Self-adapting double and triple-lift absorption cycles for low-grade heat driven cooling

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A B S T R A C T

Multiple-lift absorption cycles are an interesting option for cooling and refrigeration driven by waste or renewable heat. Compared with single effect cycles, they allow higher thermal lift or lower thrust, but they often require the use of controlled valves, which can cause stability and control issues. The self-adapting concept, firstly introduced in the two-pump series-flow double-lift cycle, replaces the valve with a phase separator, overcoming this drawback. In this work, five new cycle layouts, incorporating the self-adapting concept, are presented: the one-pump series-flow double-lift cycle and four triple-lift cycles. The cycles are compared in terms of COP and heat duties under various conditions, using NH_3-H_2O and NH_3-LiNO_3 as working pairs. It is found that the double-lift cycles have a COP in the range 0.35–0.20, about 0.1 higher than the triple-lift cycles. However, triple-lift cycles accept cooling water temperature up to 8 °C higher. Cycles with multiple pumps have higher efficiency than single-pump cycles, especially at high lift conditions. The use of NH_3-H_2O as working pair guarantees higher COP at low thermal lift, while NH_3-LiNO_3 has wider operating range and better performances at high thermal lift.

Cycles à absorption auto-adaptatifs à deux et trois étages pour un refroidissement à basse température

1. Introduction

Absorption is considered an interesting technology to provide cooling service. This is particularly true when electrical energy is expensive or unreliable. In those cases, the use of direct fired absorption units, exploiting $H_2O-LiBr$ as working pair when producing cooling capacity or NH_3-H_2O for refrigeration purposes, is a convenient option for large application and easily available on the market. However, under the current cross-sectors requirement of reducing primary energy consumption and CO_2 emissions, the absorption technology could be applied also to produce cooling by exploiting renewable energy (e.g. solar energy) or waste heat.

Given that renewable and waste heat are more easily available at low temperatures, cycles able to cope with low temperature sources are generally of main interest. The single effect $H_2O-LiBr$ chiller has been studied extensively and its use is recommended for driving temperatures of about 85–95 °C with a cooling tower as heat rejection device (Wang et al., 2016). Single effect ammonia–water chillers typically require higher generation temperatures (above 120 °C) and have lower thermal COP (Wang et al., 2009). However, ammonia–water chiller can be air-cooled and provide refrigeration below 0 °C. Moreover, one additional advantage of ammonia–water is compactness, as also shown by recent developments in the manufacturing of monolithic microchannel absorption chillers (Garimella et al., 2016).

With respect to a single effect cycle, multiple lift cycles (Ziegler and Alefeld, 1987) are characterized by a larger temperature lift (i.e., the temperature difference between the intermediate temperature sink and the cold source) and/or a lower temperature thrust (i.e., the temperature difference between the hot source and the intermediate temperature sink). Thus, multiple lift cycles make

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Nomenclature

COP	Coefficient of performance, Dimensionless
f_{SEP}	Separation ratio, Dimensionless
f_{SPL}	Split ratio, Dimensionless
h	Specific enthalpy, J kg^{-1}
$\dot{m}$	Mass flow rate, kg s^{-1}
Np	Number of ports per domain
q	Specific heat duty, Dimensionless
Q	Heat transfer rate, W
P	Pressure, Pa
T	Temperature, °C
x	Vapor quality, Dimensionless
X	Ammonia mass fraction, Dimensionless
w_{PUMP}	Specific pumping power, W kW^{-1}
W	Mechanical power, W

Greek letters

Δ	Difference
ρ	Density, kg m^{-3}

Subscripts

conv.	Conventional cycle
EVAP	Evaporator
in	Input
inl	Inlet
out	Outlet
p	Port
ref	Refrigerant
S.A.	Self-adapting cycle
vap	Vapor
W inl	Cooling water inlet

Abbreviations

1P	One-pump
2P	Two-pump
3P	Three-pump
ABS	Absorber
COND	Condenser
DL	Double-lift
EVAP	Evaporator
GEN	Generator
HP	High pressure
IHP	Intermediate-high pressure
ILP	Intermediate-low pressure
IP	Intermediate pressure
LP	Low pressure
RCA	Refrigerant cooled absorber
RHE	Refrigerant heat exchanger
RR	Restrictor of the refrigerant
RS	Restrictor of the solution
SA	Self-adapting
SEP	Phase separator
SHX	Solution heat exchanger
TL	Triple-lift

possible the coupling of ammonia-based absorption chillers with low temperature heat sources and can provide not only cooling above 0 °C (e.g. air conditioning) but also refrigeration below 0 °C (e.g. food conservation). This feature is obtained at the cost of lower thermal COPs, which means higher heat consumptions and higher costs. Nevertheless, when renewable or waste heat is used, operating costs and CO_2 emissions are associated to the parasitic energy consumption, rather than to the heat driving the chiller. Therefore, multiple lift cycles that are combined with efficient

heat rejection devices (e.g., wet cooling towers, ground water) can provide financial benefits and CO_2 emission savings. Ammonia-based absorption chillers driven by low temperature heat sources can benefit from the superior thermodynamic performance of non-volatile absorbents such as lithium nitrate, allowing to remove the complex rectification section needed with volatile absorbents like water and improve thermal COP at low regeneration temperatures (Kaushik and Kumar, 1987; Sun, 1998). Moreover, compact stainless-steel plate heat exchangers have been successfully employed in the construction of absorption chillers utilizing ammonia–lithium nitrate as working pair (Zamora et al., 2014). These premises suggest that (i) investigating the performance of multiple-lift absorption cycles is interesting for ammonia-based chillers driven by low temperature heat and able to provide not only cooling above 0 °C (e.g. air conditioning at 7 °C) but also refrigeration below 0 °C (e.g., food conservation at −15 °C and −35 °C), (ii) alternative, non-volatile ammonia absorbents like lithium nitrate can be used to improve the thermal COP at low regeneration temperatures, and (iii) standard and compact heat exchangers can be used to simplify the construction of these cycles.

A first extensive review of double-lift cycles is provided by Erickson and Tang (1996), who proposed a comparison among various cycles in terms of overall heat duty and COP. The considered cycles are three variations of the vapour exchanger cycle, the resorption cycle, the two-pump series flow cycle, the one-pump series flow cycle, the parallel flow cycle and two versions of the semi-GAX cycle. Kim and Infante Ferreira (2009) investigated the heat-coupled half-effect parallel-flow absorption cycle for solar air-cooled air conditioning applications with water–LiBr working pair. A similar concept utilizing the ammonia–water working pair was investigated by Du et al. (2012). A recent review of cycles suitable for solar cooling systems is proposed by Xu and Wang (2018), who describe the main double-lift cycles and investigate the COP at different source and heat rejection temperatures.

The embodiments of multiple lift cycles presented in the literature usually require a split valve on a liquid stream of refrigerant or solution. Since the split ratio must be adjusted to maintain optimal operation under changing conditions, these cycles can be difficult to control. To overcome this issue, Guerra (2012) proposed a new two-pump series-flow double-lift cycle exploiting the self-adapting concept. The proposed configuration eliminates the split on the refrigerant stream, which is now routed entirely at intermediate pressure toward the Refrigerant Cooled Absorber (RCA). The fraction of refrigerant which evaporates in the RCA is absorbed by the externally cooled absorber, while the liquid fraction is throttled at low temperature toward the evaporator. This solution is called self-adapting since the amount of refrigerant evaporated in the RCA automatically matches the quantity required to absorb the vapour from the evaporator, without the need of an active control. Additionally, prototypes based on this cycle proved to run smoothly even under changing working conditions (Aprile et al., 2014; Toppi et al., 2017). A similar concept is also used by Yan et al. (2013), who explored the possibility to couple the single effect cycle in parallel with a double-lift half-effect cycle to increase the amount of recoverable heat from waste flue gases.

Purpose of this work is to investigate the possibility to exploit the self-adapting concept in different multiple lift cycles. Five new cycle configurations are presented: one double-lift series flow cycle and four triple-lift series flow cycles. For the sake of brevity, cycles are named DL (double-lift) or TL (triple-lift) followed by the number of solution pumps used (e.g. 1P, 2P etc.). As all the presented cycles exploit a series flow configuration, from this point ahead the flow configuration will not be specified. After a description of the self-adapting concept, the five new cycles will be described along with the modelling approach used to evaluate their performances. Then, the effectiveness of the self-adapting concept to automatically perform the optimal separation of the refrigerant
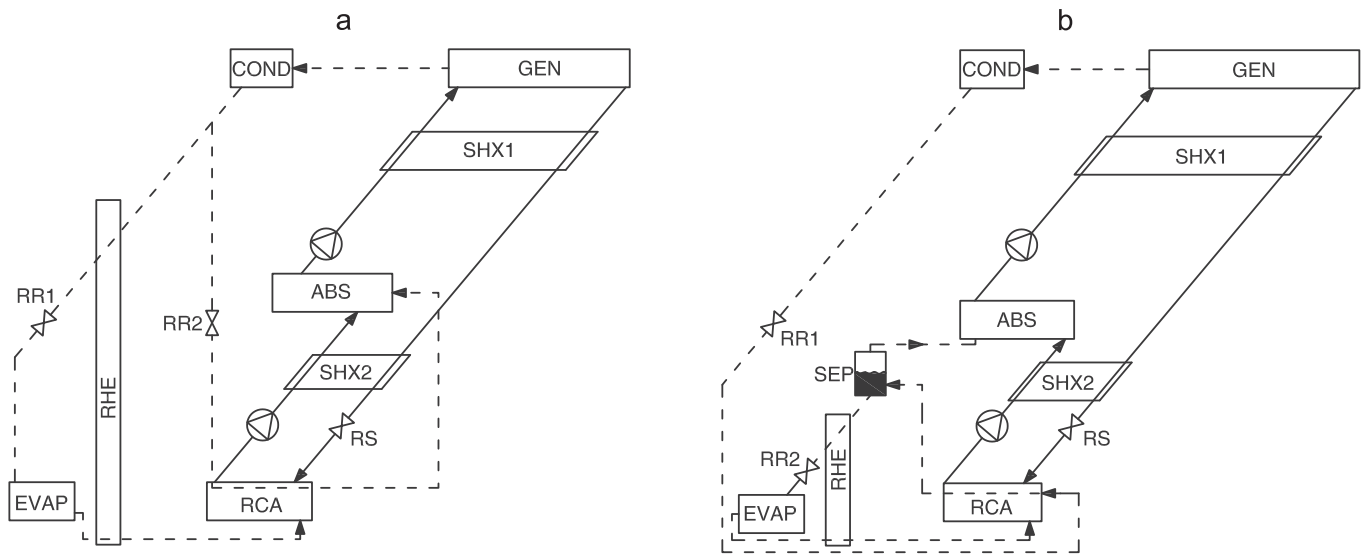
Fig. 1. Comparison between the original (a) and the self-adapting (b) versions of the DL-2P cycle.

between evaporator and RCA streams is shown by numerically comparing the split valve-based and self-adapting configurations of the DL-2P series flow cycle. Finally, the performances of the new cycles are compared numerically in terms of thermal COP and specific heat duties, using ammonia as refrigerant and either water (volatile) or lithium nitrate (non-volatile) absorbent.

2. Cycles description

After a first overview of the self-adapting concept, the five new cycles configurations are presented in this section.

The cycles are built using counter-current heat exchangers for all the components, including the generator. Additionally, a phase separator divides vapour from the poor solution and, for cycles with volatile absorbent, the generator is completed with a tray column, where the vapour flow upward meeting the rich solution flowing downward. Given the low driving temperature used within this study, the use of a rectifier after the tray column was not necessary.

2.1. The self-adapting concept

The self-adapting concept was introduced by Guerra (2012) modifying the DL-2P cycle. The layouts of the original DL-2P cycle and the corresponding cycle exploiting the self-adapting concept can be found in Fig. 1.

In the original version of the cycle, the liquid refrigerant leaving the condenser is divided in two streams: the first is routed toward the evaporator to provide the cooling capacity, the second is used in the RCA to absorb the vapour from the evaporator. The valve splitting the refrigerant must be controlled in order to achieve optimal performances under changing temperatures. In fact, the amount of refrigerant required in the RCA depends on the cycle conditions. Moreover, if the amount of refrigerant at the RCA is insufficient, the absorption process cannot be completed, causing a reduction of the cooling capacity and possible unstable operation due to the flooding of the evaporator. On the other end, it can be expected that if an excess of refrigerant is sent at the RCA, it would leave partially evaporated. This would not cause cycle instability, but would reduce the cycle performances, since the amount of not evaporated refrigerant could have been routed toward the evaporator to produce cooling capacity.

The self-adapting concept is obtained by removing the controlled valve and throttling the entire refrigerant stream at

intermediate pressure. Here it flows through the RCA, providing the cooling capacity required to absorb the vapour from the evaporator. In the process, part of the refrigerant evaporates allowing liquid and vapour to be separated in a phase-separator, i.e. a small tank which divides by gravity the two phases. The liquid fraction is routed toward the evaporator, where it provides the cooling effect, while the vapour is absorbed in the absorber. Unlike the original version, the self-adapting does not require an active control. Indeed, the exact quantity of refrigerant needed to absorb the vapour evaporates in the RCA. An increase of the vapour coming from the evaporator, which requires a higher cooling capacity at the RCA, causes more evaporation of the intermediate pressure stream, which leaves with higher vapour quality. This automatically reduces the flow rate of liquid from the separator to the evaporator, establishing a new equilibrium.

This feature makes real appliances easy to control and stable. Thus, in the following five cycles based on the self-adapting concept are proposed: double-lift one-pump (DL-1P), triple-lift one pump (TL-1P), triple-lift three-pump (TL-3P), triple-lift two-pump type a (TL-2Pa) and triple-lift type b (TL-2Pb).

2.2. The double-lift one-pump cycle (DL-1P)

Advanced cycles, with multiple pressure levels, usually require more than one solution pump. Since pumps are one of the most expensive part of an appliance, especially if of low capacity, the number of solution pumps is one of the criteria used to evaluate an absorption cycle. However, in some cases, cycles with more than one pressure level can be built exploiting a single pump. The double-lift series-flow cycle can be designed with two pumps or a single pump (Erickson and Tang, 1996). The introduction of the self-adapting concept to the DL-1P cycle provides the layout in Fig. 2. Unlike in the DL-2P cycle by Guerra, with this new configuration the poor solution leaving the generator (state point 1 in Fig. 2) is not throttled directly at low pressure, but it is first routed at intermediate pressure toward the absorber (2). Here it is enriched by the absorption of the vapor coming from the separator before being furtherly throttled at low pressure toward the RCA (3). Then, the rich solution leaving the RCA (4) is directly pumped at high pressure (5), avoiding the need for a two-stage compression, which would require an additional pump. As in the DL-2P cycle, the self-adapting concept is introduced to avoid the
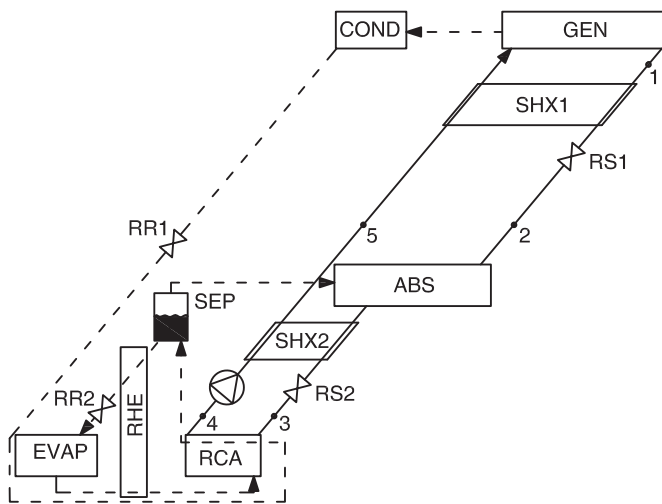
Fig. 2. Layout of the DL-1P self-adapting cycle.

split on the refrigerant, which is replaced by the phase separator after the RCA.

Fig. 3 reports the P-T-X diagram for the two cycles, calculated according to the assumption described in Section 3, for the inlet/outlet temperatures of 90/80 °C at the hot source, 12/7 °C at the cold source and 35/45 °C at the heat rejection sink.

2.3. Triple-lift cycles

Four versions of the triple-lift cycle are presented in this paper. As in the case of the double-lift cycle presented above, also in the triple-lift cycles different approaches can be followed for the management of the solution at the different pressure levels. A triple-lift cycle has four pressure levels, which will be named as High Pressure (HP), Intermediate-High Pressure (IHP), Intermediate Low Pressure (ILP) and Low pressure (LP). The self-adapting concept is implemented through the use of two refrigerant cooled absorbers: the Low Pressure RCA (LP_RCA) and the Intermediate Pressure RCA (IP_RCA). The former performs the absorption of the vapor at LP,

being cooled by a stream of refrigerant at ILP, the latter absorbs vapor at ILP, being cooled by a refrigerant stream at IHP.

Depending on the solution layout, four configurations of the self-adapting triple-lift cycle are proposed:

- TL-3P: triple-lift with three pumps (Fig. 4)), which follows the same approach as the DL-2P, with the poor solution throttled directly at low pressure and three different pumping stages for the rich solution, located after each enriching process at the LP-RCA, at the IP-RCA and at the absorber (ABS).
- TL-1P: triple-lift with one pump (Fig. 5), which follows the same approach as the DL-1P, with the poor solution undergoing multiple throttling as it passes through the absorber, the IP-RCA and the LP-RCA. The rich solution is then directly pumped from the low pressure to the high pressure.
- TL-2Pa: triple-lift with two pumps, type a (Fig. 6). This cycle is obtained by a combination of the two approaches used in the former cycles. In fact, the poor solution is firstly throttled at the IHP, where it is enriched in the absorber, similarly to what happens in the TL-1P cycle. Then, instead of passing through the IP_RCA, it is routed directly at low pressure. Consequently, the rich solution needs to be firstly pumped at the ILP and then at HP.
- TL-2Pb: triple-lift with two pumps, type b (Fig. 7). The layout of this cycle follows a similar approach as the corresponding type a, with the difference that the poor solution is firstly throttled at ILP, where it absorbs vapour in the IP-RCA and then at LP, where it flows through the LP-RCA. Consequently, within the first pumping stage the rich solution reaches the absorber, where it is further enriched, and with the second pumping stage it is pumped at HP.

The differences among the four configurations can be appreciated in Fig. 8, which reports the P-T-X diagrams for the triple lift cycles, calculated with inlet/outlet water temperatures of 90/80 °C at the hot source, 12/7 °C at the cold source and 45/55 °C at the heat rejection sink.

3. Modeling approach

The calculations to assess the performances of the proposed cycles are carried out with the support of STACY (Aprile et al., 2018), a mathematical modelling framework for steady-state simulation
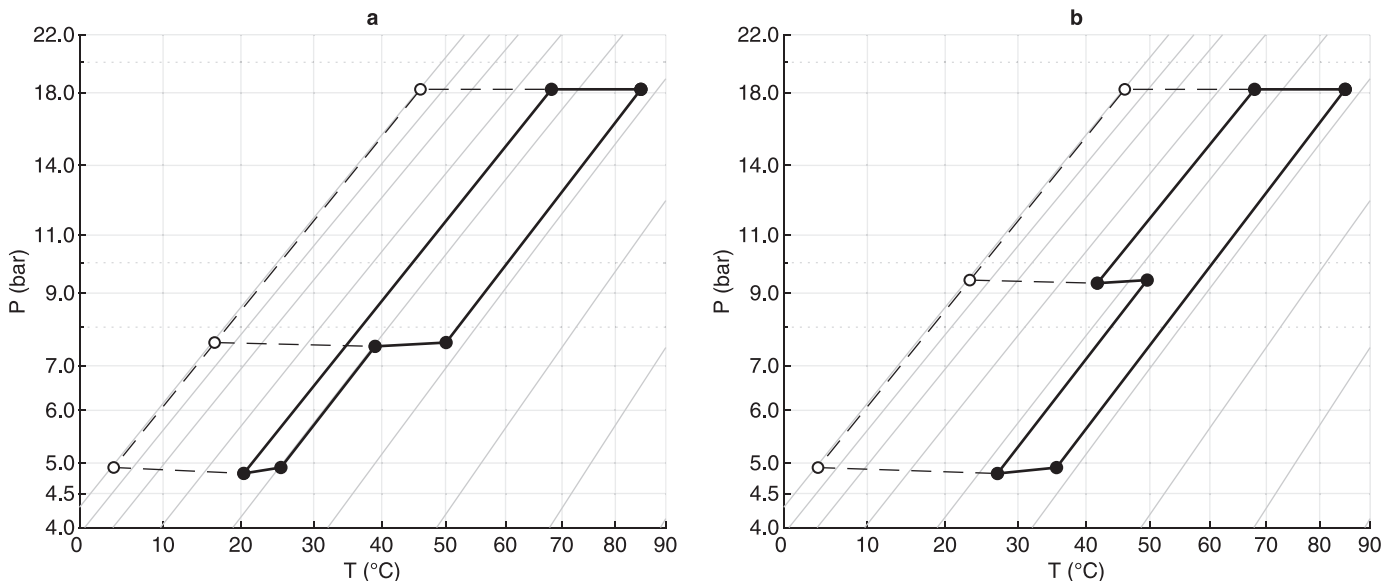

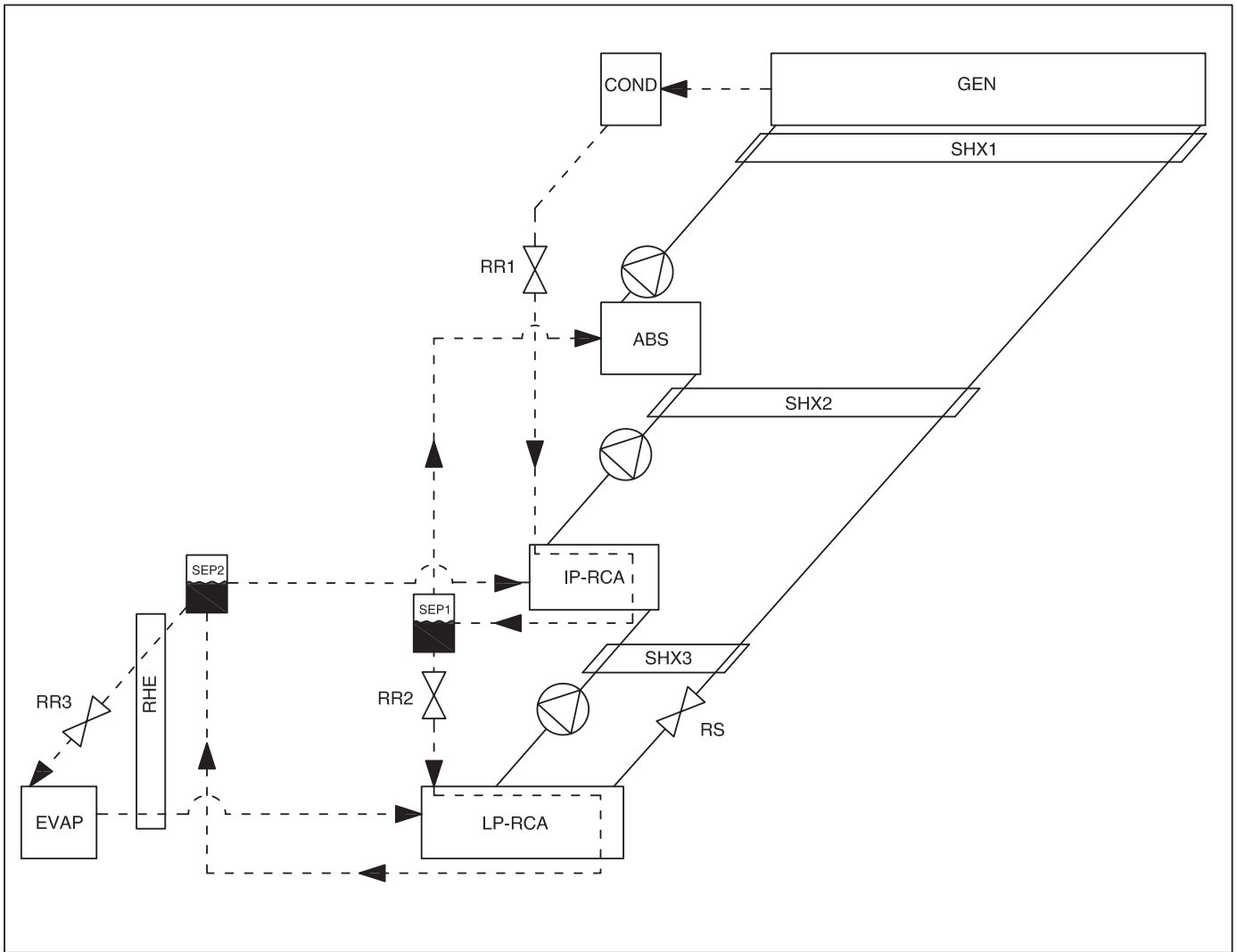Fig. 3. P-T-X diagram of the DL-1P (a) and DL-2P (b) cycles.

Fig. 4. Layout of the three-pump triple-lift (TL-3P) self-adapting cycle.

of absorption cycles, experimentally validated and based on a modular approach. Mass, species, and energy balances are written for each component domain (i.e., the volume occupied by the same fluid) based on the stream flow rates and thermodynamic states at its ports ($p = 1,…,N_p$) and the resulting redundant equations are deleted:

$$\sum_{p=1}^{N_p} \dot{m}_p = 0 \qquad (1)$$

$$\sum_{p=1}^{N_p} \dot{m}_p X_p = 0 \qquad (2)$$

$$\sum_{p=1}^{N_p} \dot{m}_p h_p + Q_{in} + W_{in} = 0 \qquad (3)$$

The system of algebraic equations is completed by adding heat transfer relationships, auxiliary conditions as mass flow rates (fixed or variable according to e.g. a throttling condition), independent species mass fractions (e.g. based on saturation or subcooling conditions), pressure levels (e.g. condensation or evaporation pressure) and outlet temperatures of source components.

In particular, the following assumptions are used in the present study.

- Pressure losses are negligible in the pipes and in all heat exchangers but the absorbers, where pressure drops play a role in defining the performance of the cycle. The actual pressure drop in the absorbers depend on the heat exchanger type, dimensioning and operation conditions. In the present study, considering the assumption made in previous studies (Xu and Wang, 2018; Toppi et al., 2017), a fixed value of 10 kPa is assumed.
- Heat losses are negligible (Gebreslassie et al., 2010).
- Throttling are isenthalpic (Xu and Wang, 2018).
- In the generator, saturation condition is set for the liquid and vapour leaving the phase-separator and a minimum pinch of 5 °C is set at the heat exchanger between hot water and solution (Xu and Wang, 2018).
- Constant efficiency of 0.8 for the internal heat exchangers (SHX and RHE).

The conditions at the absorber, condenser and evaporator have been set based on the following assumptions (Toppi et al., 2016):

- at the absorbers, a fixed subcooling of 1 °C at the solution outlet and a temperature difference of 3 °C between solution outlet and cooling fluid inlet are imposed;
- at the condenser, a fixed subcooling of 8 °C is set at the refrigerant outlet and the minimum temperature difference of 1 °C is imposed between refrigerant and cooling water;
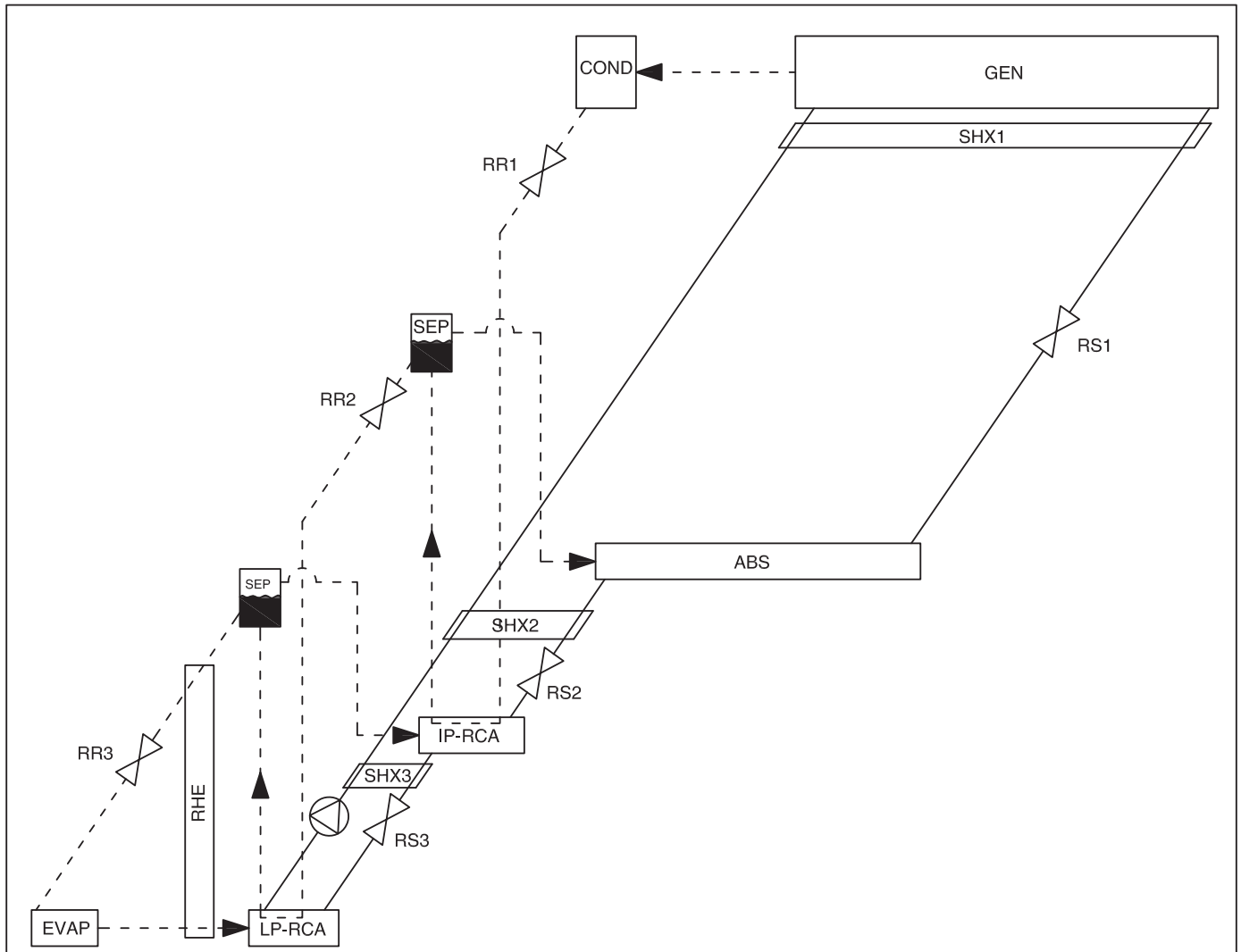
Fig. 5. Layout of the one-pump triple-lift (TL-1P) self-adapting cycle.

- in the evaporator, the refrigerant enters 3 °C below the heat source fluid (chilled water or brine) outlet temperature and leaves 1 °C below the heat source fluid inlet temperature.

The mass flow rate of the external circuits has been set in order to maintain fixed temperature differences:

- condenser and absorbers are connected in parallel, and each flow rate is set in order to achieve a difference of 10 °C between inlet and outlet;
- a temperature difference of 10 °C is set between inlet and outlet of the hot water at the generator;
- a temperature difference of 5 °C is set between inlet and outlet of the heat source fluid at the evaporator.

The main performance indicators are the cooling coefficient of performance of the system (COP) and the specific heat duty (q_j) of the involved subsystems grouped according to the main processes that characterize the analyzed cycles, i.e. generation (GEN), condensation (COND), externally cooled absorption (ABS), refrigerant cooled absorption (RCA) and solution heat recovery (SHX).

$$COP = \frac{Q_{EVAP}}{Q_{GEN}} \qquad (4)$$

$$q_j = \frac{Q_j}{Q_{EVAP}} \qquad (5)$$

When many heat exchangers are involved in one process, Q_j represents the sum extended to the heat duties of each heat exchanger involved (e.g., $Q_{SHX} = Q_{SHX,1} + Q_{SHX,2}$ for the DL-1P cycle). The heat duty of the refrigerant heat recovery (RHE) and of the tray column are not considered in the analysis. The former because it is both significantly smaller than the heat duties at the other heat exchangers and barely affected by the cycle layout. The latter because the simultaneous heat and mass transfer makes difficult a sound comparison with the heat duty of the other heat exchangers.

Since a low electrical consumption is one of the most interesting features of thermally driven cycles, the specific pumping power, defined as the ratio between power input to all solution pumps and the evaporator heat capacity (see Eq. (6)), is included in the analysis. The fluid is considered incompressible, a sound hypothesis for the liquid solution, and its density is calculated at the inlet conditions. The pump efficiency is not included in the calculation, since this parameter is independent on the cycle layout and is mainly influenced by the pump size, type and working conditions.

$$w_{PUMP} = \frac{W_{PUMP}}{Q_{EVAP}} = \frac{\frac{\dot{m}}{\rho_{inl}}\Delta P}{Q_{EVAP}} \qquad (6)$$
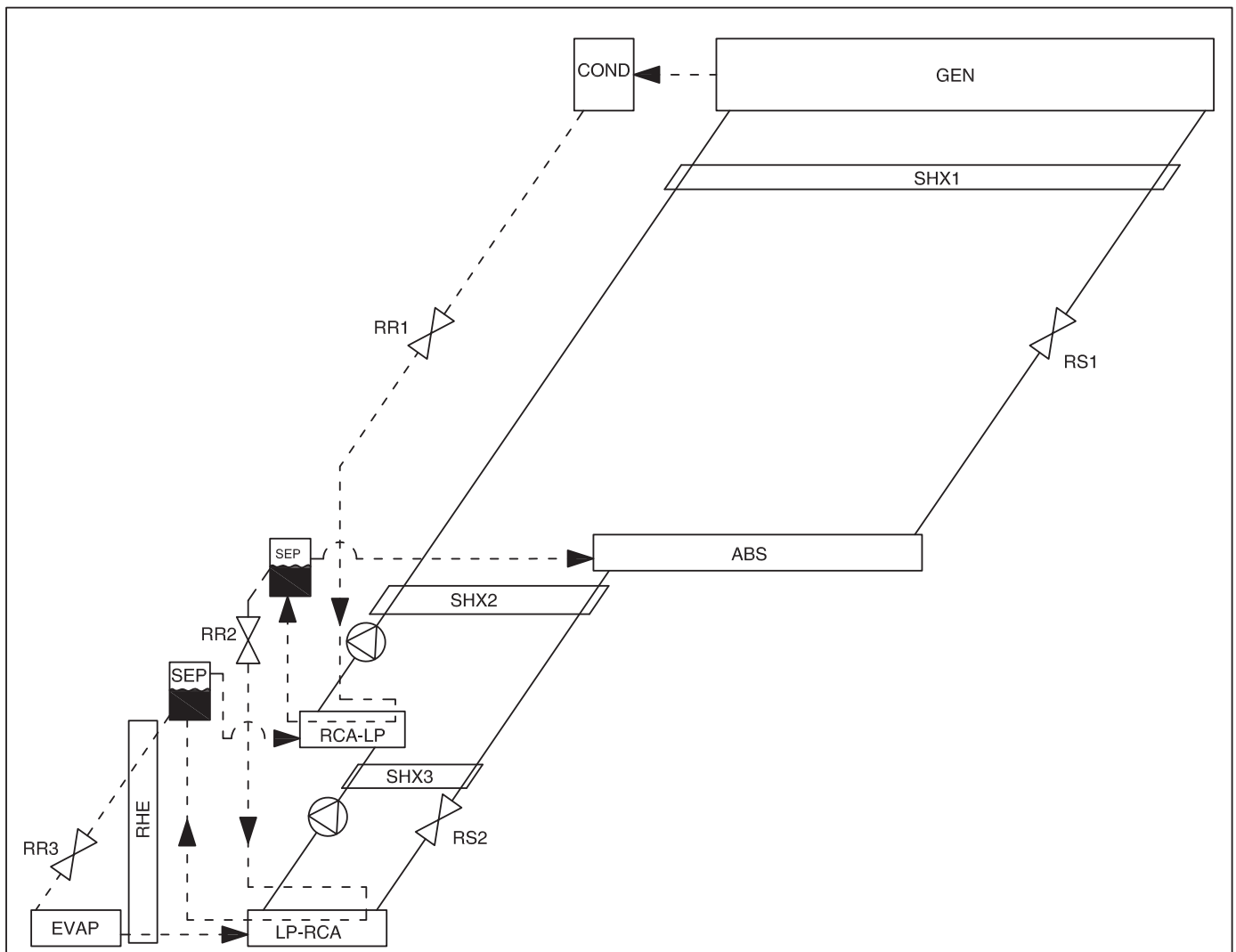
Fig. 6. Layout of the two-pump triple-lift type a (TL-2Pa) self-adapting cycle.

4. Results

In this section at first it is shown that the self-adapting layout can automatically operate in the same conditions of the conventional cycle when the split valve is optimally controlled. This is done by means of numerical calculation performed on the conventional and self-adapting DL-2P cycles. Then the performances of the five new configurations are calculated under different working conditions and the impact of the working pair are investigated. Finally, the heat duties of the cycles are compared under given conditions.

The calculations have been carried out using hot water temperatures at the generator inlet and outlet of 90 °C and 80 °C respectively, i.e. the temperatures compatible with renewable heat (e.g. solar) or waste heat recovery from internal combustion engines. The analysis includes different cooling water temperatures at the absorber and condenser, maintaining a temperature difference of 10 °C between outlet and inlet. Similarly, a constant temperature difference of 5 °C has been imposed between inlet and outlet of the heat source fluid at the evaporator.

4.1. Self-adapting optimal operation

The regulated split on the liquid refrigerant of the conventional double-lift cycles requires adjustment in order to obtain optimal

cycle operation. In fact, if an insufficient amount of refrigerant is routed toward the RCA, the complete absorption of the vapour from the evaporator cannot be performed, with a loss of cooling capacity.

On the other end, if more refrigerant than the amount required to sustain the absorption process is sent to the RCA, part of it leaves as a liquid. This excess of refrigerant simply mixes with the solution in the absorber, without providing useful effects to the cycle, while it could have been used in evaporator to provide cooling capacity. The self-adapting cycle has the advantage of operating without the need of an active control to perform the optimal separation. To support this statement, two quantities are defined to express the amount of condensed refrigerant used to sustain the low-temperature absorption in the RCA in the two cycle configurations: the separation ratio (f_{SEP}) for a self-adapting cycle (see Eq. (7)) is the ratio between the flow rates of the vapour leaving the separator and of the overall incoming refrigerant; the split ratio (f_{SPL}) for the conventional cycle is the fraction of the mass flow rate of condensed refrigerant routed toward the evaporator (see Eq. (8)).

$$f_{SEP} = \frac{\dot{m}_{SEP\ vap\ out}}{\dot{m}_{SEP\ ref\ in}} \tag{7}$$

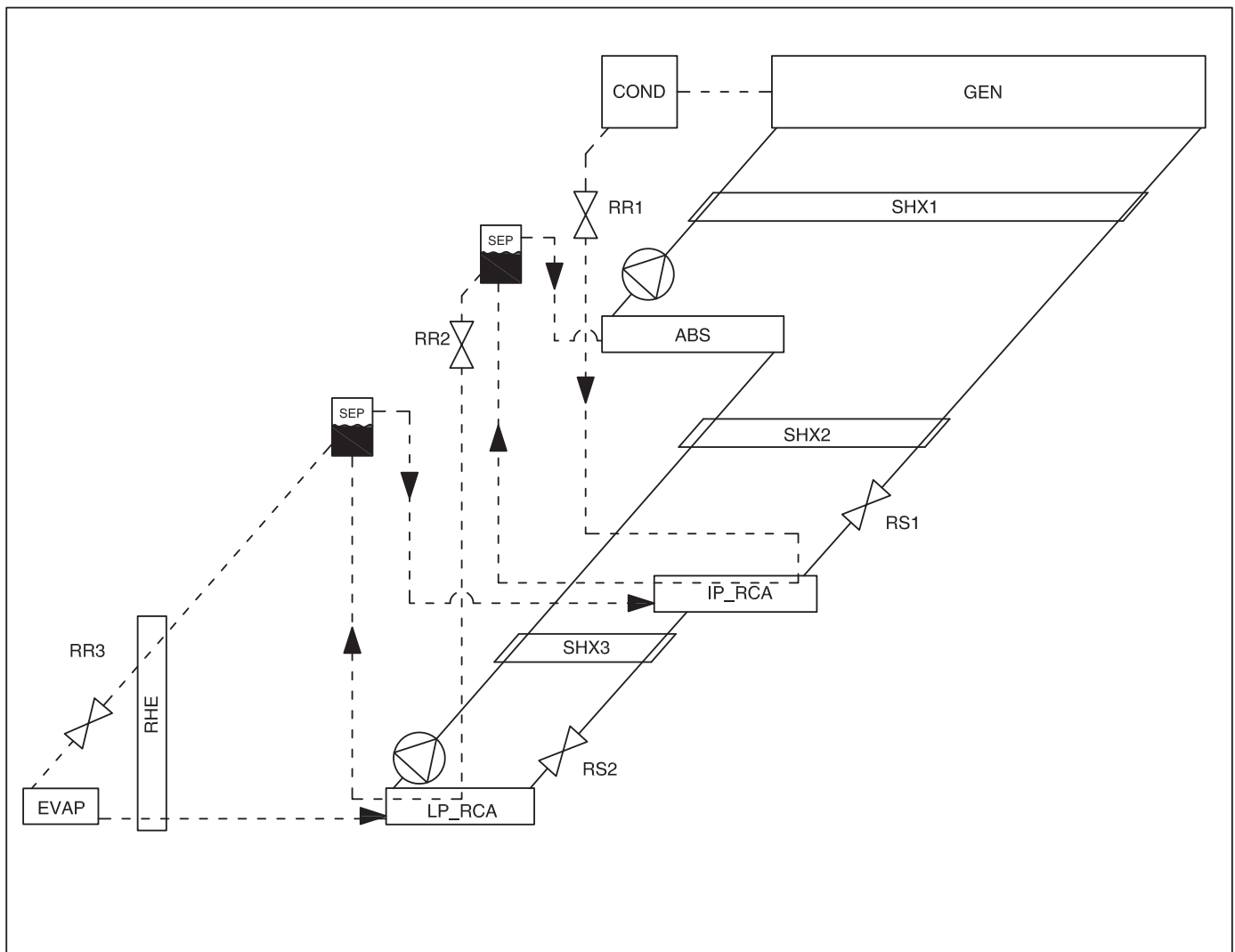$$f_{SPL} = \frac{\dot{m}_{RCA\ ref\ in}}{\dot{m}_{COND\ ref\ out}} \tag{8}$$

Fig. 7. Layout of the two-pump triple-lift type b (TL-2Pb) self-adapting cycle.

In Fig. 9 the COP of the conventional DL-2P cycle (COP$_{conv.}$) and the vapour quality of the refrigerant at the RCA outlet ($x_{RCA\ out,\ conv.}$) are reported for tree heat source fluid temperatures, varying the split ratio. The chart shows that the maximum COP is achieved when the vapour quality is 1, i.e. when the refrigerant at the RCA outlet is completely evaporated. As discussed above, a lower vapour quality means that an excess of refrigerant has been subtracted to the evaporator, where it would had provided additional cooling capacity. No data are reported for split ratios lower than the optimum because, in those conditions, the cycle does not work in a stable way since the refrigerant at the RCA is not enough to sustain the absorption of the vapour coming from the evaporator. From the numerical point of view, the calculation does not converge, because a steady state solution cannot be reached. From the physical point of view, this would result in a higher low pressure and in a flooded evaporator, which compromise the cycle cooling capacity.

The calculated COP is also reported for the corresponding DL-2P self-adapting cycle (COP$_{S.A.}$), at the separation ratio automatically set by the phase separator. It can be seen that for all the three conditions the f$_{SEP}$ of the self-adapting cycle corresponds to the split ratio that gives the highest COP for the conventional cycle. Moreover, the COP of the former is equal to the maximum COP of the latter. For what concerns the quality of the vapour leaving the

separator ($x_{RCA\ out,\ S.A.}$), given the nature of the separation process itself, it is always 1.

4.2. Comparison among different configurations

In this section, the COP of the five new configurations are compared with one another and with the self-adapting DL-2P cycle. In particular, the COP is calculated with different inlet water temperatures at the condenser and absorber (T$_{W\ inl}$) and for three heat source fluid temperature levels, i.e. inlet/outlet of 12/7 °C, −10/−15 °C and −30/−35 °C. As anticipated, the inlet/outlet temperatures of hot water driving the generator are 90/80 °C and the outlet temperature of the cooling water is 10 °C higher than the inlet temperature (T$_{W\ inl}$). The results are summarized in two charts (see Fig. 10 and Fig. 11), the first one using NH$_3$–H$_2$O as working pair and the second one with NH$_3$-LiNO$_3$.

As expected, with both working pairs and for all the brine temperature levels, the COP of the double-lift cycles is usually higher than the COP of the triple-lift cycles, even if the difference becomes smaller at low brine temperatures. On the other hand, at constant heat source fluid temperature, the triple-lift cycles are able to operate with cooling water temperatures from 5 to 8 °C higher than the double-lift cycles. This feature can be either negligible or very valuable, based on the application. As an example,
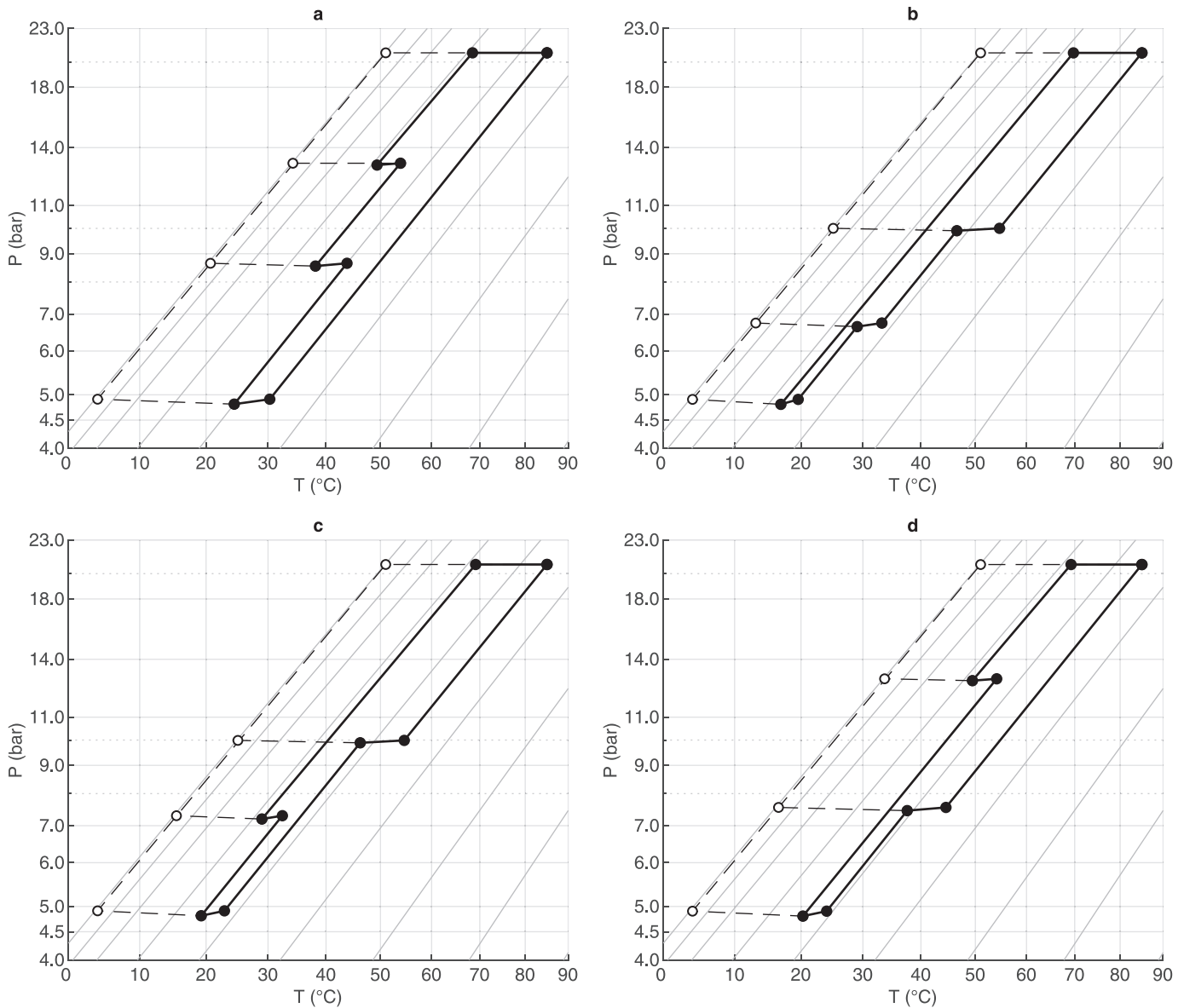
Fig. 8. P-T-X diagram of the TL-3P (a), TL-1P (b), TL-2Pa (c) and TL-2Pb (d) cycles.
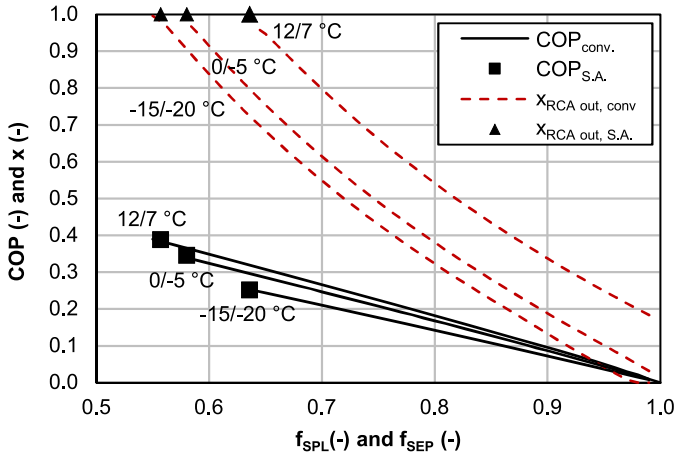

Fig. 9. comparison of the COP and vapour quality after the RCA of the conventional and self-adapting DL-2P cycle.

a higher heat rejection temperature may allow the use of dry cooling instead of wet cooling, which represents an advantage for small size applications and where the access to water is limited. Moreover, the wider operating range of the triple lift cycles may be relevant for heat recovery, where the possibility to run the appliance is more valuable than its efficiency.

Comparing the performances of the two double-lift cycles, the two-pump version has a higher COP and a slightly wider operating range. This is more relevant at brine temperature $-30/-35$ °C, while the difference become small at 12/7 °C. Additionally, larger difference between the two configurations are found with the NH_3-$LiNO_3$ than with NH_3–H_2O working pair.

The same trend is found in the triple-lift cycles, with the largest differences at low brine temperatures and for the NH_3-$LiNO_3$ rather than for the NH_3–H_2O pair. Moreover, the differences among the triple-lift configuration increase with the cooling water temperature, i.e. with the thermal lift. In particular, at high lift conditions the best performances are found for the three-pump version, while the lowest are found for the one-pump cycle, with the two-pump configurations laying in between. At low water
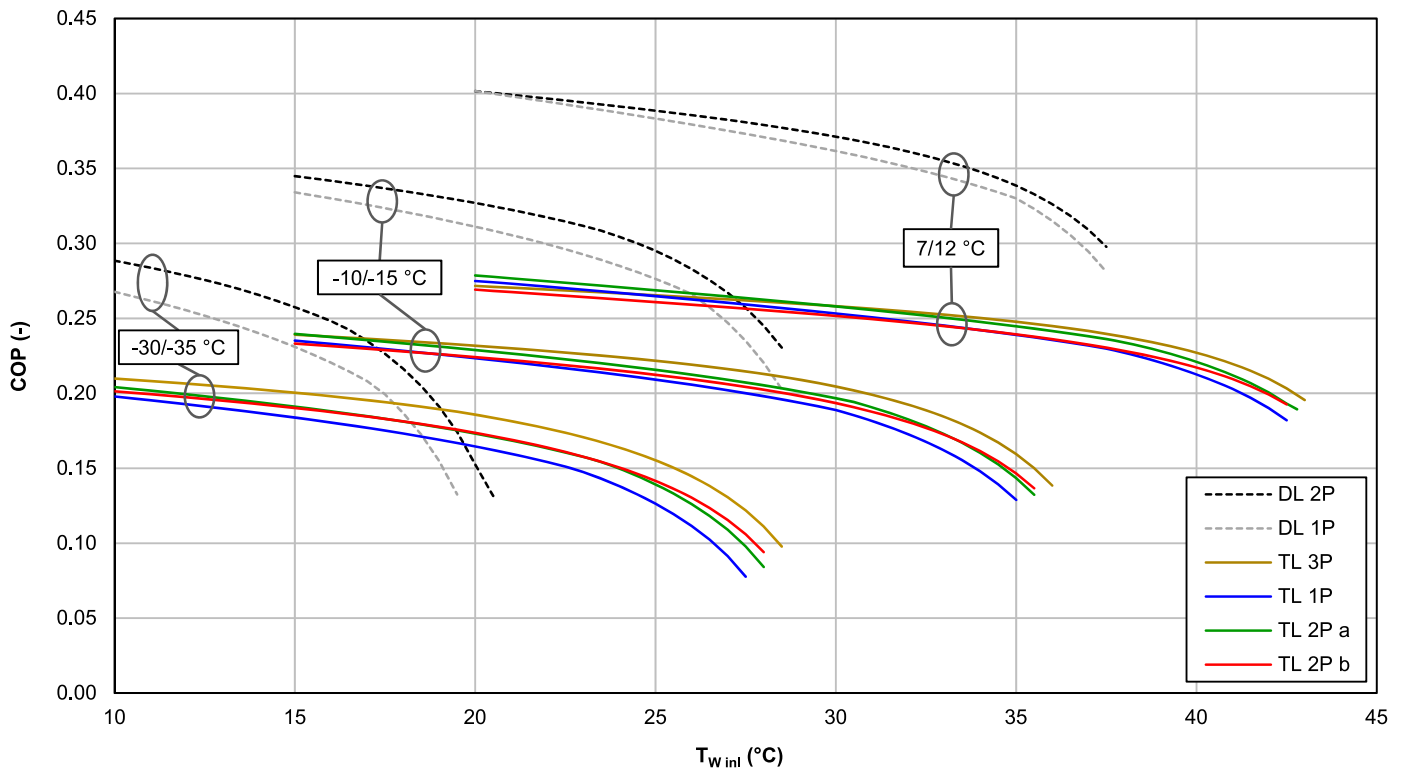
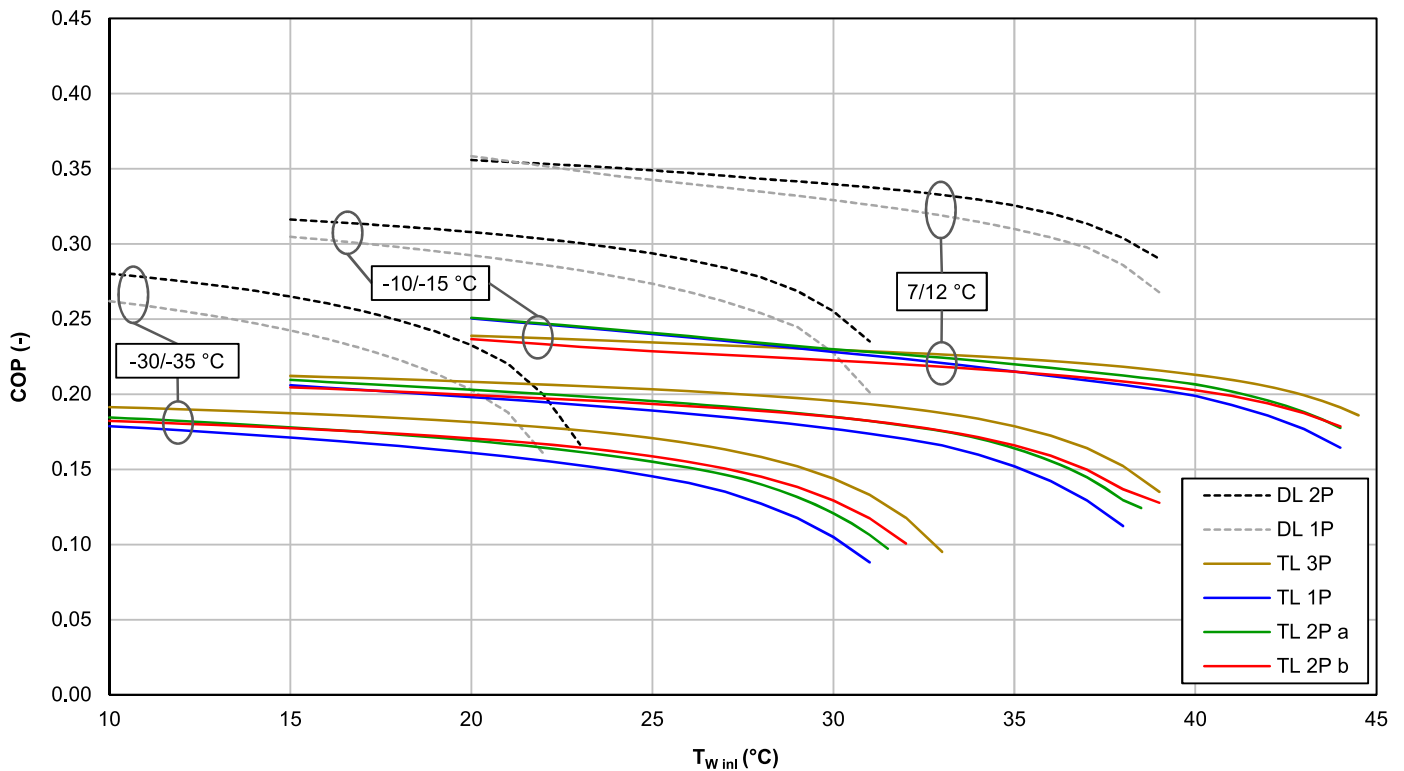**Fig. 10.** COP at various heat rejection and heat source temperatures with NH_3-H_2O as working pair.

Fig. 11. COP at various heat rejection and heat source temperatures with NH_3-LiNO_3 as working pair.

temperature, the differences among the cycles are small, but at chilled water temperature of 12/7 °C the TL-1P cycle or the TL-2Pa become the most efficient by a narrow margin.

The different performance between the one-pump and the multiple-pump cycles can be explained considering two main counteracting factors. In particular, the temperature variation across the absorber is higher for one-pump cycles than for multiple pump cycles. This influences the position of the pinch between solution and cooling water, which is located at the cooling water inlet/solution outlet side for the former cycle and at the solution inlet/cooling water outlet side for the latter. This causes a lower outlet solution temperature for the one-pump cycles, which
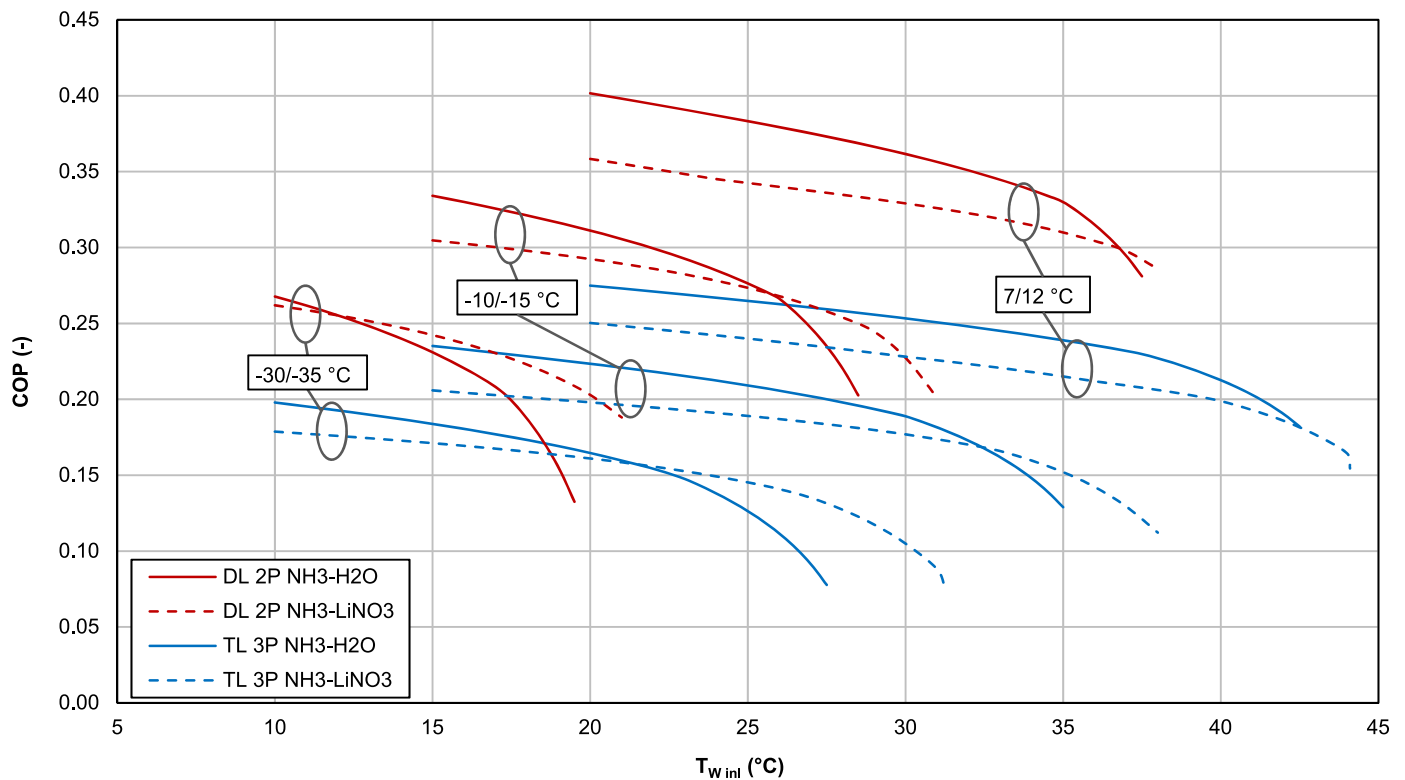
Fig. 12. COP at various heat rejection and heat source temperatures for the cycles DL-1P and TL-1P with NH_3-H_2O and NH_3-$LiNO_3$ as working pairs.

is beneficial for the cycle performance. However, the pressure staging is favorable for the multiple pumps cycles, which have higher intermediate pressures and, consequently, lower refrigerant expansion losses. The impact of this two factor changes with the working conditions. In particular, the advantage of the one-pump cycles decreases when the thermal lift increases, while the impact of the pressure distribution increases with the thermal lift. Consequently, one-pump cycles have a COP similar or slightly higher than multiple pump cycles at low thermal lift (both high cold source temperatures and low cooling water temperature), while multiple pump cycles have higher performances at high thermal lift.

4.3. Impact of the working pair

Given the similar relative trends found in the previous section, the impact of the working pairs on the cycles performances is evaluated based on a representative double-lift and a representative triple-lift cycle. In particular, the DL-2P and the TL-3P cycles are selected, given that they are the most efficient in the high lift operation, i.e. when the multiple-lift cycles are mostly needed. The results of the comparison are reported in Fig. 12 where the COP of the cycles is plotted against the inlet water temperature. For both double and triple-lift cycles the NH_3-H_2O pair displays a higher COP at low cooling water temperature, with a wider gap at high heat source fluid temperature and a smaller difference when the brine temperature is low. On the contrary, as the water temperature increases, the COP of the NH_3-LiNO_3 becomes higher than the COP of the NH_3-H_2O. Moreover, the slope of the COP curve of the NH_3-LiNO_3 pair is lower than the slope of NH_3-H_2O, which experiences a more rapid drop in performances and reaches earlier the cut-off conditions. In fact, the cut-off temperature of the double-lift cycle is 0–3 °C higher with the NH_3-LiNO_3 than with the NH_3-H_2O, while the difference becomes of 2–4 °C in the case of the triple-lift cycle. The advantage of the $LiNO_3$ against

H_2O at high cooling water temperatures is more pronounced at low brine temperatures rather than at high brine temperatures, i.e. at high thermal lifts.

Summarizing, the comparison between the two working pair shows an advantage of the NH_3-H_2O pair at relatively high brine temperatures and when the cycle operates far from its cut-off conditions, thanks to the higher COP. On the other hand, the NH_3-LiNO_3 displays a higher COP with low brine temperatures and high water temperatures, also benefitting from higher cut-off temperatures.

4.4. Heat duties

When dealing with heat recovery, other parameters besides the COP and the number of pumps must be taken into account when comparing different cycles. A possible parameter for cycle comparison is the size of the heat exchangers. Following the approach proposed by Erickson and Tang (1996), a first estimation of the size of the heat exchangers is provided by their heat duties, normalized based on cooling capacity. A more detailed approach would require the estimation of the heat transfer rate for the different heat exchangers based on fluid properties, geometry and flow pattern, which is outside the scope of this work.

The information related to the heat duties of the heat exchangers is given for the two double-lift cycles and for the TL-1P and the TL-3P, which may be considered as the upper and lower limits for the triple-lift cycles in terms of both efficiency and characteristics. Given that double-lift and triple-lift target different thermal lifts, the heat duties are provided for two sets of three operating conditions each. The heat source fluid temperatures are the same for the two sets (12/7 °C, −10/−15 °C and −30/−35 °C), while different cooling water temperature are used: 35/45 °C, 25/35 °C and 15/25 °C for the double-lift cycles and 43/53 °C, 33/43 °C and 23/33 °C for the triple-lift cycles. The choice of the water temperature levels has been done with the purpose of representing
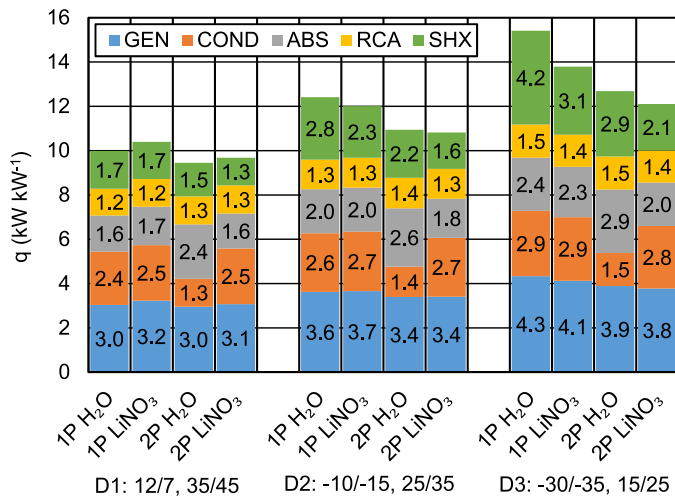
Fig. 13. Heat duties of the DL-1P and DL-2P cycles with NH_3-H_2O and NH_3-LiNO_3 as working pairs.
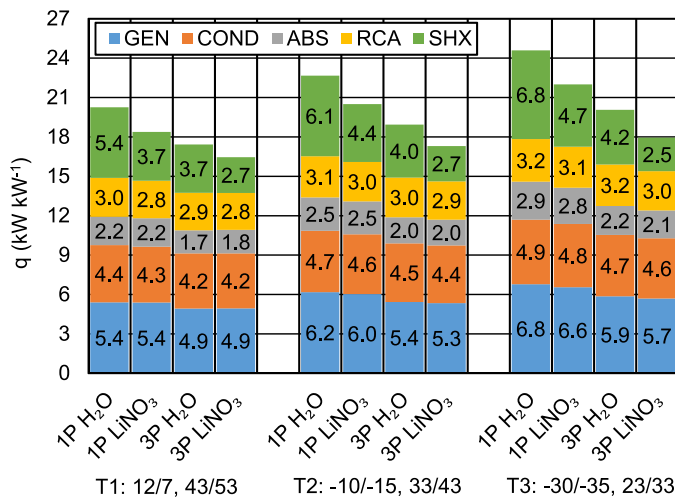

Fig. 14. Heat duties of the TL-1P and TL-3P cycles with NH_3-H2O and NH_3-LiNO_3 as working pairs.

interesting working conditions, i.e. with a thermal lift suitable to display some differences among cycles and working pairs, without moving too close to the cut-off conditions, where those differences increase significantly and unevenly. The conditions for the double-lift cycle will be identified as D1, D2 and D3, while the conditions for the triple-lift cycles will be referred as T1, T2 and T3. The heat duties reported in Fig. 13 and Fig. 14 are subdivided into the contributions given by the different heat exchangers: generator (GEN), condenser (COM), absorbers (ABS), refrigerant cooled absorbers (RCA) and solution heat exchangers (SHX).

At condition D1, with brine temperature of 12/7 °C and water temperature of 35/45 °C, few differences exist between the two double-lift cycles, with both working pair (see Fig. 13). The highest value is found for the DL-1P with LiNO$_3$ as absorbent, which has, in the given conditions, the lowest COP and, consequently, the highest heat duty at the generator. Moving to brine temperature of −10/−15 °C and cooling water temperatures of 25/35 °C (condition D2), the heat duty of the DL-1P cycle increases respect the DL-2P cycle and for both the cycles a higher value is found with NH_3-H_2O as working pair. This trend, which is confirmed at brine temperature of −30/−35 °C and water temperatures of 15/25 °C (condition D3), is due to the lower COP of the one-pump cycle, especially at high lifts, as discussed above. In fact, as the heat
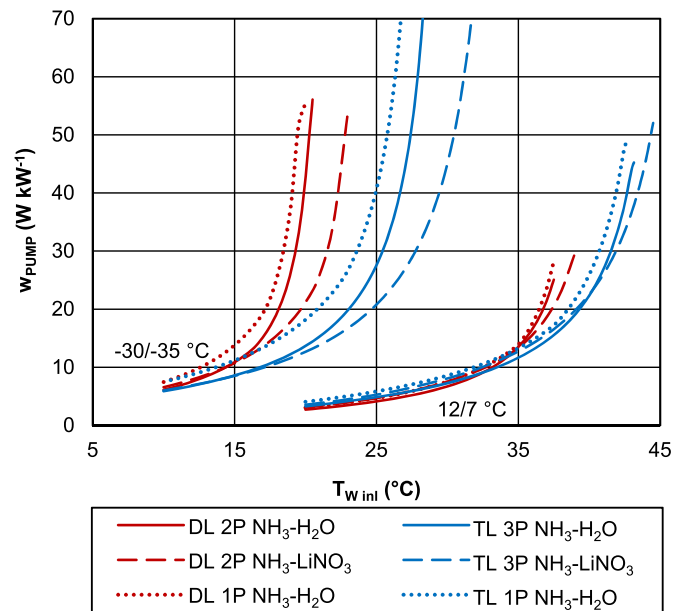

Fig. 15. Specific pumping power vs. inlet water temperature as function of the cycle layout and working pair.

duty at the generator is the inverse of the COP, a reduction of the efficiency automatically causes a growth of the heat duty of generator, absorber and condenser.

Moreover, moving from condition D1 to condition D3, the heat duty at the SHX of the 1P cycles becomes higher than the one of the 2P cycle and the same happens moving from NH_3-LiNO_3 to NH_3-H_2O. While the differences among the conditions can again be explained with the different COP, the impact of the working pair is also due to the lower specific heat capacity of the NH_3-LiNO_3 solution. At condition D1, the differences between the working pairs are mitigated by the higher COP of the NH_3-H_2O couple, while at condition D3 the effect of lower COP and higher specific heat capacity sum up and give heat duties at the SHX significantly higher than for NH_3-LiNO_3.

The same discussion can be done for the triple effect cycles, where the higher heat duty of the one-pump cycle and of the NH_3-H_2O pair can be already found at condition T1 (see Fig. 14). Moreover, since the weight of the SHX is higher in the triple-lift than in the double-lift cycles, the differences among the heat duties of these heat exchangers is more impacting on the overall heat duty.

4.5. Specific pumping power

As anticipated in Section 3, in the present work the comparison of the cycles based on the required mechanical power input is limited to the work for pumping the solution. In particular, specific pumping power of the DL 2P cycle and the TL 3P cycle using NH_3-H_2O is reported in Fig. 15, changing the inlet water temperature of the heat rejection circuit. Moreover, the impact of cycle layout is investigated adding respectively the DL 1P and TL 1P cycles, while the effect of the working pair is explored with the corresponding NH_3-LiNO_3 cycles. To keep the chart simple, the data are limited to cold source temperatures of 12/7 °C and −30/−35 °C.

As expected, the pumping power increases with the heat rejection temperature, due to the increase of circulation ratio and pressure difference. The values of w_{PUMP} of the selected cycles are quite close when the heat rejection temperature is low, while the differences increase at higher temperatures. Moreover, smaller differences among cycles and working pairs are found for cold source

temperature of 12/7 °C than for −30/−35 °C. This can be explained looking at the differences in terms of COP (see Sections 4.2 and 4.3), which are larger when the cold source temperature is lower.

Comparing double-lift with triple-lift cycles, the former have lower w_{PUMP} values than the latter at low inlet water temperature, but experience a more rapid growth as the temperature increases. This can be justified considering that double-lift cycles approach cut-off condition at lower heat rejection temperature than the triple-lift cycles.

The comparison between single-pump with multiple-pump cycles shows higher w_{PUMP} for the former layout both for the double and the triple-lift cycles. This can be explained considering that even if the overall pressure difference that the pump(s) have to overcome is independent on the cycle layout, the low-pressure pump of the multiple-pump cycles have a slightly lower flow rate than the pump of one-pump cycles. Moreover, multiple-pump cycles have a higher COP, which implies a lower circulation ratio.

As the heat rejection temperature increases, w_{PUMP} of the cycles using NH_3–$LiNO_3$ is lower than the corresponding value with NH_3–H_2O. This can be explained considering that, as discussed in Section 4.3, with the former working pair the cycles maintain a higher COP than with the latter when the heat rejection temperature increases.

5. Conclusions

Five new cycles have been presented, introducing the self-adapting concept in the one-pump double-lift cycle and in four triple-lift cycles. By means of numerical calculation it has been shown that the use of a phase separator allows the cycle to automatically operate at the conditions which would have been achieved in the original cycles only with the optimal regulation of the split valve.

The performances of the new cycles have been numerically compared at three heat source fluid temperature levels and different cooling water temperature, using NH_3–H_2O and NH_3–$LiNO_3$ as working pairs. The results of the comparison can be summarized as follows:

- the COP is about 0.1 higher for double-lift than for triple-lift cycles;
- cycles with one pump have lower COP than cycles with two or three pumps, especially under condition of high lift and low thrust. Moreover, they have a slightly narrower operating range, since they reach cut-off condition at lower water temperatures;
- if NH_3–H_2O is used as working pair, higher efficiency can be reached but the operating range is more limited. Additionally, the performance with NH_3–H_2O is more affected by the thermal lift than with NH_3–$LiNO_3$, meaning that the latter is more suitable at low heat source fluid temperature or high cooling water temperature.
- the heat duties are affected by the COP, therefore heat duty of the one pump cycles becomes higher than the one of the multiple pump cycles at high thermal lift. The same applies for the NH_3–H_2O pair, which, regardless the cycle, requires higher heat duty than NH_3-$LiNO_3$, except for some low lift conditions.
- Larger differences in the specific pumping power are found at high heat rejection temperature, with the multiple-pump cycles performing better than single-pump cycles and NH_3-$LiNO_3$ better than NH_3–H_2O.

Based on these results, it can be summarized that even if the use of a single pump is interesting because it could reduce the complexity and the cost of the chiller, the cycles with one pump have lower performances and higher specific pumping power than the cycles with more than one pump, especially under high lift conditions, where multiple lift cycles are expected to be used. The comparison between the two working pair suggest that the choice should be dependent on the application: at very low heat source fluid temperature NH_3–$LiNO_3$ is the best option in terms of operating conditions, while at higher temperature, NH_3–H_2O may be preferred thanks to the higher COP.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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