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Energy-exergy efficiencies analyses of a waste-to-power generation system combined with an ammonia-water dilution Rankine cycle

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HIGHLIGHTS

- A Novel waste-to-power generation system combines with an ammonia dilution Rankine cycle is investigated.
- The energy-exergy analysis is utilized to study the effects of critical parameters on the efficiency.
- Engineering Equation Solver (EES) software is used for delivering such accurate outlines.
- Includes providing a solution for calculating the calorific value of waste.

ARTICLE INFO

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Waste incinerator
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ABSTRACT

This research investigates a waste-to-energy system combined with an ammonia dilution Rankine cycle. It uses cooling-energy recycling of dissolved natural-gas to reduce the working-fluid temperature. This project includes providing a solution for calculating the calorific value of waste by proposing a novel waste incineration power generation combined cycle. The energy-exergy analyses is utilized to study the effects of critical parameters on the efficiency and determines the points with higher efficiencies than the conventional steam Rankine cycle. Engineering Equation Solver (EES) software is used for delivering such accurate outlines. The ammonia dilution distillation temperature and the turbine's inlet and outlet pressures are considered vital parameters. The results revealed that mutually energy and exergy efficiencies rise as the ammonia solution's distillation temperature decreases. Besides, by increasing the inlet turbine pressure, the energy and exergy efficiencies increase. In addition, as the output pressure of the turbine increases, the combined cycle's energy efficiency decreases while the exergy efficiency increases.

1. Introduction

The forthcoming knowledge in energy consumption and ecological issues includes an unconventional energy chain system [1] based on the novel methods that recuperates heat loss from various facilities and equipment and effectively recycles energy to the demand unit. The waste ignition power cycles [2] are studied as promising methods for the waste energy consumption and

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Nomenclature

air A
city gas CG
specific heat (kJ/kgK) Cp
Combined Heat & Power CHP
Compressed Natural Gas CNG
Exergy (kJ/kg) e or Ex
garbage G
Gross Heating Value (kJ/kg) GHV
Gas to Liquid GTL
Enthalpy (kJ/kg) h
Heat exchanger HX
Higher Heating Value (kJ/kg) HHV
component i
Incinerator INC
Liquid Natural Gas LNG
LNG cycle L
Lower Heating Value (kJ/kg) LHV
Liquid Petroleum Gas LPG
Moisture and Ash Free MAF
Moisture Free MF
Mass flow rate (kg/s) m
NH3/H2O cycle N
natural gas N.G
Natural Gas Liquids NGLs
reference (ambient) O
Pressure (kPa) P
pump 1 P1
pump 2 P2
Heat transfer rate (kW) Q
Entropy (kJ/kgK) S
Temperature (K) T
turbine 1 TN1
turbine 2 TN2
Theoretical Flame Temperature TFT
thermal th
Work (kW) W
ηEfficiency η η

environmental protection [3]. Indeed, incineration of waste materials converts them into ashes accumulated in the incinerator floor, flue gases, fine particles, and, most importantly, heat, that can be used for the electricity generation. Also, the chimney's pollution of exhaust gases is taken before discharge into the atmosphere [4]. Therefore, waste incineration power cycles are strengthening energy systems [5] for the use of waste energy and environmental protection. In order to optimize the heat generated by burning waste in a waste incineration plant, it is necessary to design a steam generating system following the operating conditions. The design and performance evaluation process includes choosing several design factors that independently affect the converting process (e.g., output steam temperature and pressure of super-heater) [6–8]. It is generally possible to raise the efficiency of the waste incinerator power generation cycles by developing materials that prevent temperance corrosion or by reheating the ultra-hot steam to a temperature above 300 °C by means of a waste heat source from a gas turbine. Modern incinerators may emit small amounts of fine particles, heavy metals, dioxins, and acid gases. Proper management of hazardous toxic ash residues should be taken to install, operate, and dispose of incinerators. Waste incinerators produce electrical efficiencies of between 14 and 28% [9,10]. Due to the use of high temperature in the incinerator, the formation of dioxin and eruption in the exhaust gases is prevented. In this research, in order to use this high temperature's potential energy, the incinerator is equipped with two heat exchangers

Lately, there is a study in analysis of incinerator power generation cycles under various circumstances. Bernt Johnke et al. [11] studied the burning of municipal waste in Europe and presented that the heat generated during the burning process has the highest efficiency just when the combustion is in control and the steam generated can be continuously accessible to supply electricity and heat to the industrial plant. Latthaphonh Kythavone et al. [12] studied a very slight biological Rankine cycle mingled with a municipal solid waste incinerator and designed and constructed a novel cycle for infectious medical waste. Mingzhang Pan et al. [13] proposed a novel organic Rankine cycle and heat pump system in a surplus to energy combined heat and power plant and conducted a multi-objective optimization analysis [14]. Xiaohuang He et al. [15] used flame emission spectroscopy for the size of the temperature and alkali metal concentration in a municipal solid waste incinerators. Jun Liu et al. [16] calculated the efficiency and heat losses under various unit loads of an incinerator-waste heat boiler. Maryam Mohammadi et al. [17] developed and analysis of the performance of a waste-to-energy technology for sustainable energy generation and presented a sustainable solution. Bahram Ghorbani et al. [18] interspersed Kalina/Rankine/Gas/Steam mixed power cycles which used LNG regasification as a cooling source. The energy and Exergy efficiency [19–21] and also irreversibility of the hybrid system were studied. Heng Chen et al. [22] designed a new waste incineration power system integrated with a coal fired power plant and a supercritical CO2 power cycle. They discovered that a waste-to-electricity efficiency is unusually raised up to 8.34 ratio points. M. Sadi et al. [23] proposed a hybrid concentrating solar power-plant and examined the environmental impact and thermodynamic performance of the system. ShimaYazdani et al. [24] used exergy analysis to compare a municipal solid waste incineration power plant and a natural gas power plant and the best Emergy Sustainability Index (ESI) obtained. Aline Bhering Trindade et al. [25] conducted the advanced exergy analysis of an incineration system of municipal solid waste with energy recovery. The main task was cost decrement of a power plant from urban solid waste.

This paper includes providing a solution for calculating the calorific value of waste by proposing a novel waste incinerator power generation combined with an ammonia water dilution Rankine cycle. In order to cool the operating fluid, the cooling energy recycling of LNG is used. Also, natural gas is used as additional waste incinerator fuel. The 2E analyses is utilized to study the effects of significant parameters on the efficiency and determine the points with higher efficiencies. Engineering Equation Solver (EES) software is applied for such detailed analyses. The distillation temperature of the ammonia dilution and the turbine's inlet and outlet pressures are considered vital parameters [26].

2. Calorific value of the waste components

To estimate the calorific rate of the waste component [27,28], the results of their chemical analysis are used with the use of Dulong formula [29]:

$$HHV = 337.8C + 1418.8\left(H - {}^{1}\!/\!{}_{8}\,O\right) + 93S + 23.2N \tag{1}$$

The calorific unit is kj/kg. In the above relation, C: carbon, N: nitrogen, H: hydrogen, O: oxygen and S is the Sulphur. In energy estimation calculations, the following equation is used to obtain the gross calorific value of waste:

$$HHV_{MSW} = \sum_{i=1}^{n}\left(\frac{HHV_i.p_i}{100}\right) \tag{2}$$

HHV_{MSW} indicates the gross calorific value in the wet state. P_i is the percentage of the component in the total waste. HHV_i is the calorific value of the component, n is the number of combustible components in the waste. The user enters the calorific value, or the default values are used. The gross energy or the primary energy of the waste is the mass of the received waste multiplied by its total calorific value. However, in practice only part of this energy is recyclable since a part is used to evaporate the moisture in the waste, one part is used to evaporate the water that is created by the combustion reaction, and some of the heat energy is wasted due to the losses of the furnace. By subtracting these non-useful energies from the primary energy, the useful energy of the waste that can be recycled is obtained. The heat required to evaporate the moisture in the waste is obtained by multiplying the buried heat of evaporation of water by the weight percentage of the component by the percentage of moisture in that component. The concealed heat of water evaporization is assumed to be 2595 [30].

$$\Delta Q_1 = 2595\sum_{i=1}^{n}\frac{p_i}{100}.\frac{w_i}{100} \tag{3}$$

According to the latent heat of water evaporation, the heat required to evaporate this amount of water is obtained from the following equation:

$$\Delta Q_2 = 2595\sum_{i=1}^{n}9.\left(\frac{p_i}{100}\right).\left(\frac{H_i}{100}\right) \tag{4}$$

LHV is obtained by subtracting the amount of energy required to evaporate water from HHV. Therefore, if we subtract the values ΔQ_1 and ΔQ_2 from the HHV_{MSW}, the value (LHV (HHV_{net} or the lower thermal value is obtained, which is considered as useful input energy in thermal calculations.

$$HHV_{net} = HHV_{MSW} - \Delta Q_1 - \Delta Q_2 \tag{5}$$

Therefore, the useful energy of the waste will be calculated from the following equation:

$$E_{net} = m_{MSW}.HHV_{net} \tag{6}$$

In the above relation, m_{MSW} is the mass of waste received per unit (kg/day). If the waste incinerator unit is combined with a power plant boiler and steam turbine, the electricity that can be generated is estimated according to the useful energy.

$$E_{Electric} = E_{net}.\eta_{Electric} \tag{7}$$

We can now calculate the calorific rate of the surplus used in the incinerator of the power generation cycle considered in this research. The composition of waste components in terms of their constituent chemicals [3] are reported as: Humidity 37%, Carbon 27.21%, Oxygen 23.45%, Hydrogen 3.88%, Nitrogen 0.72%, Sulphur 0.03%, Ash 7%, Chlorine 0.71%. In the analysis of this study, these numbers are considered as the useful calorific value of the waste.

In order to provide the computer codes to calculate the calorific value of waste, we have assumed that the physical and/or chemical compositions of the waste component are known. If the physical composition of the waste components [31] is known, the considered inputs are; the mass percentage of moisture in each component (*HHV*), humidity mass percentage of each component (*Wi*), mass percentage in total waste (*Pi*), percentage of hydrogen in each component (*H*), garbage received per day (m_{MSW}). The program must take the above inputs from the user and calculate the $HHV_{net}, \Delta Q_2, \Delta Q_1, HHV_{MSW}$. If the chemical compositions of waste component [32] are known, the considered inputs are; the percentage of total carbon in waste (*C*), percentage of total hydrogen in waste (*H*), percentage of total oxygen in the waste (*O*), percentage of total sulphur in waste (*S*), percentage of total nitrogen in waste (*N*), percentage of total moisture in the waste (*H2O*), garbage received per day (m_{MSW}). The program must take the above inputs from the user and calculate the value using the Dulong formula. Other values are also calculated similarly to those described earlier. It should be noted that the amount of waste received by the system is considered 16,000 kg per day.

3. Cycle description, methodology and energy-exergy analyses

The suggested and considered waste incinerator power cycle with cold LNG energy recovery and city gas use as additional fuel in the incinerator is shown in Fig. 1. However, to have more accurate calculation and better efficiency, the schematic is divided and studied part by part, and then each cycles are compared separately.

Regarding the cycle description, the cycle No.1 represents the power cycle with a waste incinerator, composed of heat exchanger No.1, a waste incinerator, a turbine, a pump, and heat exchanger No.2. The working fluid is water. The waste incineration process heats the air to more than 950 °C, which in turn supplies heat to exchanger No.1 and exits at 183 °C. The working fluid is super-heated to greater than 300 °C and, after reaching a pressure of 3 MPa in HX No.1, enters the turbine. After leaving the turbine, the operating fluid is distilled in HX No.2 and, at a temperature of 40 °C, enters the pump. The operating fluid in the pump is increased in pressure and passes into exchanger No.1 in order to overheat. It is assumed that the outlet composition of the exhaust from the incinerator is air with a temperature of 950 °C. This air then enters a heat exchanger in which the temperature reaches 183 °C. It should be noted that this amount of hot air (183 °C) is itself a good option for CHP work. In this research, some solutions have been considered for using this heat. In the conventional power cycle and in HX No.2, a cooler fluid is used to decrease the temperature of the operating fluid. In order to obtain the energy efficiency in this cycle, the following method is considered:

$$W_{net} = (W_{TN1} - W_{P1}) \tag{8}$$

$$\eta_{th} = \frac{W_{net}}{Q} \tag{9}$$

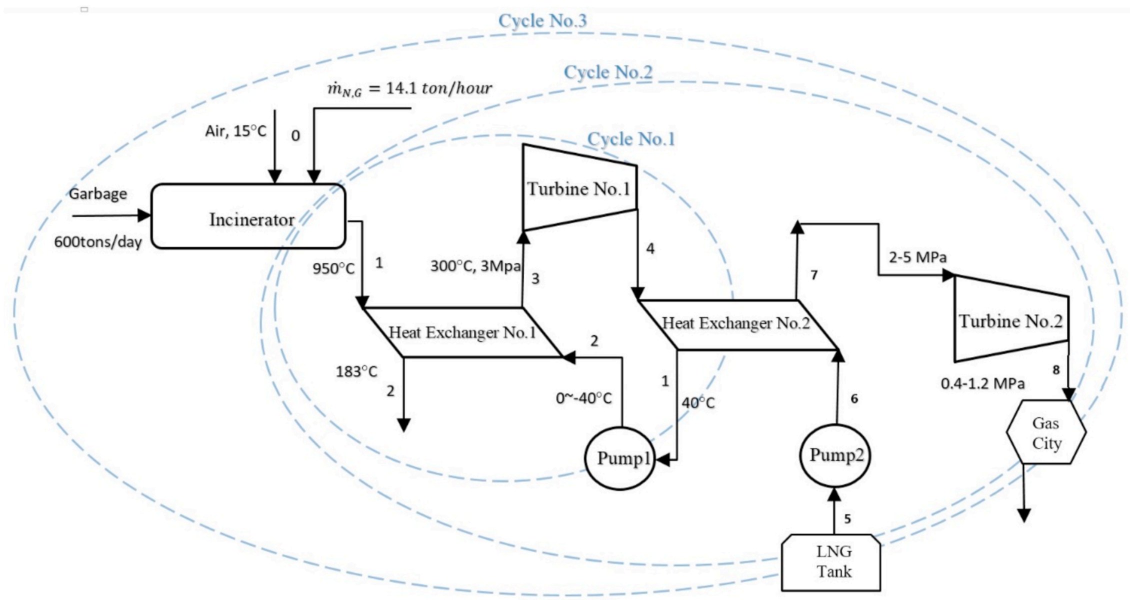

Fig. 1. The combined power cycle using waste incinerator with cold LNG energy recovery and use of city gas produced as additional fuel in the incinerator.

The exergy of the incoming air flow into the cycle is calculated as follows [33]:

$$Ex_Q = Q\left(1 - \frac{T_0}{T_b}\right) \tag{10}$$

$$T_b = \frac{(H_{in} - H_{out})}{S_{in} - S_{out}} \tag{11}$$

In this case, it should be noted that the boundary temperature (T_b) in the HX is a variable, so we use the average thermodynamic temperature. The exergy of the inlet cooling water is as follows:

$$Ex_{Water,in} = (H - H_0) - T_0(S - S_0) = 0kW \tag{12}$$

The amount of work done is equal to the amount of exergy produced and it is equal to $Ex_W = 12569.73kW$. In the case of cooling water exergy, we have:

$$Ex_{Water,out} = (H - H_0) - T_0(S - S_0) \tag{13}$$

$$\eta_{Ex} = \frac{E_{OUT}}{E_{IN}} \tag{14}$$

In cycle No.2, the operating fluid changes to a mixture of water and ammonia. The freezing point temperature of this mixture is below zero. Since in this cycle and in some parts of it, the temperature of the working fluid drops below zero, if the working fluid is water, the cycle will be disrupted. Therefore, the main reason for using a mixture of water and ammonia is its low freezing point.

As can be seen, the liquefied gas is used to cool the turbine outlet. In a separate cycle, liquefied petroleum gas (assumed to be a combination of methane gas) is stored at −162 °C and atmospheric pressure in a storage tank. A pump pumps liquefied gas from the tank to the second HX, and because of the increased pressure, the temperature of the gas changes slightly, however the gas is still liquid. Due to the heat exchange in the HX No.2, the gas temperature increases and the liquefied gas is converted to the vapor phase. In the next step, a turbine is used to reduce the pressure to the normal level, which also produces a significant amount of work. The composition of the operating fluid percentage is (water = 0.3) and (ammonia = 0.7). To calculate the energy efficiency as in cycle No.1, we have:

$$W_{net} = (W_{TN1} - W_{P1}) + (W_{TN2} - W_{P2}) \tag{15}$$

$$\eta_{th} = \frac{(W_{TN1} - W_{P1}) + (W_{TN2} - W_{P2})}{Q} \tag{16}$$

Exergy of the incoming air to the HX No. 1 and the exergy of the liquid gas entering the cycle No. 2 are calculated as fuel. The exergy of the incoming air flow into the cycle is calculated as follows:

$$Ex_Q = Q\left(1 - \frac{T_0}{T_b}\right) \tag{17}$$

$$T_b = \frac{(H_{in} - H_{out})}{S_{in} - S_{out}} \tag{18}$$

In this case, it should be noted that the temperature of the current boundary in the HX is a variable, so we use the average thermodynamic temperature. The exergy of the inlet liquefied gas is as follows:

$$Ex_{LNG,in} = (H - H_0) - T_0(S - S_0) \tag{19}$$

In the case of the outlet gas exergy, we have:

$$Ex_{LNG,out} = (H - H_0) - T_0(S - S_0) \tag{20}$$

$$\eta_{Ex} = \frac{E_{OUT}}{E_{IN}} \tag{21}$$

Cycle No.3, which is structurally similar to cycle No.2, is assumed that the part of generated city gas is returned to the incinerator and consumed there as fuel. As a result, the flow of hot gas produced at the incinerator output increases. Therefore, in other parts of the cycle, we will have changes in temperature, pressure and fluid flow. With these assumptions, an increase in the cycle productive work will not be unexpected. In the simulation performed with software, it is considered that 10% of the produced gas in cycle number 2 is sent to the incinerator and used as fuel. In addition, the mass production of city gas in cycle number 2 is assumed 141,000 kg/h. In the cycle 2, it is assumed that the incinerator is removed and the energy from the combustion is transferred to the air, thus increasing the air temperature. As a result of adding 10% of energy released from the combustion of the city gas to the waste incinerator, the produced hot air increases. If it is assumed the combustion chamber is adiabatic, the temperature of the produced flame, which calls the theoretical flame temperature, is assumed 1825°C. The energy released by combustion is as follows:

$$Q_{added-gas} = \dot{m}_{CH4} * \frac{C_{P,CH4}}{M_f}(T_{TFT} - T_0) \tag{22}$$

In the above relation, it is considered that the efficiency of this combustion is 80%. By assuming that 80% of the heat transfers to the air inside the incinerator, the hot air flow increment rate from the incinerator is obtained as follows:

$$\dot{m}_{A,Excess} = \frac{C_{PA}(T_{A1} - T_{A0})}{Q_{added-gas} * 0.8} \tag{23}$$

To calculate the energy efficiency, we have:

$$W_{net} = (W_{TN1} - W_{P1}) + (W_{TN2} - W_{P2}) \tag{24}$$

$$\eta_{th} = \frac{(W_{TN1} - W_{P1}) + (W_{TN2} - W_{P2})}{Q} \tag{25}$$

The exergy of the input air to HX No.1 and exergy of the liquid gas entering the cycle No.3 are calculated and considered as fuel, and the net exergy of output along with cold natural gas exergy of the second cycle are considered as reaction sections. The exergy of the incoming air flow into the cycle is calculated as follows:

$$Ex_Q = Q\left(1 - \frac{T_0}{T_b}\right) \tag{26}$$

$$T_b = \frac{(H_{in} - H_{out})}{S_{in} - S_{out}} \tag{27}$$

In this case, it should be noted that the temperature of the current boundary in the HX is a variable, so we use the average thermodynamic temperature. The exergy of the inlet liquefied gas is as follows:

$$Ex_{LNG,in} = (H - H_0) - T_0(S - S_0) \tag{28}$$

The amount of work done is equal to the amount of exergy produced and is $Ex_W = 16561.479kW$. Regarding the exergy of the exhaust gas from the cycle:

$$Ex_{LNG,out} = (H - H_0) - T_0(S - S_0) \tag{29}$$

$$\eta_{Ex} = \frac{E_{OUT}}{E_{IN}} \tag{30}$$

The energy and exergy equation details are presented in Table .1 and Table .2 and are conducted on the considered cycle in this research.

The energy and exergy efficiency outcomes are accessible in the Table 3 as follow:

4. Discussions & results

According to the achieved results, changes in energy efficiency and exergy account increase the cycle's efficiency. The rate of increase of energy and exergy efficiency between the second and third round is insignificant, but it should be noted that with increasing gas to the incinerator, the rate of hot fluid flow increases, and therefore the work production increases. Besides, with increasing the

Table 1
Energy balance equations.

Waste incinerator	$\eta_{INC}\dot{m}_G LHV_G + \dot{m}_{N.G} LHV_{N.G} = \dot{m}_A C_{PA}(T_{A1} - T_{A0})$ (31)
Heat exchanger No.1	$\dot{m}_A C_{PA}(T_{A1} - T_{A2}) = \dot{m}_N(h_{N3} - h_{N2})$ (32)
Turbine No. 1	$W_{TN1} = \dot{m}_N(h_{N3} - h_{N4}) = \eta_{TN1}\dot{m}_N(h_{N3} - h'_{N4})$ (33)
Turbine No. 2	$W_{TN2} = \dot{m}_L(h_{L3} - h_{L4}) = \eta_{TN2}\dot{m}_L(h_{L3} - h'_{L4})$ (34)
Heat exchanger No. 2	$\dot{m}_N(h_{N4} - h_{N1}) = \dot{m}_L(h_{L3} - h_{L2})$ (35)
Pump No. 1	$W_{P1} = \dot{m}_N(h_{N2} - h_{N1}) = \frac{1}{\eta_{P1}}\dot{m}_N(h'_{N2} - h_{N1})$ (36)
Pump No. 2	$W_{P2} = \dot{m}_L(h_{L2} - h_{L1}) = \frac{1}{\eta_{P2}}\dot{m}_L(h'_{L2} - h_{L1})$ (37)
City gas	$Q_{CG} = \dot{m}_L(h_{L5} - h_{L4})$ (38)
Overall energy balance	$\eta_{INC}\dot{m}_G LHV_G + \dot{m}_{N.G} LHV_{N.G} = W_{TN1} - W_{P1} + W_{TN2} - W_{P2} - Q_{CG} + \dot{m}_L(h_{L5} - h_{L1}) + \dot{m}_A C_{PA}(T_{A2} - T_{A0})$ (39)
Energy efficiency	$\eta_{th} = \frac{(W_{TN1} - W_{P1}) + (W_{TN2} - W_{P2})}{\dot{m}_G LHV_G + \dot{m}_{N.G} LHV_{N.G}}$ (40)

Table 2
Exergy balance equations.

Exergy of air	$e = C_P\left[(T - T_0) - T_0 \ln\left(\frac{T}{T_0}\right)\right]$ (41)
Exergy of ammonia mixture	$e = (h - h_0) - T_0(s - s_0)$ (42)
Inlet exergy to the cycle	$e_{IN} = e_G + e_{L1} + e_{N.G}$ (43)
Outlet exergy from the cycle	$e_{OUT} = e_{OUT.N} + e_{OUT.L} + e_{OUT.CG}$ (44)
	$e_{OUT.N} = e_{N3} - e_{N4} - (e_{N2} - e_{N1})$ (45)
	$e_{OUT.L} = e_{L3} - e_{L4} - (e_{L2} - e_{L1})$ (46)
	$e_{OUT.CG} = e_{L5}$ (47)
Total exergy balance	eIN = eOUT + eLoss (48)
Exergy efficiency	$\eta_{Ex} = \frac{e_{OUT}}{e_{IN}}$ (49)

Table 3
Energy & Exergy efficiency of the presented cycles.

Cycle No	Energy efficiency	Exergy efficiency
1	%28.1	%29.1
2	%30.1	%31.2
3	30.7%	%32

efficiency of the cycles which converts the liquefied gas to the urban gas, the amount of produced urban gas also increases.

Regarding the sensitivity analysis, we have assumed all the main parameters except one constant, and in a logical interval we have changed the desired parameter and examine the results of these changes in the whole set. The sensitivity analysis is primary directed toward the pump inlet temperature within the first cycle while the flow warmth changes in the range (0 to −40) degrees Celsius. The ammonia concentration of the operating fluid varies from (0.4) to (1). All these changes are made for the purpose of producing the cycle performance diagram. The detailed results are shown in Figs. 2 and 3.

5. Discussions by the path No.1 flow temperature change

1. By decreasing the flow's temperature within path number 1, the exergy and energy efficiency of the cycle both increase, which is the reason why a temperature of −40 °C yields the greatest efficiency among the three studied temperature groups. The reason is that by lowering the flow temperature, higher heat recovery takes place in HX No.2 and increases the energy and exergy efficiency.
2. The highest energy efficiency is perceived in an ammonia concentration of 1 MPa, while the lowest energy efficiency is seen in an ammonia concentration of 0.8 MPa.
3. The highest exergy efficiency is seen in an ammonia concentration of 1 MPa and the lowest exergy efficiency was reported in an ammonia concentration of 0.6 MPa.
4. In some parts, the chart has a fracture in the ascent process, which is due to thermodynamic constraints. These limitations include: steam entering the pump, non-observance of the minimum cross-flow temperature [34] (Cross Pinch).

Besides, we have examined the changes in the cycle efficiency related to the change in inlet gas pressure of the second turbine. The flow pressure of Path 7 changes in the range of 2–5 MPa, yielding the following results. The cycle's remaining basic conditions are

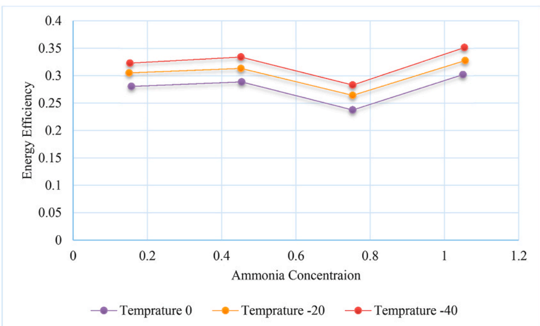

Fig. 2. The temperature effect (T_{N1}) on the energy efficiency of cycle No.3 ($P_{L3} = 4MPa$, $P_{L4} = 0.4MPa$).

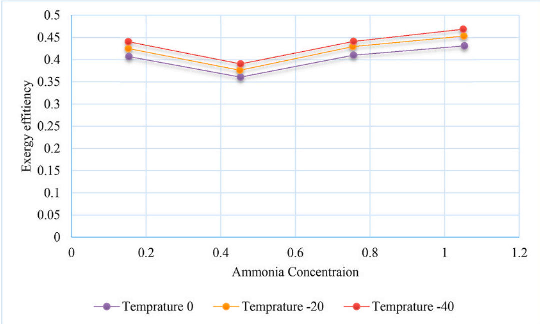

Fig. 3. The effect of temperature (T_{N1}) on the exergy efficiency of cycle No.3 ($P_{L3} = 4MPa$, $P_{L4} = 0.4MPa$).

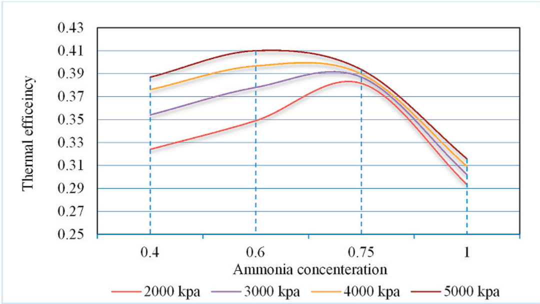

Fig. 4. The pressure P_{L3} outcome on the energy efficiency of cycle No.3,
$T_{N1} = -40^oC$, $P_{L4} = 0.4MPa$

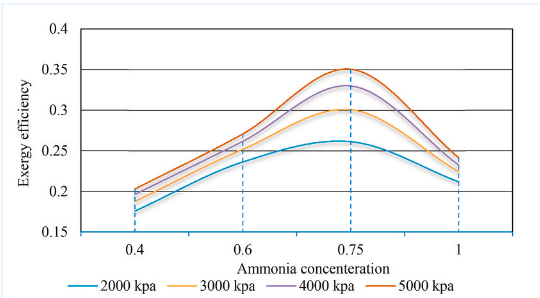

Fig. 5. The pressure P_{L3} outcome on the exergy efficiency of cycle No.3.$T_{N1} = -40^oC$, $P_{L4} = 0.4MPa$

assumed to be unchanging (Figs. 4 and 5).

6. Discussions by the path No.7 flow pressure changes

1) The energy and exergy efficiency increase in the set as a result of increasing the pressure of flow path number 7. This is precisely the reason that the diagrams with 5 MPa pressure yield the highest efficiency of the four studied compression groups. As the inlet gas pressure to the turbine increases, the amount of output exergy increases and, then the efficiency increases.
2) The highest energy efficiency is reported at an ammonia concentration of 0.6 MPa, while the lowest energy efficiency is reported at an ammonia concentration of 1 MPa.
3) The highest exergy efficiency is seen at an ammonia concentration of 0.75 MPa, while the lowest exergy efficiency is seen at an ammonia concentration of 0.4 MPa.
4) In some areas, the chart is in the ascent process, which is due to thermodynamic limitations. These restrictions include: steam entering the pump, non-observance of the minimum cross-flow temperature (Cross Pinch).

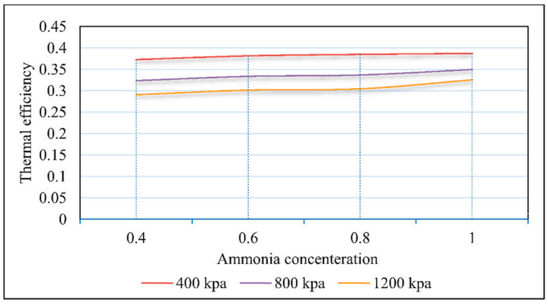

Fig. 6. The pressure P_{L4} outcome on the energy efficiency of set No.3.$T_{N1} = -40^oC$, $P_{L3} = 4MPa$

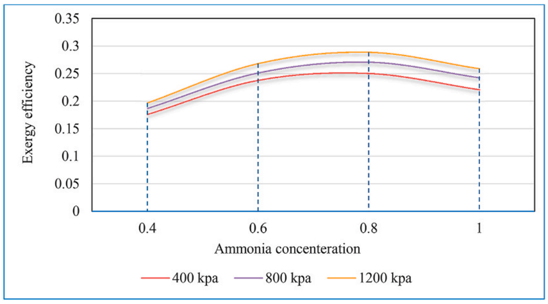

Fig. 7. The pressure P_{L4} outcome on the exergy efficiency of set No.3.$T_{N1} = -40^oC$, $P_{L3} = 4MPa$

We also have examined the changes in cycle efficiency related to the change in exhaust gas pressure from the turbine No.2. The pressure of flow path number 8 changes in the range of 0.4–1.2 MPa, producing the following results. The cycle's remaining basic conditions are assumed to be unchanging (Figs. 6 and 7).

7. Discussions by the path No.8 flow pressure changes

1. The cycle's energy efficiency experiences a decrease after increasing the pressure of flow path number 8, while the cycle's exergy efficiency increases. Increasing the flow pressure No. 8 means producing less work, however in practice, the output exergy of the cycle is increased.
2. The highest energy efficiency is yielded in an ammonia concentration of 1 MPa, and the lowest is reported in an ammonia concentration of 0.4 MPa.
3. The greatest exergy efficiency is yielded at an ammonia concentration of 0.8 MPa, and the lowest exergy efficiency is reported at an ammonia concentration of 0.4 MPa.
4. In some areas, the chart has a fracture in the ascent process, which is due to thermodynamic limitations. These restrictions include: steam entering the pump, non-observance of the minimum cross-flow temperature.

8. Conclusion

This study investigated the energy and exergy efficiencies of a waste incinerator power generation cycle when joint with an ammonia dilution Rankine cycle, a process which uses cold energy retrieval of liquefied natural gas along with the use of natural gas as additional fuel for waste incineration. A combined cycle is proposed and is studied with part by part better analysis illustration, and the energy and exergy efficiencies are thus investigated. Also, by changing crucial parameters such as the ammonia solution, distillation temperature of the ammonia solution, inlet and outlet pressures of turbine No.2, the energy and exergy efficiencies are thereby studied. The resulting data indicate that both energy and exergy efficiencies increase with decreasing distillation temperature of ammonia solution. Energy and exergy efficiencies both increase by cumulating the inlet pressure of turbine No.2. Moreover, as the output turbine pressure No.2 increases, the energy efficiency of the combined cycle experiences a reduction while the exergy efficiency experiences a rise.

As the suggestions for future researches in this area the following items are listed:

1. Thermo-economic analysis, increase the economic justification of the cogeneration systems. Therefore, it is recommended that the thermodynamic analysis be combined with economic analysis.
2. It is possible to raise the mixture temperature of ammonia-water by using materials with high anti-corrosion properties in superheat steam transmission lines, and try to study changes in energy and exergy efficiencies.

3. The suggested combined cycle in this research can be coupled with municipal waste incineration systems. Having a justification plan for the construction of this type of incinerator can turn this project into the beginning of a large national plan to improve energy consumption.
4. The ability to combine hospital incinerators with a cogeneration cycle to supply some of the hospital's energy can be investigated. The important point about these incinerators is that their emissions should not enter the atmosphere due to their high pollution.

Credit authorship contribution statement

Soheil Mohtaram: Conceptualization, Validation, Writing - original draft, Visualization. Yonghui Sun: Validation, resources, review & editing, project administration, Funding acquisition. Mohammad Omidi: Validation, Writing, review, editing and formal analysis. Ji Lin: Writing, review and revision.

Declaration of competing interest

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

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