

Research Paper

Performance investigation and numerical evaluation of a single-effect double-lift absorption chiller

Abdelmajid Saoud^a, Joan Carles Bruno^{b,*}, Yasmina Boukhchanaa^a, Ali Fellah^c^a University of Gabes, National School of Engineers of Gabes, Laboratory of Applied Thermodynamic, LR18ES33, 6029 Gabes, Tunisia^b Universitat Rovira I Virgili, Mechanical Engineering Dept, CREVER-Research Group on Applied Thermal Engineering, Av. Paisos Catalans 26, 43007 Tarragona, Spain^c University of Gabes, Higher Institute of Applied Sciences and Technology of Gabes, Laboratory of Applied Thermodynamic LR18ES33, 6029 Gabes, Tunisia

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ABSTRACT

This study deals with innovative integrated absorption refrigeration systems. This cycle could be useful for converting low-temperature energy sources into useful outputs like cooling. The Single Effect Double Lift Absorption Chiller (SE-DL AC) was introduced and energetically evaluated. A detailed modeling based on the first law of thermodynamics is built to investigate the performance of this novel system. This model was initially validated considering a reference SE-DL AC from the Hitachi company with a nominal cooling capacity of 106.1 kW. This system is compared to a conventional single-effect absorption chiller while maintaining the same operating parameters and boundary conditions in both configurations to ensure a fair comparison in terms of the internal and external baseline operating conditions. A parametric study of the driving hot water inlet temperature, the solution mass flow rate, the effectiveness of the solution heat exchanger, and internal and external temperatures on the Coefficient of Performance (COP), Cooling capacity (CC), and Glide Temperature (ΔT) are conducted. The final results demonstrate that the cooling production capability of the SE-DL is increased by an average of about tenfold when compared to the conventional SE AC at a thermal source temperature of 88.7 °C and an inlet hot water mass flow rate of 1 kg/s. The COP results produced under the same operating conditions are quite similar. The cooling generation capability of the SEDL is, on average, 10 times greater than that of the traditional SE AC. This explains that the SEDL-AC is well suited for a wide range of low-temperature heat sources (e.g., solar energy). Furthermore, the SE-DL AC enables efficient use of low-grade heat with a high glide temperature in the driving heat stream of about 10 times (36 °C instead of 3.5 °C). This large temperature difference exemplifies that the SE-DL AC absorption chiller is an alternative for extracting as much heat as possible, thereby improving cold production.

1. Introduction

Promoting energy security, production, conversion, and management are among the most critical flaws in today's energy debate. According to the U.S. International Energy Outlook (IEO2021), total energy consumption is projected to rise by roughly 50% over the next 30 years [1]. The volatile energy rates and environmental concerns, such as the depletion of fossil fuels and greenhouse gas (GHG) emissions, create a stumbling block in the way of producing clean and green energy with low CO₂ footprints [2,3].

In this respect, buildings, or the tertiary sector, are considered one of the world's largest energy drivers, accounting for approximately 40% of total energy consumption [4]. According to the 2016 EU Heating and

Cooling Strategy Report, heating and cooling contribute to about 80% of the total energy consumption of buildings [5]. As a result, the integration of innovative energy conversion systems is becoming increasingly important. This solution provides the best compatibility between energy sources supply and load requirements. The transition to sustainable energy is the most effective policy for reducing the overconsumption of fossil fuels [6,7]. Biomass, wind, hydro, and solar appear to be the sole sustainable solution to meet various engineering and comfort standards [8,9].

In the area of heating, ventilation, and air conditioning (HVAC), absorption refrigeration systems (ARS) attract a great deal of attention. These systems can balance the growing demand for indoor heating and air conditioning. Furthermore, they can be "thermally activated" by using low-grade heat sources (solar, waste heat, geothermal)[10].

* Corresponding author.

E-mail address: juancarlos.bruno@urv.cat (J.C. Bruno).<https://doi.org/10.1016/j.applthermaleng.2023.120369>

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Nomenclature

h	Enthalpy (kJ/kg)
$\dot{Q}$	Heat transfer rate (kW)
$T_{h,in}$	Hot side inlet temperature ($^{\circ}\text{C}$)
$T_{h,out}$	Hot side inlet temperature ($^{\circ}\text{C}$)
$T_{c,in}$	Cold side inlet temperature ($^{\circ}\text{C}$)
$T_{c,out}$	Cold side outlet temperature ($^{\circ}\text{C}$)
LMTD	Logarithmic Mean Temperature Difference ($^{\circ}\text{C}$)
$\dot{m}$	Mass flowrate (kg/s)
P	Pressure (KPa)
P_{high}	High Pressure (KPa)
P_{int}	Intermediate Pressure (KPa)
P_{Low}	Low Pressure (KPa)
SCR	Solution Circulation Ratio (-)
T	Temperature ($^{\circ}\text{C}$)
UA	Overall heat transfer coefficient-area product (kW/ $^{\circ}\text{C}$)
$\dot{W}$	Work done by the pump (kW)
X_{LiBr}	Lithium bromide mass fraction (kg LiBr / kg solution)

Abbreviations

AC	Absorption chiller
ARS	Absorption Refrigeration System
ASEV	Auxiliary Solution Expansion Valve
ASHX	Auxiliary Solution Exchanger
ASPP	Auxiliary Solution Pump
COP	Coefficient of performance
EES	Engineering Equation Solver
HSEV	High Solution Expansion Valve
HSHX	High Solution Heat Exchanger
LiBr	Lithium bromide

LSEV	Low Solution Expansion Valve
LSHX	Low Solution heat Exchanger
LSP	Low Solution Pump
REV	Refrigerant Expansion Valve
SE AC	Single Effect Absorption chiller
SE-DL AC	Single effect double lift absorption chiller
SHX	Solution heat exchanger
Str	Stream

Subscripts and superscripts

AA	Auxiliary Absorber
AG	Auxiliary Generator
Aux	auxiliary
C	Condenser
E	Evaporator
HG	High generator
H	High
i	stream number
in	inlet
int	intermediate
LA	Low-Pressure Absorber
LG	Low-Pressure Generator
L	Low
out	Outlet
r	Refrigerant
ss	Strong Solution
ws	Weak Solution

Greek Symbols

ϵ	Solution Heat Exchanger efficiency (%)
ΔT	Temperature difference ($^{\circ}\text{C}$)

Owing to their high reliability and eco-friendliness, these cycles have a lot of advantages over vapor compression systems. For instance, being operated with suitable and environmentally friendly refrigerants.

Two working pairs are commonly used in ARS, namely water-ammonia ($\text{H}_2\text{O}/\text{NH}_3$), which are mainly utilized for refrigeration temperatures below 0°C . For higher refrigeration temperatures, using Water/LiBr is recommended [11,12]. Water/Lithium bromide chillers are preferred in solar absorption refrigeration applications as they are more efficient, have a simple design, and can operate at lower generating temperatures than ammonia/water chillers.

However, despite the evident advantages of ARS, some inconveniences associated with its electric counterparts, such as lower efficiency, higher costs, or a larger footprint, have prevented its widespread application in many sectors. To address the aforementioned issues, innumerable academics and researchers have carried out extensive experimental and numerical research on absorption refrigeration technologies. [13–17]. Numerous features and criteria, such as energy, exergy [18], economic evaluation [19], the influence of heat exchangers [20], and others, have been included in the ARS analysis approach. Hence, the absorption refrigeration system represents a multidisciplinary research area. The methodology of analysis and modeling is one of the ongoing research hotspots. Jie Ren et al. [21] have presented a novel calculation method, rarely used in literature, which is based on mass and energy balances and the Duhring equation. This procedure demonstrates relevant reliability when comparing the $\text{H}_2\text{O}-\text{LiCl}$ and the $\text{H}_2\text{O}-\text{LiBr}$ ARS efficiencies. In another work, using a characteristic equation, Puig-Arnavat et al. [22] compared two approaches based on the characteristic equation and then extended the selected approach to a commercial double-effect chiller. Gogoi et al. [23] performed an exergetic analysis of $\text{H}_2\text{O}/\text{LiCl}$ single-effect vapor absorption refrigeration using the inverse technique to estimate operating temperatures.

The impact of alternative working pairs is another area that has been thoroughly studied [24]. Lithium chloride water ($\text{LiCl}-\text{H}_2\text{O}$) and ionic liquid (IL) appear to be competitive ways to improve the performance of ARS. Gunhan et al. [25] presented an experimental-based exergy analysis of a novel $\text{H}_2\text{O}-\text{LiCl}$. Weng et al. [26] investigated nine working fluids based on ammonia-ionic (NH_3/IL) liquid for an absorption heat pump intended for heating purposes in the residential sector. According to their findings, NH_3/IL beats $\text{NH}_3/\text{H}_2\text{O}$ with a higher COP of 1.84.

As far as configurations are concerned, many studies have been conducted. Saleh et al. [27] conducted a study to optimize a single-effect water/lithium bromide absorption refrigeration system driven by a flat plate collector in hot regions. The major results reveal that the heat source temperature plays a major role in designing and optimizing the overall efficiency of the chiller. The performance of this system could reach a COP of 0.8 when providing the appropriate range of hot sources, cooling, and chilled temperatures. In recent work, a single-effect absorption chiller has also been integrated with an adsorption cycle to improve the overall system performance. Nikbakhti et al. [28] first investigated this cycle integration and later, the same authors continued to research and optimize a similar configuration of the adsorption-absorption (AD-AB) cycle using new key parameters [29]. The results showed that the proposed system could reach a COP of 0.4 with a heat source temperature of 50°C . In another research work, Nikbakhti and Iranmanish [30] investigated the application of solar energy within the AD-AB cycle to evaluate cooling production under several weather conditions in Australia. Two different layouts of solar systems (with and without storage tanks) were examined. According to their results, it was disclosed that including a storage tank improved the daily cooling capacity by 15% with a slightly stable COP of about 0.37. For an optimum solar collector area of 34 m^2 , a payback period of 10.46 years was obtained.

The double-effect absorption system is another ARS configuration that has been, broadly, evaluated in the literature. Kaynakli et al. [31] performed a parametric analysis of a double-effect chiller to investigate the impacts of operating temperatures on the thermal capacity, exergy destruction, and mass flow rate of the heat source on the system performance. Moreover, several common configurations have been reported in the literature, such as double-effect in series, double-effect inverse, and triple effects [32,33]. Likewise, the half-effect chiller has been cited in the literature. Hernández-Magallanes et al. [34] carried out a thermodynamic analysis of a half-effect absorption-cooling system powered by a low-geothermal energy source. The results show that the system can operate at a low generation temperature supplied by a geothermal source but provide a low cooling effect.

Nevertheless, the single-effect double-lift absorption chiller (SE-DL AC) is a new generation of water/lithium bromide absorption systems that have been studied in a limited way [35]. In regards to the commercialization of this cycle arrangement, a couple of companies have moved towards the conception and manufacturing of such chillers. Hitachi Johnson Controls Air Conditioning Co., Ltd. has industrialized a SE-DL AC driven by hot water [36,37]. World Energy is also a leader in the development and creation of novel technologies in the field of absorption cooling chillers. Various single-effect double-lift absorption chillers for cogeneration systems and industrial applications that are powered by hot water have been manufactured by this company [38]. The 2AB single effect double lift hot water absorption chiller is mainly dedicated to district heating and industrial use, with a capacity range of 105 to 7033 kW.

According to this review, the cited studies emphasize the importance of investigating the single-effect double-lift absorption chiller technology. There is a significant gap regarding this novel cycle evaluation.

In this paper, novel integrated absorption chillers based on the double-lift cycle and the single-effect absorption chiller is studied in detail. This system is allowing for better use of solar thermal or waste heat energy by increasing the temperature glide or difference of the driving heat. This glide usually in the range of 5 to 10 °C could reach around 50 °C with the proposed configuration while maintaining a similar chiller efficiency. Although a few commercial units are available and their performance has not been modeled and studied in detail in the open literature. In the present work, this absorption cooling system is investigated numerically with a detailed thermal model developed under steady-state conditions using the commercial software Engineering Equation Solver (EES). Parametric analysis of the cooling chiller performance is validated with the available commercial SE-DL AC from the Hitachi company. The overall performance of the system is compared to that of a basic single-effect absorption chiller under the same operating conditions.

2. Description of the single-effect double-lift absorption chiller configuration

The SE-DL cycle is a modification of the well-known basic single-effect cycle. Just to provide a complete description of the whole scenario, the configuration for both chillers is presented in this section. Figs. 1 and 2 show the single-effect and single-effect double-lift absorption chiller configurations considered in this study.

The single-effect chiller operates on a straightforward principle. It is made up of four heat exchangers: a generator, an evaporator, a condenser, and an absorber. Its operation begins when heat is applied to the generator, causing the refrigerant (water) to partially separate from the mixture (water/LiBr). The refrigerant will pass through the condenser before entering the evaporator, where it will cool. The refrigerant (Str. 10) is then absorbed in the absorber using a solution of concentrated LiBr (Str. 6). The weak solution leaving the absorber (Str. 1) is pumped to the generator using a solution pump. At the generator, the solution is separated into refrigerant (Str. 7) and strong solution (Str. 4), which is concentrated in the absorbent (LiBr), and the cycle starts

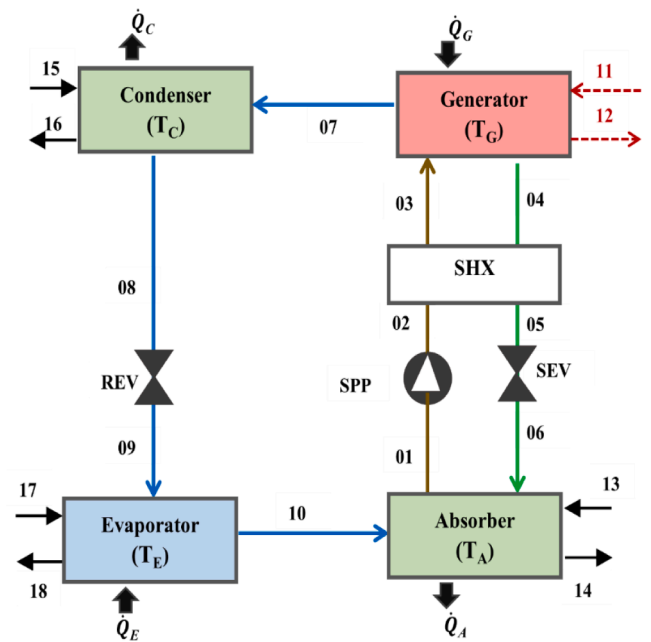


Fig. 1. Schematic representation of the single-effect absorption chiller (SE AC).

again. A solution heat exchanger (SHX) is used for internal heat recovery.

Fig. 2 depicts a schematic representation of the single-effect double-lift absorption chiller (SE-DL AC) investigated in this study. This system could be seen as an integration of a single-effect absorption chiller (SE AC) into a double-lift absorption chiller (DL AC). SE-DL AC is a multi-component and complex system that includes mainly 10 heat exchangers: three generators (high, auxiliary, and low), two absorbers (low and auxiliary), an evaporator, a condenser, three solution heat exchangers (low, high, and auxiliary), two solution expansion valves, one refrigerant expansion valve, and two circulating solution pumps.

This cycle runs under three pressure levels: the high pressure (P_C) at the high auxiliary generators and condenser, an intermediate pressure (P_{int}) at the low generator and auxiliary absorber, and the evaporator and low absorber operated at low pressure (P_E). The entire cycle is made up of two closed solution circuits, the first of which begins with the low absorber and proceeds through the low SHX, the high SHX, the high generator, and then the low generator before returning to the low absorber.

At the outlet of the low-pressure absorber (Str. 1), a weak LiBr solution is driven through the low-circulation pump (Str. 2) to a high-pressure level and reaches the high-pressure generator. To improve the efficiency of the system, two solution heat exchangers were incorporated in-between (Str. 3 and Str. 4). Leaving the low-pressure absorber, the dilute solution will be preheated in the low solution heat exchanger (L_{SHX}) to flow later to the high solution heat exchanger (H_{SHX}) before entering the high generator. The driving heat (Q_{LG}) produces partial evaporation of the solution, and the refrigerant vapor (Str. 1a) is sent to the condenser, where it is condensed. The mid-strong solution leaves the high-pressure generator (Str. 5), passes through the high solution heat exchanger, and reaches the low generator (Str. 7). The auxiliary absorber (Str. 3a) absorbs the refrigerant produced at the low-pressure generator. The strong solution at the intermediate pressure level (Str. 8) leaves the low generator, is cooled by the low solution heat exchanger, and expanded (Str. 9) before returning to the low absorber. In this way, the first solution loop is completed.

The primary components of the second solution circulation circuit are the auxiliary absorber, the auxiliary solution heat exchanger, and the auxiliary generator. The weak solution ($X_{w,s}$) at the outlet of the auxiliary absorber (Str. 14) flows to the auxiliary generator (Str. 16)

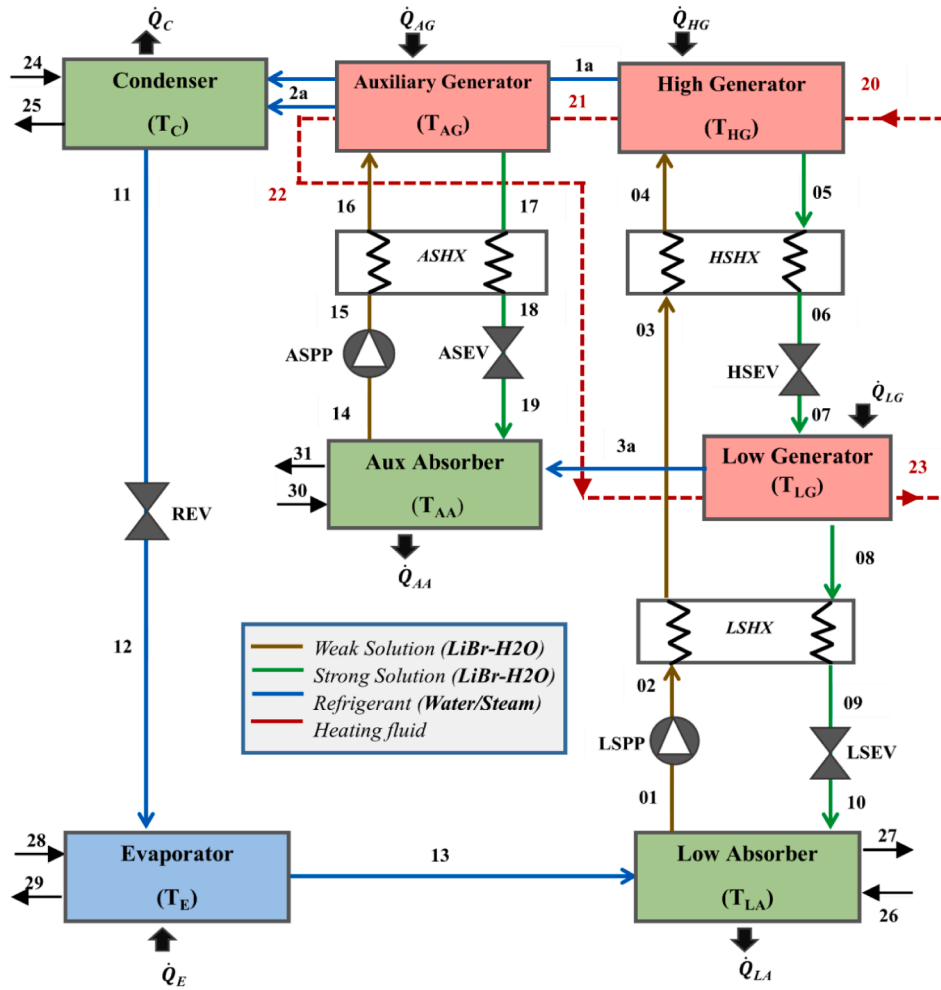


Fig. 2. Schematic representation of the single effect double lift absorption chiller (SE-DL AC).

through an auxiliary solution heat exchanger (Str. 15). Overheated water vapor is produced at the auxiliary generator (Str. 2a). The liquid refrigerant produced at the high and auxiliary generators is condensed (Str. 11). The water at the condenser outlet continues to flow over the throttling valve, where the temperature will drop (Str. 12), and thus, the water/steam mixture comes about. This mixture flows and enters the evaporator (Q_E), where the cooling effect occurs. The refrigerant at the outlet of the evaporator (Str. 13) is absorbed by the low-pressure absorber, where it will form a solution with the residual LiBr mixture. Heat will be released into the environment, closing the thermodynamic cycle of the SE-DL AC.

3. Modeling of the Single-effect double-lift absorption chiller

The commercial software Engineering Equation Solver (EES) [39] is used to solve the thermodynamic equations of the single effect double lift absorption chiller. EES is a powerful tool for solving problems related to thermodynamics and heat transfer applications. It has an extensive data bank of built-in libraries of thermodynamic, thermophysical, and transport properties. In addition, it includes a wide number of fluids (water, air), organic fluids, and mixtures (Water/LiBr) among others [40].

3.1. Assumptions

In the present section, the governing equations of the absorption chiller (Figs. 1 and 2) are described in significant detail. The adopted

assumptions for this study are conducted as follows:

- Each component of SE-DL AC and SE AC runs under steady-state working conditions.
- Pipelines and the pressure losses of heat exchangers are negligible, as they are very small in both chillers [41]. Only pressure drops across the expansion and throttling valves are taken into account. The work generated by the solution pumps (SPPs) is neglected.
- Pumping and throttling are assumed to be isenthalpic processes [41].
- The low and auxiliary absorbers and the condenser are supposed to work at the same temperature level (35 °C), to allow heat rejection to the ambient [42].
- All heat exchangers are assumed to be adiabatic (without heating loss to the ambient) [42].
- The refrigerant at the outlet of the condenser (str 11 and str 8) is taken to be saturated liquid water ($Q_{11}, Q_8 = 0$), while at the outlet of the evaporator, it is considered saturated vapor ($Q_{13}, Q_{10} = 1.0$).
- For the SE-DL AC, streams 1, 5, 8, 14, and 17 are considered to be saturated liquid solutions with a title (vapor quality) equal to 0. In the SE AC (Fig. 1), streams 1 and 4 are assumed to be saturated liquids.
- The intermediate pressure of the SE-DL AC is calculated using the auxiliary absorber concentration and temperature under saturated conditions.

3.2. Thermodynamic formulation

Thermodynamic modeling is performed under the first law of thermodynamics and the laws of mass and energy conservation. Given that each device of the AC is held as a control volume with inlet and outlet streams, heat transfer, work exchanges, and the thermophysical properties of the working fluid. The enthalpy change per component of the chiller at various state points is a critical parameter that can be calculated by combining the temperature (T), pressure (P), and lithium bromide mass fraction (X_{LiBr}). The equation that describes this correlation is presented as follows:

$$h_i = f(P_i, T_i, X_{LiBr,i}) \quad (1)$$

Based on the equation above (equation (1)), many of the state properties of the lithium bromide equation are derived using the equation established by Pátek and Klomfar [43].

3.2.1. Mass conservation

For steady-state and steady flow, the general governing equation for mass conservation in the SE-DL AC consists of the mass balance of the total mass and each species of solution.

- Conservation of total mass

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (2)$$

- Conservation of mass for each specie

$$\sum \dot{m}_{in} X_{in} = \sum \dot{m}_{out} X_{out} \quad (3)$$

$\dot{m}$ is the mass flow rate of the low and high solution pumps, respectively, and $\times$ is the concentration of the lithium bromide in the solution.

3.2.2. First law analysis

The following equation is used to calculate the energy balance rates in the SE-DL AC thermal device:

$$\sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} + [\sum \dot{Q}_{in} - \sum \dot{Q}_{out}] \pm \dot{W} \quad (4)$$

3.2.2.1. Generators. The present SE-DL AC describes two pressure levels for generators: high pressure (high and auxiliary generators); and an intermediate pressure level (low-pressure generator).

Referring to Fig. 2, the quantity of heat delivered by thermal energy (solar, waste heat, etc.) to all these generators represents the energy input to the chiller cycle, and the conservation of total mass and energy at the generators is stated as follows:

High-pressure generator

$$\dot{m}_4 = \dot{m}_5 + \dot{m}_{1a} \quad (5)$$

$$\dot{m}_4 = \dot{m}_5 \quad (6)$$

$$\dot{Q}_{HG} = (\dot{m}_{1a} h_{1a} + \dot{m}_5 h_5) - \dot{m}_4 h_4 \quad (7)$$

Auxiliary generator

$$\dot{m}_{16} = \dot{m}_{17} + \dot{m}_{2a} \quad (8)$$

$$\dot{m}_{16} = \dot{m}_{17} \quad (9)$$

$$\dot{Q}_{AG} = (\dot{m}_{2a} h_{2a} + \dot{m}_{17} h_{17}) - \dot{m}_{16} h_{16} \quad (10)$$

Low-pressure generator

$$\dot{m}_7 = \dot{m}_8 + \dot{m}_{3a} \quad (11)$$

$$\dot{m}_7 = \dot{m}_8 \quad (12)$$

$$\dot{Q}_{LG} = (\dot{m}_{3a} h_{3a} + \dot{m}_8 h_8) - \dot{m}_7 h_7 \quad (13)$$

According to the previously stated assumptions, saturated water is present at the outlet of each generator (Str 1a, 2a, and 3a), which is 100% water with no salt content.

To avoid over-specification of the governing equation in the SE-DL AC model, it is useless to apply the total mass conservation equation (equation (2)) and the conservation equation in each species (equation (3)) across the absorbers (low and auxiliary) because those equations are the same through the three generators (equations (5), 6, 8, 9, 11, and 12).

3.2.2.2. Solution heat exchangers. There are three solution heat exchangers (SHX) in the current chiller configuration. Two equations must be defined for their analysis. The first equation is the energy balance of the hot and cold streams for each SHX, as given by equations (2) and (3). The second equation defines effectiveness as the ratio of actual to maximum possible heat transfer and is described by equations (15), 17, and 19).

Low Solution Heat Exchanger

$$\dot{m}_{w,s}(h_3 - h_2) = \dot{m}_{ss}(h_8 - h_9) \quad (14)$$

$$\varepsilon_{L.SHX} = \frac{Q_{act}}{Q_{max}} = \frac{T_8 - T_9}{T_8 - T_2} \quad (15)$$

High Solution Heat Exchanger

$$\dot{m}_{w,s}(h_4 - h_3) = \dot{m}_{ss}(h_5 - h_6) \quad (16)$$

$$\varepsilon_{H.SHX} = \frac{Q_{act}}{Q_{max}} = \frac{T_5 - T_6}{T_5 - T_3} \quad (17)$$

Auxiliary Solution Heat Exchanger

$$\dot{m}_{w,s}(h_{16} - h_{15}) = \dot{m}_{ss}(h_{17} - h_{18}) \quad (18)$$

$$\varepsilon_{A.SHX} = \frac{Q_{act}}{Q_{max}} = \frac{T_{17} - T_{18}}{T_{17} - T_{15}} \quad (19)$$

3.2.2.3. Absorbers. The SE-DL chiller has two absorbers: a low and an auxiliary absorber. The two absorbers and condenser temperatures are set to operate at the same temperature (35 °C), which is a logical hypothesis that should be taken into account.

Equations (20) and (21) define the energy balances in the low and auxiliary absorbers, respectively.

$$\dot{Q}_{LA} = (\dot{m}_{13} h_{13} + \dot{m}_{10} h_{10}) - \dot{m}_1 h_1 \quad (20)$$

$$\dot{Q}_{AA} = (\dot{m}_{3a} h_{3a} + \dot{m}_{19} h_{19}) - \dot{m}_{14} h_{14} \quad (21)$$

3.2.2.4. Condenser. The rate of heat rejection out of the condenser is written as follows:

$$\dot{Q}_C = \dot{m}_{1a} h_{1a} + \dot{m}_{2a} h_{2a} - \dot{m}_{11} h_{11} \quad (22)$$

$$\dot{Q}_C = \dot{m}_r \cdot ((h_{1a} + h_{2a}) - h_{11}) \quad (23)$$

The mass balance equation required to represent energy balances through the condenser is given by equation (24):

$$\dot{m}_{11} = \dot{m}_{1a} + \dot{m}_{2a} \quad (24)$$

$$\dot{m}_r = \dot{m}_{11} = \dot{m}_{1a} + \dot{m}_{2a} \quad (25)$$

3.2.2.5. Evaporator. The quantity of energy added to the evaporator is determined by the energy balance, as below:

$$\dot{Q}_E = \dot{m}_{13} \cdot h_{13} - \dot{m}_{12} h_{12} \quad (26)$$

3.2.2.6. Pumps. Aside from modeling the primary SE-DL AC heat exchangers (absorbers, generators, condenser, evaporator, and solution heat exchangers), the equations (27)–(34) must be considered. There are two solution pumps: low and high solution pumps, where the first one is used to pick up the mixture from low to high pressure, and the second solution pump is used to take the intermediate mass flow rate of the auxiliary absorber to a high-pressure level.

The following equations describe the basic governing equations of solution pumps.

Low SPP

$$\dot{W}_{LSPP} = \dot{m}_2 h_2 - \dot{m}_1 h_1 \quad (27)$$

Auxiliary SPP

$$\dot{W}_{ASPP} = \dot{m}_{15} h_{15} - \dot{m}_{14} h_{14} \quad (28)$$

The enthalpy across the low and auxiliary circulating pumps is quite low in comparison to the heat transfer rate of the primary heat exchangers, which results in a negligible amount of work being delivered by the pump.

$$h_2 \approx h_1 \quad (29)$$

$$h_{15} \approx h_{14} \quad (30)$$

Moreover, four expansion or reducing valves were introduced within the SE-DL circuit, with their enthalpies considered to be constant.

$$h_6 \approx h_7 \quad (31)$$

$$h_8 \approx h_9 \quad (32)$$

$$h_{18} \approx h_{19} \quad (33)$$

$$h_{11} \approx h_{12} \quad (34)$$

3.2.3. COP and heat exchange modeling

After determining the amount of heat transfer in the heat exchangers, the coefficient of performance can be calculated (equation (35)). The system COP is defined as the heat load added to the evaporator per unit of heat input to the three generators while ignoring the work done by pumps (low and auxiliary).

$$COP = \frac{\dot{Q}_E}{(\dot{Q}_{HG} + \dot{Q}_{AG} + \dot{Q}_{LG})} \quad (35)$$

The introduction of the heat exchange process output among the internal and external streams that are responsible for the heat exchange is another crucial element of thermodynamic modeling in the numerical simulation of the system. Throughout the complete heat transfer coefficient-area product (UA) formulation, the linked heat and mass transfer processes at the different units are modeled using the logarithmic mean temperature difference (LMTD) method presented in equations (36) and (37). H and C refer to the hot and cold sides of a heat exchanger, respectively.

$$Q_i = LMTD \cdot UA \quad (36)$$

$$LMTD = \frac{(T_{H,in} - T_{C,out}) - (T_{H,out} - T_{C,in})}{\ln \left[\frac{T_{H,in} - T_{C,out}}{T_{H,out} - T_{C,in}} \right]} \quad (37)$$

Because EES solves the LMTD equation iteratively, it may attempt negative values during the simulation, causing EES to stop. To avoid this, a useful function for stabilizing the logarithmic mean temperature difference calculation was implemented. This function restricts the null LMTD values by constraining the calculation for the null numerator. This way, there will be no issues with UA values when searching in broad ranges of EES code variables to investigate the solution. Following these boundary conditions, it is simple to calculate the thermal loads and

product heat transfer area with the logarithmic mean temperature difference.

The logarithmic mean temperature difference and heat transfer rates for the SE-DL AC are calculated using Equations (38)–(51).

For the high generator

$$Q_{HG} = \dot{m}_{20}(h_{20} - h_{21}) = LMTD_{HG} \cdot UA_{HG} \quad (38)$$

$$LMTD_{HG} = \frac{(T_{20} - T_5) - (T_{21} - T_{1a})}{\ln \left[\frac{T_{20} - T_5}{T_{21} - T_{1a}} \right]} \quad (39)$$

For the auxiliary generator

$$Q_{AG} = \dot{m}_{21}(h_{21} - h_{22}) = LMTD_{AG} \cdot UA_{AG} \quad (40)$$

$$LMTD_{AG} = \frac{(T_{21} - T_{17}) - (T_{22} - T_{2a})}{\ln \left[\frac{T_{21} - T_{17}}{T_{22} - T_{2a}} \right]} \quad (41)$$

For the low generator

$$Q_{LG} = \dot{m}_{22}(h_{22} - h_{23}) = LMTD_{LG} \cdot UA_{LG} \quad (42)$$

$$LMTD_{LG} = \frac{(T_{22} - T_8) - (T_{23} - T_{3a})}{\ln \left[\frac{T_{22} - T_8}{T_{23} - T_{3a}} \right]} \quad (43)$$

For the Low absorber

$$Q_{LA} = \dot{m}_{26}(h_{27} - h_{26}) = LMTD_{LA} \cdot UA_{LA} \quad (44)$$

$$LMTD_{LA} = \frac{(T_{10} - T_{27}) - (T_1 - T_{26})}{\ln \left[\frac{T_{10} - T_{27}}{T_1 - T_{26}} \right]} \quad (45)$$

For the auxiliary absorber

$$Q_{AA} = \dot{m}_{30}(h_{31} - h_{30}) = LMTD_{AA} \cdot UA_{AA} \quad (46)$$

$$LMTD_{AA} = \frac{(T_{19} - T_{32}) - (T_{14} - T_{31})}{\ln \left[\frac{T_{19} - T_{32}}{T_{14} - T_{31}} \right]} \quad (47)$$

For the condenser

$$Q_C = \dot{m}_{24}(h_{25} - h_{24}) = LMTD_C \cdot UA_C \quad (48)$$

$$LMTD_C = \frac{(T_{11} - T_{25}) - (T_{11} - T_{24})}{\ln \left[\frac{T_{11} - T_{25}}{T_{11} - T_{24}} \right]} \quad (49)$$

For the evaporator

$$Q_E = \dot{m}_{28}(h_{28} - h_{29}) = LMTD_E \cdot UA_E \quad (50)$$

$$LMTD_E = \frac{(T_{12} - T_{29}) - (T_{13} - T_{28})}{\ln \left[\frac{T_{12} - T_{29}}{T_{13} - T_{28}} \right]} \quad (51)$$

4. Simulation methodology

To fully understand the computational process followed during this simulation, Fig. 3 shows the simulation flowchart of the model that was implemented in EES.

Because the proposed model is complex and contains many equations, two simulation steps have been defined to evaluate the overall energy performance of the system. The first one focuses on the modeling and simulation of the internal stream properties of the chiller cycle, including the three pressure levels (P_{high} , P_{int} , and P_{low}) that define the working pressure for the entire system. The concentration of the system is also defined, allowing for the calculation of the enthalpy of each stream (Equation (1)). After designing all the internal operating parameters, the overall performance of the chiller can be calculated.

In the second step of the simulation, the sizes of the various heat

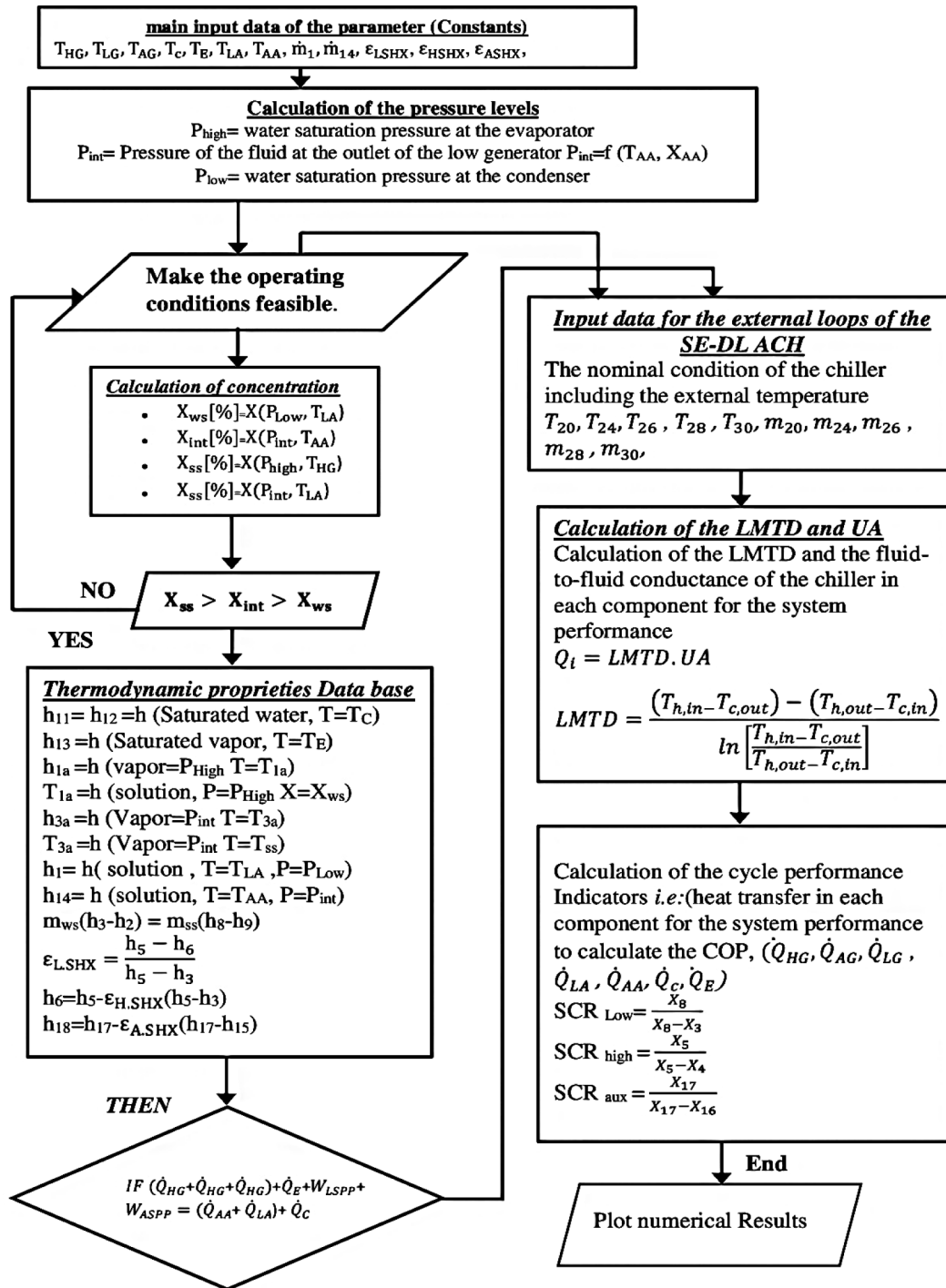


Fig. 3. Flowchart of the developed methodology and modeling in EES software.

exchangers that make up the SE-DL AC are calculated (Table 5). The heat transfer-area product values (UA values) of the heat exchangers (generators, absorbers, condenser, and evaporator) are determined considering both the nominal conditions and the preliminary simulated internal operating conditions in the first simulation step. The calculated UA values were later used as operating parameters to analyze the cycle.

The SE-DL simulation model was validated using data from a similar chiller manufactured by Hitachi [36]. The results will be presented later on in Section 5.1. Furthermore, to calculate the absorption chiller's reference or nominal operation point (Fig. 2), the mass flow rate of all the external heat sources, cold sinks, and chilled water streams was calculated using the available technical specifications of the Hitachi SE-

DL AC.

5. Results and discussion

This section is devoted to highlighting the main findings of this study. First, section 5.1 determines the performance of the SE-DL chiller under the expected nominal conditions. In section 5.2, a sensitivity analysis concerning the driving heat temperature is carried out to determine the most favorable driving temperature for the new cycle, and, finally, the performance is compared to a conventional single-effect absorption chiller (section 5.3).

5.1. Model validation and determination of nominal operation of the SE-DL AC

Table 1 summarizes the input parameters of the simulation. The results obtained are the low and auxiliary mass flow rates and the internal temperatures of the main heat exchangers.

As stated in section 4, the internal and external streams are linked using the UA values of the main chiller's heat exchangers to simulate the entire SE-DL AC.

The single-effect double lift absorption chiller manufactured by the Hitachi Company was used to validate the developed model. A similar set of operating parameters (external temperatures) has been used to confirm the accuracy of the current model. The results of this analysis are listed in Table 2. It includes, Specifically, the data currently available from one manufacturer [36]. A comparison of the results of the model that was developed for this study.

This comparison was conducted using the cooling capacity and the temperatures of the hot, cold, and chilled water as reference values.

The developed model produces reliable results, with the maximum deviation being 3.7% for external temperatures and 0.6% for cooling capacity. According to a comparison of Hitachi technical specification data, these values are acceptable.

Based on the input parameters reported in Tables 1 and 2, the thermodynamic properties of the SE-DL AC (mass flow rate, temperature, enthalpy, and LiBr concentration) for each state point have been calculated and summarized in Table 3. The baseline condition or nominal point of the operation was set at 88.7 °C, 35 °C, and 12 °C for the inlet temperatures of the external streams at the high temperature generator, absorbers/condenser, and evaporator, respectively.

The heat transfer rates of the different heat exchangers (Q), solution circulation ratio (SCR), heat transfer characteristics (UA), and coefficient of performance (COP) for the nominal conditions are calculated and presented in Table 4 and Table 5 using the data reported in Table 3.

5.2. Parametric analysis

The developed simulation model was used to determine the effect of the driving heat temperature on the coefficient of performance, cooling capacity, internal temperatures, and temperature glide (the difference between the supply and return temperatures of the driving heat).

5.2.1. Effect of the driving heat temperature on COP and cooling capacity

Usually, the thermal energy source used to drive the SE-DL AC will be a relatively low-enthalpy renewable energy source or recovered waste heat. The temperature of this source is an important factor in determining the overall performance and efficiency of the refrigeration cycle.

Fig. 4 depicts the effect on the COP and cooling capacity produced by a driving heat temperature ranging between 65 °C and 100 °C while maintaining all other operating parameters fixed. According to this figure, it is disclosed that the system delivers maximum performance at heat source temperatures between 65 and 85 °C. This demonstrates that

Table 2

Validation of the developed model.

Input parameters	Reference Values [33]	Present work	Deviation (%)
Nominal cooling capacity	106.1 kW	106.7 kW	0.6
Hot source to HG, (T_{20})	88.7 °C	88.7 °C	–
Hot source out of HG (T_{23})	53.7 °C	52.36 °C	2.45
Cooling water to LA(T_{26}), AA (T_{30}), C(T_{24})	30.9 °C	30.9 °C	–
Cooling water out of LA (T_{27})		36.66 °C	0.16
Cooling water out of AA (T_{31})	36.6 °C	37.97 °C	3.7
Cooling water out of LA (T_{25})		36.7 °C	0.27
Chilled water to E (T_{28})	12 °C	12 °C	–
Chilled water out of E (T_{29})	7 °C	6.917 °C	1.19

the SE-DL AC system is capable of operating effectively with a variety of low-temperature heat sources. Moreover, the system COP increases up to a maximum of 0.787 at about 70 °C and later decreases as the temperature continues rising. This behavior is explained by the evolution of the solution circulation ratio, which decreases at temperatures higher than the optimum ones.

In the first SE-DL loop (1–2; 3–4; 5–6; 7–8; 9–10), both low and high solution circulation ratios affect the system, COP. At a driving temperature of 70 °C, the low and high SCR shows a maximum value of 118.1 and 769.6. The solution circulation ratios exhibit the same trend as the COP with the driving heat. In addition, the amount of strong solution decreases, resulting in a reduction in the COP and an improvement in the cooling load.

The cooling generation capability is also investigated in Fig. 4. This parameter shows a continuous increase from 59.26 kW to reach a maximum value of 120.5 kW. Except for the evaporator, all temperatures show a nearly linear increase as the heat source's generator temperature goes up. This was primarily due to the strong solution and refrigerant temperatures rising at the outlets of the generator as the driving heat source temperature increased, which caused an increase in the average temperatures of the condenser and absorbers (low and auxiliary). Given that the inlet cooling water is constant, the temperature difference of heat transfer between solution and water increases for the condenser and absorbers, which result in improving the heat transfer. Following this, the capacity of the evaporator is increased.

5.2.2. Effect of the driving heat temperature on internal cycle temperatures

The influence of the inlet temperature of the driving heat (T_{20}) on the SE-DL internal temperatures is important to be investigated in the present analysis. Figs. 5a and 5b present a plot of some inlet and outlet stream temperatures for the main heat exchangers, for a range of the driving temperature from 65 °C to 100 °C. It is observed that the internal cycle temperatures (Figs. 5a and 5b) increase with the inlet-driven temperature (T_{20}), except for the outlet of the low-pressure absorber (T_1), which moves similarly to the COP. Moreover, the outlet of the evaporator (T_{13}), declines from 7.83 °C to 3.53 °C. This behavior is the same as that already found in single-effect absorption chillers[44].

5.2.3. Effect of the internal solution heat exchangers efficiency

Fig. 6 illustrates the variation of the system COP in the cycle as a function of the efficiency of the internal solution heat exchangers. The SHX effectiveness is changed from zero (no SHX) to a value of 100%, which represents the ideal case. The purpose of a solution heat exchanger in a chiller circuit is to improve the refrigeration cycle COP by increasing the internal heat recovery. Table 1 shows the default SHX efficiency for heat exchangers whose values are not varied. According to Fig. 6, the heat exchanger that shows a higher impact on the performance is the auxiliary SHX, which raises the COP from 0.65 to 0.75. The impact of the other two SHXs is minimal.

Table 1

Input values for the first simulation step (internal cycle parameters).

Input parameters	Values
High generator temperature (T_{HG})	82 °C
Auxiliary generator temperature (T_{AG})	63 °C
Low generator temperature (T_{LG})	63 °C
Auxiliary absorber temperature (T_{AA})	35 °C
Low absorber temperature (T_{LA})	35 °C
Evaporator temperature (T_E)	4.5 °C
Condenser temperature (T_C)	35 °C
Solution pumps mass flow rate	0.26 kg/s
Low SHX effectiveness(e_{LSHX})	40%
High SHX effectiveness(e_{HSHX})	40%
Auxiliary SHX effectiveness (e_{ASHX})	65%

Table 3

Thermodynamic properties in each stream of the SE-DL AC cycle.

Reference Stream	$\dot{m}_i$ (kg/s)	P (kPa)	T (°C)	h (kJ/kg)	X_{LiBr} (%)	State
1	0.26	0.8426	35	89.76	55.82	Solution-saturated liquid
2	0.26	5.629	35	89.77	55.82	Solution-Compressed liquid
3	0.26	5.629	44.07	108	55.82	Solution -compressed liquid
4	0.26	5.629	56.96	134.2	55.82	Solution-compressed liquid
5	0.2357	5.629	82	206.4	61.57	Concentrated solution-saturated liquid
6	0.2357	5.629	66.83	177.5	61.57	Concentrated solution- subcooled liquid
7	0.2357	1.766	59.92	177.5	61.57	Concentrated solution and water vapor
8	0.2297	1.766	63	179.6	63.19	Concentrated solution-saturated liquid
9	0.2297	1.766	51.8	158.9	63.19	Concentrated solution-subcooled liquid
10	0.2297	0.8426	50.27	158.9	63.19	Concentrated solution and water vapor
11	0.04514	5.629	35	146.6	0	Water-saturated liquid
12	0.04514	0.8426	4.5	146.6	0	Vapor-liquid water
13	0.04514	0.8426	4.5	2509	0	Saturated water vapor
14	0.26	1.766	35	71.85	48.00	Solution-saturated liquid
15	0.26	5.629	35	71.86	48.00	Solution-Compressed liquid
16	0.26	5.629	50.95	107.6	48.00	Solution-Compressed liquid
17	0.2391	5.629	63	138.6	52.2	Concentrated solution-saturated liquid
18	0.2391	5.629	44.8	99.67	52.2	Concentrated solution- subcooled liquid
19	0.2391	1.766	41.41	99.67	52.2	Concentrated solution and water vapor
1a	0.02428		69.72	2630	0	Overheated water vapor
2a	0.02086		55.21	2605	0	Overheated water vapor
3a	0.006056		62.86	2612	0	Overheated water vapor
20	1.00		88.7	371.6		
21			70.21	294		
22			56	234.5		
23			52.36	219.2		
24	4.6		30.9	129.5		
25			36.70	153.8		
26	5.25		30.9	129.5		
27			36.66	153.6		
28	5.00		12	50.41		
29			6.917	29.08		
30	0.71		30.9	129.5		
31			37.97	159		

Table 4

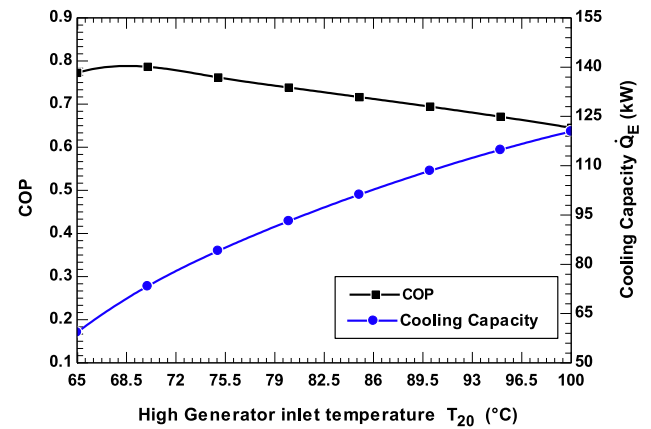
SE-DL AC performance parameters (heat capacity, COP, and SCR).

Components	Heat transfer rates
High Generator ($\dot{Q}_{HG}$)	77.61 kW
Low Generator ($\dot{Q}_{LG}$)	15.23 kW
Auxiliary Generator ($\dot{Q}_{AG}$)	59.51 kW
Low absorber ($\dot{Q}_{LA}$)	126.4 kW
Auxiliary absorber ($\dot{Q}_{AA}$)	20.97 kW
Condenser ($\dot{Q}_C$)	111.6 kW
Evaporator ($\dot{Q}_E$)	106.7 kW
High SHX ($\dot{Q}_{HSHX}$)	6.81 kW
Auxiliary SHX ($\dot{Q}_{ASHX}$)	9.303 kW
Low SHX ($\dot{Q}_{LSHX}$)	4.753 kW
Overall energetic performance indicators	
Coefficient of Performance (COP)	0.7
High SCR	10.71
Auxiliary SCR	12.46
Low SCR	8.571

Table 5

Calculated UA values for the SE-DL absorption chiller components.

Components	Heat transfer characteristics (kW /°C)
High-Pressure Generator (UA_{HG})	31
Low-Pressure Generator (UA_{LG})	2.441
Auxiliary Generator (UA_{AG})	5.946
Low-Pressure absorber (UA_{LA})	15.95
Auxiliary absorber (UA_{AA})	5.573
Condenser (UA_C)	40.89
Evaporator (UA_E)	23.76

**Fig. 4.** Effect of the inlet high-temperature generator on the coefficient of performance and cooling capacity of the SE-DL AC.

5.2.4. Effect of the driving heat temperature on the external streams

This subsection has investigated the effect of the driving heat temperature on the return temperatures of the external streams of the chillers to calculate the temperature glide, or difference, between the inlet and outlet of each external stream. These streams are the return temperature of the driving heat (T_{23}), the return temperature of the cooling water from the condenser and the two absorbers (T_{25} , T_{27} , and T_{31}) considering an inlet temperature of 35 °C, and the return temperature of the chilled water from the evaporator (T_{29}) of 12 °C.

The aim is to check the effect of the driving heat temperature on the mentioned outlet temperatures from each main component to learn about the possible interaction with other upstream and downstream

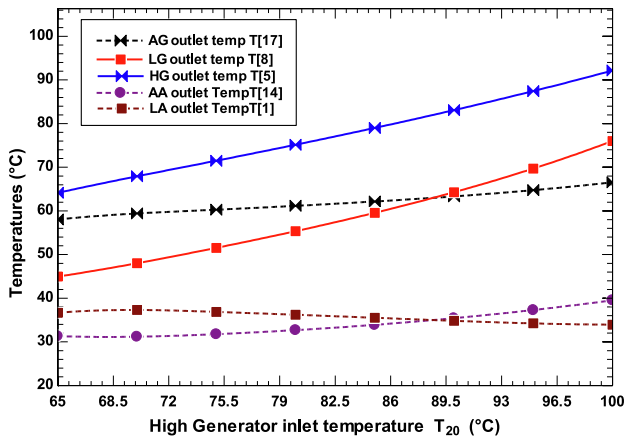


Fig. 5a. Internal temperatures of the working fluid at the outlet of the AG, LG, HG, AA, and LA as a function of the hot water inlet temperature for the SE-DL AC.

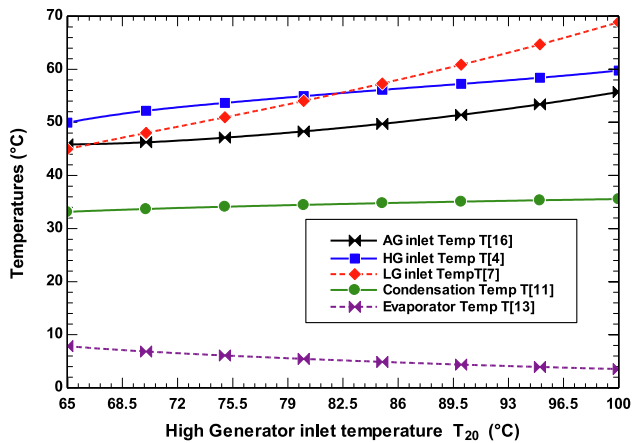


Fig. 5b. Temperatures of the working fluid at the inlet of the AG, HG, LG, and at the outlet of the condenser and evaporator as a function of the hot water inlet temperature for the SE-DL AC.

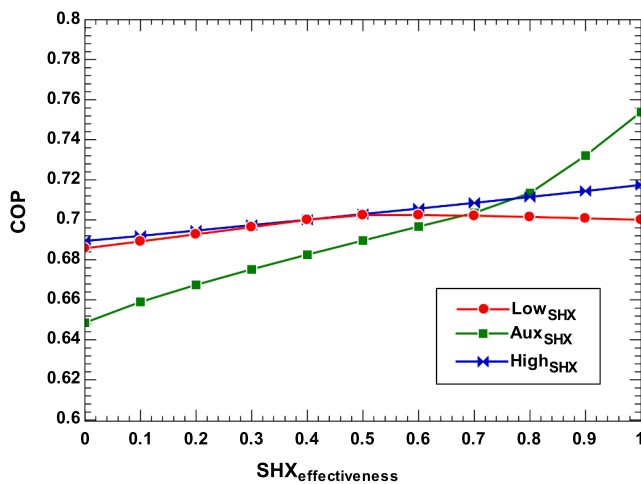


Fig. 6. Coefficient of performance as a function of the solution heat exchanger's effectiveness (ϵ_{SHX}) for the low (LSHX), high (HSHX), and auxiliary (ASHX) solution heat exchangers.

systems such as solar collectors, district heating networks, cooling towers, or chilled water users.

The results are shown in Fig. 7, where the inlet temperature of the hot source has been varied from 65 to 100 °C. As seen in this figure, all the return temperatures increase with the driving temperature except the evaporator outlet which, drops from 9.1 to 6.3 °C. The reason for this is the increase in the mean temperature in the different components. In the case of the evaporator, it is seen that the evaporator's cooling capacity increases. Thus, keeping the mass flow rate and chilled water inlet temperature constant causes the evaporator outlet temperature to decrease.

5.2.5. Effect of the diluted solution mass flow rates on COP and cooling capacity

The influence of the mass flow rates of the diluted solution leaving the low-pressure absorber (Str. 1) and the auxiliary absorber (Str. 14) on the SE-DL chiller performance is studied in this subsection. Based on the nominal condition (Table 1), the mass flow rate has been changed from 0.2 to 0.32 kg/s for both solutions. The results are given in Fig. 8 for the (a) low-pressure absorber and (b) auxiliary absorber.

The COP shows a modest but gradual increase for the low-pressure solution (Fig. 8a), whereas, for the mass flow rate of the auxiliary absorber (Fig. 8b), the COP drops from 0.72 to 0.67. Increasing the mass flow through the low-pressure absorber produces a benefit for the capacity of the evaporator. However, there is a slight decrease when the mass flow through the auxiliary absorber is increased because of the higher impact on the required energy to drive the system.

5.3. Comparison of the single-effect and single-effect double-lift absorption chillers

5.3.1. Comparison of nominal conditions

Following the previous parametric analysis of the SE-DL AC, the performance of this chiller will be compared with that of the conventional SE AC operating at the same conditions. The internal temperatures used for the analysis of both chillers are the generator (T_G) at 82 °C, the condenser and absorber ($T_A = T_C = 35$ °C), and the evaporator ($T_E = 4.5$ °C).

The efficiency of the solution heat exchanger has been set to 0.60 in the case of the SE AC. The external inlet temperatures for the generator, condenser, absorber, and evaporator are set to 88.7 °C, 30.9 °C, and 12 °C, respectively. The mass flow rate for the generator's inlet hot water is fixed at 1 kg/s. The equivalent values for the SE DL absorption chiller were already presented above in Tables 3 and 4. Table 6 shows the results of the thermodynamic performance of the single-effect absorption chiller.

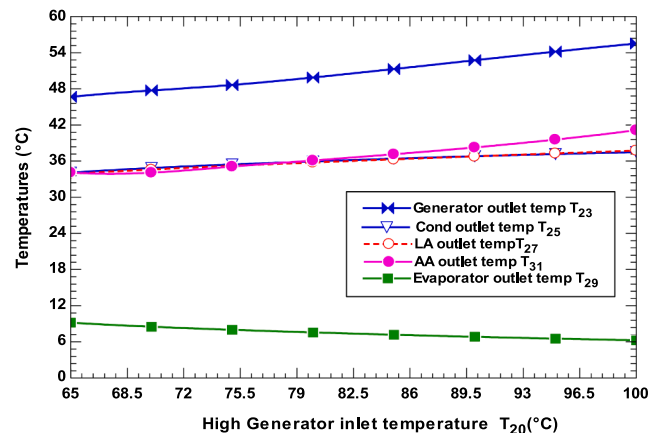


Fig. 7. Impact of the driving heat temperature on the return temperature of the SE-DL chiller's external streams.

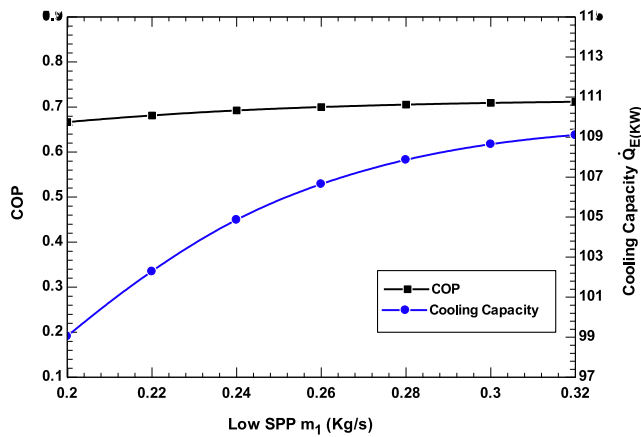


Fig. 8a. The effect of the low solution mass flow rate on the SE-DL absorption chiller performance.

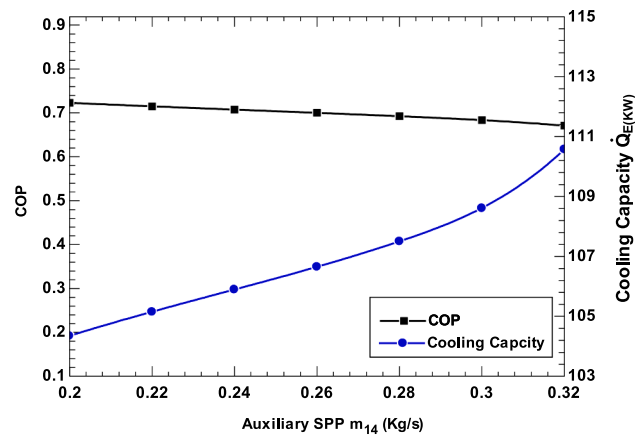


Fig. 8b. The effect of the auxiliary solution mass flow rate on the SE-DL absorption chiller performance.

Table 6

SE AC's main performance parameters.

Components	Heat transfer rates
Generator ($\dot{Q}_G$)	14.72 kW
Absorber ($\dot{Q}_A$)	14.50 kW
Condenser ($\dot{Q}_C$)	11.59 kW
Evaporator ($\dot{Q}_E$)	11.03 kW
SHX ($\dot{Q}_{SHX}$)	2.427 kW
Overall energetic performance indicators	
Coefficient of performance (COP)	0.749
Solution Circulation Ratio (SCR)	10.71
Generator temperatures	
Generator inlet temperature (T_{11})	88.7 °C
Generator outlet temperature (T_{12})	85.2 °C

Comparing the nominal conditions for the SE-DL AC given in Tables 3 and 4 with those of the SE AC given in Table 6, it is shown that the SE-DL AC outperforms the SE AC, primarily in terms of cooling production when both systems are operated under the same conditions. The SE AC produces only 11.03 kW compared to the 106.7 kW of cooling of the SE-DL AC. The reason is that at nominal conditions, the SE-DL AC extracts much more heat from the high-temperature energy source than the SE AC. The temperature glide of the driving heat, which is responsible for the higher cooling capacity, is almost 36 °C, while it is only 3.5 °C in the case of the single-effect chiller. Regarding the nominal COP given in Tables 4 and 6 (SE-DL and SE), it is only slightly higher for the

SE chiller, 0.75 versus 0.7.

5.3.2. Comparison of the effect of the driving heat temperature on the performance of the SE and SE-DL absorption chillers

A sensitivity analysis was performed to compare the effect of the driving heat temperature on the cooling capacity and temperature glide in SE AC and SE-DL AC configurations.

Fig. 9 shows the system performance in terms of cooling capacity as a function of the driving heat temperature in the two chiller configurations, (T_{11}) in the case of the SE and (T_{20}) for the SE DL.

In both configurations, the cooling capacity increases with the driving temperature, but the amount of cooling produced at a certain temperature is significantly larger in the case of the SE DL chiller. In both configurations, the higher the heat source temperature, the slower the rate of increase of the cooling capacity.

At the lowest driving temperature (65 °C), the SE AC produces only 4.0 kW for 59.3 kW in the case of the SE-DL AC, which goes up to 120.5 kW for the highest external inlet temperature of 100 °C. At the maximum considered driving temperature of 100 °C, the SE-DL AC produces around 8 times more cooling than the SE AC. When this temperature decreases, the difference in cooling capacity is even higher in favor of the SE-DL AC.

According to the thermodynamic data at nominal conditions reported in tables 6 and 4, the temperature glide of the generator in the SE AC is only 3.5 °C, whereas, for the same conditions, the SE-DL AC produces a much larger glide temperature of about 36 °C. In this case, the hot-driven stream enters at 88.7 °C and exits at 52.4 °C. Therefore, the glide temperature has improved approximately 10 times in comparison to the single-effect chiller.

A parametric analysis has been carried out (Fig. 10) to further evaluate the effect of varying the external generator inlet temperature on the glide temperatures for both chillers. As shown in this figure, by increasing the inlet temperatures of the heat source water (T_{11}) and (T_{20}) (depending on the chiller configuration), the outlet temperatures of the generators (T_{12} and T_{23}) show a rise in both types of chillers, but with a larger slope for the single effect. For instance, at the lowest driving temperature of 65 °C, the temperature glide is just a little bit higher than 1 °C, while for the SE-DL AC, it is about 18 °C. Furthermore, as the temperature rises, at the maximum driving temperature considered (100 °C), the SE DL can extract heat to cool down the heat input to 55.5 °C while the heat extracted by the conventional SE chiller reduces its temperature only to 95.5 °C. To summarize, a SE DL absorption chiller could be a very competitive way to use the driving heat effectively.

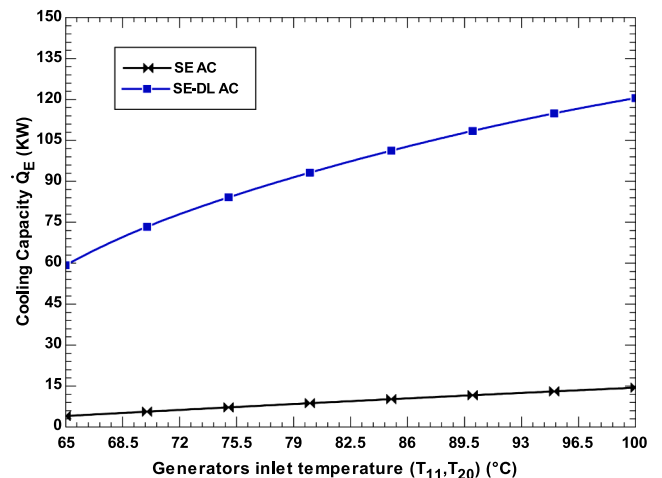


Fig. 9. Effect of the heat source inlet temperature on the cooling capacity for both chillers (the SE and the SE-DL AC).

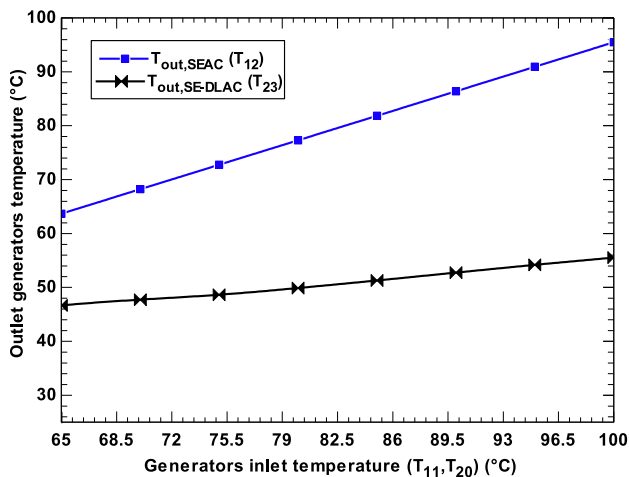


Fig. 10. Effect of the heat source inlet temperature on the return temperature in both SE and SE-DL AC for a given driving heat temperature (T_{11} and T_{20}).

6. Conclusions

In this study, a new configuration of an integrated absorption refrigeration system was introduced to utilize low-temperature thermal energy sources that could reach 65 °C. The new absorption cooling system is defined as an integration of the basic single effect chiller (SE AC) with the Double Lift (DL) cycle. The proposed H₂O–LiBr single effect double lift absorption chiller was simulated under steady-state conditions, considering a reference commercialized SE-DL AC. Comprehensive thermodynamic modeling was developed and validated accordingly. A sensitive evaluation of the overall performance of the system was carried out, followed by a realistic comparison with the single-effect absorption chiller based on the same operating conditions. The main outcomes of this study are summarized as follows:

- The single effect double lift absorption chiller is found to exhibit higher performance in a broad range of hot water thermal sources. This system exhibits a Coefficient of Performance (COP) up to 0.75 at a hot water driving temperature of 65 °C and a driving heat mass flow rate of 1 kg/s. Furthermore, the cooling capacity is found to be 59 kW at the same operating conditions.
- The SE-DL AC outperforms the conventional SE AC as the cooling capacity increased from 59.26 kW to 120.5 kW for a driving temperature going from 65 °C to 100 °C whereas, it is only 4 kW to 14 kW for the SE AC case at the same range of driving temperatures. The COP in both cases showed a quite similar tendency. Indeed, the coefficient of performance demonstrated a stable efficiency between 65 and 85 °C of a wide range of thermal driving temperatures with a value of up to 0.75. Thus, the single-effect double-lift absorption chiller can be operated with low-temperature heat sources by utilizing low-grade heat energy sources (solar energy) and thermal collectors (e.g., a flat plate collector).
- The optimum temperature should be a trade-off between the COP and the amount of cooling produced. When it comes to cooling production, its optimum value (120.5 kW) is reached at a driving hot water temperature of 100 °C. Whereas the optimal COP (0.78) is reached at a temperature of 70 °C. However, it must be taken into consideration that this temperature will be constrained by the kind of heat source (solar energy, recovered waste heat from a prime mover, etc.).
- One of the most important benefits of the preference for SE-DL AC over SE AC is that this novel configuration performs better in terms of heat recovery from low-grade heat sources and low-temperature thermal solar collectors. Indeed, unlike the single-effect, the glide temperature of the single-effect double-lift AC is ten times higher

(36 °C instead of 3.5 °C), working under the same operating conditions (thermal source temperature of 88.7 °C and fluid mass flow rate of 1 kg/s). The SE-DL could be used for a higher variety of applications of its high-temperature differences between supply and return pipes.

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.

Data availability

Data will be made available on request.

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