significant decrease of the performances when $T_{c,i} > 20$°C is critical for the implementation of AHT technologies. The need for a low-temperature cooling source introduces additional constraints and limits the range of potential applications. Consequently, alternative architectures and thermodynamic cycle types should be investigated to enable the upgrading of industrial waste heat with less constraints.

3. Hybrid reversible cycle

3.1. Numerical modelling

The NH_3-H_2O hybrid reversible cycle can operate in 2 modes with the same architecture comprising two HX, a pump, a reciprocating piston compressor/expander and an expansion valve as described in Fig 3. In heat production mode, this specific design allows for upgrading excess heat to higher temperature with the generator at low pressure and the compressor flowing the refrigerant vapor to the absorber at high pressure. Inversely, in power production mode, the high-pressure HX can operate as the generator and the refrigerant is expanded to the low-pressure HX being in that configuration the absorber. Thus, the work produced by the expander generates power. Either the poor or rich solution flow into the pump and the expansion valve depending on the operating mode of the cycle, heat or power production.

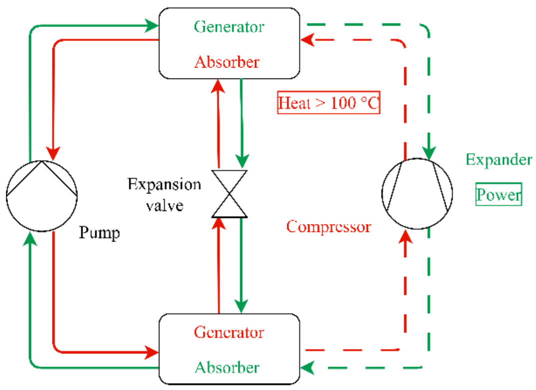

Figure 3. Simplified schematic of a hybrid reversible cycle producing either high-temperature heat or power.

The cycle for the heat production mode was conceptualized with numerical modelling in stationary conditions using the *Engineering Equation Solver* (EES v.9) software [12]. The performances expected are a $COP_{elec} \geq 4$ for a heat output of 10 kW_{th} at temperatures between 100 and 150°C. The two HX are plates HX, which will be sized and designed to operate reversibly, as both absorber and generator at the two pressure levels. The high-pressure HX, in both modes, operate with an HTF inlet temperature of 90 – 100°C. The low-pressure HX HTF inlet temperature varies depending on the mode: waste heat temperature (80°C) in heat production mode, and ambient temperature in power production mode (20°C). The compressor/expander is a custom-made reciprocating piston compressor compatible with NH_3.

The numerical model for the heat production mode converged based on thermodynamic equilibrium and efficiencies to the different components. The parameters imposed are summarized in Table 2. To consider the heat and mass transfers occurring in the absorber, the model developed by Collignon *et al.* [9] was applied. This model assumes that a first section of the absorber is adiabatic, resulting in a saturated rich solution at higher temperature than the poor solution inlet. Then, this hotter rich solution continues to absorb NH_3 vapor as its cooling down by transfer to the HTF in the cooled section of the absorber. The HTF outlet temperature is deduced from the absorber thermal efficiency, the HTF inlet temperature and the temperature of the hotter saturated rich solution exiting the adiabatic section of the absorber. The NH_3 fractions of the vapor and the poor solution are calculated considering that both streams are saturated at the temperature of the fluid leaving the generator at the low-pressure. The enthalpy of the compression is calculated using the compressor efficiency and the enthalpy considering an isentropic compression. The expansion through the valve is assumed adiabatic.

Table 2. Imposed parameters for the numerical modelling of the heat production mode.

Parameters	Set value
Pump efficiency	0.6
Compressor efficiency	0.65
HX pinch points / thermal efficiencies	5°C / 0.8
Absorber mass efficiency	0.7

Primary results from the numerical modelling in heat production mode are presented in Fig 4. The absorber and generator HTF inlet temperatures were set to 90 and 80°C, respectively, and the heat output at the absorber fixed to 10 kW_{th}. Data are obtained at different compressor speed (400 to 800 rpm) for different compression ratio (CR). Fig 4a shows that $COP_{elec} \geq 4$ and outlet temperature $T_{a,out}$ up to 120°C are reachable. However, what stands out in Fig 4b is that the lower the high-pressure level P_{high}, the lower the NH_3 fraction x_{vap} in the refrigerant vapor.

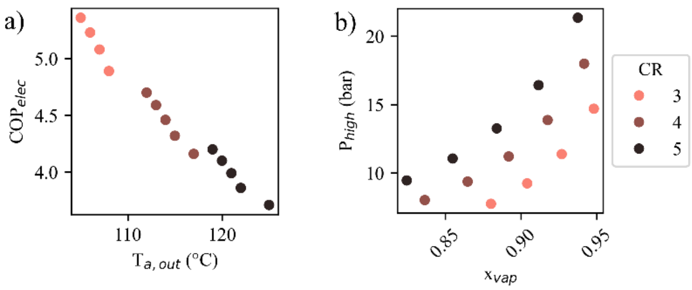

Figure 4. Performances in heat production mode at different CR: a) COP_{elec} function of the absorber HTF outlet temperature and b) high-pressure level function of the NH_3 fraction in the refrigerant vapor.

3.2. Discussions and perspectives

This first modelling analysis suggests that, in the studied operation range, the best optimal performances occur for a CR of 4, compromising with a COP_{elec} between 4 – 5 and $T_{a,out} \geq 110$°C. Higher CR allows for higher temperatures but is detrimental to the COP_{elec}. Also, the refrigerant fraction in the vapor must be taken into account because the compressor would not tolerate high amount of water because of condensation that could damage the component. Thus, only $x_{vap} \geq 0.97$ are acceptable to avoid jeopardizing the compressor. According to the aforementioned results, higher NH_3 fraction implies higher pressure levels and CR which would increase drastically the energy consumption of the system. The addition of a rectifier in the cycle architecture could be considered to overcome this issue. Further investigations on the reversibility of this cycle must be carried out to frame the optimal and realistic operating conditions and revise the performances expectation of the experimental prototype that may arise from future study.

4. Conclusions

An operational NH_3-H_2O AHT prototype reached the targeted performances of producing 10 kW of high-temperature heat (> 100°C) from excess heat at 80°C with COP_{elec} up to 30. The prototype performances under various conditions must be further analyzed to have a more complete characterization of its hydrodynamics behavior. But these first experimental results already show that the low-temperature (< 20°C) needed at the condenser is a major drawback for implementing AHT technologies in industrial facilities.

Therefore, an innovative hybrid architecture is under investigation to improve the versatility and resilience of NH_3-H_2O thermodynamic cycles upgrading excess heat. The reversibility of the components, including the reciprocating piston compressor/expander and the absorber/generator HX are critical technological challenges. The results from the experimental behavior drawn for the AHT and the numerical modelling will assist in prototyping the reversible hybrid cycle required to assess the relevance of the system.

Moreover, industries must undergo Life Cycle Analysis of their process to evaluate precisely their environmental impacts and compare the effect of implementing such thermodynamic systems; as well as, the environmental footprints particularly if alteration to biodiversity and ecosystems occur.

Acknowledgements

The authors acknowledge Fabien Bruyat, Bertrand Chandez and Florent Lefrançois for building the AHT prototype and assisting the experimental tests, and Romain Collignon for the preliminary modelling work and sizing of the AHT.

The authors acknowledge that this work has been supported by the French Alternative Energies and Atomic Energy Commission (CEA) and the Carnot Énergies du futur institute.

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