Computational Sciences and Engineering 2(2) (2022) 251-274

University of Guilan

Computational Sciences and Engineering

journal homepage: https://cse.guilan.ac.ir/

Energy, exergy and economic analysis of a multi-generation system using the waste heat of a wind turbine

Kazem Zanganeh mahalleh, Mina Kohansal Vajargah *

Department of Mechanical Engineering, Langarud Branch, Islamic Azad University, Langarud, Iran

ARTICLE INFO

Article history:
Received 11 January 2023
Received in revised form 24 January 2023
Accepted 13 February 2023
Available online 13 February 2023

Keywords:
Wind Turbine
Waste energy recovery
Adsorption cycle
Exergy analysis
Economic analysis

ABSTRACT

Due to the increasing consumption of fossil fuels and the resulting environmental pollution, using renewable and clean energy has been considered and various research has been conducted for investigating the possibility of using them as a stimulus for different energy conversion systems. Compared to other renewable energy sources, wind energy has the least harm to the environment. The waste heat energy in the wind turbine gearboxes in the temperature between 100^o C to 150^o C can be used as a stimulus for low-temperature systems. In the present study, using this low-temperature energy has been studied as an energy source for a power-cooling absorption cycle and a domestic hot water heat exchanger. At first, the energy, exergy, and exergy-economic formulation of the system is presented. Then, in the results section after the validation study, a comprehensive parametric analysis has been conducted to investigate the influence of changing the effective parameters such as the average wind speed, and the temperature on the performance of the proposed system. The obtained results show that the effect of changing the wind speed on the performance of the system is greater than other parameters.

1. Introduction

In recent decades, the economic development of countries in the world has depended on finite fossil fuels. The oil crisis in 1976, adverse climate changes, the destruction of the ozone layer due to air pollution, and increasing fossil fuel prices led to more attention to the use of renewable energy sources in many countries [1]. In recent years, wind power recognized as one of the main power sources of clean and renewable energy that can decrease dependence on fossil fuels using [2]. Also in recent years, with considering suitable configurations the waste heat of various sectors of industries is deliberated to supply drinkable water in the same sector. However, waste heat capturing

* Corresponding author.
kohansal.vajargah@gmail.com

https://doi.org/10.22124/cse.2023.23837.1050

of the gearbox of a wind turbine for potable water and cooling production has paid less attention [3]. Due to the simultaneous necessity for electricity and heat energy in different industries, using simultaneous heat and electricity production systems has been noted, and because of the advantages of these systems in reducing energy consumption and cost, it is considered one of the suitable economical solutions [4].

In the last decade, usually the required cooling for the air conditioning process has been provided by the vapor compression cycle, and heat pump, which requires a large value of electrical energy. Recently, cycles have been introduced to produce refrigeration that uses low-temperature or medium-temperature thermal energy as a stimulus, which is called refrigeration production cycles with thermal energy which can be noted in absorption and volumetric chillers, ejector refrigeration cycle, and combined Rankine cycle and heat pump [5-7]. In these systems, the need for external electrical energy as a stimulus is eliminated, and low-temperature or medium-temperature renewable energy can be used as a stimulus for the refrigeration production cycle. In recent years, different studies have been conducted on the use of wind turbines in energy systems and absorption cycles, in which the thermodynamic characteristics of the cycles have been investigated and their performance and efficiency have been optimized. Some of these are mentioned following.

Nematollahi et al. [3] used the waste energy of wind turbines as the driver of an organic Rankine cycle. The results indicated that the maximum output power is 7.1 kW and was achieved using a134R fluid. In this case, the energy efficiency is 14.7% and the total cost of the system is 7050 dollars per year. From the energy and economic viewpoints, Khalilzadeh and Hosseinnejad [8] investigated using the waste energy of a wind turbine to produce potable water in a multiple-effect distillation system.

Rostamzadeh and Rostami [9] studied the use of waste heat extraction from the generator of a wind turbine in desalination. Five different nanoparticles in water were used as the cooling fluid of the wind turbine. In order to produce drinkable water and cooling capacity, Rostami et al. [10] used the waste energy of the wind turbine as the stimulus of a steam compression system. Nanofluid was used as the cooling fluid of the wind turbines.

Using a hot water heat exchanger, Rankine cycle and absorption chiller, Khalilzadeh and Hosseinnejad [11] used the waste energy in the wind turbine in a triple production system. The results showed that in the proposed system, the exergy yield and capital return period are 38.61 and 1.9 years, respectively.

Makkah et al. [12] investigated a co-generation system of electricity and potable water with a co-stimulator of solar energy and wind turbine. The solar collector was used as the stimulant of the organic Rankine cycle, and the production power in the wind turbine and Rankine was used as the stimulant of the reverse osmosis desalination. From the exergy-economic viewpoint, Li et al. [13] studied a multi-generation systems with a wind power stimulant and coal combustion chamber. The proposed system included a Rankine cycle, an absorption chiller, and an alkaline electrolyzer.

Sezer and Koç [14] investigated a multi-generation system with wind and solar energy stimulants. The system included an electrolyzer, a refrigeration cycle, a multi-stage flash desalination plant, a reverse osmosis desalination plant, and a fuel cell. From a thermodynamic viewpoint, Ozlu and Dincer [15] investigated the performance of a multi-generation system with a simultaneous stimulant of wind turbine and solar collector. The solar collector is used as the stimulant of the

Rankine cycle and absorption chiller, and the power of the wind turbine is used to produce hydrogen. The maximum energy efficiency of the proposed system was obtained 43%.

In order to create three pressure levels, Ventas et al. [16] used a compressor between the evaporator and the absorber of the absorption cycle. The results indicated the improvement of absorption cycle performance. Bellos and Tzivanidis [17] used an ejector in the absorption cycle and a solar collector as a stimulant. The results showed that the performance of the proposed system improved compared to the simple system. Also, in this case, the coefficient of performance of the absorption chiller was achieved 1.65.

From the viewpoint of energy and using low temperature renewable energies, Li et al. [18] investigated the performance of the combined system of transcritical organic Rankine cycle and reverse osmosis desalination. Then a parametric analysis was performed to determine the effect of the high pressure of the Rankine cycle on the output parameters. The results showed that by using R245fa and R152a fluids, the efficiency of the system was 15.86% and 13.47%, respectively.

In the current study, the waste energy of the wind turbine system is used in the new absorption cycle and a hot water heat exchanger. In the absorption cycle, the output steam of the generator is divided into two parts and used to produce power and cooling. Also, energy dissipation in the absorption cycle condenser is used as the hot fluid of a thermoelectric generator. The output power of the system is supplied from the wind turbine and the output power of the absorption cycle is used to produce fresh water in an osmosis desalination plant. In this study, in order to obtain the highest energy efficiency and exergy, a suitable configuration has been considered and also in each part of the combined system, the latest technology introduced according to the existing research has been used. Also, by using a comprehensive energy, exergy, economic and exergy-economic analysis, the system is examined and the components with the highest amount of exergy destruction and cost rate imposed on the system are identified and methods to reduce the waste and corresponding costs are proposed. In this research, the equations have been solved using Engineering Equation Solver (EES) software.

2. System description

In this section, first, the required formulation for the performance analysis of the proposed system is presented. Then the multi-generation cycles are introduced. Also, in order to study the performance of the cycle from the viewpoints of energy, exergy, and exergy-economic for different components of the cycle, the initial assumptions and input data of the problem are presented.

2.1. Thermodynamic analysis

Conservation equations in terms of mass, energy, entropy and exergy are employed through the investigation of system as below:

Mass balance equation [19]:

$$\sum_k \dot{m}_i - \sum_k \dot{m}_e = \frac{dm_{cv}}{dt} \tag{1}$$

in which $\dot{m}_i$ and $\dot{m}_e$ denote input and output mass flow rates, respectively.

Energy balance equation [19]:

$$\dot{Q} - \dot{W} + \sum_i \dot{m}_i (h_i + \frac{{v_i}^2}{2} + gZ_i) - \sum_e \dot{m}_e (h_e + \frac{{v_e}^2}{2} + gZ_e) = \frac{dE_{cv}}{dt} \quad (2)$$

Entropy balance equation [19]:

$$\dot{S}_{gen} = \sum_e \dot{m}_e s_e - \sum_i \dot{m}_i s_i - \sum_k \frac{\dot{Q}}{T_k} + \frac{dS_{cv}}{dt} \quad (3)$$

where $\dot{Q}$ is the value of heat transfer between heat source and working fluid. Also, T_k denote heat source temperature.

Exergy balance equation for a system in steady state, [19]:

$$\dot{Ex}_i + \dot{Ex}_Q = \dot{Ex}_e + \dot{Ex}_w + \dot{Ex}_{dest} \quad (4)$$

in which $\dot{Ex}_i$ and $\dot{Ex}_Q$ are the input and output system exergy flows, respectively. Also, $\dot{Ex}_Q$ is Exergy rate corresponding to heat transfer and $\dot{Ex}_w$ denote Exergy rate corresponding to work transfer. $\dot{Ex}_{dest}$ is exergy destruction rate.

Each component of equation (4) is expressed as follows [19]

$$\dot{Ex}_i = \dot{m}_i ex_i \quad (5)$$

$$\dot{Ex}_Q = (1 - \frac{T_0}{T_i})\dot{Q}_i \quad (6)$$

$$\dot{Ex}_e = \dot{m}_e ex_e \quad (7)$$

$$\dot{Ex}_w = \dot{W} \quad (8)$$

$$\dot{Ex}_{dest} = T_0 \dot{S}_{gen} \quad (9)$$

$$ex = ex_{ph} + ex_{ch} \quad (10)$$

In which T_0 is dead state temperature and $\dot{S}_{gen}$ is entropy generation in irreversible process. Also, ex_{ph} and ex_{ch} are physical and chemical exergy, respectively. Also, the exergy efficiency of the system is defined as follows [19]

$$\eta_{ex} = \frac{\dot{Ex}_P}{\dot{Ex}_F} = 1 - \frac{\dot{Ex}_{dest}}{\dot{Ex}_F} \quad (11)$$

Where $\dot{Ex}_P$ and $\dot{Ex}_F$ denote product exergy flow and system fuel, respectively.

2.2. Exergy–economic analysis

In exergy-economic analysis, the cost rate associated with each material or energy flow is used to calculate the exergy-economic variables of the system components. These variables are due to investment and thermodynamic inefficiency costs. In the exergy costing process, a cost is assigned to each exergy flow. The cost rate of $i\ th$ material flow is defined as $\dot{C}_i\left(\frac{\$}{hr}\right)$ and calculated in the following form [19]

$$\dot{C}_i = c_i \dot{Ex}_i \tag{12}$$

In which $\dot{Ex}_i$ denote Flow exergy rate and c_i is the cost of one exergy unit. Also, a cost is considered for exergy flow corresponding to heat transfer and work. It is defined as follow [19]

$$\begin{aligned} \dot{C}_q &= c_q \dot{Ex}_q = c_q \dot{Q}_k \left(1 - \frac{T_0}{T_k}\right) \\ \dot{C}_w &= c_w \dot{W} \end{aligned} \tag{13}$$

The exergy costing process includes cost balance equations that are usually written separately for each system component. The cost balance for the $k\ th$ component of the system indicates that the total cost of the outgoing flows is equal to the costs of the incoming flows, investment, operation, and maintenance of the same component and is written as follows [19]

$$\sum (c_e \dot{Ex}_e)_k + c_{w,k} \dot{W}_k = c_{q,k} \dot{Ex}_{q,k} + \sum (c_i \dot{Ex}_i)_k + \dot{Z}_k \tag{14}$$

In the ***Eq. (14)***, $\dot{Z}_k$ is the cost rate for the $k\ th$ component and it is obtained from the following relation

$$\dot{Z}_k = \frac{Z_k . CRF . \varphi}{N} \tag{15}$$

where Z_k is the purchase price of the component, φ is the operation and maintenance cost coefficient, N is the number of annual operating hours of the component, and CRF is the investment return coefficient, which is obtained from the following equation [19]

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \tag{16}$$

where i is the capital interest rate and n is the number of years of system operation. Also, the cost of exergy recovery can be predicted by the cost of additional fuel that is required to compensate for exergy destruction and simultaneous production of product exergy.

$$\dot{C}_{dest,k} = c_{F,k} \dot{E}_{dest,k} \tag{17}$$

For a component in the system, the exergy-economic factor is equal to the exergy independent cost to total cost ratio and is obtained as follow [19]

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{dest,k}} \tag{18}$$

3. multi-generation system description

The schematic of the proposed combined system with wind energy drive is shown in ***Figure (1).***

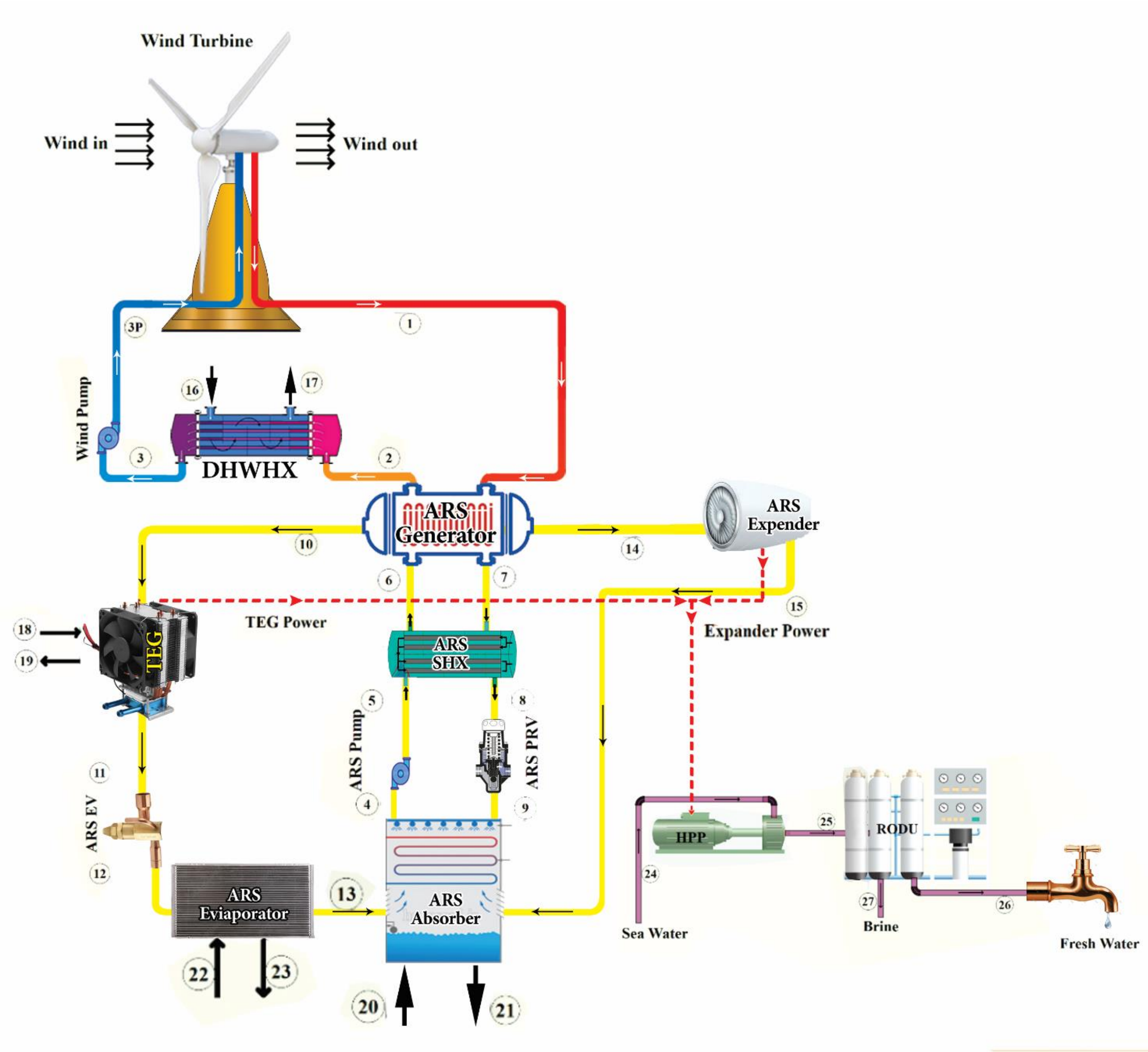

Figure 1. Schematic of the proposed system

By using wind energy, wind turbine produces electricity. The waste heat of the wind turbine in points 1 and 2 is considered as the driver of the absorption cycle and the heat exchanger. The waste heat of the wind turbine in points 1 and 2 is considered as the driver of the absorption cycle and the heat exchanger. The first part, point 10, is transferred to the thermoelectric generator in order to dissipate heat, and the second part, point 14, is transferred to the expander in order to generate power. In the proposed configuration, using the output power of the wind turbine electricity is produced. Also, fresh water is produced in reverse osmosis by using the output power of the expander and thermoelectric generator.

Also, in order to simulate the combined system, the following assumptions are considered:

- System works at steady state
- Kinetic and potential energy changes in different components are ignored.

- The pressure loss in the connecting pipes of components and heat exchangers is insignificant.
- There is no heat transfer between heat exchangers and the surroundings.
- The efficiency of turbines and pumps is isentropic and constant.
- • In the absorption cycle, the output fluid of the absorber and generator is in equilibrium and at a temperature corresponding to the mentioned components.
- In the absorption cycle, the refrigerant fluid at the output of the thermoelectric generator is a saturated liquid and at the output of the absorption evaporator, it is a saturated vapor.
- In the absorption cycle, at points 7 and 4, the fluid is saturated.
- For exergy analysis, surrounding environment temperature and pressure are considered as reference temperature and pressure.

3.1. Formulation of energy and exergy of combined system

The output power of the wind turbine is calculated from the following equation [3]

$$P_{avg,w,tur} = P_{mR}\left(\frac{\exp\left(-\left(\frac{u_c}{c}\right)^k\right) - \exp\left(-\left(\frac{u_R}{c}\right)^k\right)}{\left(\frac{u_R}{c}\right)^k - \left(\frac{u_c}{c}\right)^k} - \exp\left(-\left(\frac{u_F}{c}\right)^k\right)\right) \tag{19}$$

where different velocities are presented in Table 1. Also, the constants are calculated as follows [3]

$$k = d_1\sqrt{u_{avg}}$$
$$C = \frac{u_{avg}}{\Gamma\,(1+\frac{1}{k})} \tag{20}$$

that d_1 is constant and equal to 0.93. The gamma function is also obtained as follows [3]

$$\Gamma(y) = \int_0^\infty e^{(-x)}x^{(y-1)}dx \tag{21}$$

Table 1. Initial input values to simulate the wind turbine and its cooling subsystem [3,8]

Parameter	Description	Value
$\boldsymbol{T_1}$(°C)	Temperature at point 1	125
$\boldsymbol{T_2 - T_6}$(°C)	Temperature difference of the cold side of the generator heat exchanger	5
$\boldsymbol{T_3}$(°C)	Temperature at point 3	30
$\mathbf{u_{avg}}\ (\boldsymbol{m/s})$	Average wind speed	10
$\mathbf{u_c}(\boldsymbol{m/s})$	cut in speed	3.5
$\mathbf{u_r}(\boldsymbol{m/s})$	Nominal speed	15
$\mathbf{u_f}(\boldsymbol{m/s})$	Furling speed	28
$\mathbf{C_{pR}}$	power factor	0.281
$\mathbf{A_{wt}}(\boldsymbol{m^2})$	Wind turbine area	12668
$\boldsymbol{\eta_{tR}}$	Transmission efficiency	1
$\boldsymbol{\eta_{gen}}$	Generator efficiency	0.93

Also, P_{mR} is the maximum power produced by the wind turbine and calculated as follows [3]

$$P_{mR} = (\rho/2)A_{w,tur}{u_R}^3(C_{pR}\eta_{tR}) \tag{22}$$

Finally, the injected waste energy into the underlying system is calculated from the following equation

$$\dot{Q}_{waste} = \left(1-\eta_{gear}\right)\left(1-\eta_{gen}\right)P_{avg,w,tur} = \dot{m}_{waste}(h_1 - h_{3p}) \tag{23}$$

Also, exergy destruction in wind turbine is calculated from the following equation [3]

$$i_{wind,tur} = \left(\dot{Ex}_{in,wind} + \dot{Ex}_{3p}\right) - (\mathrm{P}_{avg,w,tur} + \dot{Ex}_1) \tag{24}$$

The initial values for the turbine simulation are presented in Table 1. The mass and energy balance equations in the absorption cycle evaporator are as follows [20]

$$\begin{gathered} \dot{m}_{12} = \dot{m}_{13} \\ \dot{Q}_{eva,ACH} = \dot{m}_{12}(h_{12} - h_{13}) \end{gathered} \tag{25}$$

where $\dot{Q}_{eva,ACH}$ is the rate of heat transfer in the evaporator. Exergy destruction in the evaporator is defined as follows [20]

$$i_{eva,ACH} = \left(\dot{Ex}_{12} - \dot{Ex}_{13}\right) + (\dot{Ex}_{22} - \dot{Ex}_{23}) \tag{26}$$

The mass and energy balance equations in the absorption cycle absorber are as follows [20]

$$\begin{gathered} \dot{m}_4 = \dot{m}_9 + \dot{m}_{13} + \dot{m}_{15} \\ \dot{m}_4 x_4 = \dot{m}_9 x_9 \\ \dot{Q}_{abs,ACH} = \dot{m}_9 h_9 + \dot{m}_{13} h_{13} + \dot{m}_{15} h_{15} - \dot{m}_4 h_4 \end{gathered} \tag{27}$$

The heat transfer rate in the absorber is obtained from the following equation [20]

$$\dot{Q}_{abs,ACH} = \dot{m}_{20}(h_{21} - h_{20}) \tag{28}$$

Also, exergy destruction in the absorber is calculated from the following relationship [20]

$$i_{abs,ACH} = (\dot{Ex}_9 + \dot{Ex}_{15} + \dot{Ex}_{13} - \dot{Ex}_4) + (\dot{Ex}_{20} - \dot{Ex}_{21}) \tag{29}$$

And $\dot{m}_{20}$ is the flow rate of input cool water from the cooling tower to the absorption chiller absorber. The energy balance equation in the absorption system generator is as follows [20]

$$\dot{Q}_{gen,ACH} = \dot{m}_7 h_7 + \dot{m}_{10} h_{10} + \dot{m}_{14} h_{14} - \dot{m}_6 h_6 \tag{30}$$

$\dot{Q}_{gen,ACH}$ is the heat transfer rate in the generator and obtained from the following equation

$$\dot{Q}_{gen,ACH} = \dot{m}_{waste}(h_1 - h_2) \tag{31}$$

The exergy destruction in the generator is achieved from the following relation

$$i_{gen,ACH} = (\dot{E}x_6 - \dot{E}x_7 - \dot{E}x_{10} - \dot{E}x_{14}) + (\dot{E}x_1 - \dot{E}x_2) \tag{32}$$

The production power in the absorption system condenser is calculated from the following relations

$$\begin{gathered} \dot{W}_{TEG} = \eta_{TEG} \cdot \dot{Q}_{ELEGANT} \\ \dot{Q}_{ELEGANT} = \dot{m}_{10}(h_{10} - h_{11}) \\ \eta_{TEG} = \eta_{Carnot} \frac{\sqrt{1+ZT_M}-1}{\sqrt{1+ZT_M}+\frac{T_L}{T_H}} \end{gathered} \tag{33}$$

That ZT_M is equal to 0.93. Exergy destruction is also obtained from the following relationship

$$i_{gen,ACH} = (\dot{E}x_6 - \dot{E}x_7 - \dot{E}x_{10} - \dot{E}x_{14}) + (\dot{E}x_1 - \dot{E}x_2) \tag{34}$$

The efficiency coefficient, energy balance equation and exergy destruction of the heat exchanger are obtained from the ***Eqs. (35)*** to ***(37)***

$$\varepsilon_{shx} = \frac{T_7 - T_5}{T_7 - T_8} \tag{35}$$

$$\dot{m}_7(h_7 - h_8) = \dot{m}_5(h_6 - h_5) \tag{36}$$

$$i_{shx,ACH} = \left(\dot{E}x_5 - \dot{E}x_6\right) + (\dot{E}x_7 - \dot{E}x_8) \tag{37}$$

The isentropic efficiency, output work rate and irreversibility in the expander are obtained from the following relations

$$\eta_{\exp} = \frac{h_{14} - h_{15}}{h_{14} - h_{15s}} \tag{38}$$

$$\dot{W}_{\exp} = \dot{m}_{14}(h_{14} - h_{15}) \tag{39}$$

$$i_{\exp} = (\dot{E}x_{14} - \dot{E}x_{15}) - \dot{W}_{\exp} \tag{40}$$

The initial input values for simulating the absorption cycle are presented in the Table 2.

Table 2. Initial input values for simulating the absorption cycle [20]

Parameter	Description	Value
T_{13}(°C)	Absorption evaporator temperature	6
T_{11}(°C)	Absorption condenser temperature	35
T_{4}(°C)	Absorption absorber temperature	35
T_{7}(°C)	Absorption generator temperature	70
η_{sp}	Absorption solution pump efficiency	0.9
ε_{shx}	Efficiency of absorption solution heat exchanger	0.7
$\eta_{\exp}$	Isentropic efficiency of the expander	0.8

In reverse osmosis desalination, recovery ratio and desalination percentage are calculated from the following relationships, respectively.

$$RR = \frac{\dot{Q}_p}{\dot{Q}_f} \tag{41}$$

$$\mathrm{SR} = 1 - \frac{x_p}{x_f} \tag{42}$$

In which $\dot{Q}$ is volumetric flow rate, x mount of salt concentration in the flow and the subscripts f and P correspond to the input salt water and the output fresh water, respectively. The consistency of volume flow and salt flow in desalination water is written according to the following relationships [21]

$$\dot{Q}_f = \dot{Q}_p + \dot{Q}_b \tag{43}$$

$$\dot{Q}_f x_f = \dot{Q}_p\, x_p + \dot{Q}_b\, x_b \tag{44}$$

The required power of the reverse osmosis high pressure pump is calculated from the following equation [21]

$$\mathrm{HPP} = \frac{1000 \cdot \dot{Q}_f \cdot \Delta P}{3600 \cdot \rho_f \cdot \eta_p} \tag{45}$$

The required power is supplied from turbine number 2. In ***Eq. (45)***, ρ_f, η_p, and ΔP denote input flow density, the pump efficiency and the pressure difference, respectively. Which ΔP is calculated from the following equation [21]

$$\Delta P = \frac{\dot{Q}_p}{3600 \cdot TFC \cdot FF \cdot A_e \cdot n_e \cdot n_v \cdot k_w} + \Delta\pi \tag{46}$$

where TFC is the temperature correction factor, FF is the fouling factor, A_e is the area of each element, n is the number of elements, n_v is the number of pressure channels, k_w is water permeability and $\Delta\pi$ is the total osmotic pressure of desalination water. TFC, k_w and $\Delta\pi$ are calculated from ***Eqs. (47)*** to ***(50)*** and other parameters are presented in Table 2. [21]

$$TFC = \exp\,(2700 \times (\frac{1}{273 + t_f} + \frac{1}{298}) \tag{47}$$

where t_f is the operating temperature of the desalination water.

$$k_w = 6.84 \times 10^{-8}(\frac{18.68 - 0.177x_b}{273 + t_f}) \quad (48)$$

To calculate $\Delta\pi$, first the osmotic pressure in different sections is calculated from the following relations [21]

$$z_f = 75.84 \cdot x_f$$

$$z_p = 75.84 \cdot x_p \quad (49)$$

$$z_b = 75.84 \cdot x_b$$

The average osmotic pressure is calculated from the following equation [22]

$$\bar{z} = 0.5(z_f + z_b) \quad (50)$$

and the total osmotic pressure is obtained as below [22]

$$\Delta\pi = \bar{z} - z_p \quad (51)$$

The input data for reverse osmosis desalination simulation are presented in Table 3. [21, 22].

Table 3. Input data for desalination system simulation

Parameter	Description	Value
$\boldsymbol{x_f}\ (\frac{\boldsymbol{kg}}{\boldsymbol{m^3}})$	The salt concentration of input water	45
$\boldsymbol{p_f}\ (\boldsymbol{kPa})$	Pressure of the input salt water to the desalination water system	6700
$\boldsymbol{t_f}$ (℃)	Operating temperature of desalination water	25
$\boldsymbol{RR}$	Recovery ratio	0.3
SR	Desalination ratio	0.994
$\boldsymbol{A_e}\ (\boldsymbol{m^2})$	The area of the element	35.4
$\boldsymbol{FF}$	Fouling factor	0.85
$\boldsymbol{\eta_p}$	Pump efficiency	0.8
$\boldsymbol{n_e}$	Number of element	7
$\boldsymbol{n_v}$	Number of pressure ducts	42

The heat transfer rates in the hot water heat exchanger and evaporator are obtained from the ***Eqs. (52), (53)***

$$\dot{Q}_{DHWHX} = \dot{m}_{waste}(h_2 - h_3) \quad (52)$$

$$\dot{Q}_{DHWHX} = \dot{m}_{16}(h_{17} - h_{16}) \quad (53)$$

Also, exergy destruction in the evaporator is defined as follow

$$i_{DHWHX} = \left(\dot{Ex}_2 - \dot{Ex}_3\right) + (\dot{Ex}_{16} - \dot{Ex}_{17}) \tag{54}$$

The equations used for the exergy-economic analysis of the system are presented in the Table 4.

Table 4. Required data for the exergy-economic analysis of the system

Component	Cost balance relationship	Complementary relationship
Wind Turbine	$c_{wind}\dot{Ex}_{in,wind} + \dot{C}_{3p} + \dot{Z}_{w,tur} = \dot{C}_1 + c_{\mathrm{el,w}}\, P_{avg,w,tur}$	$c_1 = c_{3p}\, , c_{wind} = 0$
Wind pump	$\dot{C}_3 + c_{\mathrm{el,w}}\dot{W}_{\mathrm{w,pum}} + \dot{Z}_{\mathrm{w,pum}} = \dot{C}_{3p}$	
Hot water heat exchanger	$\dot{C}_2 + \dot{C}_{16} + \dot{Z}_{\mathrm{DHWHX}} = \dot{C}_3 + \dot{C}_{17}$	$c_2 = c_3\, , c_{16} = 0$
Absorption generator	$\dot{C}_1 + \dot{C}_6 + \dot{Z}_{\mathrm{ARS,gen}} = \dot{C}_2 + \dot{C}_7 + \dot{C}_{10} + \dot{C}_{14}$	$c_1 = c_2\, , \frac{\dot{C}_{10}-\dot{C}_6}{\dot{Ex}_{10}-\dot{Ex}_6} = \frac{\dot{C}_7-\dot{C}_6}{\dot{Ex}_7-\dot{Ex}_6} = \frac{\dot{C}_{14}-\dot{C}_6}{\dot{Ex}_{14}-\dot{Ex}_6}$
Thermoelectric generator	$\dot{C}_{10} + \dot{C}_{18} + \dot{Z}_{\mathrm{ARS,con}} = \dot{C}_{11} + \dot{C}_{19} + \mathrm{c}_{\mathrm{el,ARS}}\dot{\mathrm{W}}_{\mathrm{TEG}}$	$c_{10} = c_{11}\, , c_{18} = 0$
Absorption absorber	$\dot{C}_9 + \dot{C}_{13} + \dot{C}_{15} + \dot{C}_{20} + \dot{Z}_{\mathrm{ARS,abs}} = \dot{C}_4 + \dot{C}_{21}$	$c_{20} = 0\, , \frac{\dot{C}_9+\dot{C}_{13}+\dot{C}_{15}}{\dot{Ex}_9+\dot{Ex}_{13}+\dot{Ex}_{15}} = c_4$
Absorption evaporator	$\dot{C}_{12} + \dot{C}_{22} + \dot{Z}_{\mathrm{ARS,eva}} = \dot{C}_{13} + \dot{C}_{23}$	$c_{12} = c_{13}, \mathrm{c}_{22} = 0$
Solution heat exchanger	$\dot{C}_5 + \dot{C}_7 + \dot{Z}_{\mathrm{ARS,SHX}} = \dot{C}_6 + \dot{C}_8$	$c_7 = c_8$
Solution pump	$\dot{C}_4 + c_{\mathrm{el,w}}\dot{W}_{\mathrm{ARS,pum}} + \dot{Z}_{\mathrm{ARS,pum}} = \dot{C}_5$	
Refrigerant pressure-reducing valve		$c_{11} = c_{12}$
Solution pressure-reducing valve		$c_8 = c_9$
Expander	$\dot{C}_{14} + \dot{Z}_{\mathrm{ARS,tur}} = \dot{C}_{15} + c_{\mathrm{el,ARS}}\, \dot{W}_{\mathrm{ARS,exp}}$	$c_{14} = c_{15}$
reverse osmosis	$c_{\mathrm{el,ARS}}\, \dot{W}_{\mathrm{HPP}} + \dot{Z}_{\mathrm{RODU}} = \dot{C}_{26}$	

Also, the required equations to calculate the cost of system components are presented in Table 5.

Table 5. Required equations to calculate the cost of system components

Component	Equation
Wind turbine	$Z_{\mathrm{w,tur}} = 3000 P_{avg,w,tur}$
Wind pump	$Z_{\mathrm{w,pum}} = 1120\ \dot{W}_{\mathrm{w,pum}}{}^{0.8}$
Hot water heat exchanger	$Z_{\mathrm{DHWHX}} = 130\ \left(\frac{A_{\mathrm{DHWHX}}}{0.093}\right)^{0.78}$
Absorption generator	$Z_{\mathrm{ARS,gen}} = Z_{\mathrm{R,gen}}\left(\frac{A_{\mathrm{ARS,gen}}}{A_{\mathrm{R}}}\right), Z_{\mathrm{R,gen}} = 17500\ \$, A_{\mathrm{R}} = 100\ m^2$
Thermoelectric generator	$Z_{\mathrm{TEG}} = 1500\ \dot{W}_{\mathrm{TEG}}$
Absorption absorber	$Z_{\mathrm{ARS,abs}} = Z_{\mathrm{R,abs}}\left(\frac{A_{\mathrm{ARS,abs}}}{A_{\mathrm{R}}}\right), Z_{\mathrm{R,abs}} = 16500\ \$, A_{\mathrm{R}} = 100\ m^2$
Absorption evaporator	$Z_{\mathrm{ARS,eva}} = Z_{\mathrm{R,eva}}\left(\frac{A_{\mathrm{ARS,eva}}}{A_{\mathrm{R}}}\right), Z_{\mathrm{R,eva}} = 16000\ \$, A_{\mathrm{R}} = 100\ m^2$
Solution heat exchanger	$Z_{\mathrm{ARS,SHX}} = Z_{\mathrm{R,SHX}}\left(\frac{A_{\mathrm{ARS,SHX}}}{A_{\mathrm{R}}}\right), Z_{\mathrm{R,SHX}} = 12000\ \$, A_{\mathrm{R}} = 100\ m^2$
Solution pump	$Z_{\mathrm{ARS,pum}} = Z_{\mathrm{R,pum}}\left(\frac{\dot{W}_{\mathrm{ARS,pum}}}{\dot{W}_{\mathrm{R,pum}}}\right)^{0.26}\left(\frac{1-\eta_{\mathrm{pum}}}{\eta_{\mathrm{pum}}}\right)^{0.5}, Z_{\mathrm{R,pum}} = 2100\ \$, \dot{W}_{\mathrm{R,pum}} = 10\ \mathrm{kW}$
Refrigerant pressure-reducing valve	$Z_{\mathrm{ARS,EV}} = 114.5\ \dot{m}_{main}$
Solution pressure-reducing valve	$Z_{\mathrm{ARS,PRV}} = 114.5\ \dot{m}_{strong}$
Expander	$Z_{\mathrm{ARS,exp}} = 4405\ \dot{W}_{\mathrm{ARS,exp}}{}^{0.7}$
reverse osmosis	$Z_{RODU} = 52\ \dot{Q}_{fe}\left(P_{fe} - P_0\right) + 996\ \dot{m}_{fe}{}^{0.8} + C_k n_e n_v + C_{pv} n_v$ $C_k = 1200\$, C_{pv} = 7000\$$

The work of absorption system and the total work are calculated from the following relations

$$\dot{W}_{ARS} = \dot{W}_{\mathrm{ARS,exp}} + \dot{W}_{TEG} \tag{55}$$

$$\dot{W}_{net} = P_{avg,w,tur} - (\dot{W}_{w,pum} + \dot{W}_{ARS,pum}) \tag{56}$$

The production flow rate of potable water is obtained from the following equation

$$\dot{m}_{freshwater} = \dot{V}_{fr} . \rho_{fr}/3600\ (kg \cdot s^{-1}) \tag{57}$$

The total exergy efficiency of the system is determined using the following equation

$$\eta_{ex,poly} = \frac{\dot{W}_{net} + \dot{Ex}_{P,DHWHX} + \dot{Ex}_{P,ARS,eva} + \dot{Ex}_{26}}{\dot{Ex}_{in,wind}} \tag{58}$$

Using ***Eq. (58),*** the wind input exergy is calculated as follows [25]

$$\dot{Ex}_{in,wind} = (\rho/2) A_{w,tur} u_{avg}{}^3 \tag{59}$$

The costs of the exergy of different products and multi-generation system are calculated as below

$$c_{el} = \frac{\dot{C}_{W,net}}{\dot{W}_{net}}$$
$$c_{heating} = \frac{\dot{C}_{17}}{\dot{Ex}_{17}} \tag{60}$$

$$c_{cooling} = \frac{\dot{C}_{23}}{\dot{E}x_{23}}$$

$$c_{freshwater} = \frac{\dot{C}_{26}}{\dot{E}x_{26}}$$

$$c_{poly} = \frac{\dot{C}_{W,net} + \dot{C}_{17} + \dot{C}_{23} + \dot{C}_{26}}{\dot{W}_{net} + \dot{E}x_{17} + \dot{E}x_{23} + \dot{E}x_{26}}$$

Period of Return the capital and the value of system capital are obtained as below

$$PP = \frac{ln\left(\frac{CF}{CF - i \cdot Z_{tot}}\right)}{ln(1+i)} \tag{60}$$

$$NPV = CF\frac{(1+i)^n - 1}{i(1+i)^n} - Z_{tot} \tag{61}$$

Where

$$CF = N \cdot \sum c_p\, \dot{E}n_p \tag{62}$$

3. Validation

In this section for validation study, the obtained results of wind turbine and reverse osmosis desalination are compared with the results of previous works. Comparison is presented in Tables 6, 7

Table 6. Validation study of wind turbine results

Average wind velocity (m/s)	$P_{avg,w,tur}(kW)$			$\dot{Q}_{waste}(kW)$		
	Current work	Ref [9]	Difference (%)	Current work	Ref [9]	Difference (%)
9	2230	2219	0.49	156.1	155.3	0.51
10	2749	2745	0.14	192.4	192.2	0.1
11	3291	3300	0.27	230.3	231	0.12
12	3829	3856	0.7	268	269.9	0.7

Table 7. Validation study of reverse osmosis desalination results

Parameter	Current work	Ref [23]	Ref [37]
	Input result		
SR	0.994	0.994	0.994
$\dot{V}_{fe}(m^3\ \mathrm{h}^{-1})$	485.9	485.9	485.9
	Output result		
$\dot{W}_{HPP}$ (kW)	1126	1185	1131
x_b (kg m^{-3})	64.17	64.16	64.18
x_{fr} (kg m^{-3})	0.27	0.252	0.25
$\Delta P(kPa)$	6843	6843	6850

As shown in Tables 6 and 7, there is a very good agreement between the results of the present study and previous references.

4. Results and discussion

The output parameters of the system are presented in Table 8. In this case, the recovery of waste wind turbine energy leads to the production of $73.25\ kW$ heat, $45.86\ kW$ cooling and $4.49\ kW$ output power. This output power in the reverse osmosis desalination produces $0.274\ kg/s$ of fresh water. For the investigated system, the exergy of the wind input fuel was $7497\ KW$, of which $2756\ KW$ convert to the output product, and as a result, the exergy efficiency of the system is equal to 36.76%.

Table 8. Output parameters of the system

Parameter	Value
$P_{avg,w,tur}(\mathbf{KW})$	2749
$\dot{\mathbf{W}}_{\mathbf{ARS}}(\mathbf{KW})$	4.494
$\dot{\mathbf{W}}_{\mathbf{net}}(\mathbf{KW})$	2749
$\dot{\mathbf{Q}}_{\mathbf{DHWHX}}(\mathbf{KW})$	73.25
$\dot{\mathbf{Q}}_{\mathbf{cooling}}(\mathbf{KW})$	45.86
$\dot{\mathbf{m}}_{\mathbf{freshwater}}(\frac{\mathbf{kg}}{\mathbf{s}})$	0.274
$\dot{\mathbf{Ex}}_{\mathbf{fuel,tot}}(\mathbf{kW})$	7497
$\dot{\mathbf{Ex}}_{\mathbf{product,tot}}(\mathbf{kW})$	2756
$\dot{\mathbf{Ex}}_{\mathbf{dest,tot}}(\mathbf{kW})$	4737
$\dot{\mathbf{Ex}}_{\mathbf{loss,tot}}(\mathbf{kW})$	4.882
$\boldsymbol{\eta}_{\mathbf{ex,multi}}(\%)$	36.76
$\dot{\mathbf{Z}}_{\mathbf{tot}}(\$/\mathbf{h})$	202.6
$\dot{\mathbf{C}}_{\mathbf{dest,tot}}(\$/\mathbf{h})$	319.1
$\dot{\mathbf{C}}_{\mathbf{tot}}(\$/\mathbf{h})$	521.8
$\mathbf{f}_{\mathbf{tot}}\ (\%)$	38.84
$\mathbf{c}_{\mathbf{el}}(\$/\mathbf{GJ})$	18.58
$\mathbf{c}_{\mathbf{heating}}\ (\$/\mathbf{GJ})$	63.49
$\mathbf{c}_{\mathbf{cooling}}\ (\$/\mathbf{GJ})$	64.13
$\mathbf{c}_{\mathbf{freshwater}}(\$/\mathbf{GJ})$	5938.72
$\mathbf{c}_{\mathbf{multi}}(\$/\mathbf{GJ})$	20.25
$\mathbf{PP}\ (\mathbf{years})$	2.301
$\mathbf{NPV}\ (\mathbf{M\$})$	32.46

Also, $4737\ KW$ of input exergy, 63.18%, was wasted in the form of exergy destruction in various components, and $4.88\ KW$, $0.065\ \%$, was wasted in the form of water flow in thermoelectric, absorber, and salt water flow in desalination. In the proposed system, the total initial cost rate is 202.6 and the initial exergy destruction cost rate is 319.1. As a result, the total cost rate is 521.8

and the economic exergy factor is 38.84. Also, the costs of the exergy unit and the cost of the freshwater production unit are equal to 5938.72 $(\$/Gj)$ and are significantly higher than other costs, which is due to the high cost of the electricity unit of the absorption system. Also, the initial cost of reverse osmosis is higher, too.

The unit cost of exergy production of cooling and heating is almost equal and is about 64. In addition, the cost of multi-generation units is equal to 20.25 $(\$/Gj)$. Finally, the capital return period is equal to 2.301 years and the NPV of the system is equal to 32.46 million dollars over 20 years.

4.1.Parametric analysis results

In this section, the effect of changing the influencing parameters on the system performance is investigated. The effect of changing the average wind speed on the output parameters is shown in ***Figures 2 to 7.*** As shown in ***Figure 2*** increasing the average wind speed leads to increasing the output power of the wind turbine, thus the total output work increases. On the other hand, based on ***Figure 3,*** with the increase in the output power of the wind turbine, the flowrate of the cooling system increases, which increases the flowrate in the hot water production exchanger and absorption system. Despite the increase in the output products, the increase in the average speed increases the exergy of the wind fuel system, which has a dominant effect on the exergy efficiency, and according to ***Figure 4*** ,it causes a significant decrease in the exergy efficiency from 45.9% to 23.4%. Also, increasing the flow rate of all system points increases the total irreversibility from 2074 to 15750.

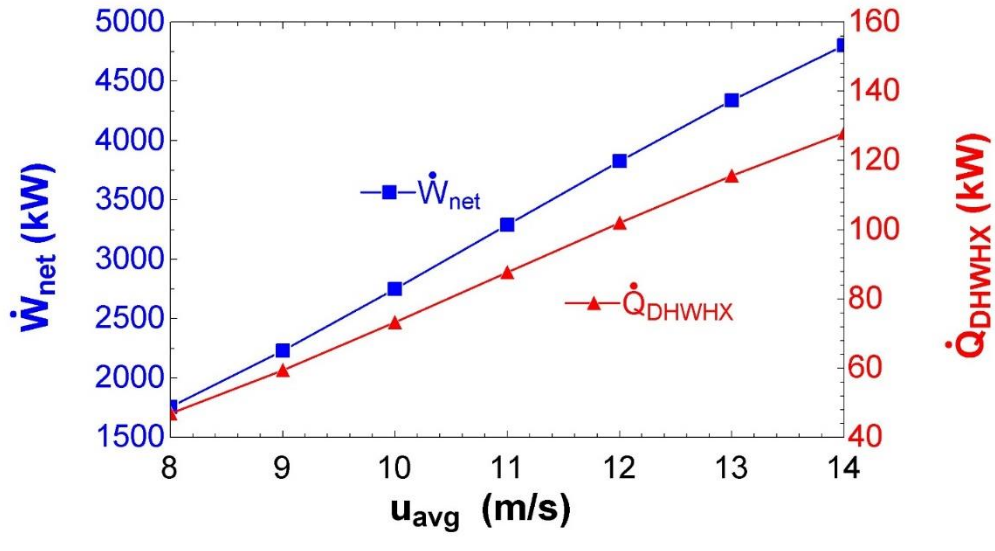

Figure 2. The effects of wind speed variations on the work and output heat

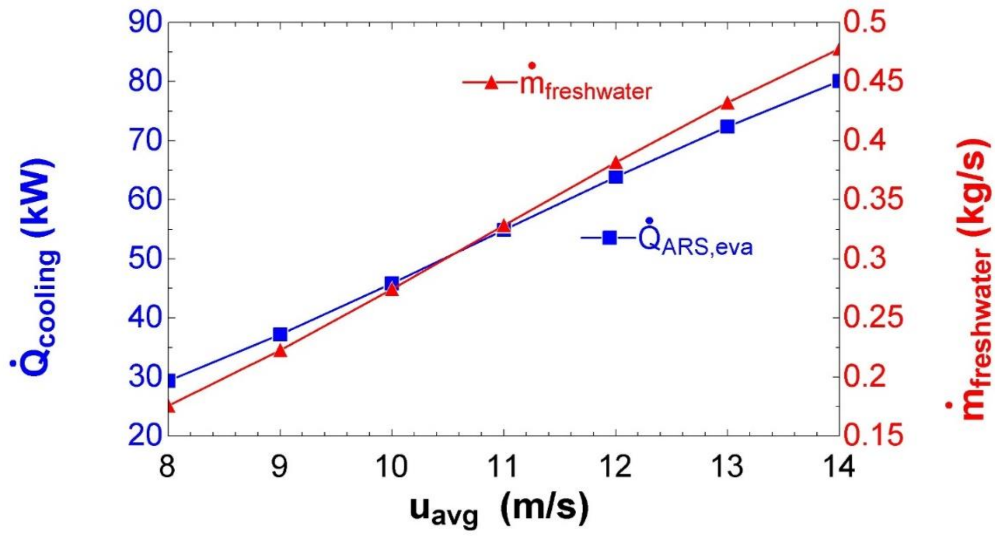

Figure 3. The effects of wind speed variations on output cooling and freshwater flowrate

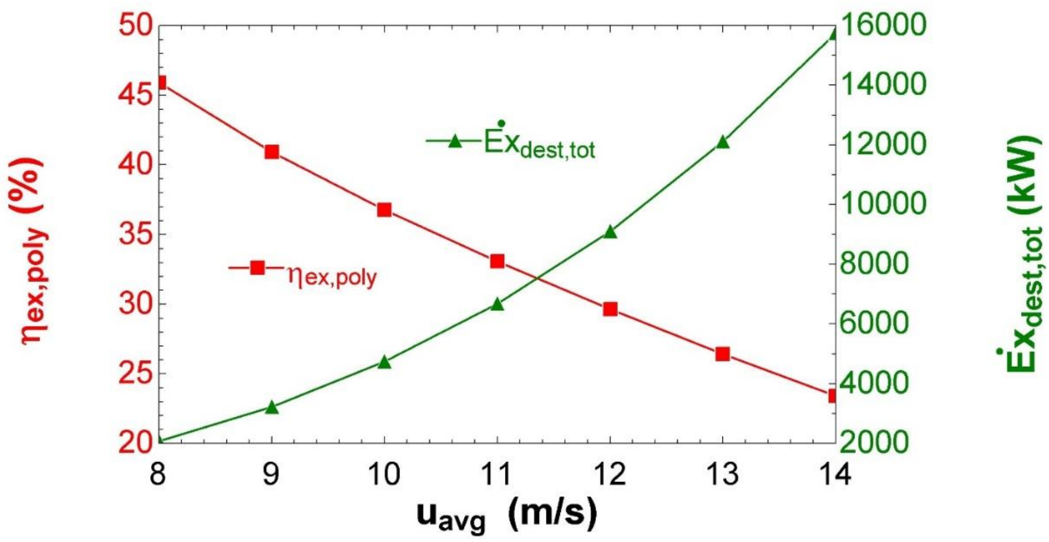

Figure 4. The effects of wind speed variations on exergy efficiency and exergy destruction

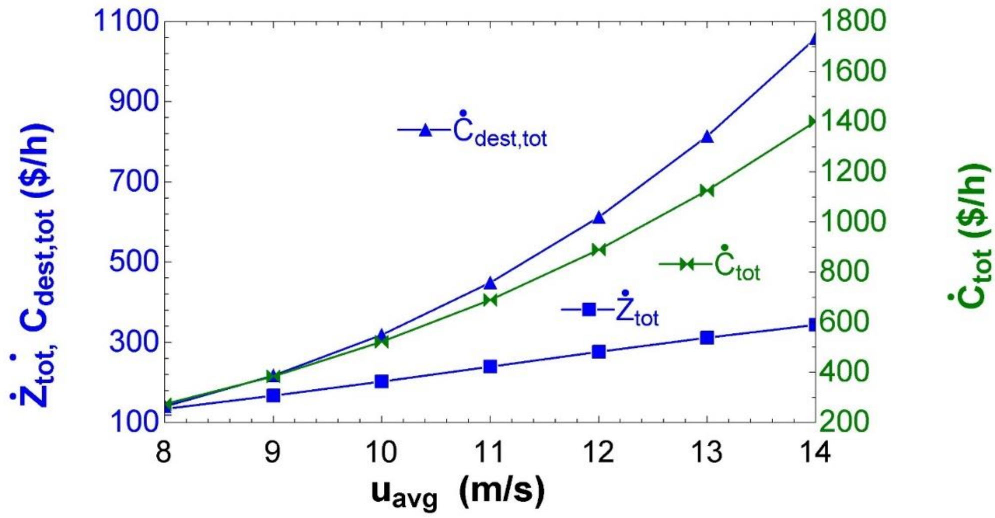

Figure 5. The effects of wind speed variations on cost rates

According to ***Figure 5***, increasing the average speed causes a simultaneous increase in the initial cost rate and exergy destruction cost rate, which increases the total cost rate from 274.6 to 1402 which is due to the significant increase in the initial cost rate and exergy destruction in the wind turbine. According to ***Figure 6***, as the average wind speed increases, the unit cost of electricity is almost constant. But by reducing the cost of the cooling, heating and desalination unit, the cost of the multi-generation unit decrease to 6.42%. As shown in ***Figure 7***, the capital return period increases from 1.68 to 3.28 years, and the NPV increases from 30.67 to 35.96 million dollars.

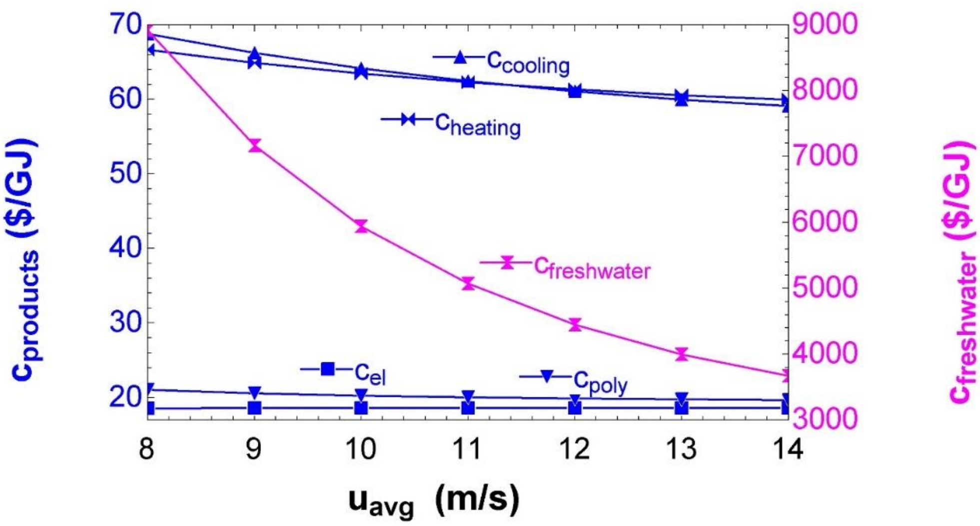

Figure 6. The effect of wind speed variations on the unit cost of products exergy

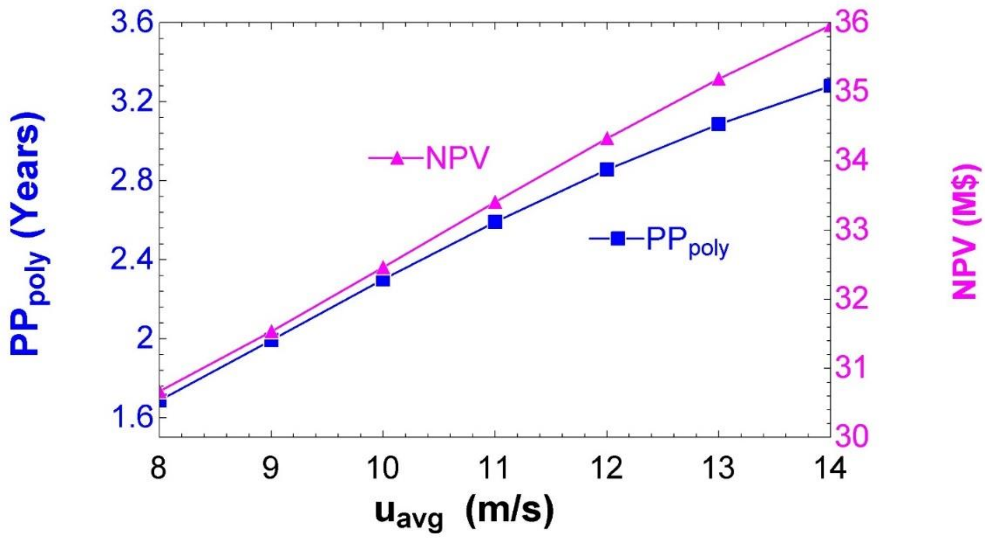

Figure 7. The effects of wind speed variations on economic parameters

The effects of temperature variations in point 1 on the system outputs are shown in ***Figures (8)*** to ***(13).*** Increasing the temperature of point 1 has no effect on the wind turbine. On the other hand, based on ***Figure (8)***, the conservation of energy in the turbine cooling sub-system reduces the water flow, which leads to the reduction of the heat produced in the hot water exchanger. Also, according to ***Figure (9)***, increasing the temperature of point 1, in the generator causes an increase in the flow rate of the absorption cycle and an increase in the production cooling and fresh water.

In terms of exergy efficiency, the effect of reducing the output heat is greater, which results in the reduction of exergy efficiency from 36.77% to 36.75%. On the other hand, exergy destruction decreases in wind turbine and hot water heat exchanger and increases in other system components. As shown in Figure 11, increasing the temperature of point 1 leads to a decrease in the total cost from 2.522 to 4.521 $(\$/hr)$

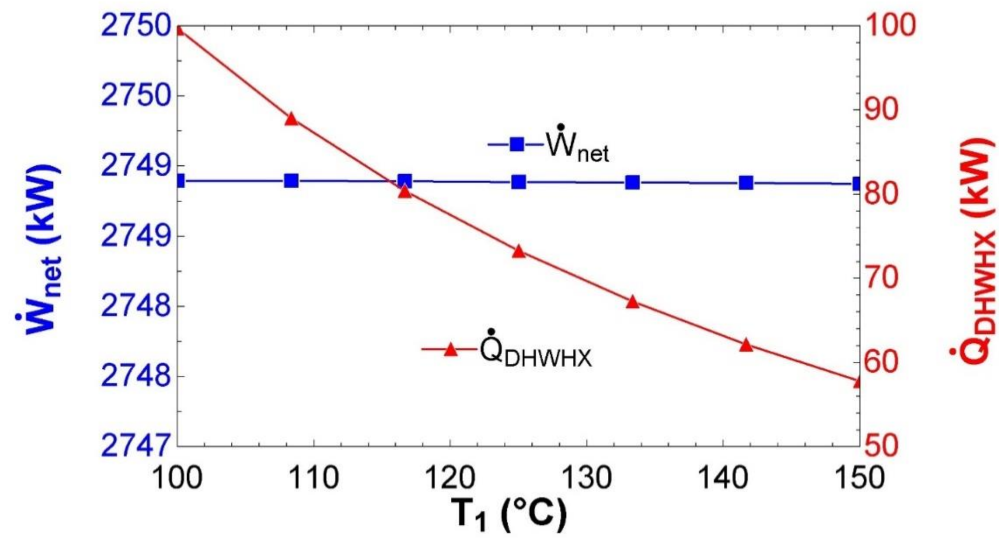

Figure 8. The effects of the temperature variations in point 1 on work and output heat

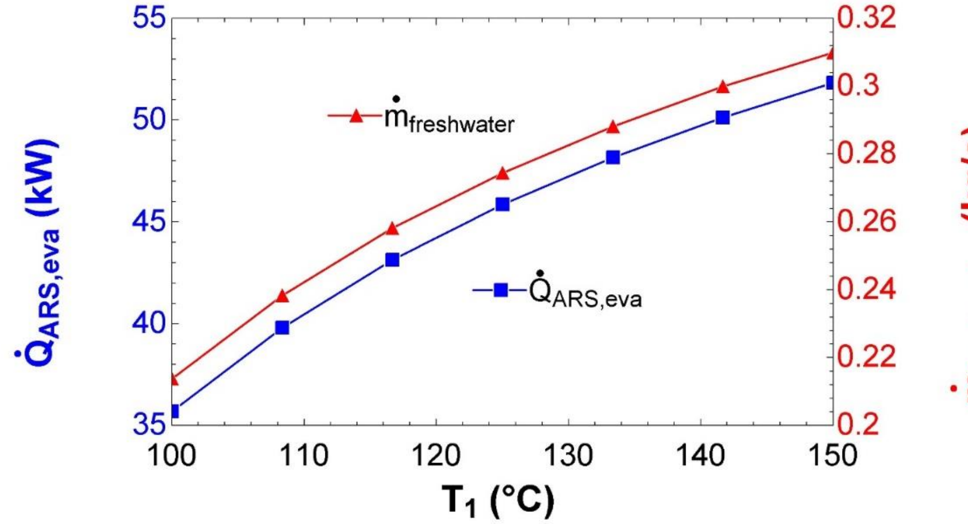

Figure 9. The effect of the temperature variations in point 1 on the cooling and freshwater flowrate

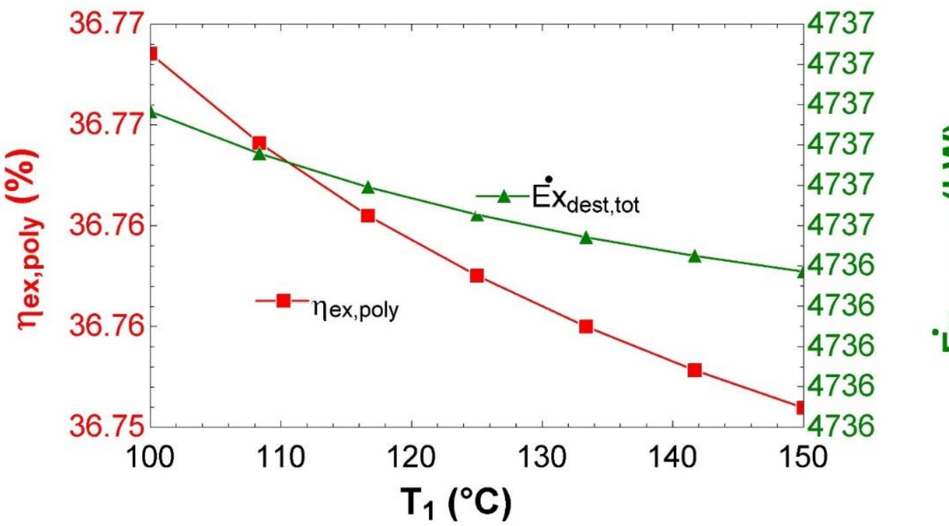

Figure 10. The effects of temperature variations in point 1 on exergy efficiency and total exergy destruction

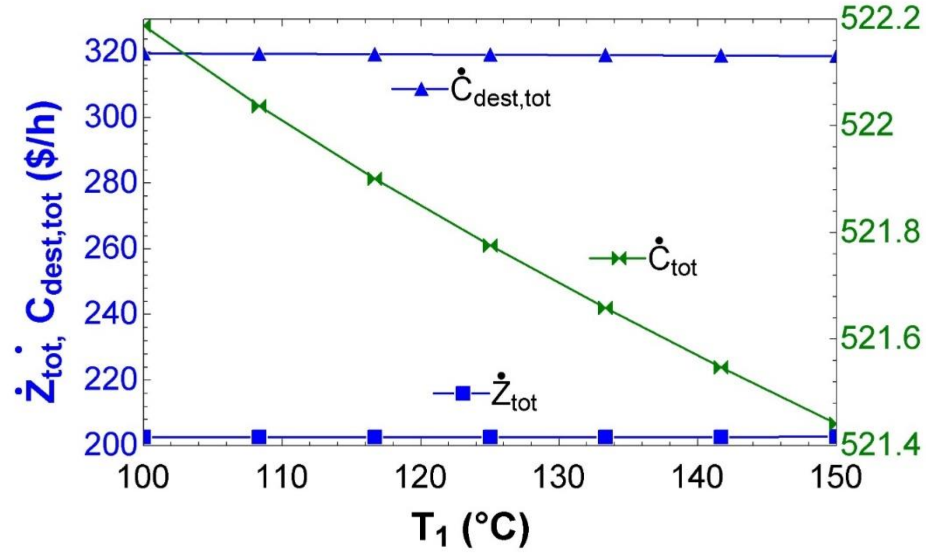

Figure 11. The effects of temperature variations in point 1 on cost rates

According to ***Figure (12)***, the cost of the exergy unit of multi-generation system is reduced slightly from 20.29 to 20.22 dollars per gigajoule due to the reduction in the cost of the cooling unit and fresh water and the increase in the cost of the heating unit. Finally, according to ***Figure (13)***, the capital return period decreases from 2.345 years to 2.266 years and the NPV increases from 31.73 to 33.03 million dollars.

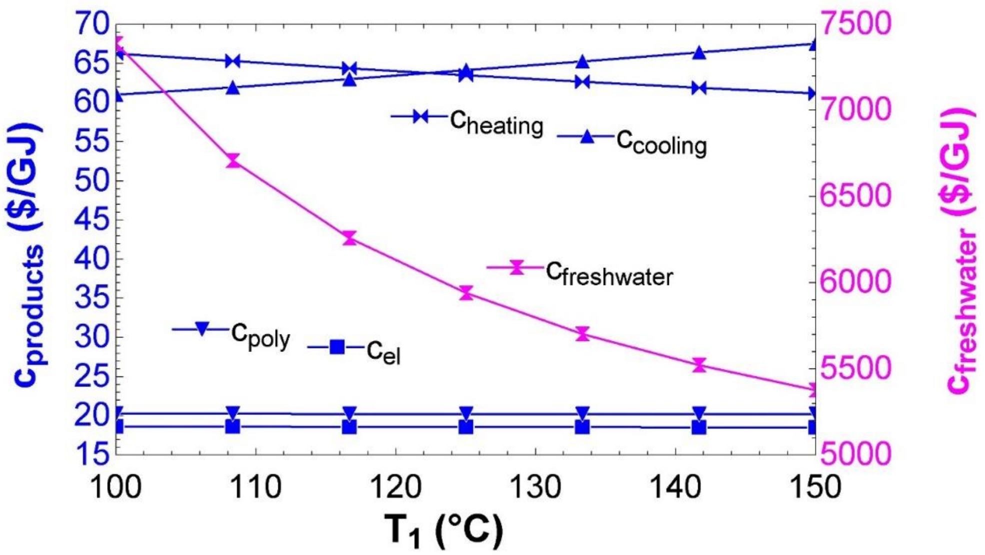

Figure 12. The effect of the temperature variations in point 1 on the costs of the exergy

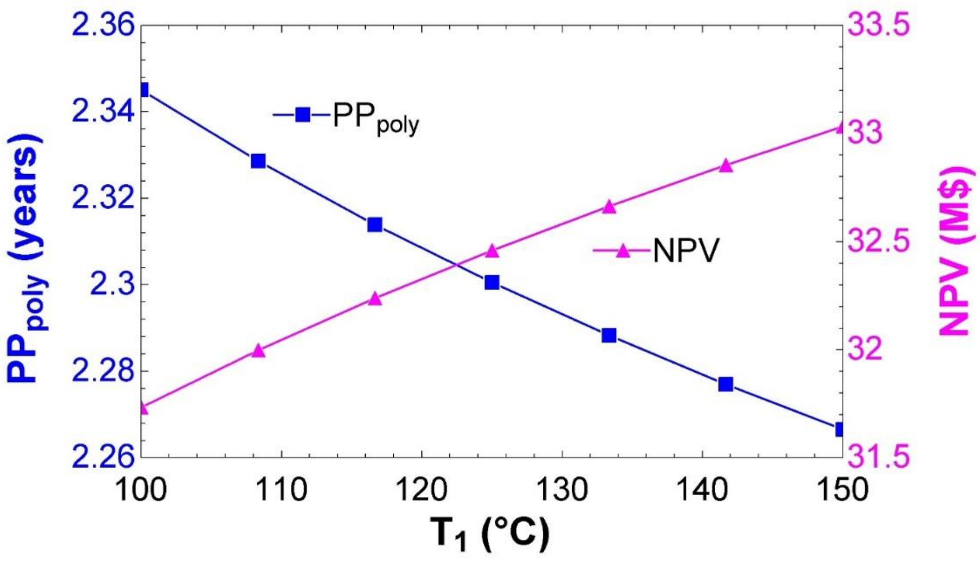

Figure 13. The effect of the temperature variations in point 1 on the economic parameters

5. Conclusion

In the current work, a new multi-generation system was introduced to produce power, heating, cooling, and fresh water, using the maximum recovery of waste wind turbine energy. In order to recover the waste energy of the wind turbine, in the proposed system, an absorptive power-cooling configuration and a heat exchanger for hot water production were used. Then the output power in the absorption system was used to produce fresh water in the reverse osmosis desalination plant. A comprehensive analysis of energy, exergy, economic and exergy was conducted on the system in the form of base state output and parametric analysis. The important results of this research are as follows:

- The proposed system is able to produce 2749 kW of power, 73.25 kW for heating, 45.86 kW for cooling and 0.274 kg/s of fresh water.
- Exergy efficiency of 36.76% was obtained for the proposed system.
- Wind turbine is the most effective component of the system, which includes the major part of exergy destruction and cost rates.
- In the parametric analysis, the change in average wind speed had the greatest effects on the output performance of the system, while the effect of other parameters was less.
- Increasing the heat source temperature at point 1 reduced the exergy efficiency but improved other system parameters.

References

[1]. Wang, Z., Danish, Zhang, B., Wang, B. (2018). Renewable energy consumption, economic growth and human development index in Pakistan: evidence form simultaneous equation model. J. Clean. Prod. 184, 1081e1090.

[2]. Leung, D.Y., Yang, Y. (2012). Wind energy development and its environmental impact: a review. Renew. Sustain. Energy Rev. 16 (1), 1031e1039.

[3]. Nematollahi O, Hajabdollahi Z, Hoghooghi H, Kim KC. (2019) An evaluation of wind turbine waste heat recovery using organic Rankine cycle. Journal of Cleaner Production. 214:705-16.

[4]. Anvari S, Taghavifar H, Parvishi A. (2017) Thermo-economical consideration of Regenerative organic Rankine cycle coupling with the absorption chiller systems incorporated in the trigeneration system. Energy Conversion and Management.148:317-29.

[5]. Zeyghami M, Goswami DY, Stefanakos E. (2015) A review of solar thermo-mechanical refrigeration and cooling methods. Renewable and Sustainable Energy Reviews.51:1428-45.

[6]. Allouhi A, Kousksou T, Jamil A, Bruel P, Mourad Y, Zeraouli Y. (2015) . Solar driven cooling systems: An updated review. Renewable and Sustainable Energy Reviews.

[7]. Ghafoor A, Munir A. (2015). Worldwide overview of solar thermal cooling technologies. Renewable and Sustainable Energy Reviews.43:763-74.

[8]. Khalilzadeh S, Nezhad AH. (2018). Utilization of waste heat of a high-capacity wind turbine in multi effect distillation desalination: Energy ,exergy and thermoeconomic analysis. Desalination.439:119-37.

[9]. Rostamzadeh H, Rostami S. (2020). Performance enhancement of waste heat extraction from generator of a wind turbine for freshwater production via employing various nanofluids. Desalinatio.

[10]. Rostami S, Rostamzadeh H, Fatehi R. (2021). A new wind turbine driven trigeneration system applicable for humid and windy areas, working with various nanofluids. Journal of Cleaner Production.296:126579.

[11]. Khalilzadeh S, Nezhad AH. (2020). Using waste heat of high capacity wind turbines in a novel combined heating, cooling, and power system. Journal of Cleaner Production. 276:123221.

[12]. Makkeh SA, Ahmadi A, Esmaeilion F, Ehyaei M. (2020). Energy, exergy and exergoeconomic optimization of a cogeneration system integrated with parabolic trough collector-wind turbine with desalination. Journal of Cleaner Production.273:123122.

[13]. Li Z, Ehyaei M, Ahmadi A, Jamali D, Kumar R, Abanades S. (2020) Energy, exergy and economic analyses of new coal-fired cogeneration hybrid plant with wind energy resource. Journal of Cleaner Production.269:122331.

[14]. Sezer N, Koç M. (2019). Development and performance assessment of a new integrated solar, wind, and osmotic power system for multi-generation, based on thermodynamic principles. Energy Conversion and Management.188:94-111.

[15]. Ozlu S, Dincer I. (2015). Development and analysis of a solar and wind energy based multi-generation system. Solar Energy.122:1279-95.

[16]. Ventas R, Lecuona A, Zacarías A, Venegas M. (2010). Ammonia-lithium nitrate absorption chiller with an integrated low-pressure compression booster cycle for low driving temperatures. Applied Thermal Engineering.30(11-12):1351-9.

[17]. Bellos E, Tzivanidis C. (2018). Parametric analysis and optimization of a cooling system with ejector-absorption chiller powered by solar parabolic trough collectors. Energy Conversion and Management.168:329-42

[18]. Li C, Besarati S, Goswami Y, Stefanakos E, Chen H. (2013). Reverse osmosis desalination driven by low temperature supercritical organic rankine cycle. Applied energy.102:1071-80.

[19]. Bejan A, Tsatsaronis G, Moran MJ. (1995).Thermal design and optimization: John Wiley & Sons.

[20]. Kerme ED, Chafidz A, Agboola OP, Orfi J, Fakeeha AH, Al-Fatesh AS. (2017).Energetic and exergetic analysis of solar-powered lithium bromide-water absorption cooling system. Journal of Cleaner Production.151:60-73.

[21]. Xia G, Sun Q, Cao X, Wang J, Yu Y, Wang L. (2014) hermodynamic analysis and optimization of a solar-powered transcritical CO2 (carbon dioxide) power cycle for reverse osmosis desalination based on the recovery of cryogenic energy of LNG (liquefied natural gas). Energy.66:643-53.

[22]. Nemati A, Sadeghi M, Yari M. (2017). xergoeconomic analysis and multi-objective optimization of a marine engine waste heat driven RO desalination system integrated with an organic Rankine cycle using zeotropic working fluid. Desalination.422:113-23.

[23]. Nami H, Akrami E. (2017). Analysis of a gas turbine based hybrid system by utilizing energy, exergy and exergoeconomic methodologies for steam, power and hydrogen production. Energy Conversion and Management.143:326-37.

[24]. Zhang M, Chen H, Zoghi M, Habibi H. (2022). Comparison between biogas and pure methane as the fuel of a polygeneration system including a regenerative gas turbine cycle and partial cooling supercritical CO2 Brayton cycle: 4E analysis and tri-objective optimization. Energy.257:124695.

[25]. Ehyaei M, Ahmadi A, Rosen MA. (2019). Energy, exergy, economic and advanced and extended exergy analyses of a wind turbine. Energy conversion and management.183:369-81.