# Comparison of Artificial Intelligence Methods on the Performance of an Energy Conversion Cycle

Sarang Ezazi[1]
[1]Department of Mechanical Engineering, Ahar Branch, Islamic Azad University, Ahar, Iran.
Email: sa.ezazi@iau.ac.ir(Corresponding author)
Receive Date: 31July 2024 Accept Date:10Sep 2024

**Abstract**

*In the present research, the performance of a double absorption heat transformer is investigated through the Variable Metric and the Conjugate Directions methods. The main desired parameters to optimize are COP, ECOP, and exergy destruction. The results showed no considerable difference in COP, ECOP, and total exergy destruction rate achieved by the two methods. Besides, it was clear that, the design of the condenser and the absorber is very important. On the other hand, for the exergy destruction rate of components in some cases, the first algorithm performed better, while in other cases, the second algorithm did. Computer programming is accomplished through the EES.*

**Keywords:** COP, DAHT, ECOP, exergy destruction, optimization

## 1. Introduction

In recent decades, the optimal using of energy has been an essential issue due to their limited resources. One of the most desirable energy-saving devices that run on thermal energy at low or medium temperatures is the Absorption Heat Transformer [1]. Absorption heat converters have low electrical energy consumption, which leads to a reduction in the emission of pollutants such as carbon dioxide [2].

A type of double absorption heat transformer, which had been experimentally constructed by Mostofizadeh et al. [3], and is known as the third type DAHT, was theoretically investigated by Zhao et al. [4], which performed a higher COP when compared to formers. A new cycle for mono-absorption and double-absorption systems based on heat reabsorption from solution and refrigerant lines was introduced by Hernandez et al. [5], which has a lower working pressure than the existing converters, but its performance factor is also reduced. Wakim et al. [6], simulated the single absorption model to eliminate the contradictions of previous studies and obtained different results, including the presence of an extremum point in the curve of the coefficient of performance towards the increase in the temperature of the absorber. Wang et al. [7], through an advanced and economic exergy analysis by simulation method showed that only 21.28% of the exergy destruction rate could be avoided by improving the cycle efficiency. Mahmoudi et al. [8], performed a thermodynamic analysis on a proposed cogeneration cycle, and they estimated the minimum cost of the total product unit of \$42.6/GJ, the maximum exergy efficiency of 27.9%, and the maximum mass flow rate of distilled water of 0.53kg/s. Tang et al. [9], investigated the performance improvement of a new cycle, which is designed by adding an AHT to the supercritical carbon dioxide Brayton cycle, and showed 4.96% higher exergy efficiency when compared to the critical carbon dioxide cycle. Ezazi et al. [10], proposed five new cycles for the double absorption heat transformer and analyzed their energy efficiency and exergy efficiency compared with the third type of double absorption heat transformer.

A cycle including single absorption heat transformer, vapor absorption refrigeration, and a humidification-dehumidification-desalination system was presented by Beniwal et al. [11]. The thermodynamic performance of the system has been investigated in terms of coefficient of performance, refrigeration temperature, and the amount of distilled water production. Aulido et al. [12], used the artificial intelligence technique to calculate the absorption temperature in the second stage in a two-stage absorption converter by controlling the refrigerant flows based on the calculation of the flow ratio in both stages. Liu et al. [13] proposed a new cycle including heat exchanger and refrigeration cycles for cooling using a low-temperature source, so that, results showed the proposed system has a great potential to receive low-grade heat and create refrigeration below 15°C. With a broader look at the technical literature, it is clear that finding the optimal working conditions has always been on the minds of researchers.

In the present research, the third type double absorption heat transformer has been chosen due to its wide application. Then the optimization has been conducted to determine the optimal working conditions from COP, ECOP, and exergy destruction aspects, which is a concern for researchers.

## 2. Performance Description

An impure water desalination system attached to a third type of DAHT introduced by Khamooshi et al., [14] is shown in Fig. 1.

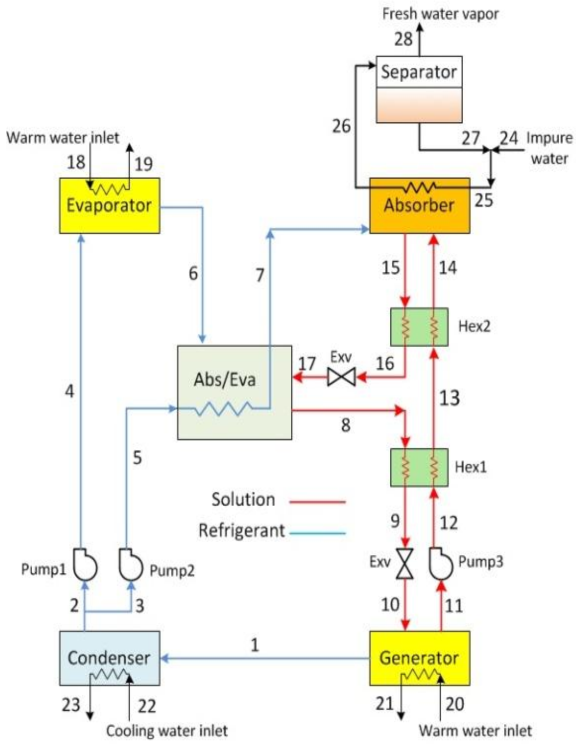


**Fig.1**.Schematic diagram of a DAHT coupled to a desalination system.

In the mentioned system, the wasted heat energy from industrial processes, $Q_{Gen}$, feeds generator at a relatively low temperature, $T_{Gen}$, to vaporize a portion of water refrigerant from $LiBr\_H_2O$ solution. The refrigerant flows to condenser where a phase change to saturated liquid occurs. Then, the condensed refrigerant is divided into two flow lines. One portion is pumped to evaporator, where receives a quantity of the waste heat, $Q_{Eva}$, at an intermediate temperature, $T_{Eva}$, and becomes saturated vapor.

The other portion flows to Abs\Eva via a pump at a higher pressure to produce steam at a higher temperature, $T_{Abs\backslash Eva}$. The steam in absorber will be absorbed to strong solution coming from the generator at a higher temperature, $T_{Abs}$. This is an exothermic process in which, desired thermal energy, $Q_{Abs}$, at higher temperature, $T_{Abs}$, is released. In addition, the strong solution already recovered some heat through two heat exchangers towards the absorber. The output weak solution from the absorber flows to the absorber-evaporator to absorb the saturated vapor that comes from the evaporator. The recent process, is exothermic and some heat is produced, $Q_{Abs\backslash Eva}$, through it too. Impure water from outside of DAHT receives the $Q_{Abs}$ and becomes partially evaporated. Finally, desired pure fresh water vapor is extracted through a separator.

## 3. Simulation

Assumptions, input parameters, and basic equations needed for modeling are presented in this part.3.1. Assumptions

Some assumptions are consideredas follows:

(1) All processes are under steady conditions [2,3,5,10,13,14].
(2) Changes in kinetic and potential energies are neglected.
(3) The Pressure drop because of friction in the piping system and components is ignored [2,3,4,5,14].
(4) Heat transfer to the environment is ignored for all components.
(5) The Solution is saturated at the generatorand the absorber outlet. Refrigerant is saturated at the evaporator and condenser outlet [2,3,4,5,10].
(6) The input energy of the cycles issupplied from a heat source withtemperature range of 90~100°C [3,10].
(7) The amount of input heat energysupplied to the evaporator is assumed to be the same in all cycles andtemperatures of the generator and theevaporator are equal [3,10].

The input parameters of modeling besides related ranges are presented inTable 1.

Table 1. Input parameters

| parameter | value |
|---|---|
| $T_{con}$ (°C) | 20~30 |
| $T_{eva}$= $T_{gen}$ (°C) | 80~90 |
| $T_{abs/eva}$ (°C) | 105~115 |
| $T_{abs}$ (°C) | 130~165 |
| $T_{heat\ source}$ (°C) | $T_{eva}$+10 |
| Max. mass rate of heat source (kg/s) | 20 |
| $T_{cooling\ water}$ (°C) | $T_{con}$-4 |
| Mass flow rate of cooling water (kg/s) | 13.8 |
| $\eta_{HEX}$ (%) | 80 |

### 3.2.Evaluation of performance

It should be noticed that, each component of the cycle, is considered as a control volume to evaluate the thermodynamic performance. Subsequently,the principle of conservation of mass and energy is considered in each case through equations 1 to 3.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

$$\sum \dot{m}_{in} x_{in} = \sum \dot{m}_{out} x_{out} \tag{2}$$

$$\dot{Q}_{cv} + \sum \dot{m}_{in} h_{in} = \sum \dot{m}_{out} h_{out} + \dot{W}_{cv} \tag{3}$$

Based on the first law of thermodynamics, the energy performance coefficient, as in equation 4indicates the ability to upgrade the input thermal energy, in absorption cycles [7,10].

$$COP = \frac{\dot{Q}_{abs}}{\dot{Q}_{eva} + \dot{Q}_{gen}} \quad (4)$$

Based on the second law of thermodynamics, the ratio of output thermal exergy to input thermal exergy, is defined as the exergy performance coefficient, as in equation 5 [15,16].

$$ECOP = \frac{\dot{Q}_{abs}(1 - \frac{T_\circ}{T_{abs}})}{\dot{Q}_{eva}(1 - \frac{T_\circ}{T_{eva}}) + \dot{Q}_{gen}(1 - \frac{T_\circ}{T_{gen}})} \quad (5)$$

The amount of heat energy required for water desalination is another significant parameter, which is given through relation 6 [14].

$$\dot{Q}_{utilized} = \dot{m}_{25}(h_{26} - h_{25}) \quad (6)$$

Another important parameter in the performance evaluation of the absorption heat transformer is the exergy destruction rate, which always should be tried to be minimized and is defined by equation number 7 [17].

$$E\dot{x}_D = T_o\dot{S}_G \quad (7)$$

The equations related to energy conservation of the cycle of Figure 1 are expressed in Table 2.

Table 2. Energy equations

| Item | Equation |
|---|---|
| Con | $\dot{Q}_{con} = \dot{m}_2 h_2 + \dot{m}_3 h_3 - \dot{m}_1 h_1 = \dot{m}_{22}(h_{23} - h_{22})$ |
| Eva | $\dot{Q}_{eva} = \dot{m}_4(h_6 - h_4) = \dot{m}_{18}(h_{18} - h_{19})$ |
| Gen | $\dot{Q}_{gen} = \dot{m}_1 h_1 + \dot{m}_{11} h_{11} - \dot{m}_8 h_{10} = \dot{m}_{20}(h_{20} - h_{21})$ |
| Abs\Eva | $\dot{Q}_{abs\_eva} = \dot{m}_8 h_8 - \dot{m}_2 h_6 - \dot{m}_{15} h_{17} = \dot{m}_3(h_7 - h_5)$ |
| Abs | $\dot{Q}_{abs} = \dot{m}_{15} h_{15} + \dot{m}_3 h_7 - \dot{m}_{11} h_{14} = \dot{m}_{25}(h_{26} - h_{25})$ |
| Hex 1 | $T_{13} = T_{12} + (T_8 - T_{12})\eta_{Hex1}$ <br> $\dot{Q}_{Hex1} = \dot{m}_8(h_8 + h_9) = \dot{m}_{12}(h_{13} - h_{12})$ |
| Hex2 | $T_{14} = T_{15} + (T_{15} - T_{13})\eta_{Hex2}$ <br> $\dot{Q}_{Hex2} = \dot{m}_{15}(h_{15} + h_{16}) = \dot{m}_{13}(h_{14} - h_{13})$ |
| Exv | $h_i = h_e$ |

Equations of exergy destruction rate of the cycle are expressed in Table3.

Table 3. Energy destruction rate equations

| Item | Equation |
|---|---|
| Con | $E\dot{x}_D = T_o(\dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_{22} s_{22} + \dot{m}_{23} s_{23})$ |
| Eva | $E\dot{x}_D = T_o(\dot{m}_6 s_6 + \dot{m}_{19} s_{19} - \dot{m}_4 s_4 - \dot{m}_{18} s_{18})$ |
| Gen | $E\dot{x}_D = T_o(\dot{m}_1 s_1 + \dot{m}_{11} s_{11} - \dot{m}_{10} s_{10} + \dot{m}_{21} s_{21} - \dot{m}_{20} s_{20})$ |
| Abs\Eva | $E\dot{x}_D = T_o(\dot{m}_8 s_8 + \dot{m}_7 s_7 - \dot{m}_5 s_5 - \dot{m}_{17} s_{17} - \dot{m}_6 s_6)$ |
| Abs | $E\dot{x}_D = T_o(\dot{m}_{15} s_{15} + \dot{m}_{26} s_{26} - \dot{m}_{25} s_{25} + \dot{m}_{14} s_{14} - \dot{m}_7 s_7)$ |
| Hex 1 | $E\dot{x}_D = T_o(\dot{m}_{13} s_{13} + \dot{m}_9 s_9 - \dot{m}_{12} s_{12} - \dot{m}_8 s_8)$ |
| Hex2 | $E\dot{x}_D = T_o(\dot{m}_{13} s_{13} + \dot{m}_9 s_9 - \dot{m}_{12} s_{12} - \dot{m}_8 s_8)$ |
| Exv | $E\dot{x}_D = T_o(\dot{m}_{10} s_{10} - \dot{m}_9 s_9)$ |
| Exv | $E\dot{x}_D = T_o(\dot{m}_{17} s_{17} - \dot{m}_{16} s_{16})$ |

3.3. Optimization

The optimization is carried out to achieve the maximum amount of COP, ECOP, and the minimum amount of exergy destruction rate of the DAHT cycle in the present study. Two different optimization algorithms are used for this purpose.The first algorithm is the Variable Metric method and the second algorithm is the Conjugate Directions method.

The temperature of the absorber, evaporator, generator, and condenser are considered as independent variables by the limits defined in Table 1. The simulation has been done through the EES [18].

## 4.Result

4.1. Validation of the results

In order to validate the thermodynamic analysis, the results of the present study in a specific case are compared to the results of Khamooshi et al. [14], and are presented in Figure 2.

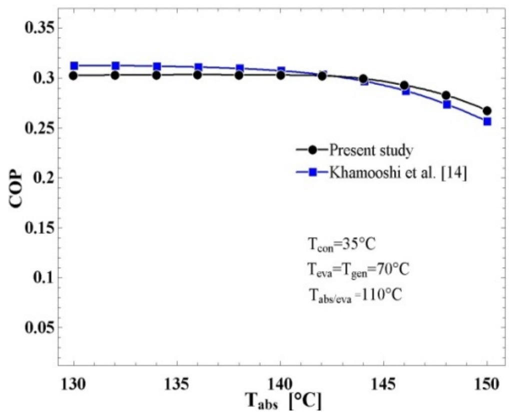


**Fig. 2.**Validation chart for thermodynamic analysis

4.2. Results of thermidynamic analysis

This section provides an in-depth analysis of how the operating conditions of the cycle components influence its overall performance.

Additionally, it includes a comprehensive presentation of the optimization results derived from these operating conditions.

Figure 3 illustrates how the coefficient of performance is affected by raising the absorber temperature. It is clear that the maximum value of the COP of the third type DAHT occurs in the temperature range between 150-165°C.

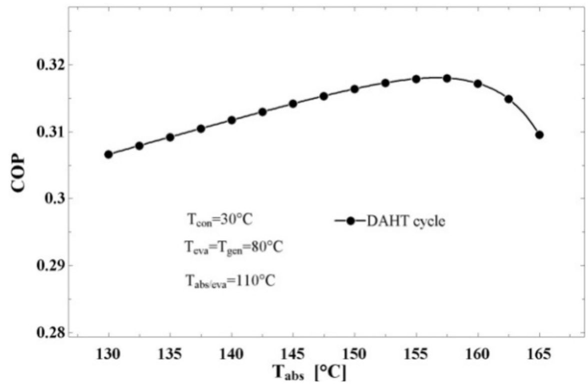


**Fig. 3.**The effect of the absorber temperature on the COP

Figure 4, shows the effect of absorber temperature increase on ECOP. It is observed that the exergy coefficient of performance has a similar behavior tothe energy coefficient of performance.

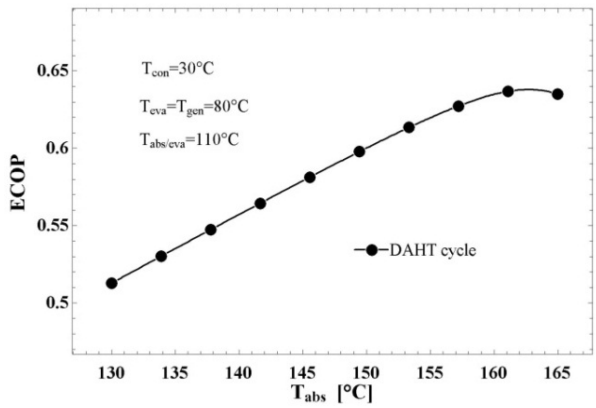


**Fig. 4.** The effect of the absorber temperature on the ECOP

Figure 5, shows how produced heat will be affected by absorber temperature.

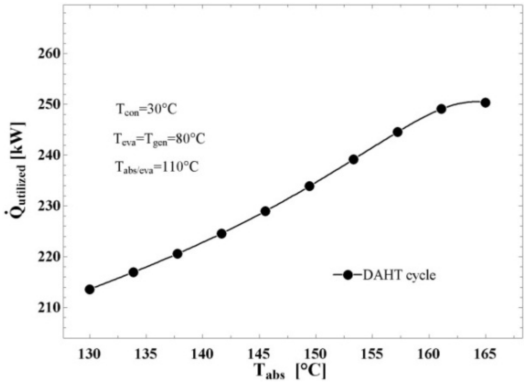


**Fig. 5.** The effect of the absorber temperature on the produced heat

A summary of the operation of the cycle is shown in Figures 3 to 5. But, the main aim of the study is to find optimum performance conditions through two different methods. Subsequently, the results are shown in Tables 4 to 10.

The maximum COP that is achieved through optimization is shown in Table 4.

Table 4. Simulation results for COP

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | COP |
|---|---|---|---|---|
| V.M. meth. | 165 | 90 | 20 | 0.3317 |
| C.D. meth. | 165 | 90 | 22.38 | 0.3318 |

As observed, there is a negligible difference in COP between two methods. However, the temperature of condenser has an increase of about 11.9% revealing the importance of condenser design.

Table 5. Simulation results for ECOP

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | ECOP |
|---|---|---|---|---|
| V.M. meth. | 165 | 80 | 21.16 | 0.66787 |
| C.D. meth. | 165 | 80 | 20 | 0.669 |

The results of Table 5 show the difference in ECOP is negligible through two methods. However, the temperature of condenser has decreased by about 5.48% which reveals the importance of the condenser design.

Table 6. Simulation results for exergy destruction rate of absorber

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | $Ex_{D, abs}$ (kW) |
|---|---|---|---|---|
| V.M. meth. | 160 | 80 | 25 | 1.621 |
| C.D. meth. | 130 | 80 | 30 | 0.9677 |

From Table 6, the exergy destruction rate of the absorber has an increase of about 40.3% and the condenser temperature difference is about 20%. But the point is that the absorber temperature is low by conjugate direction algorithm that is more logical, Because, it is clear that the temperature increase results in more exergy destruction.

Table 7. Simulation results for exergy destruction rate of evaporator

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | $Ex_{D, eva}$ (kW) |
|---|---|---|---|---|
| V.M. meth. | 160 | 80 | 30 | 0.2562 |
| C.D. meth. | 163 | 80 | 30 | 0.2562 |

According to Table 7, it is clear that there is no difference in exergy destruction rate of the evaporator. But the temperature of the absorber has an increase of 1.87% which is sufficiently little.

Table 8. Simulation results for exergy destruction rate of generator

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | $Ex_{D, gen}$ (kW) |
|---|---|---|---|---|
| V.M. meth. | 160 | 80 | 25.03 | 1.112 |
| C.D. meth. | 165 | 80 | 30 | 0.6472 |

From Table 8, the conjugate direction method resulted in, the higher temperature in the absorber and condenser the less exergy destruction in the generator, while, the generator temperature is constant.

This is not a correct scientific fact. Therefore, the first algorithm gives better results in this case.

Table 9. Simulation results for exergy destruction rate of condenser

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | $Ex_{D, con}$ (kW) |
|---|---|---|---|---|
| V.M. meth. | 160 | 80 | 25 | 0.04138 |
| C.D. meth. | 165 | 80 | 30 | 0.00152 |

Results show in Table 9, that the higher the temperature in the condenser the less energy destruction in the condenser itself. This is strongly false.

Table 10. Simulation results for exergy destruction rate of total cycle

| | $T_{Abs}$ (°C) | $T_{Gen}$ (°C) | $T_{Con}$ (°C) | $Ex_{D, con}$ (kW) |
|---|---|---|---|---|
| V.M. meth. | 130 | 80 | 30 | 3.393 |
| C.D. meth. | 129 | 80 | 30 | 3.394 |

According to Table 10, the same results are achieved for the exergy destruction rate of the total cycle. In other words, both algorithms demonstrated similar performance.

## 5. Conclusion

In the present research, the performance of a double absorption heat transformer has been optimized from COP, ECOP, and exergy destruction rate points of view through two different algorithms. The results showed no considerable difference in COP, ECOP, and total exergy destruction rate achieved by the two methods. However, for the exergy destruction rate of components in some cases, the first algorithm performed better, while in other cases, the second algorithm did. Therefore, a consensus on the effectiveness of the optimization methods could not be reached at the district level. Therefore, it is recommended that these methods be evaluated on a case-by-case basis.

## Refrences

[1]. K. Parham, M.Khamooshi, D.B.K. Tematio, M.Yari, and U. Atikol, "Absorption Heat Transformers– A Comprehensive Review," *Renewable and Sustainable Energy Reviews*, vol. 34, pp. 430-452, 2014.

[2]. Z. Zhao, F. Zhou, X. Zhang, and S. Li, "The Thermodynamic Performance of a New Solution Cycle in Double Absorption Heat Transformer Using Water/Lithium Bromide as the Working Fluids," *International Journal of Refrigeration*, vol. 3, no. 26, pp. 315-320, 2003.

[3]. C. Mostofizadeh, and C. Kulick, "Use of a new type of heat transformer in process industry," *Applied Thermal Engineering*, vol. 18, pp. 857-874, 1998.

[4]. Z. Zhao, Y. Ma, and J. Chen, "Thermodynamic performance of a new type of double absorption heat transformer," *Applied Thermal Engineering*, vol. 23, pp. 2407-2414, 2003.

[5]. J. A. Hernández-Magallanes, W. Rivera, and A. Coronas, "Comparison of single and double stage absorption and resorption heat transformers operating with the ammonia-lithium nitrate mixture," *Applied Thermal Engineering*, vol. 125, pp. 53-68, 2017.

[6]. Wakim M., and Rivera-Tinoco R., "Absorption heat transformers: Sensitivity study to answer existing discrepancies," *Renewable Energy*, vol. 130, pp. 881-890, 2019.

[7]. Wang Y., Liu Y., Liu X., Wanxiang Z., Cui P., Mengxiao Y., Zhiqiang L., and Sheng Y., "Advanced exergy and exergoeconomic analyses of a cascade absorption heat transformer for the recovery of low grade waste heat," *Energy Conversion and Management*, vol. 205, pp. 112392, 2020.

[8]. Mahmoudi S. M. S., Akbari A. D., and Rosen M. A., "A novel combination of absorption heat transformer and refrigeration for cogenerating cooling and distilled water: Thermoeconomic optimization," *Renewable Energy*, vol. 194, pp. 978-996, 2022.

[9]. Tang J., Zhang Q., Zhang Z., Li Q., Wu C., and Wang X., "Development and performance assessment of a novel combined power system integrating a supercritical carbon dioxide Brayton cycle with an absorption heat transformer," *Energy Conversion and Management*, vol. 251, pp. 114992, 2022.

[10]. Ezazi S., Khalilarya Sh., and Jafarmadar S., "Thermodynamic Analysis and Comparative Assessment of Proposed Schemes of Double Absorption Heat Transformer in Water Vaporization Process for Desalination," *Journal of Mechanical Engineering*, vol. 50,

no. 4, pp. 11-20, 2021.
[11]. R. Beniwal, K. Garg, and H. Tyagi, "Thermodynamics analysis of a novel absorption heat transformer-driven combined refrigeration and desalination system" *Energy Conversion and Management*, vol. 277, pp. 116597, 2023.
[12]. S. Vázquez-Aveledo, R.J. Romero, M. Montiel-González, and J. Cerezo, "Control Strategy Based on Artificial Intelligence for a Double-Stage Absorption Heat Transformer" *Processes*, vol. 11, pp. 1632, 2023.
[13]. Z. Liu, D. Lu, L. Shen, H. Guo, Y. Bai, L. Wang, and M. Gong, " New configurations of absorption heat transformer and refrigeration combined system for low-temperature cooling driven by low-grade heat", *Applied Thermal Engineering*, vol. 228, pp. 120567, 2023.
[14]. M. Khamooshi, K. Parham, I. Roozbeh, and H. Ensafisoroor, "Applications of innovative configurations of double absorption heat transformers in water purification technology," *Desalination and Water Treatment*, vol. 57, pp. 8204-8216, 2016.
[15]. Qin X., Chen L., and Xia S., "Ecological performance of four-temperature-level absorption heat transformer with heat resistance, heat leakage and internal irreversibility", *International Journal of Heat and Mass Transfer*, vol. 114, pp. 252-257, 2017.
[16]. Parham K., Khamooshi M., Daneshvar S., Assadi M., and Yari M., "Comparative assessment of different categories of absorption heat transformers in water desalination process", *Desalination*.
[17]. Ezazi S.,Sadighmanesh A., "Performance Optimization of Double Absorption Heat Transformers Using Genetic Algorithm in Order to Minimize Physical Exergy Destruction", *Journal of Artificial Intelligence in Electrical Engineering*, Vol. 9, No. 36, pp. 15-23, 2021.
[18]. S.A. Klein, Engineering Equation Solver, version 9.43, F-Chart Software, 2013.