Analysis of Hybrid Ejector Absorption Cooling System

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In this paper, a hybrid ejector single-effect lithium-bromide water cycle is theoretically investigated. The system is a conventional single-effect cycle activated by an external steam-ejector loop. A mathematical model of the whole system is developed. Simulations are carried out to study the effect of the major parameters of the hybrid cycle on its performances and in comparison with the conventional cycle. The ejector performance is also investigated. Results show that the entrainment ratio rises with steam pressure and condenser temperature, while it decreases with increasing generator temperature. The effect of the evaporator temperature on ejector performance is negligible. It is shown also that the hybrid cycle exhibits better performances than the corresponding basic cycle. However, the performance improvement is limited to a specific range of the operating parameters. Outside this range, the hybrid system behaves similar to a conventional cycle. Inside this range, the COP increases, reaches a maximum, and then decreases and rejoins the behavior of the basic cycle. The maximum COP, which can be as large as that of a conventional double-effect cycle, about 1, is obtained at lower temperatures than in the case of single-effect cycles.

1. Introduction

Cooling and air conditioning are essential for small scale and large industrial process applications. While systems applying the vapor-compression technique use environmental harmful refrigerants (FCC, FCHC, etc.), absorption technique for production of cold is based on environment friendly working fluids, namely, aqueous lithium bromide solutions with water as refrigerant or water-ammonia mixtures with ammonia as refrigerant. This technique however suffers from low performances. That is the reason why new hybrid and combined configurations are proposed, implying the integration of new components, particularly ejectors, in order to enhance the performances.

Various configurations incorporating ejectors were studied. Exhaustive review of the literature on this subject can be found in Besagni et al. [1, 2]. Elaborated CFD-models of ejectors developed to evaluate the ejector performances in both on-design and off-design conditions have been also published [3]. Combined cycles were investigated with ejector set at the absorber inlet [4–9]. COP of such cycles are reported to be higher by about 2–4% than that of conventional cycles. Principally, investigations indicate that COP of the combined configuration are greater or equal to that of single-effect cycles, but reached at lower generator temperatures.

Other configurations are discussed where the ejector is located at the condenser inlet of single-effect systems [10–14]. Theoretical investigations confirm the improvement of the performances in comparison with basic single-effect cycles. Experimental studies [15] show that this combined cycle is 30-60% more performant than conventional absorption cycles and almost reaches the COP of double-effect systems. Besides modifying configurations, adding a flash tank between ejector and evaporator was also proposed [16, 17].

Ejector improved double-effect absorption system was also investigated [18–20]. The COP of the proposed refrigeration scheme was found to increase with the temperature of the heat source until this temperature reaches 150°C. Beyond that value, the new cycle worked as a conventional double-effect cycle. Another configuration was studied with an ejector coupled to vapor generator [21–23]. This procedure is intended to enhance the concentration process by compressing the vapor produced from the lithium bromide solution in order to reheat the solution from which it came. Results showed that COP of the new cycle increases especially with the heat source temperature.
In this paper, an ejector-activated single-effect LiBr-water cycle is proposed and theoretically investigated. The objective is to assess the feasibility and limits of performance of this new cycle scheme. If the COP of the proposed system could reach that of a conventional double-effect cycle, this would mean obtaining high performance by avoiding the configuration complexity of double-effect cycles. We investigate the evolution of the COP of the hybrid cycle with the steam generator temperature and the main factors of the cooling machine, i.e., desorber, condenser, and evaporator temperature. The behavior of the entrainment ratio as ejector performance criterion is also investigated for various primary and secondary flow pressure and backpressure.

2. System Description

Figures 1(a) and 1(b) are schematics of a conventional single-effect absorption cycle and an ejector-enhanced single-effect absorption system. A conventional single-effect absorption chiller (Figure 1(a)) is composed of evaporator, absorber, condenser, generator, solution expansion-valve, pump, solution heat exchanger, and refrigerant expansion-valve. In a hybrid system (Figure 1(b)) a steam-generator-ejector loop is coupled to the conventional single-effect installation via the machine generator. This extra circuit is constituted of an ejector, a steam generator, a water pump, and an expansion valve.
The ejector loop is intended to improve the cycle performance by enhancing the concentration process in the machine generator. A high-pressure flow (18) coming from the external steam generator enters the primary nozzle of the ejector where its pressure drops while it is accelerated. At the nozzle exit section (i) (Figure 2) its velocity becomes supersonic and high enough to entrain a secondary flow (19) generated in the desorber. The two streams mix in the mixing chamber and the resulting gas, after undergoing a shockwave that reduces its velocity to subsonic, is compressed in the diffuser forming the last segment of the ejector. The exiting vapor (12) condenses in the coil placed inside the solution generator, liberating thus condensation heat used to concentrate the saline solution by desorbing vapor from the water-rich solution (3) entering the generator. Part of the condensate flows, after appropriate pressure reduction, to the condenser, and the rest is pumped back to the steam generator.

3. Chiller Model

Basing on mass and energy balances written for every machine element a mathematical model of the installation is set up. For the numerical simulations, a computer code of the machine model is realized using the software Engineering Equations Solver, EES [24].

The model is elaborated under the following assumptions:

(i) Steady state conditions

(ii) Negligible heat losses to the surroundings at generator, condenser, absorber, and evaporator

(iii) Negligible pressure losses in pipes and components

(iv) Saturated refrigerant exiting condenser and evaporator

(v) Isenthalpic flow in solution and refrigerant valves

(vi) Phase equilibrium between solution entering refrigerant generator and vapor leaving

(vii) Constant solution flow-rate leaving the absorber, specifically 2 kg/s

(viii) Heat exchanger effectiveness, $\varepsilon_{HX} = 80\%$

In the following major elements of the model are presented.

3.1. Ejector Loop. This loop includes steam generator, ejector, heating coil placed in solution generator, expansion valve, and water pump.

(i) Steam Generator

The mass and energy balances on steam generator write, respectively,

$$\dot{m}_{17} = \dot{m}_{18} \quad \text{(1)}$$

$$Q_{SG} = \dot{m}_{17} (h_{18} - h_{17}) \quad \text{(2)}$$

The properties of exiting saturated vapor (18) are:

$$P_{SG} = P_{18} = P_{W-sat} \left( T_{18} \right) \quad \text{(3)}$$

$$h_{18} = h_{W-sat} \left( T_{18}, X_{18} = 1 \right) \quad \text{(4)}$$

Further, $P_{17} = P_{18}$.

Properties with index W for water refer to pure water properties as given in steam tables.

(ii) Ejector

The ejector performance depends on the backpressure $P_{bp}$—the pressure of the exiting (supposed saturated) steam flowing in the heating coil—, the primary pressure, $P_{18}$, and the secondary pressure, $P_{19}$. The relations between the different pressures around the ejector are

$$P_{bp} = P_{13} = P_{W-sat} \left( T_{13} \right) \quad \text{(5)}$$

$$P_{19} = P_{1} = P_{b} \quad \text{(6)}$$

The mass balance for the ejector writes

$$\dot{m}_{12} = \dot{m}_{18} + \dot{m}_{19} = (1 + \omega) \dot{m}_{18} \quad \text{(7)}$$
where $\omega$ stands for the entrainment ratio

$$\omega = \frac{\dot{m}_{19}}{\dot{m}_{18}}$$  \hspace{1cm} (8)

The enthalpy of exiting flow (12) can be deduced from the energy balance

$$h_{12} = \frac{h_{18} + \omega h_{19}}{1 + \omega}$$  \hspace{1cm} (9)

(iii) Heating Coil

Assuming a difference of 5 K between the temperatures of the heat source and that of the refrigerant generator solution, we get

$$T_{13} = T_{12} = T_{G} + 5 = T_{4} + 5$$  \hspace{1cm} (10)

$$h_{13} = h_{W-sat}(T_{13}, X_{13} = 0)$$  \hspace{1cm} (11)

The mass balance writes

$$\dot{m}_{12} = \dot{m}_{13}$$  \hspace{1cm} (12)

(iv) Water Pump

We suppose approximately isothermal pumping

$$T_{17} = T_{16} = T_{13} = T_{14}$$  \hspace{1cm} (13)

The mass and energy balances write, successively,

$$\dot{m}_{17} = \dot{m}_{16}$$  \hspace{1cm} (14)

$$h_{17} = h_{16} + \frac{(P_{17} - P_{16})}{\rho_{17}}$$  \hspace{1cm} (15)

where the term $[(P_{17} - P_{16})/\rho_{17}]$ in the last equation represents the specific pump work (kJ/kg), with $(\rho_{17} = \rho_{W}(T_{17}, P_{17})).$

(v) Expansion Valve

The expansion is isenthalpic, i.e.,

$$h_{14} = h_{15} = h_{13}$$  \hspace{1cm} (16)

$$\dot{m}_{14} = \dot{m}_{15}$$  \hspace{1cm} (17)

3.2. Liquid Solution Loop. The absorber-generator loop comprises absorber, solution valve, solution pump, solution heat exchanger, and refrigerant generator.

(i) Refrigerant Generator

With $\xi$ denoting the lithium bromide concentration in the liquid solution, the mass balances for this machine element write

$$\dot{m}_3 = \dot{m}_{17} + \dot{m}_4$$  \hspace{1cm} (18)

Solving for $\dot{m}_7$ yields

$$m_7 = \dot{m}_4 \frac{\xi_4 - \xi_3}{\xi_3}$$  \hspace{1cm} (20)

For the energy balance we get

$$\dot{m}_1 h_4 + \dot{m}_7 h_7 = \dot{m}_3 h_3 + \dot{m}_{12} (h_{12} - h_{13})$$  \hspace{1cm} (21)

from which we deduce

$$\dot{m}_4 = \frac{\dot{m}_{12} (h_{12} - h_{13}) - (\dot{m}_1 h_7 - \dot{m}_3 h_3)}{h_4}$$  \hspace{1cm} (22)

The properties of water-weak solution (4) exiting the generator are determined as follows:

$$P_G = P_{CD} = P_4$$  \hspace{1cm} (23)

$$\xi_4 = \xi_{SOL-sat}(T_G, P_G)$$  \hspace{1cm} (24)

$$h_4 = h_{SOL-sat}(T_G, \xi_4)$$  \hspace{1cm} (25)

For known solution temperature and pressure, the saturation concentration can be deduced from solution property relations. Following equations fix the properties of exiting vapor at (7)

$$P_7 = P_G$$  \hspace{1cm} (26)

$$T_7 = T_{SOL-sat}(P_7, \xi_3)$$  \hspace{1cm} (27)

$$h_7 = h_W(T_7, P_7)$$  \hspace{1cm} (28)

(ii) Solution Heat Exchanger

Besides the trivial relations

$$P_5 = P_4$$  \hspace{1cm} (29)

$$P_3 = P_2$$

Mass and energy balance equations write

$$\xi_5 = \xi_4$$  \hspace{1cm} (30)

$$\xi_6 = \xi_2$$

$$\dot{m}_3 = \dot{m}_2$$

$$\dot{m}_5 = \dot{m}_4$$

$$h_3 = h_2 + \frac{\dot{m}_1}{\dot{m}_2} (h_4 - h_5)$$  \hspace{1cm} (31)

Considering the heat exchanger effectiveness, $\varepsilon_{Hx}$, we have the following further relations:

$$T_5 = \varepsilon_{Hx} T_2 + (1 - \varepsilon_{Hx}) T_4$$  \hspace{1cm} (32)

$$h_5 = h_{SOL-sat}(T_5, \xi_5)$$  \hspace{1cm} (33)
\[ h_3 = h_{\text{SOL-sat}}(\xi_3, T_3) \]  
(34)

(iii) Solution Valve

Through the solution valve, the pressure is reduced from condenser to evaporator pressure. In addition to the usual mass balance-equations \((\xi_6 = \xi_3)\) and \((\dot{m}_6 = \dot{m}_3)\) we have the relations

\[ h_6 = h_5 \]  
(35)
\[ T_6 = T_{\text{SOL-sat}}(\xi_4, h_6) \]  
(36)

(iv) Solution Pump

Again, we have the trivial mass balances \(\dot{m}_2 = \dot{m}_1\) and \(\xi_5 = \xi_1\). As for the water-pump, the pumping process is assumed isothermal \(T_2 = T_1\). During pumping, the enthalpy of the refrigerant-rich solution from absorber is increased by \(\frac{P_2 - P_1}{\rho_2}\), with \(\rho_2 = \rho_{\text{SOL-sat}}(\xi_2, h_2)\),

\[ h_2 = h_1 + \frac{P_2 - P_1}{\rho_2} \]  
(37)

(v) Absorber

Per definition, \((T_{AB} = T_1)\) and \((P_{AB} = P_1)\). For the liquid solution \((1)\) exiting the absorber we get in addition to the mass and energy balance equations

\[ \dot{m}_1 = \dot{m}_6 + \dot{m}_{11} \]  
(38)
\[ \dot{m}_1 \xi_1 = \dot{m}_6 \xi_6 \]  
\[ Q_{AB} = (\dot{m}_{11} h_{11} + \dot{m}_6 h_6) - \dot{m}_1 h_1 \]  
(39)

The property relations are

\[ \xi_1 = \xi_{\text{SOL-sat}}(P_1, T_1) \]  
(40)
\[ h_1 = h_{\text{SOL-sat}}(T_1, P_1) \]  
(41)

3.3. Refrigerant Loop

(i) Condenser

Streams \((8)\) and \((15)\) flow in the condenser where they condensate. Condensing temperature and pressure are \(T_{CD} = T_9\) and \(P_{CD} = P_9\), respectively. The mass and energy balances around the condenser write

\[ \dot{m}_9 = \dot{m}_8 + \dot{m}_{15} = \dot{m}_8 + \dot{m}_{19} = \dot{m}_7 \]  
(42)
\[ \dot{Q}_{CD} = m_9 (h_8 - h_9) \]  
(43)

Knowing the condensation temperature \(T_9\), pressure \(P_9\), as well as the enthalpy of exiting liquid can be deduced as

\[ P_9 = P_{\text{W-sat}} (T_9) \]  
(44)

\[ h_9 = h_{\text{W-sat}} (T_9, X_9 = 0) \]  
(45)

(ii) Refrigerant Expansion Valve

Liquid refrigerant \((9)\) undergoes a pressure reduction before it enters the evaporator. Evaporation temperature and pressure are \(T_{EV} = T_{11} = T_{10}\) and \(P_{EV} = P_{10}\), respectively.

For fixed evaporator temperature \(T_{EV}\) and assuming saturated vapor at exit, we can write \(P_{EV} = P_{W-sat}(T_{EV})\).

The mass and energy balances for the valve write

\[ h_{10} = h_9 \]  
(46)
\[ m_{10} = m_9 \]  
(47)

(iii) Evaporator

The evaporator equations are

\[ \dot{Q}_{EV} = m_{11} (h_{11} - h_{10}) \]  
(48)
\[ \dot{m}_{11} = \dot{m}_{10} \]  
\[ h_{11} = h_{W-sat} (T_{EV}, X_{11} = 1) \]  
(49)

The COP_{hybrid} of the proposed absorption system, when neglecting all pump work, can be expressed as

\[ \text{COP}_{hybrid} = \frac{\dot{Q}_{EV}}{\dot{Q}_{SG}} \]  
(50)

4. Ejector 1D Model and Analysis

Because the performances of the proposed cycle depend largely on ejector performances, a reliable ejector model is necessary for the cycle simulations. In this paper, the ejector is modelled basing on the 1D analyses in [25, 26].

In this type of model, it is assumed that

(i) primary fluid expands isentropically in nozzle, and the exiting flow compresses isentropically in diffuser

(ii) inlet velocities of primary and entrained fluids are insignificant

(iii) velocity of the compressed mixture at ejector outlet is neglected

(iv) mixing of primary and secondary fluids in the suction chamber occurs at constant pressure

(v) flow in ejector is adiabatic

Isentropic efficiencies are introduced in the model to account for eventual irreversibility in the expansion process in primary nozzle, \((\eta_n)\), in the mixing process of primary and secondary flow in the mixing chamber, \((\eta_m)\), and finally in the compression process in the diffuser, \((\eta_d)\). For the numerical simulations we set \(\eta_n = 0.95, \eta_m = 0.95, \text{ and } \eta_d = 1\).
4.1. Primary Nozzle. In the nozzle, the primary vapor (18) expands and accelerates. The Mach number of the fluid at nozzle outlet plane, deduced from energy balance, writes

\[ M_{18i} = \sqrt{\frac{2\eta_n}{y-1} - 1} \left( \frac{P_{19}}{P_i} \right)^{(y-1)/y} \]  

(51)

In this equation, \( \eta_n \) is the isentropic nozzle efficiency, defined as the ratio between actual enthalpy change and enthalpy change undergone during an isentropic process.

The expression for \((A_j/A_i)\) the area ratio at nozzle throat and outlet is

\[ A_i = \sqrt{1 - \frac{2}{y+1} \left( \frac{P_{19}}{P_i} \right)^{(y-1)/y} - 1} \left( \frac{M_{18i}}{M_i} \right)^{(y-1)/(y+1)} \]  

(52)

4.2. Suction Chamber. Because \( P_i < P_{19} \), the secondary fluid (19) expands in the suction chamber and is entrained by the high-speed primary flow. The Mach number of the entrained fluid at nozzle exit plane writes

\[ M_{19i} = \sqrt{\frac{2}{y-1} - 1} \left( \frac{P_{19}}{P_i} \right)^{(y-1)/y} \]  

(53)

4.3. Mixing Chamber. Here, primary and secondary fluids are mixed. The properties of the resulting stream at section (j) are deduced from continuity, momentum, and energy equations and expressed as function of the critical Mach number of the original streams, \( M_{j}^* \),

\[ M_j^* = \eta_m \frac{M_{18i}^* + \omega M_{19i}^* \sqrt{\tau}}{\sqrt{(1 + \omega \tau)(1 + \omega)}} \]  

(54)

As can be noticed, the mixture \( M_j^* \) is written as a combination of critical Mach numbers of the original streams, \( M_{18i}^* \) and \( M_{19i}^* \). \( \tau \) in this equation stands for the temperature ratio of incoming streams (19) and (18):

\[ \tau = \frac{T_{19}}{T_{18}} \]  

(55)

The relationship between \( M \) and \( M^* \) at any point of the ejector is given by the equation

\[ M = \sqrt{\frac{2M_j^*}{(y+1) - (y-1)M_j^*}} \]  

(56)

By the end of the mixing chamber, a shock wave occurs at section (k). The flow changes from supersonic to subsonic conditions, producing simultaneously a sudden rise in the static pressure. The relation between the Mach number upstream and downstream of the shock wave is given by

\[ M_k = \sqrt{\frac{2/(y-1) + M_j^2}{(2y/(y-1))M_j^2 - 1}} \]  

(57)

The corresponding pressure increase writes

\[ \frac{P_k}{P_j} = \frac{M_k^2}{M_j^2} \frac{1 + (1/2)M_j^2(y-1)}{1 + (1/2)M_k^2(y-1)} \]  

(58)

4.4. Diffuser. The expression of the pressure lift in the diffuser is

\[ \frac{P_{12}}{P_k} = \left( 1 + \frac{1}{2} \eta_d M_j^2(y-1) \right)^{(y-1)/y} \]  

(59)

The ejector area ratio \((A_j/A_c)\), i.e., the ratio of nozzle throat area and diffuser constant area section, writes

\[ \frac{A_j}{A_c} = \frac{P_{12}}{P_{18}} \left( \frac{P_k}{P_{12}} \right)^{1/y} \cdot \sqrt{\frac{1 - \left( \frac{P_{12}}{P_k} \right)^{(y-1)/y}}{(1 + \omega \tau)(1 + \omega) \left( \frac{y+1}{y-1} \right)^{(y-1)/y}}} \]  

(60)

5. Results and Discussion

The EES machine model program is run to thermodynamically analyze the proposed hybrid single-effect absorption refrigeration system. The thermophysical properties of LiBr-H2O solution are estimated using the software property data and model-bank.

The simulations are performed for the conditions given in Table 1. Evaporator temperature \( T_C \) is set to 4°C, condenser temperature \( T_{CD} \) to 37°C, and absorber temperature \( T_{AB} \) to \( T_{CD} - 2 \). Condenser and absorber are both supposed water-cooled. The cooling medium is processed thereafter in a cooling tower.

5.1. Program and Machine Model Validation. The simulation program is first validated by comparing our simulation results for a conventional single-effect cycle with the results published by Somers (2009) [27] for the same operating conditions: evaporator temperature 1.5°C; condenser and absorber temperatures at 40.2°C and 32.7°C, respectively; effectiveness of solution heat exchanger, 0.5; mass flow rate of solution leaving absorber, 1 kg/s. As can be noticed when comparing the results in columns 2 and 3 of Table 2, both sets of data are in very good agreement. Therefore, we can now proceed to the simulations of the proposed hybrid cycle with some confidence.

The next step was to validate the adequacy of the conventional model by comparing the predicted, calculated performance with experimental data reported in [28] concerning a large capacity LiBr-chiller. Two different sets of operating conditions are considered. As can be observed when studying columns 4 to 7 in Table 2, the calculated data is for both tests very close to the reported data in [28]. Finally, the proposed ejector configuration model is validated using
Table 1: Simulation input data.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Variation range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam generator pressure, (P_{SG}) bar</td>
<td>15</td>
<td>10–15</td>
</tr>
<tr>
<td>Generator temperature, (T_{GD}) °C</td>
<td>80</td>
<td>65–90</td>
</tr>
<tr>
<td>Evaporator temperature, (T_{EV}) °C</td>
<td>4</td>
<td>2–12</td>
</tr>
<tr>
<td>Condensation temperature, (T_{CD}) °C</td>
<td>37</td>
<td>28–37</td>
</tr>
<tr>
<td>Absorber temperature, (T_{AB}) °C</td>
<td>(T_{CD} - 2)</td>
<td></td>
</tr>
</tbody>
</table>

Table 2: Program and machine model validation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Data 1 [27]</th>
<th>Present work</th>
<th>Data 2 [28]</th>
<th>Present work</th>
<th>Data 3 [28]</th>
<th>Present work</th>
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<tbody>
<tr>
<td>(T_{GD}) °C</td>
<td>90</td>
<td>101.6</td>
<td>83</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(T_{EV}) °C</td>
<td>1.3</td>
<td>5</td>
<td>12.3</td>
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<td></td>
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<tr>
<td>(T_{CD}) °C</td>
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<td>43</td>
<td>42</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(T_{AB}) °C</td>
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<td>38.3</td>
<td>39</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>(\dot{Q}_{GD}), kW</td>
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<td>15.00</td>
<td>1150</td>
<td>1143</td>
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<td>1105</td>
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<tr>
<td>(\dot{Q}_{EV}), kW</td>
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<td>10.80</td>
<td>843</td>
<td>842.5</td>
<td>842.7</td>
<td>842.5</td>
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<tr>
<td>COP</td>
<td>0.73</td>
<td>0.72</td>
<td>0.73</td>
<td>0.74</td>
<td>0.76</td>
<td>0.76</td>
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<tr>
<td>(\xi_4), %</td>
<td>62.6</td>
<td>62</td>
<td>65.5</td>
<td>65.8</td>
<td>57.2</td>
<td>58.5</td>
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<tr>
<td>(\xi_3), %</td>
<td>57.4</td>
<td>56.3</td>
<td>56.5</td>
<td>57.4</td>
<td>53.1</td>
<td>53.4</td>
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</tbody>
</table>

Figure 3: Hybrid cycle model validation basing on experimental data of ref. [29].

5.2. Comparison of Hybrid and Conventional Cycle Performances. For purpose of illustration, the chiller cycle is represented in Figure 4 in the usual Oldham-diagram and in the water \((P – h) – diagram in Figure 5.

We now proceed to the comparison of the performances of the proposed cycle and the conventional basic cycle (without ejector) for varying machine generator called also desorber-temperature \((T_{GD})\), condenser temperature \((T_{CD})\), and evaporator temperature \((T_{EV})\).

As depicted in Figures 6–8, the coefficient of performance of the hybrid cycle is in all cases larger than the COP of the conventional cycle for the same operating conditions.

However, this performance enhancement is restricted to a specific interval of machine-generator temperature, as Figure 6 clearly shows. Outside this temperature interval, both cycles are practically equivalent. Figure 6 shows also that with growing desorber temperature \(T_{GD}\) the COP curve of the hybrid cycle first exceeds that of the basic cycle, reaches a maximum than decreases gradually, and resumes the curve of the conventional cycle COP. It is also worth noticing that the COP of the hybrid cycle under optimal conditions approaches the COP of double-effect conventional cycle.

Figures 7 and 8 depict the evolution of the COP of both cycles with condenser and evaporator temperature, respectively, for \((P_{18} = 15\) bar; \(T_{18} = 200\) °C). Note that \(T_{18}\) is the steam generator temperature, not the chiller desorber temperature, the abscissa in Figures 6–14. Both COP are expectably decreasing in the first case and increasing in the second. COP of hybrid is always larger than COP of conventional cycle because the constant maintained desorber-temperature is set to 80° C, i.e., in the favourable interval 70° C–90° C. In conclusion of this section we notice that an ejector incorporated in the hybrid cycle (i) improves the cycle performances and (ii) the maximal COP is reached at lower machine generator temperature.

5.3. Performances of the Hybrid Cycle. The effect observed previously in Figure 6 (enhancement of the cycle performance due to the incorporation of ejector in the driving
Figure 4: Chiller cycle representation in the Oldham-diagram \( (T_{SG} \approx 200^\circ C; T_G = 85^\circ C; \ T_{EV} = 4^\circ C; \ T_{CD} = 37^\circ C) \).

Figure 5: Chiller cycle representation in the water \((P - h)\)-diagram \( (T_{SG} \approx 200^\circ C; T_G = 85^\circ C; \ T_{EV} = 4^\circ C; \ T_{CD} = 37^\circ C) \).

Figure 6: \( \text{COP} \) of hybrid and conventional cycle vs. machine generator temperature, \( T_{cl}(P_{18} = 15 \text{ bar}; \ T_{18} \approx 200^\circ C) \).

Figure 7: \( \text{COP} \) of hybrid and conventional cycle vs. condenser temperature, \( T_{CD} \).

The compartment of the machine (and consequently temperature) depends on the primary flow pressure \( P_{SG} = P_{18} \) used to activate the ejector. Increasing this pressure expands this effect in magnitude and amplitude as Figure 9 shows: the higher the steam-generator pressure (and consequently temperature), the larger the machine-generator temperature range where the cycle performance is improved, and the higher the maximum \( \text{COP} \) that could
be reached inside this interval. On the opposite, when the steam generator pressure $P_{SG}$ is decreased to 10 bar, practically no improvement more of the cycle performance is observed under the prevailing conditions.

Figure 10 depicts the evolution of $COP_{hybrid}$ with $T_G$ by varying the condenser temperature, $T_{CD}$. It is observed that the typical pink curve of Figure 6 is expectedly shifted to lower machine-generator temperatures (with lower condenser temperature, less high desorber temperature is needed to activate the cycle) with however concomitantly increased maximal $COP$ and enlarged favorable temperature interval, where the cycle performance is improved.

Similar effects are observed in Figure 11 depicting the evolution of $COP_{hybrid}$ with $T_G$ by varying evaporator temperature. Here, the typical $COP$—improved portion of the curve is shifted to lower $T_G$—values when the evaporator temperature is increased, a thermodynamically more favourable situation. The $COP$ of the hybrid cycle rises from 0.85 to 1.12 for generator temperature decreasing from 78°C to 67°C when the evaporator temperature increases from 4°C to 12°C.

5.4. Ejector Performance. The ejector model presented in Section 4 will help us interpret the represented simulation results in Figures 7–11 and assess the beneficial effect—and
limits—of integration of an external ejector loop to a conventional absorption cycle. We first investigate the relation between the performance of the incorporated ejector, i.e., its entrainment ratio $\omega$, and significant absorption machine parameters, namely, desorber temperature $T_G$, evaporator temperature $T_{EV}$, and condenser temperature $T_{CD}$. Figure 12 depicts the evolution of $\omega$ with $T_G$. For a given primary pressure $P_{SG}$, the entrainment ratio decreases monotonously with $T_G$ and finally vanishes for a maximal value of the desorber temperature; i.e., secondary flow (19) is no more entrained inside the ejector. The ejector is then off-design and its geometry should be changed. Same behaviour of $\omega$ vs. $T_G$ is noticed if the steam pressure $P_{SG}$ is increased. However, in this case the curve is shifted upwards to larger values of $\omega$; i.e., more secondary vapour is sucked in the ejector for a given temperature $T_G$, and the limit value of $T_G$ where the entrainment ration vanishes is pushed farther away.

Similar behaviour is observed in Figure 13, when for fixed primary pressure the condenser temperature (secondary pressure) is varied. If the condensation temperature is reduced (or alternatively enlarged), the entrainment ratio is also decreased (or increased, respectively). However, the curves $\omega$ vs. $T_G$ for the various condenser temperatures all converge to the same point on the temperature-axis where $\omega$ vanishes. This temperature depends solely on the primary steam pressure.

Finally, Figure 14 shows that the evaporator temperature has practically no effect on the ejector performance by fixed $P_{SG}$ and $T_{CD}$, as all $\omega$ vs. $T_G$ for the various tested $T_{EV}$ are superimposed.

According to the ejector model presented in Section 3 of the present paper, the entrainment ratio depends on six independent parameters: nozzle area ratio, primary flow and secondary flow properties, and backpressure, i.e., $\omega = f(A_i/A_t, P_{18}, T_{18}, P_{19}, T_{19}, P_{12})$. The results presented in the foregoing sections are obtained for simulations with the specific conditions: (i) constant ejector nozzle ratio set to $(A_i/A_t) = 17.3$; (ii) saturated ejector-driving steam; i.e., $T_{18}$ and $P_{18}$ are then no more both independent; (iii) pressure of secondary flow $P_{19}$ equals condenser pressure, an independent parameter; (iv) temperature $T_{19}$ of flow $P_{19}$ is not an independent variable. It depends on the processes taking place in rest of the absorption chiller and in particular on the backpressure, $P_{12}$, which is considered here as an independent parameter.
The COP curves represented in Figures 6–9 depict its evolution when the effects of both the ejector and the single-effect absorption chiller are combined. By increasing the backpressure and, consequently, the desorber temperature, the COP tends first to increase as it does for a conventional cycle. The entrainment ratio however is decreasing. The resulting outcome is then first an increase of COP and then a decrease after passing a maximum where opposed effects cancel each other.

6. Conclusion

A hybrid single-effect cycle with water lithium-bromide as working fluid and activated by a steam-ejector loop is proposed and theoretically investigated. Mathematical models of the hybrid cycle and the ejector are detailed. Results show that entrainment ratio of the ejector depends on activating-steam pressure, on condenser temperature, and only slightly on evaporator temperature. For a fixed steam pressure, the COP of the hybrid cycle first surpasses that of the corresponding conventional cycle when the desorber temperature is increased, passes by a maximum, and then resumes the performance of the basic cycle. The maximum COP of an ejector-activated cycle is obtained at lower temperatures than that of a conventional system and can reach that of a double-effect basic scheme. The span of machine generator temperature where the COP is enhanced depends on the primary ejector pressure: it is larger for higher pressure. The entrainment ratio of the ejector is found to increase with the steam pressure and to decrease with rising backpressure. However, the performance of the ejector is confined to a specific region of the parameter-surface. Outside this domain, the entrainment ratio vanishes and the ejector is off-design.

### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Area</td>
</tr>
<tr>
<td>Ai</td>
<td>Nozzle area ratio (A_i/A_j)</td>
</tr>
<tr>
<td>AcAi</td>
<td>Ejector area ratio (A_c/A_i)</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy (kJ/kg)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate (kg/s)</td>
</tr>
<tr>
<td>M</td>
<td>Mach number</td>
</tr>
<tr>
<td>M*</td>
<td>Critical Mach number</td>
</tr>
<tr>
<td>P</td>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>Q</td>
<td>Heat transfer rate (kW)</td>
</tr>
<tr>
<td>R</td>
<td>Universal gas constant (kJ/(kg K))</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>W</td>
<td>Work transfer rate (kW)</td>
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<tr>
<td>X</td>
<td>Steam quality</td>
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### Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>γ</td>
<td>Ratio of steam specific heats (C_p/C_v)</td>
</tr>
<tr>
<td>εHX</td>
<td>Heat exchanger effectiveness</td>
</tr>
<tr>
<td>η</td>
<td>Nozzle, mixing, and diffuser efficiency</td>
</tr>
</tbody>
</table>
ξ: LiBr concentration in solution (mass. %)
ρ: Density (kg/m³)
τ: T₁₉/T₁₈
ω: Entrainment ratio (n₁₉/n₁₈).

Subscripts

AB: Absorber
bp: Backpressure
c: Constant section area (ejector)
CD: Condenser
d: Diffuser (ejector)
EV: Evaporator
G: Generator
i: Nozzle exit plane (ejector)
j: Plane in mixing chamber (ejector)
k: Shockwave plane
m: Mixing chamber (ejector)
n: Nozzle (ejector)
sat: Saturation
SOL: Solution
SG: Steam generator
W: Water
1–19: Referred state points.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

References


[28] M. L. Chougui, Simulation et étude comparée de cycle à absorption (LiBr/H2O) à usage de froid. Cas de l’unité de production de Henkel [Master], Université Mentouri, Constantine-Algérie, 2010, https://scholar.google.com/scholar?hl=en&as_sdt=0%2C5&q=Simulation+et+%C3%A9tude+compar%C3%A9e+de+cycle+%C3%A0+absorption+%28LiBr%2FH2O%29+%C3%A0+usage+de+froid.+Cas+de+l’unit%C3%A9+de+production+de+Henkel&btnG=.
