

New Power and Cooling Absorption Cycles

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Abstract

In this study, new absorption cycles for the combined cooling and power production are presented. These cycles are designed by modifying basic (single-stage) and two-stage double-effect absorption cooling cycles, which use ammonia/water mixture as a working fluid. The performance and operating conditions of the cycles are analysed systematically through thermodynamic cycle simulation and sensitivity study. The proposed absorption cycles are capable of providing different ratio of power and cooling to cover varying demand profile for power and cooling. They can be activated by low and medium temperature heat sources such as solar thermal energy, biomass, geothermal energy or industrial waste heat. The performances of these cycles were evaluated by several performance indicators, based on energy and exergy concepts. It is found that the hybrid two-stage absorption cycle has an overall thermal efficiency of 55%, effective first law and exergy efficiencies of 13% and 40%, respectively, for the base-case conditions considered (at a split ratio of 50% and a heat source inlet temperature of 170°C). The effect of the non-conventional working fluid mixtures (NH₃/LiNO₃ and NH₃/NaSCN) on the performance and operational working range of the single-stage hybrid cycles has been evaluated for different modes of operation, from pure refrigeration cycle mode to a combined cycle that give high priority to power production.

Keywords: *Combined cooling and power, hybrid absorption cycles, working fluid mixtures, performance indicators*

1. Introduction

Refrigeration and power (mechanical or electrical) are useful form of energy, usually produced using separate energy conversion systems. Most end-users need both products. The combined production of these useful outputs, refrigeration and power, can improve significantly the primary energy utilization efficiency than the separate production. Moreover, minimized utilization of primary energy such as fossil fuels could be reflected through lower carbon emission and ozone depletion. Absorption cycles are capable of producing both refrigeration and power by utilizing low and medium temperature heat sources such as solar thermal energy, geothermal energy, biomass or industrial waste heat.

Absorption refrigeration is a well-known technology for refrigeration applications. Various absorption cycle configurations with ammonia/water (NH₃/H₂O) mixture as a working fluid, has been suggested in the literature for the combined production of refrigeration and power using the same thermodynamic cycle [1-4]. Some of these cycles are hybrid Rankine and absorption chiller cycles, modification of the Kalina cycle [5] or derived from absorption refrigeration cycles [2]. Most of them are characterized by either higher driving heat source temperature, very complex cycle configuration or

low refrigeration output capability as in the case of Goswami Cycle [1]. Although the Goswami cycle can be driven by a heat source temperature below 200°C, the refrigeration output is relatively small.

In order to overcome some of the drawbacks associated with the conventional NH₃/H₂O working fluid and to reduce the activation temperature of the cycle, mixtures of NH₃ as refrigerant and inorganic salts such as lithium nitrate (LiNO₃) or sodium thiocyanate (NaSCN) as absorbent have been investigated for absorption refrigeration cycles over the previous years. The main advantage of these mixtures is that no rectification process is necessary due to the fact that there is a high boiling temperature difference between the absorbent and refrigerant.

The objective of this study is to systematically analyze the performance and operating conditions of the new proposed absorption cycles: single-stage hybrid absorption cycles and a two-stage hybrid absorption cycle for the combined production of cooling and power. The effect of the non-conventional working fluid mixtures (NH₃/LiNO₃ and NH₃/NaSCN) on the performance and operational working range of the single-stage hybrid cycles has been evaluated for different modes of operation, from pure absorption refrigeration cycle mode to a combined cycle mode that give high priority to power production (Goswami cycle) .

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2. Working fluids

The performance (efficiency) of a reversible thermodynamic cycle is independent of the working fluid used in the cycle. However, the efficiency, design and operational characteristics (such as the temperature and pressure working ranges) of real (irreversible) cycles are determined to a larger extent by the properties of the working fluid which causes various internal irreversibilities. For the thermodynamic performance analysis of the proposed cycle configurations presented here, the thermodynamic properties of the conventional $\text{NH}_3/\text{H}_2\text{O}$ mixtures were obtained from the work of Tillner-Roth and Friend [6]. Previously, at the Group of Applied Thermal Engineering-CREVER, the thermophysical properties of the non-conventional working fluid mixtures ($\text{NH}_3/\text{LiNO}_3$ and NH_3/NaSCN) necessary for the thermodynamic evaluation of the absorption cycles were measured and correlated [7-9].

3. Cycle Configurations

In general, absorption cycle configurations are designed for serving one purpose such as cooling, heating or power generation. Two or more independent flow streams exist in absorption cycle configurations. Consequently, two purposes (useful outputs) can be fulfilled simultaneously or alternatively by incorporating additional heat exchange units and mechanical (pressure) devices. Here, exchange unit refer to heat and/or mass exchanger whereas pressure devices refer to expander, compressor and ejector. Dual-purpose absorption cycle configurations have advantages from technical and performance perspectives, especially when cooling demand fluctuating while the power supply demand remains constant. For exploiting low-grade and midgrade heat sources, new dual-function absorption cycle configurations are proposed for the simultaneous production of cooling and power. These cycle configurations are based on the modification of basic (single-stage) and two-stage absorption cooling cycle configurations. Therefore, the proposed cycle configurations are grouped in to two parts: i) Those derived from the single-stage absorption cooling cycle for utilizing low-grade heat sources are under the category of single-stage hybrid absorption cycles and, ii) the rest which is categorized under a two-stage hybrid absorption cycle for the simultaneous production of cooling and power by utilizing midgrade heat sources. Further improvements on the performance and operational working range (such as lowering the activation temperature) of these cycles can be attained by incorporating the pressure devices such as compressor and ejector. The details of the proposed cycle configurations are described below.

3.1. Single-stage hybrid absorption cycles

There are two cycle design configurations based on the type of absorbent (for the NH_3 -refrigerant) in the working fluid mixture. Fig. 1 shows the cycle configurations using the volatile absorbent (H_2O) and non-volatile absorbents (LiNO_3 and NaSCN), respectively. The main components of these cycle configurations are the absorber (ABS), the condenser (CON), the desorber/generator (DES), the evaporator (EVA), the solution heat exchanger (SHX), the refrigerant sub-cooler (RSC) and the rectifier (REC) for a working fluid having high or moderate relative volatility between the refrigerant and the

absorbent (as in the case of $\text{NH}_3/\text{H}_2\text{O}$ working fluid). For generating mechanical power, the expander (EXP) and mostly super heater (SH) are incorporated in the power sub-cycle. The cycles also include a solution pump (SP), refrigerant and solution expansion valves (REV and SEV respectively) for sustaining the pressure difference between the high and low pressure side components.

3.1.1. Process description

As illustrated in Figs. 1, the single-stage hybrid cycle configurations have two sub-cyclic processes (namely-refrigeration and power sub-cycles) for the production of the dual-outputs. The entire cyclic process is described based on the configuration with $\text{NH}_3/\text{H}_2\text{O}$ working fluid mixture as follows: In Fig. 1a, the basic solution (strong in NH_3) leaves the ABS as saturated liquid. It is pumped to the system high pressure and then recovers heat internally in the REC and SHX. After preheating, the strong solution is further heated and partially boiled in the DES by an external heat source fluid (hot water at pressure in this study). The vapor generated in the DES split into two portions by a split ratio (SR) for the refrigeration and power sub-cyclic processes after purified in the REC. The split ratio is defined as the ratio of the vapor mass flow rate that follows the power sub-cycle to the total purified vapor from the REC in percentage. The remaining (weak in NH_3) saturated solution returns from the DES to the ABS through the SHX and SEV. The refrigeration and power sub-cyclic processes are illustrated as follows:

- **Refrigeration sub-cyclic process:** depicted by the state points 10A-11-12-13-14-15 in Fig. 1a. The refrigerant vapor first enters to the CON where totally condensed to saturated liquid, then further sub-cooled in RSC before throttling to the EVA pressure using the REV. The heat of condensation is rejected to heat sink medium (water in this study). The refrigeration effect is produced in the EVA by absorbing heat from the external secondary fluid to be chilled.
- **Power sub-cyclic process:** depicted by the state points 16-17-18-19 in Fig. 1a. The vapor that follows the power sub-cycle first superheated in the SH then expanded in the EXP for generating the mechanical power output. Depending on the operating condition (inlet condition, isentropic efficiency and exit pressure) of the EXP, some refrigeration can be obtained by sensibly heating the EXP exhaust.

The two streams (state points 15 and 19) from the refrigeration and power sub-cycles are mixed before entering the ABS (state point 20) to be absorbed by the weak solution from the DES accompanied by the rejection of heat of absorption to the heat sink. Water/ethylene-glycol mixture (with ethylene-glycol mass concentration of 38% approximately) with parallel flow arrangement is used as external secondary working fluid in the EVA and Cooler (C). The external cooling water streams for the ABS and CON are connected in parallel flow arrangement. In order to get a better temperature match between the heat source fluid (hot water at pressure) and working fluid ($\text{NH}_3/\text{H}_2\text{O}$ mixture) in the heat addition process, the SH and DES are arranged in series according to their temperature levels.

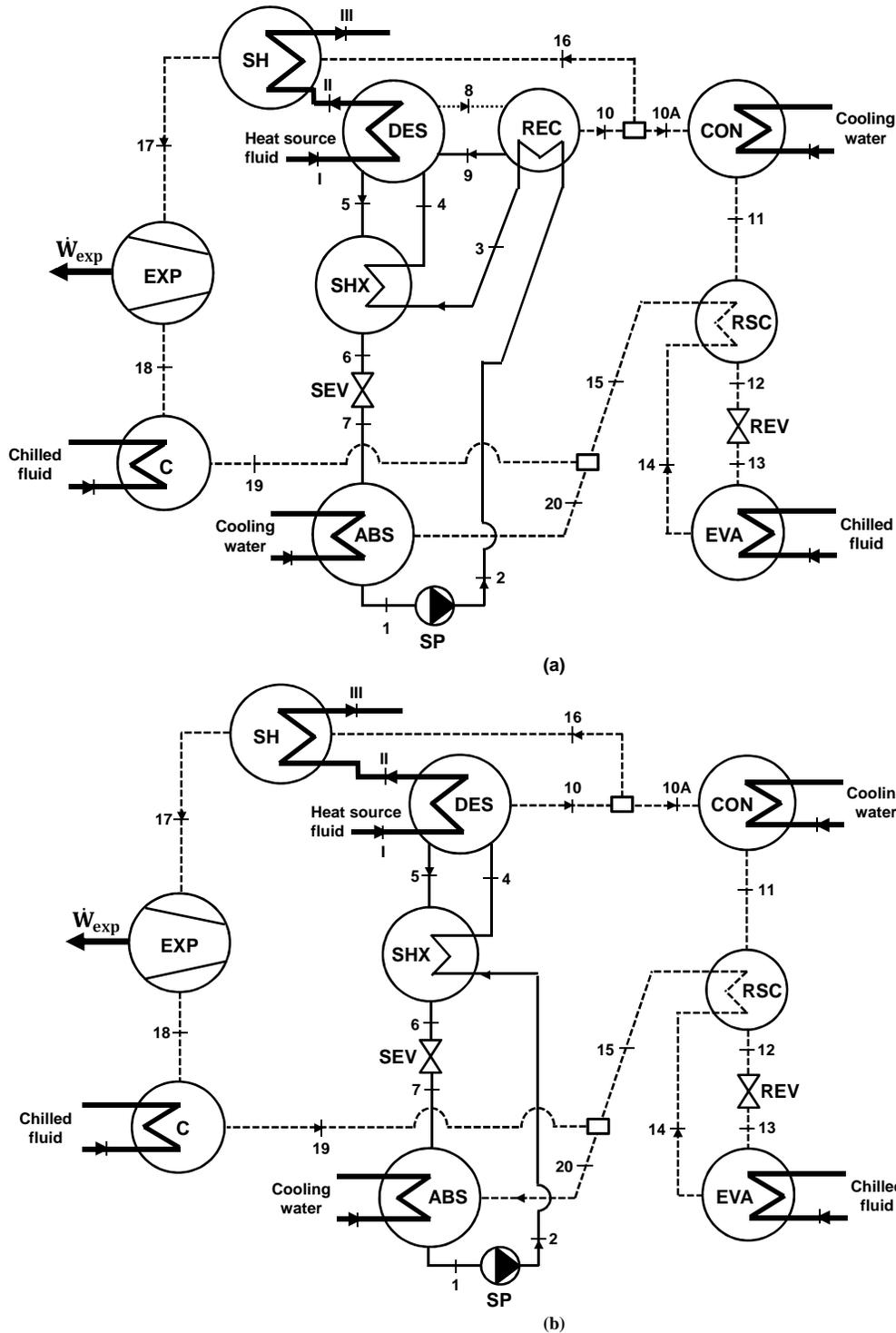


Fig. 1 Schematic diagram of single-stage hybrid cycles: (a) - $\text{NH}_3/\text{H}_2\text{O}$ mixture and (b) - $\text{NH}_3/\text{LiNO}_3$ and NH_3/NaSCN mixtures

The cyclic process for the configuration shown in Fig. 1b, with the non-conventional working fluid mixtures ($\text{NH}_3/\text{LiNO}_3$ and NH_3/NaSCN), can be described in a similar manner as the cyclic processes of Fig. 1a without considering the rectification process and associated internal heat recuperation. The proposed single-stage hybrid absorption cycles can function in different modes of operation depending on the refrigeration and power

demand. For the limiting cases when split ratio (SR) is 0 or 100% the cycle operates as pure absorption refrigeration cycle and Goswami cycle, respectively. The Goswami cycle operation is described in detail elsewhere [1]. For intermediate modes of operation ($0 < \text{SR} < 100\%$), the cycle produces the dual-outputs with different ratio of power to refrigeration (R).

3.2. Two-stage hybrid absorption cycle

The single-stage hybrid absorption cycles shown in Fig. 1 cannot take the advantage of the relatively higher temperature heat sources to achieve higher energy conversion efficiency. Thus, it is necessary to design cycle configurations that can take the advantage of the higher availability (exergy) of the heat source. Therefore, the potential of midgrade heat sources can be tapped by using a two-stage hybrid absorption cycle configuration proposed in this study. Low-grade and midgrade heat source temperature ranges considered in this study are according to the definition of Demirkaya et al. [10]. The cycle configuration shown in Fig. 2 is a modification of a two-stage double-effect absorption cooling cycle for the cogeneration of refrigeration and power. This cycle is similar to the one conceptually proposed by Ziegler [11]. Detail performance analysis of this cycle is presented in section 5.

3.2.1. Process description

Fig. 2 illustrates the flow schematic of the cycle based on two-stage double-effect absorption cycle for refrigeration and power cogeneration. Heat is transferred into the cycle in both the high temperature desorber (HTDES), superheater (SH) and the evaporator (EVA). The heat rejection from the cycle takes place from the low temperature condenser (LTCON) and absorber (ABS). The cycle includes two solution heat exchangers at low and high temperature levels in the cycle (SHX1 and SHX2 respectively) and a refrigerant sub-cooler (RSC). A series solution flow arrangement, first to HTDES, is considered for the configuration analyzed in this study. The individual components (units) in this cycle operate in a similar manner to that of the single-stage hybrid absorption cycle configuration, except the temperature, and pressure ranges for the unit in the second stage. The new feature of this two-stage cycle configuration is the internal heat transfer between the high temperature condenser (HTCON) and low temperature desorber (LTDES). The external heat transferring circuits (secondary working fluids) used are the same as the single-stage hybrid absorption cycles in sub-section 3.1. Further detail about generic two-stage absorption cycle configurations is available in Alefeld et al. [12].

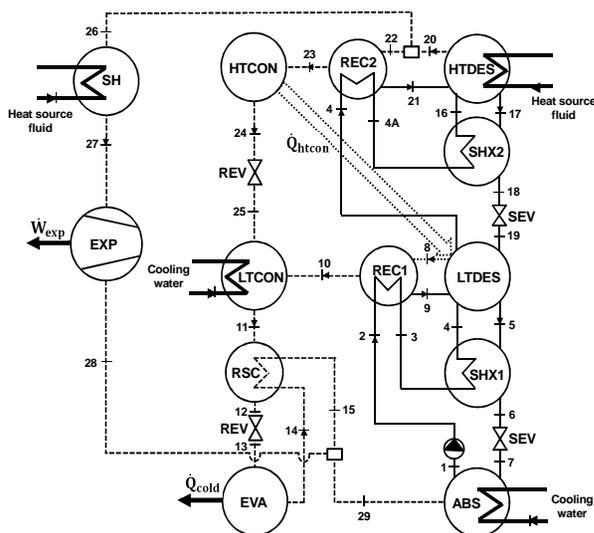


Fig. 2 Schematic diagram of a two-stage hybrid absorption cycle for simultaneous production of power and cooling.

4. Methodology

In the present study, a steady-state steady flow analysis of the proposed combined cycle systems have been performed by means of a thermodynamic cycle simulation and sensitivity study for key parameters of the system. Each component of the cycle treated as a control volume (CV, Fig. 3) with inlet and outlet mass flow streams, heat transfer and mechanical work interactions. The heat and work flow could be into or out of the control volume.

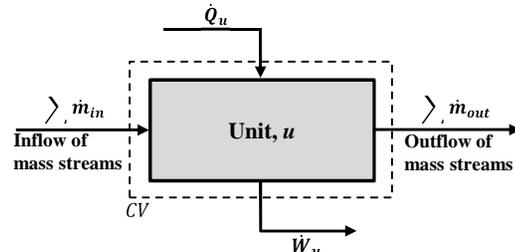


Fig. 3 Mass and energy flows in a generic unit.

The following main assumptions are considered to conduct the simulation study:

1. The frictional pressure drop in the system is neglected except through the throttling valves and expander.
2. Heat losses to the surrounding are not considered.
3. The NH₃ concentration of a purified saturated vapor leaving the rectifier (s) is 0.999.
4. Pump and expander isentropic efficiencies are 80 and 85%, respectively.
5. Constant effectiveness value for solution heat exchangers (80%) and refrigerant sub-cooler (80%).
6. Minimum closest approach temperatures of 5°C in the ABS, CON, EVA, LTCON, DES and HTDES are considered.
7. Partial vaporization is allowed in the EVA for cycle configuration with NH₃/H₂O working fluid, whereas the cycle with NH₃/LiNO₃ and NH₃/NaSCN working fluids complete vaporization attained at the exit of the EVA.

The basic models for all the units (u), in the cogeneration systems involve two balance equations: namely mass and energy equations which are subjected to conservation law. Usually, kinetic and potential energies are neglected for such types of system analysis.

The generic equations based on mass and energy balances are as follows.

Global mass balance:

$$\sum_u \dot{m}_{in} - \sum_u \dot{m}_{out} = 0, \tag{1}$$

where the subscripts in and out indicates inflow and outflow of mass streams.

Component/NH₃ mass balance:

$$\sum_u z_{in} \dot{m}_{in} - \sum_u z_{out} \dot{m}_{out} = 0, \quad (2)$$

where z is ammonia mass fraction.

Energy balance:

$$\sum_u h_{in} \dot{m}_{in} - \sum_u h_{out} \dot{m}_{out} + \dot{Q}_u - \dot{W}_u = 0, \quad (3)$$

where h is specific enthalpy of the entering and leaving mass flow streams.

Specific exergy (ex) of a stream:

$$ex = h - h_0 - T_0(s - s_0), \quad (4)$$

where h and s are the specific enthalpy and entropy of a stream respectively and T₀ is the reference environment temperature for exergy calculation.

The environment, which appears in the definition of exergy, and used in exergy calculation of the streams is a large equilibrium system in which the state variables (T₀, p₀) and the chemical potential of the chemical components contained in it remain constant when in a thermodynamic process heat and materials are exchanged between another system and the environment. The properties indicated by the subscript 0 refer to the environmental reference state, which is taken as 20°C and 101.35kPa in this work. Since there is no departure of chemical substance from the cycle to the environment, the chemical exergy is zero [13, 14]. Moreover, the exergy associated with the potential and kinetic energy is also neglected. Thus, specific exergy (ex) in Eq. 4 considers only the physical (thermomechanical) exergy.

4.1. Performance indicators

Five types of performance indicators are used to evaluate the absorption systems performance. Here, system refers to the combination of the cycle configuration and working fluid employed in it. The indicators are: overall thermal efficiency (η_{th}), effective first law efficiency (η_{1, eff}), effective exergy efficiency (η_{ex, eff}), fluid circulation ratio (SCR) and power to cooling (refrigeration) ratio (R).

- Overall thermal efficiency (η_{th}): It is defined as the useful energy outputs of the system (both net mechanical power generated, \dot{W}_{net} and cooling effect, \dot{Q}_{cold}) divided by the total thermal power input (\dot{Q}_{in}) to drive the system, given by:

$$\eta_{th} = \frac{\dot{W}_{net} + \dot{Q}_{cold}}{\dot{Q}_{in}}, \quad (5)$$

where \dot{W}_{net} is the retrieved mechanical power from the expander (\dot{W}_{exp}) reduced by mechanical power input to the solution pump (\dot{W}_{sp}), \dot{Q}_{in} is the total thermal power added to the system from external heat source fluid in the superheater \dot{Q}_{sh} and desorber (\dot{Q}_{des} or \dot{Q}_{htdes}). This energy efficiency definition is only based on the first law of

thermodynamics and from the basic concept of cost-benefit analysis for energy conversion systems. In Eq. 5, equal weight is given to the dual outputs (mechanical power and cooling effect) of the system, even though their thermodynamic quality is different. In this definition (Eq. 5), a weight factor of f = 1 is used to account the quality of the cooling output [2].

- Effective first law efficiency (η_{1, eff}): To account the thermodynamic quality of the useful outputs of the system in the energy efficiency definition Eq.5, the cooling effect is replaced by an equivalent mechanical power required to produce the same amount of cooling output (\dot{Q}_{cold}) using the conventional vapor compression refrigeration system and according to Vijayaraghavan et al. (2003) it is termed as effective first law efficiency [15]. This thermodynamically consistent and is more realistic first law (energy) efficiency definition expression and is given as follows:

$$\eta_{1, eff} = \frac{\dot{W}_{net} + \frac{\dot{E}_{cold}}{\eta_{II, ref}}}{\dot{Q}_{in}}, \quad (6)$$

where η_{II, ref} is the second law efficiency for vapor compression refrigeration system and \dot{E}_{cold} is the exergy associated with the cooling output. This exergy is given by the expression:

$$\dot{E}_{cold} = \dot{m}_{cf} [(h_{cf,in} - h_{cf,out}) - T_0(s_{cf,in} - s_{cf,out})] \quad (7)$$

\dot{E}_{cold} is calculated as the exergy change of the chilled fluid (cf). This exergy flow is equivalent to the minimum mechanical power required to produce the cooling effect (\dot{Q}_{cold}) by using a completely reversible cyclic process. The real (actual) mechanical power required to produce the same cooling effect (\dot{Q}_{cold}) is obtained by multiplying \dot{E}_{cold} with 1/η_{II, ref}. It is similar as weighing the cooling effect by an actual Coefficient of Performance (COP) of vapor compression refrigeration cycle (i.e. a weight factor, f = 1/COP is used). In this work, a typical η_{II, ref} value equal to 40% is considered.

- Effective exergy efficiency (η_{ex, eff}): It is defined as the ratio of the exergy associated with the useful energy output of the system to the exergy associated with the driving energy input of the system. The equation is formulated as:

$$\eta_{ex, eff} = \frac{\dot{W}_{net} + \frac{\dot{E}_{cold}}{\eta_{II, ref}}}{\dot{E}_{x_{hs, in}} - \dot{E}_{x_{hs, out}}}, \quad (8)$$

where the denominator of Eq. 8 is the change in exergy of the heat source (hs) fluid.

- Fluid/solution circulation ratio (SCR): It is defined as the ratio of the mass flow rate of the basic solution

through the solution pump per unit mass of the refrigerant vapor absorbed in the absorber, \dot{m}_{20}/\dot{m}_1 (Figs 1a and b) or \dot{m}_{29}/\dot{m}_1 (Fig. 2).

- Power to cooling ratio (R): This is defined as the ratio of the net mechanical power output of the system per the cooling effect produced by the system, $\dot{W}_{net} / \dot{Q}_{cold}$.

The physico-mathematical model of the proposed hybrid absorption cycles (Fig. 1 and Fig. 2) have been developed using the commercial software package EES [16].

5. Numerical Results and Discussion

5.1. Single-stage hybrid absorption cycles

The thermodynamic performance of the single-stage hybrid absorption cycles with $\text{NH}_3/\text{LiNO}_3$, NH_3/NaSCN and the conventional $\text{NH}_3/\text{H}_2\text{O}$ working fluids were evaluated for different modes of operation characterized by different vapor split ratios (SRs). The input parameters assumed for the purpose of analysis and discussion are summarized in Table 1. These input parameters were selected by considering the fact that: the cycle with $\text{NH}_3/\text{LiNO}_3$ and NH_3/NaSCN has to avoid crystallization during its operation, and maintained within the available property database correlation range. The minimum heat source inlet temperature is determined by the working fluid characteristics, absorber, condenser and evaporator operating conditions. Since the sensible part of the

refrigeration output decrease notably including a superheater, the cycle simulation was carried out without considering superheating.

Table 1. Input parameters used in the simulation.

Input parameter	Value
Heat source inlet temperature (°C)	90 – 135
System low pressure (bar)	3
Cooling water inlet/outlet temperature (°C)	30/38
Chilled fluid inlet/outlet temperature (°C)	10/3

The mix of power and refrigeration production of the systems with respect to the vapor split ratio is illustrated in Figs 4(a-d) at different heat source inlet temperatures. Since the cycle operates as a Goswami cycle when the entire vapor flowing to the power sub-cycle (SR=100%), the high pressure of the system is no longer dependent on the operating condition of the condenser and its value is between the bubble and dew pressures of the basic solution at the desorber temperature. Therefore, a pressure that maximizes the effective exergy efficiency is taken as the optimum high pressure of the cycle. The performance indicators and useful outputs of the systems at their corresponding optimum pressure are shown in Table 2.

Table 2. Performance indicators and useful outputs of the single-stage hybrid absorption cycles operating as Goswami cycle (SR=100%)

Parameter	Working fluid mixture											
	$\text{NH}_3/\text{LiNO}_3$				NH_3/NaSCN				$\text{NH}_3/\text{H}_2\text{O}$			
Heat source inlet temperature (°C)	100	110	120	130	100	110	120	130	100	110	120	130
Optimum system high pressure (bar)	12.6	15.0	17.6	20.3	11.5	13.9	16.6	19.7	10.2	11.7	13.2	15.1
Overall thermal efficiency η_{th} (%)	11.3	12.7	12.7	15.0	7.6	8.9	10.1	11.2	10.1	11.3	12.4	13.4
Effective first law efficiency $\eta_{I,eff}$ (%)	9.6	10.5	11.3	12.0	6.9	7.8	8.6	9.4	6.4	7.1	7.7	8.2
Effective exergy efficiency $\eta_{ex,eff}$ (%)	48.4	48.9	48.8	48.5	35.1	36.1	36.6	36.8	32.9	33.4	33.6	33.5
Net power output (kW)	13.2	16.7	20.4	24.2	10.1	13.0	15.9	19.0	16.3	21.2	26.6	31.7
Sensible refrigeration output (kW)	2.7	4.1	5.6	7.0	1.0	2.0	3.1	4.3	11.4	15.6	20.2	25.0

5.2. Two-stage hybrid absorption cycle

The thermodynamic performance evaluation of the two-stage hybrid absorption cycle with $\text{NH}_3/\text{H}_2\text{O}$ working fluid is performed for a relatively higher heat source inlet temperature between 130°C - 200°C, absorber and condenser cooling water inlet/outlet temperature of 30/38°C, system low pressure of 5 bar and evaporator temperature of 5°C. A degree of superheating considered in the superheater is 20°C. The expander exhaust vapor quality is constrained above 90% in the simulation to avoid the formation of liquid droplets that have an adverse effect on the performance of the expander.

5.2.1. Base-case performance analysis

The output data obtained includes:

- Thermal and mechanical power of the main units.
- Performance indicators (η_{th} , $\eta_{I,eff}$, $\eta_{ex,eff}$, SCR and R).

The computed thermodynamic performance of the system is showed in Table 3. A unit mass flow rate (1.0 kg/s) of basic solution through the pump and a split ratio (SR) of 50% are considered for the base-case analysis. For this system the split ratio is defined as $\dot{m}_{26} / \dot{m}_{20}$ (Fig. 2) in percentage. A heat source inlet temperature of 170°C, corresponds to the maximum effective exergy efficiency, is fixed for the base-case calculations presented in Table 3. A 3.27 kg/s of hot water is used as a heat source fluid in the exergy flow calculation; it is determined by adjusting the temperature of the hot water at the exit of the high temperature desorber (HTDES) in order to minimize heat capacitance rates mismatch in the HTDES.

The exergy of the cooling effect, \dot{E}_{cold} , is calculated as the exergy difference across the evaporator.

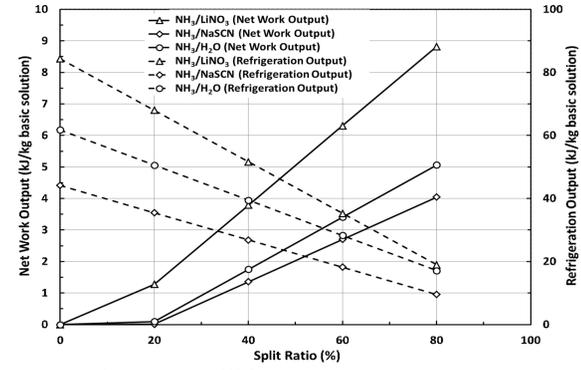
Table 3. Base-case performance summary for the proposed two-stage hybrid absorption cycle

Useful Output (kW)		
Cooling capacity	145.62	
Net mechanical power	17.12	
Thermal power (kW)		
Absorber duty	338.40	
Condenser duty	High temp.	Low temp.
Desorber duty	53.27	87.93
Rectifier duty	292.1	53.27
Solution heat exchanger duty	28.17	5.47
Refrigerant sub-cooler duty	162.9	301.9
Superheater duty	11.97	
Superheater duty	5.74	
Mechanical Power (kW)		
Expander output	25.48	
Solution pump input	8.36	
Exergy Flow (kW)		
Exergy input $\Delta \dot{E}_{hs}$	94.04	
Cooling effect exergy output \dot{E}_{eva}	8.19	
Heat source fluid exergy \dot{E}_{hs}	404.56	
Performance Indicator		
Overall thermal efficiency η_{th} (%)	54.64	
Effective first law efficiency $\eta_{I,eff}$ (%)	12.62	
Effective exergy efficiency $\eta_{ex,eff}$ (%)	39.98	
Fluid circulation ratio SCR (-)	4.87	
Power to cooling ratio R (-)	0.12	

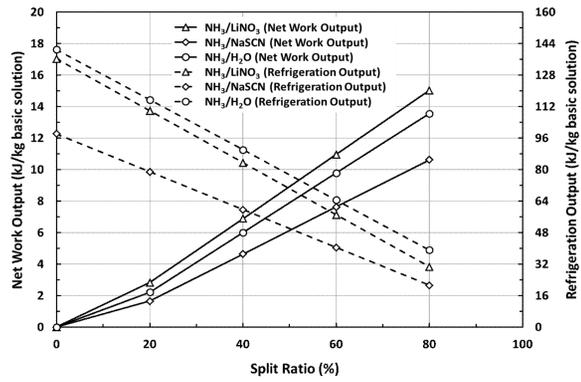
5.2.2. Sensitivity analysis

Depending on the requirement of power to cooling ratio one can adjust the split ratio of the cycle. The sensitivity of the cycle efficiencies and the useful outputs with respect to the split ratio variation are illustrated in Figs 5 and 6, respectively. As it can be seen in Fig. 5, the overall thermal efficiency of the cycle is more sensitive than the other efficiencies ($\eta_{I,eff}$, $\eta_{ex,eff}$) with respect to the variation of the split ratio. For a cycle operational mode without vapor splitting (SR=0), the cycle operates as a two-stage double-effect absorption refrigeration cycle with a COP of around 0.98.

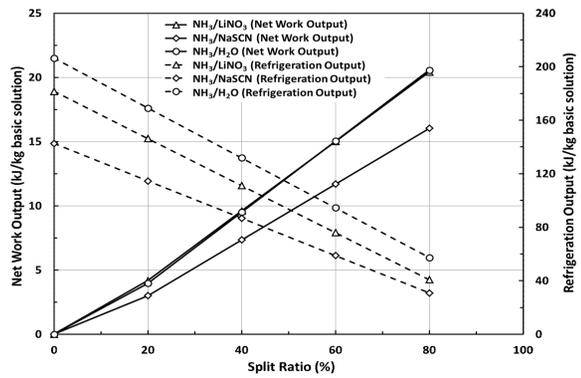
Fig. 6 shows the sensitivity of the net power and cooling outputs of the cycle with the variation of the split ratio. It is obvious that the cycle cooling output is null when the entire vapor desorbed in the high temperature desorber is flowing to the power-sub cycle (SR=100%), because at this operational mode there is no heat rejected from the high temperature condenser to drive the low temperature desorber.



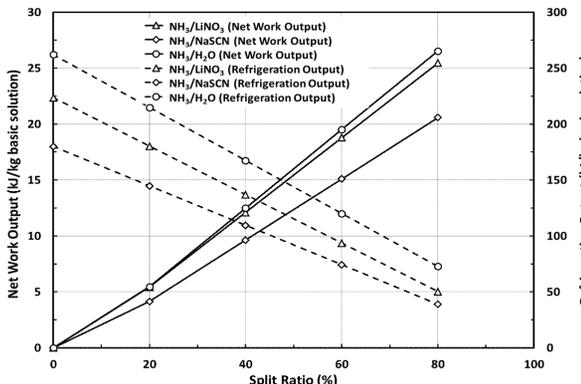
(a)-heat source inlet temperature (100°C)



(b)-heat source inlet temperature (110°C)



(c)-heat source inlet temperature (120°C)



(d)-heat source inlet temperature (130°C)

Fig.4 Cycle dual-outputs at different vapor split ratio for nh3/liho3, nh3/nascn and nh3/h2o working fluid mixtures

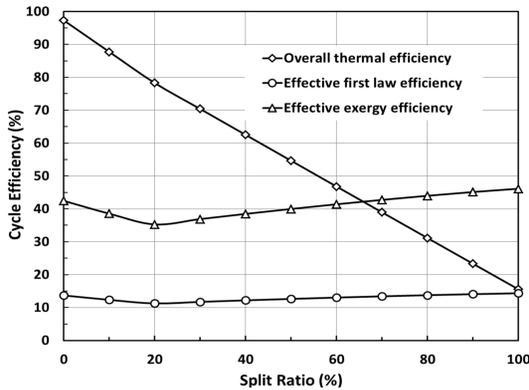


Fig. 5 Variation of the cycle efficiencies with the split ratio.

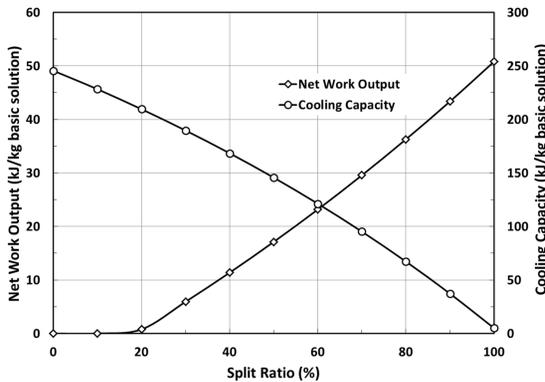


Fig. 6 Variation of the net work output and cooling capacity with the split ratio.

In order to investigate the effect of the heat source inlet temperature on the performance and operational working range of the cycle, a sensitivity analysis is conducted. When the heat source inlet temperature is varied all the other parameters are kept constant at their base-case values. The sensitivities of the cycle efficiencies and useful outputs with respect to the variation of the heat source inlet temperature are illustrated in Fig. 7 and Fig. 8, respectively. The system attained maximum effective exergy efficiency around 170°C. For a heat source temperature range, between 130 to 200°C, considered in the analysis the system produces both mechanical power and cooling outputs. However, for a heat source inlet temperature range between 130 to 135°C the net mechanical power produced was null.

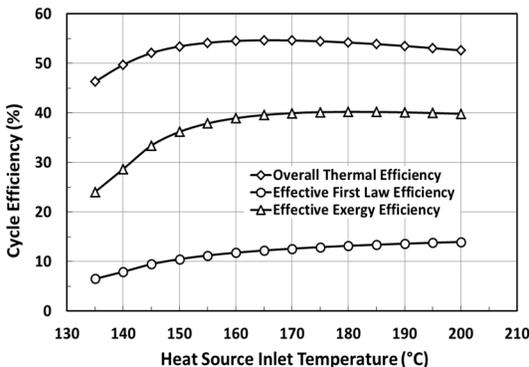


Fig. 7 Effect of heat source inlet temperature on the cycle efficiencies.

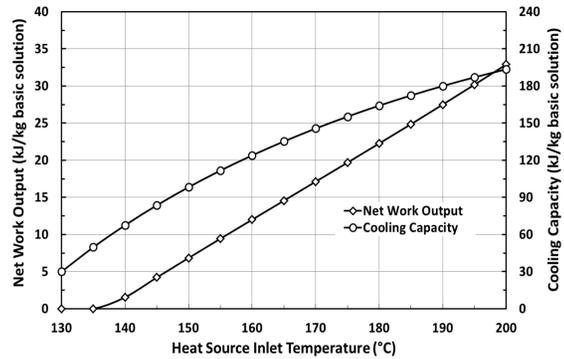


Fig. 8 Effect of heat source inlet temperature on the cycle useful outputs.

6. Conclusion

In this study, the effect of $\text{NH}_3/\text{H}_2\text{O}$, $\text{NH}_3/\text{LiNO}_3$ and NH_3/NaSCN working fluids on the performance and operational working ranges of the single-stage hybrid absorption cycles are analysed for different modes of operation from only absorption refrigeration cycle to Goswami cycle (mechanical power generation with some sensible cold production, if any) with intermediate modes of operation.

A two-stage hybrid absorption cycle with $\text{NH}_3/\text{H}_2\text{O}$ mixture as a working fluid is proposed and analyzed for the simultaneous production of mechanical power and cooling. The system performance is investigated using several performance indicators. It was found that an overall thermal efficiency of around 54%, effective first law and exergy efficiencies of 13% and 40% respectively for the base-case conditions considered (at a split ratio of 50% and a heat source inlet temperature of 170°C). The mix of power and cooling produced by the cycle can be adjusted to meet the seasonal requirement for cooling by varying the split ratio between 0 to 100%. At a split ratio between 0 and 20% the system only produce cooling with a maximum value of 245kW as a two-stage double-effect absorption refrigeration cycle. A sensitivity analysis is conducted to investigate the effect of the heat source inlet temperature on the system performances. The system attained maximum effective exergy efficiency around 170°C. For a heat source temperature range, between 130 to 200°C, considered in the analysis the system produces both mechanical power and cooling outputs. However, for a heat source inlet temperature range between 130 to 135°C the net mechanical power produced was null.

Nomenclature

- 1, 2, ..., 29 Thermodynamic state points
- COP Coefficient of performance
- \dot{E} Exergy flow, kW
- ex Specific exergy, kJ/kg
- h Specific enthalpy, kJ/kg
- hs Heat source
- \dot{m} Mass flow rate, kg/s
- \dot{Q} Thermal power, kW
- P Pressure, kPa or bar
- R Power to cooling ratio

s	Specific entropy, kJ/kg·K
SCR	Solution circulation ratio, kg/kg
SR	Split ratio, %
t	Temperature, °C
T	Temperature, K
u	Unit
\dot{W}	Mechanical power, kW
z	Ammonia mass fraction, kg/kg

Greek Symbols

Σ	Summation
η	Efficiency, %
	Difference

Subscripts

0	Reference value
I	First
II	Second
eff	Effective
ex	Exergy
ref	Refrigeration
th	overall thermal

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References

- [1] G. Demirkaya, R. Padilla, D.Y. Goswami, E. Stefanakos, M.M. Rahman. Analysis of a combined power and cooling cycle for low-grade heat sources. *Int. J. Energy Research* 2011; 35: 1145-1157, <http://dx.doi.org/10.1002/er.1750>
- [2] D.S. Ayou, R. Saravanan, J.C. Bruno, A. Coronas. An overview of combined absorption power and cooling cycles. *Renewable and Sustainable Energy Reviews*. Under review
- [3] D. Zheng, B. Chen, Y. Qi, H. Jin. Thermodynamic analysis of a novel absorption power/cooling combined-cycle. *Applied Energy* 2006; 83: 311-323, <http://dx.doi.org/10.1016/j.apenergy.2005.02.006>
- [4] C.P. Jawahar, R. Saravanan, J.C. Bruno, A. Coronas. Simulation studies on gas based Kalina cycle for both power and cooling applications. *Applied Thermal Engineering* 2011; xxx, 1-8. In press. <http://dx.doi.org/10.1016/j.applthermaleng.2011.11.004>
- [5] X. Zhang, M. He, Y. Zhang. A review on research on the Kalina cycle. *Renewable and Sustainable Energy Reviews* 2012; 16: 5309-5318 <http://dx.doi.org/10.1016/j.rser.2012.05.040>
- [6] R. Tillner-Roth, D. Friend. A Helmholtz free energy formulation of the thermodynamic properties of the mixture water+ammonia. *Journal of Physical and Chemical Reference Data* 1998; 27: 63-96. <http://dx.doi.org/10.1063/1.556015>
- [7] S. Libotean, D. Salavera, M. Valles, X. Esteve, A. Coronas. Vapour-liquid equilibrium of ammonia + lithium nitrate + water and ammonia + lithium nitrate solutions from (293.15 to 353.15) K. *J. Chem. Eng. Data* 2007; 52: 1050-55, <http://dx.doi.org/10.1021/je7000045>
- [8] S. Libotean, A. Martin, D. Salavera, M. Valles, X. Esteve, A. Coronas. Densities, viscosities, and heat capacities of ammonia + lithium nitrate and ammonia + lithium nitrate + water solutions between (293.15 and 353.15) K. *J. Chem. Eng. Data* 2008, 53, 2383-2388, <http://dx.doi.org/10.1021/je8003035>
- [9] S.K. Chaudhari, D. Salavera, A. Coronas. Densities, viscosities, heat capacities, and vapour-liquid equilibria of ammonia + sodium thiocyanate solutions at several temperatures. *J. Chem. Eng. Data* 2011; 56 (6): 2861-69, <http://dx.doi.org/10.1021/je200038n>
- [10] G. Demirkaya, S. Besarati, R.V. Padilla, A. R. Archibold, D.Y. Goswami, M.M. Rahman, E.L. Stefanakos. Multi-objective optimization of a combined cycle for low-grade and midgrade heat sources. *Journal of Energy Resources Technology- Transactions of ASME* 2012; 134 (3): art.no. 032002 (1-8).
- [11] Ziegler F. Novel cycles for power and refrigeration. *Proceeding of the 1st European Conference on Polygeneration*. Tarragona, Spain, 2007. ISBN: 978-84-96294-37-0
- [12] Alefeld, G, Radermacher R: *Heat Conversion Systems*. Boca Raton, Florida: CRC Press, Inc., 1994.
- [13] Kotas, T: *The Exergy Method of Thermale Plant Analysis*. Melbourne, Florida: Krieger Publishing Company, 1995.
- [14] A. Vidal, R. Best, R. Rivero, J. Cervantes. Analysis of a combined power and refrigeration cycle by the exergy method. *Energy* 2006; 31: 3401-3414, <http://dx.doi.org/10.1016/j.energy.2006.03.001>
- [15] S. Vijayaraghavan, D.Y. Goswami. On evaluating efficiency of a combined power and cooling cycle. *Journal of Energy Resources Technology* 2003; 125: 221-27. <http://dx.doi.org/10.1115/1.1595110>
- [16] EES. Engineering Equation Solver. F-chart software, www.fchart.com; 1992–2012