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Thermodynamic Analysis of the Integrated System that Produces Energy by Gradual Expansion from the Waste Heat of the Solid Waste Facility

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ABSTRACT

The rapid increase in consumer societies leads to a rise in waste facilities. Especially when considering the amount of power used in waste plants and the corresponding waste heat generated, an approach to recover waste heat from these facilities has been proposed. Initially, the waste heat from the solid waste facility was assessed using the Rankine cycle. Subsequently, an Organic Rankine Cycle (ORC) system was integrated into the lower cycle of the steam Rankine cycle. The integrated system was completed by harnessing waste heat from the Rankine steam cycle in the carbon dioxide cycle. These power generation systems are designed with two turbines, each with gradual expansion. Using sub-cycles, 1 kg/s of air at 873.2 K was obtained by evaluating the waste heat. In terms of energy efficiency, it can be observed that the R744 gradual expansion cycle exhibits the highest energy and exergy efficiency. Cooling with water in heat exchangers reduces exhaust efficiency. The highest mass flow requirement is found in the ORC system when the R123 fluid is used. The energy efficiency for the entire system was calculated as 22,4%, and the exergy efficiency for the entire system was calculated as 60.7%. When Exergo Environment Analysis was made, exergy stability factor was found to be %60.7, exergetic sustainability index was found to be 2.66. There is also 370K waste heat available, which is recommended for use in drying units. These calculations were performed using the Engineering Equation Solver (EES) program.

Keywords:
Energy; Exergy; Gradual expansion; Waste heat; Exergo environment analysis

INTRODUCTION

Gas turbines are thermal machines commonly used in applications such as power generation or aircraft propulsion. The principle of gradual expansion (stepwise expansion) is a design feature employed to increase efficiency and optimize the performance of gas turbines. In this context, waste gases at high temperatures and pressures, resulting from the operational principles of gas turbines in industrial processes, energy production facilities, and other similar applications, can be repurposed and converted into energy without harming the environment. Thermal energy obtained by recycling waste heat can be used in various ways, such as hot water production, steam generation, or electricity production, thereby enhancing energy efficiency and striving for more effective resource utilization. This article will explore methods for utilizing waste heat from gas turbines, discussing the advantages, challenges, and application areas of these methods. Thus, the significance of this innovative approach, which contributes to sustainable energy production, will be emphasized. Published studies in the literature, They realized an integrated Organic Rankine Cycle (ORC) to recover the waste heat of exhaust gases in the Afyon Biogas Power Plant, which produces biogas from chicken manure. They stated that waste heat recovery to the power plant greatly increases the performance parameters and economic cost savings values of the system. They calculated the maximum power capacity of the facility supported by the Organic Rankine Cycle as 4828.8 kW. They calculated the energy and exergy efficiencies as 37.4% and 32.1%, respectively, when the power plant operates under optimum operating conditions [1]. In this article, thermodynamic and thermoeconomic analyzes as well as optimization of the organic Rankine cycle (ORC) were carried

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out. The system was applied to an existing solid waste power plant with an installed power of 5.66 MW in order to generate additional power from exhaust gas. The originality of this article is that they made calculations based on the analysis of the possibility of converting the exhaust gas at 566 °C into electricity using the ORC system in the waste-to-energy concept [2]. They calculated the energetic and exergetic analysis of a multiple generation system consisting of a micro gas turbine, an organic Rankine cycle (ORC), an absorption cooler and a water heater [3]. He aimed to increase the efficiency of systems by using thermodynamic cycles from waste heat sources. The thermodynamic results of increasing the efficiency of the system by adding subcycles of the waste heat of a gas turbine to the designed system were examined [4]. Provides the design, analysis and optimization of a new municipal solid waste fueled combined cycle power plant to meet the grid electricity needs of an urban municipality [5]. They implemented a gas turbine cycle model adopting the organic Rankine cycle (ORC) in which supercritical CO_2 (S-CO_2) was used as the working fluid. Thermodynamic analysis of the system used Aspen Plus and EES programs. As a result of thermodynamic analyses, the electricity production capacity, energy and exergy efficiencies of the proposed system were found to be 1530.88 kW, 23.30% and 59.60%, respectively [6]. Thermodynamic and thermoeconomic analyzes as well as optimization of the organic Rankine cycle (ORC) were carried out. The system was adapted to an existing solid waste power plant with an installed power of 5.66 MW in order to generate additional power from exhaust gas [7]. They integrated the organic Rankine cycle (ORC) into a 2 MW natural gas engine to generate electricity by recovering the engine's exhaust heat [8]. In their study, they proposed a thermoeconomic optimization study of a vehicle-mounted ORC unit to recover waste heat from various exhaust gas conditions of a vehicle [9]. They have made a detailed comparison of the potential of ORC and S-CO_2 as bottoms of industrial gas turbines in Combined Heat and Power (CHP) system. They stated that the S- CO_2 dip cycle gives better results than ORC in both electrical and thermal efficiency, since the cycle pressure ratio is not affected by the thermal user temperature in the s-CO_2 solution examined [10]. Thermodynamic analysis of the single-stage, single-expansion S-CO_2/ORC system, which operates at the same lower and upper temperatures as the single-stage, double-expansion S-CO_2/ORC system, was examined [11]. They designed an organic Rankine cycle (ORC) waste heat recovery system with an internal heat exchanger (IHE) to recover waste heat from

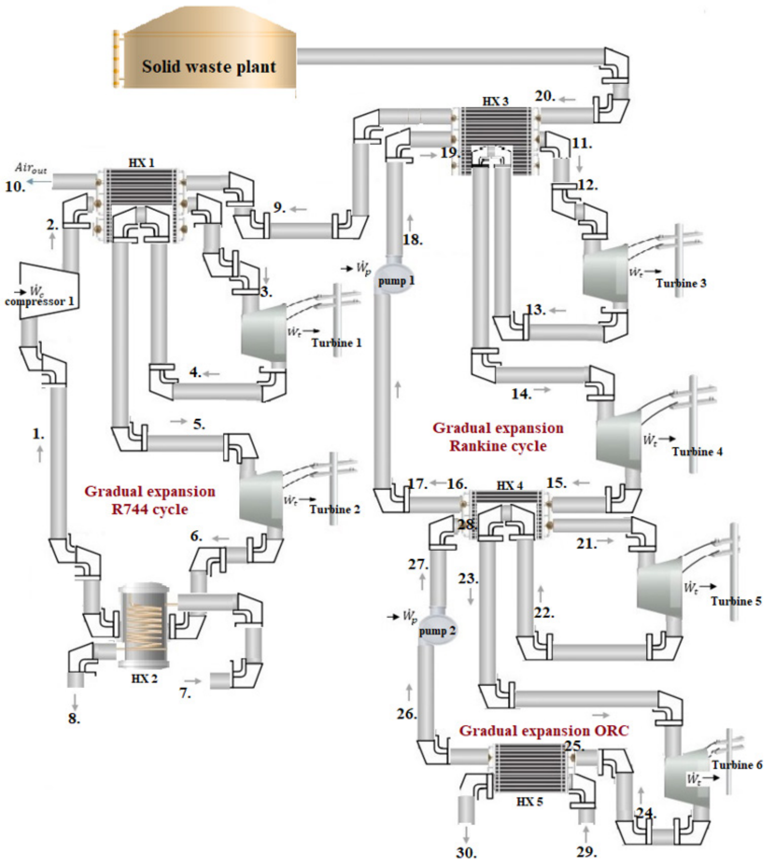

Figure 1. Integrated power generation plant with thermodynamic analysis.

diesel engine exhaust [12]. The main difference between stepwise expansion with reheat (Regenerative Rankine Cycle) and Non-Regenerative Rankine cycle energy cycles is energy recovery. Gradual expansion with reheat improves efficiency by recovering the energy of the steam after expansion and contributes to more electricity production. Stepless expansion, on the other hand, expands without this recovery process and is therefore less energy efficient. Which cycle is preferred depends on the specific requirements and cost factors of the application.

In the design of gradual expansion, the strategy of reheating and raising the fluid's temperature during pressure drops between the blades in gas turbines is employed to enhance thermal efficiency and performance. This process is often referred to as reheat or interstage heating. Consequently, reheating and increasing the fluid's temperature through reheat or intermediate stage heating in gas turbines offer several benefits, including improved thermal efficiency, enhanced performance, a broad operating range, and optimized control. This strategy represents a crucial design method used by engineers to ensure that gas turbines operate with greater effectiveness and efficiency.

MATERIAL AND METHODS

System Description

Fig. 1 shows the schematic view of the integrated power generation facility for which thermodynamic analysis was performed.

In Fig. 1, the waste heat from the solid waste facility was first evaluated in the Rankine cycle. An ORC system has been added to the lower cycle of the steam Rankine cycle. The integrated system was completed by adding the waste heat from the Rankin steam cycle to the carbon dioxide cycle. These power generating systems are designed with two turbines each with gradual expansion.

1. 2→3: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat transfer to carbon dioxide cycle 9→10 HX-1

2. 3→4: Expanding the adiabatic Turbine-1 to generate work

3. 4→5: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat to carbon dioxide cycle HX-1

4. 5→6: Reheating the working fluid to Turbine-2 inlet temperature,

5. 6→1: Heat removal by Heat Exchanger as isobar. HX-2

6. 1→2: Increasing the pressure of the with the adiabatic compressor.

7. 18→12: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat transfer to Rankine cycle 20→9 HX-3

8. 12→13: Expanding the adiabatic Turbine-3 to generate work

9. 13→14: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat to Rankine cycle HX-4

10. 14→15: Reheating the working fluid to Turbine-4 inlet temperature,

11. 15→17: Heat removal by Heat Exchanger as isobar. HX-4

12. 17→18: Increasing the pressure of the saturated liquid with the adiabatic pump.

13. 27→21: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat transfer to ORC 24→20 HX-4

14. 12→13: Expanding the adiabatic Turbine-5 to generate work

15. 13→14: Heat transfer from adiabatic and counterflow Heat Exchanger, waste heat to ORC HX-4

16. 14→15: Reheating the working fluid to Turbine-6 inlet temperature,

17. 15→17: Heat removal by Heat Exchanger as isobar. HX-5

18. 17→18: Increasing the pressure of the saturated liquid with the adiabatic pump.

Assumptions for thermodynamic analysis:

Pure substance is used in the system.

All compression processes in the system are adiabatic.

Pressure drops in system components and pipeline as well as heat transfer over the pipeline were also neglected.

All heat exchangers are counter flow. Heat exchanges are identical.

System operates in steady state.

Gravitational potential energy and kinetic energy are not taken into account.

The isentropic efficiency of compressors, pumps and turbines is 90%

The ambient temperature was taken as 293.2 K.

The exhaust exit temperature of the furnace or heating system used in the solid waste facility was taken as 873.2 K [13].

Energy and exergy analyzes

For steady state in thermodynamic analysis, the basic mass balance equation can be given as follows [14-15,16];

$$\sum \dot{m}_{in} = \sum \dot{m}_{ex} \quad (1)$$

where $\dot{m}$ is the mass flow rate, the in and ex indices represent the inlet and outlet states, respectively. The energy balance is given as:

$$\begin{aligned} &\dot{Q}_{in} + \dot{W}_{in} + \sum_{in} \dot{m}(\mathrm{h} + \frac{v^2}{2} + \mathrm{gz}) \\ &= \dot{Q}_{ex} + \dot{W}_{ex} + \sum_{ex} \dot{m}(\mathrm{h} + \frac{v^2}{2} + \mathrm{gz}) \end{aligned} \quad (2)$$

Here, $\dot{Q}$ is the heat transfer rate, $\dot{W}$ is the power, h is the specific enthalpy, v is the velocity, z is the height, and g is the gravitational acceleration. The entropy balance equation for steady-state conditions is written as:

$$\sum_{in} m_{in} s_{in} + \sum_k \frac{\dot{Q}}{T_k} + \dot{S}_{gen} = \sum_{ex} m_{ex} s_{ex} \quad (3)$$

where s is the specific entropy and $\dot{S}_{gen}$ is the entropy generation rate. The exergy balance equation can be written as:

$$\begin{aligned} &\sum \dot{\mathrm{m}}_{\mathrm{in}}\, ex_{in} + \sum \dot{E}x_{Q,in} + \sum \dot{E}x_{W,in} \\ &= \sum \dot{\mathrm{m}}_{ex}\, ex_{ex} + \sum \dot{E}x_{Q,\mathrm{ex}} + \sum \dot{E}x_{W,\mathrm{ex}} + \dot{E}x_D \end{aligned} \quad (4)$$

The specific flow exergy can be written as:

$$ex = x_{ph} + ex_{ch} + \mathrm{ex}_{\mathrm{p}t} + ex_{kn} \quad (5)$$

The kinetic and potential parts of the exergy are assumed to be negligible. Also, the chemical exergy is assumed to be negligible. The physical or flow exergy (ex_{ph}) is defined as:

$$ex_{ph} = (h - h_o) - T_o(s - s_o) \quad (6)$$

where h and s represent specific enthalpy and entropy, respectively, in the real case. h_o and s_o are enthalpy and entropy at reference medium states, respectively.

Exergy destruction is equal to specific exergy times mass;

$$E_{xd} = ex * m \quad (8)$$

$\dot{E}x_D$, are work-related exergy ratios and are given as:

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

$\dot{E}x_W$, are work-related exergy ratios and are given as:

$$Ex_W = \dot{W} \quad (10)$$

$\dot{E}x_Q$, are the exergy rates related to heat transfer and are given as below.

$$\dot{E}x_Q = (1 - \tfrac{T_o}{T})\dot{\mathrm{Q}} \quad (11)$$

$\dot{E}x_{D,syst.}$ Exergy destruction in the system;

$$\dot{E}x_{D,syst.} = \dot{E}x_{in} + \dot{E}x_{out} \quad (12)$$

What work comes out of the system;

$$\dot{W}net_{out} = \dot{Q}_{in} - \dot{Q}_{out} \quad (13)$$

system thermal efficiency (η);

$$\eta = \frac{energy\,in\,exit\,outputs}{total\,energy\,inlets} \quad (14)$$

The exergy efficiency (ψ) can be defined as follows;

$$\psi = \frac{exergy\ in\ exit\ outputs}{total\ exergy\ inlets} \quad (15)$$

Exergoenvironmental Analysis

fei shows exergoenvironmental impact factor, $\dot{E}x_{D,tot.}$ is total exergy destruction rate, $\dot{E}x_{D,in.}$ is input exergy rate [17],

$$fei = \frac{\dot{E}x_{Dtot.}}{\dot{E}x_{\mathrm{in.}}} \quad (16)$$

Cei is exergoenvironmental impact coefficient, ψ_{ex} represents exergy efficiency of the system,

$$Cei = \frac{1}{\psi_{ex}/100} \quad (17)$$

Φei is exergoenvironmental impact index,

$$\Phi\mathrm{ei} = \mathrm{fei} \times \mathrm{Cei} \quad (18)$$

Φeii represents exergoenvironmental impact improvement,

$$\Phi\mathrm{eii} = \frac{1}{\Phi\mathrm{ei}} \quad (19)$$

fes is the exergy stability factor,

$$\mathrm{fes} = \frac{\dot{E}x_{D,out.}}{\dot{E}x_{D,out.} + \dot{E}x_{D,tot.}} \quad (20)$$

Φest represents exergetic sustainability index.

$$\Phi\mathrm{est} = \mathrm{fes} \times \Phi\mathrm{eii} \quad (21)$$

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Table 1. Thermodynamic properties of the positions of the carbon dioxide cycle.

Location	T [K]	s [kJ/kg.K]	P [bar]	h [kJ/kg]	ex[kj/kg]
1.R744	313.2	-1.08	80	-103.6	214.1
2.R744	374.9	-1.072	190	-70.52	244.7
3.R744	460	-0.7164	190	75.99	287
4.R744	428.2	-0.7108	135	54.51	263.9
5. R744	460	-0.6163	135	96.41	278.8
6.R744	412.7	-0.6072	80	62.73	241.8
7. Water inlet	293.2	0.2965	1	84.01	0
8.Water outlet	303.2	0.4374	1	126	0.7042
9.Air inlet	492.2	7.371	1	495.3	47.97
10.Air outlet	370	7.08	1	370.8	8.675
To. R744	293.2	-0.01403	1	-5.168	0

RESULTS AND DISCUSSION

In one study, modeling ORC as a second stage waste heat recovery system after the primary steam cycle, the system efficiency of the steam cycle was found to be 7.63%. With the addition of ORC, this efficiency increased to 7.81% [18]. R123 has a low boiling point temperature (27.82 °C), making it a preferable fluid for aluminum cycles. They calculated the efficiency of the Basic ORC system as 6.15% and the Regenerative ORC system as 7.98% [19]. In a study optimized for reheated S-CO_2 Brayton cycle, thermal efficiency was found to be 15.2–36.3% and 14.8–35.6% [20]. In examining the Exergo Environmental Analysis, they found the exergy stability factor to be 60% and the exergetic sustainability index to be 2.62.

Thermodynamic properties of the positions of the carbon dioxide cycle are presented in Table 1.

Comparison between regenerative reheat cycle and non-regenerative non-reheat cycle is provided in Fig. 3.

Table 2 compares the thermodynamic results of the system operating with reheating and gradual expansion in the case of stepless and non-reheating operation.

Considering the carbon dioxide cycle, in case of switching from the reheated gradual expansion cycle to the conventional cycle, there will be a 71.4% decrease in exergy efficiency and a 32.5% decrease in energy efficiency. Additionally, the mass flow rate of carbon dioxide will increase by 3.7%, while the mass flow rate of water required for cooling will also increase by 4.3%.

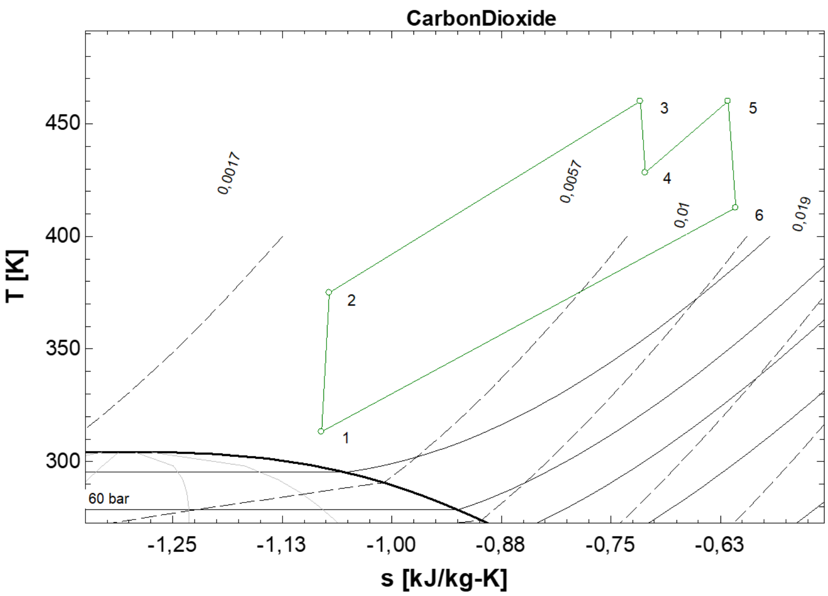

Figure 2. The temperature entropy diagram for the R744 gradual expansion.

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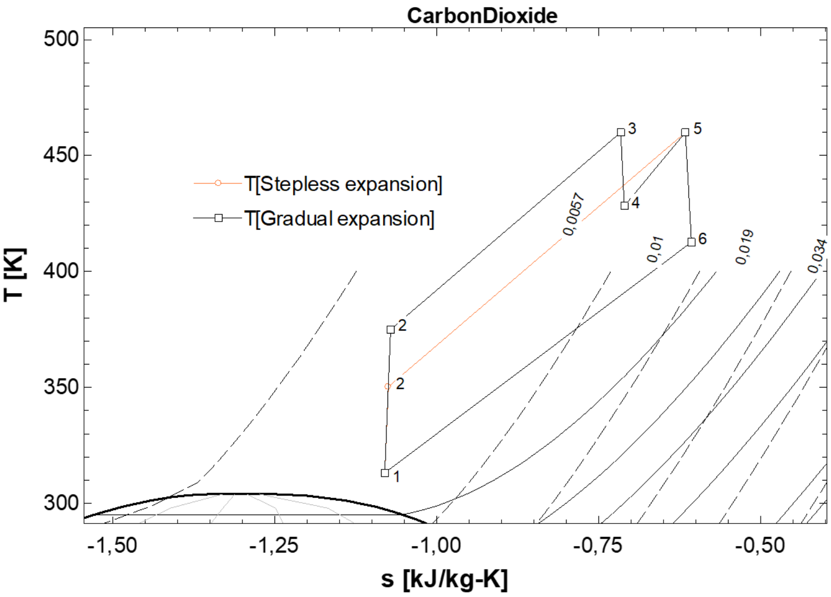

Figure 3. Comparison of regenerative and non-regenerative cycles for carbon dioxide as the working fluid.

Table 2. Thermodynamic consequences of gradual and stepless expansion.

location	R744 Cycle gradual expansion	location	R744 Cycle stepless expansion
Q in [9-10]	124,5 kW	Q in [9-10]	124.5 kW
R744exergy	78.96 %	R744exergy	22.6 %
R744energy	24.9 %	R744energy	16,8 %
Ex comp1	1.71 kW	Ex comp1	1.062 kW
Ex Turb.1	1.08 kW	Ex Turb.1	1.83 kW
Ex Turb.2	1.76 kW	Ex Turb.2	-----
Ex HX1	1.889 kW	Ex HX1	7.036 kW
Ex HX2	20.12 kW	Ex HX2	20.86 kW
HX1	95.19 %	HX1	82.1 %
HX2	10.08 %	HX2	10.03 %
comp1 [1-2]	92.17 %	comp1 [1-2]	91.62 %
Turb.1 [3-4]	92.92 %	Turb.1 [3-4]	92.6 %
Turb.2 [5-6]	92.66 %	Turb.2 [5-6]	-----
m R744	0.6608 kg/s	m R744	0.6857 kg/s
Wcomp.1 [1-2]	21.88 kW	Wcomp.1 [1-2]	1.68 kW
WTurb.1 [3-4]	14.19 kW	WTurb.1 [3-4]	23.09 kW
WTurb.2 [5-6]	22.25 kW	WTurb.2 [5-6]	-----
Q out [1-6]	109.9 kW	Q out [1-4]	114.1 kW
m su	2.616 kg/s	m su [	2.728 kg/s

Table 3. Thermodynamic properties of the positions of the Rankine cycle.

Location	T [K]	s [kJ/kg.K]	P [bar]	h [kJ/kg]	ex[kj/kg]
11. water	443.5	6.662	7.989	2768	817.8
12. water	505.5	6.966	7.989	2912	872.6
13. water	452.3	6.991	4.738	2812	765.4
14. water	505.5	7.227	4.738	2925	809.1
15. water	395.5	7.287	1.487	2716	582.9
16. water	384.2	7.226	1.487	2693	577
17. water	384.2	1.431	1.487	466	49.42
18. water	384.3	1.431	7.989	466.8	50.13
19. water	443.5	2.045	7.989	720.6	123.9
To.water	293.2	6.846	1	293.4	0
20. Air inlet	873.2	7.982	1	903.4	276.7
9.Air outlet	492.2	7.371	1	495.3	47.94

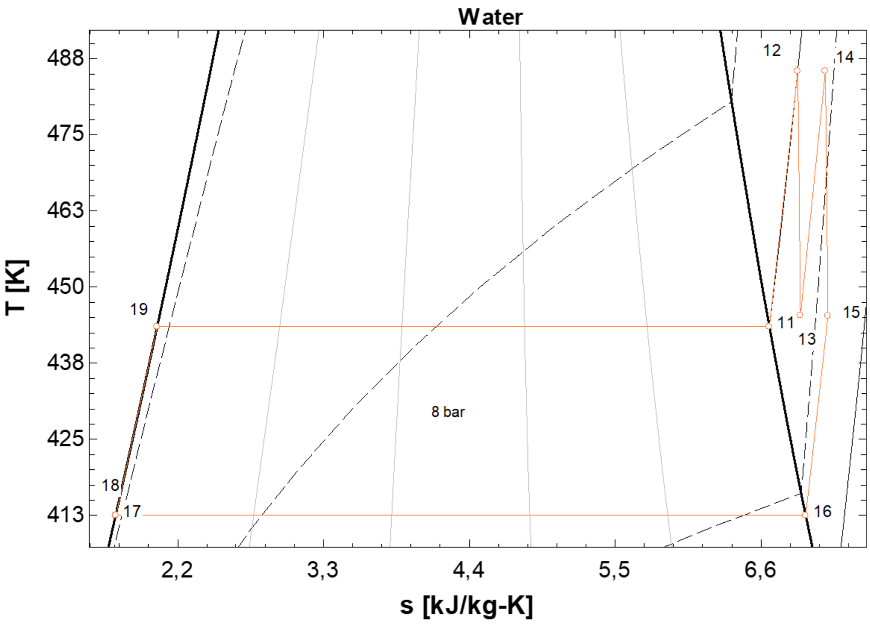

Figure 4. The temperature entropy diagram for the Rankine gradual expansion.

Table 3 gives the thermodynamic values of the positions in the Rankine cycle for Figure 4.

Table 4 gives the thermodynamic properties of the ORC positions.

The temperature entropy diagram for the ORC gradual expansion is given in Fig. 5.

In Table 5, the thermodynamic results of all subcomponents are shown for both the R744 gradual expansion transcritical cycle, the steam gradual expansion Rankine cycle, and the ORC gradual expansion for R123 refrigerant; Heat exchange, energy and exergy analysis of system components, amount of fluid used in the system, power produced in the cycles and power consumed are calculated and presented separately.

The results were obtained by evaluating 1 kg/s air waste heat at 873.2 K with sub-cycles. In terms of energy efficiency, it is seen that the R744 gradual expansion has the highest energy and exergy efficiency. Cooling with water in heat exchangers reduces the exhaust efficiency. The mass flow requirement is highest in ORC, where R123 fluid is used. The energy efficiency for the entire system was calculated as 22.4% and the exergy efficiency for the entire system was

Table 4. Thermodynamic properties of ORC positions.

Location	T [K]	s [kJ/kg.K]	P [bar]	h [kJ/kg]	ex[kj/kg]
21. R123	363.2	1.689	6.259	436.9	35.53
22. R123	348.9	1.691	3.904	429.7	27.61
23. R123	363.2	1.724	3.904	441.4	29.75
24. R123	338.9	1.729	1.549	426.5	13.38
25. R123	313.2	1.67	1.549	407.2	11.34
26. R123	313.2	1.143	1.549	241.9	0.7897
27. R123	313.4	1.143	6.259	242.3	1.123
28. R123	363.2	1.304	6.259	297	8.599
To. R123	293.2	1.074	1	221.1	0
29. Water inlet	293.2	0.2972	1	84.22	0
30. Water outlet	303.2	1.074	1	221.1	0.7042

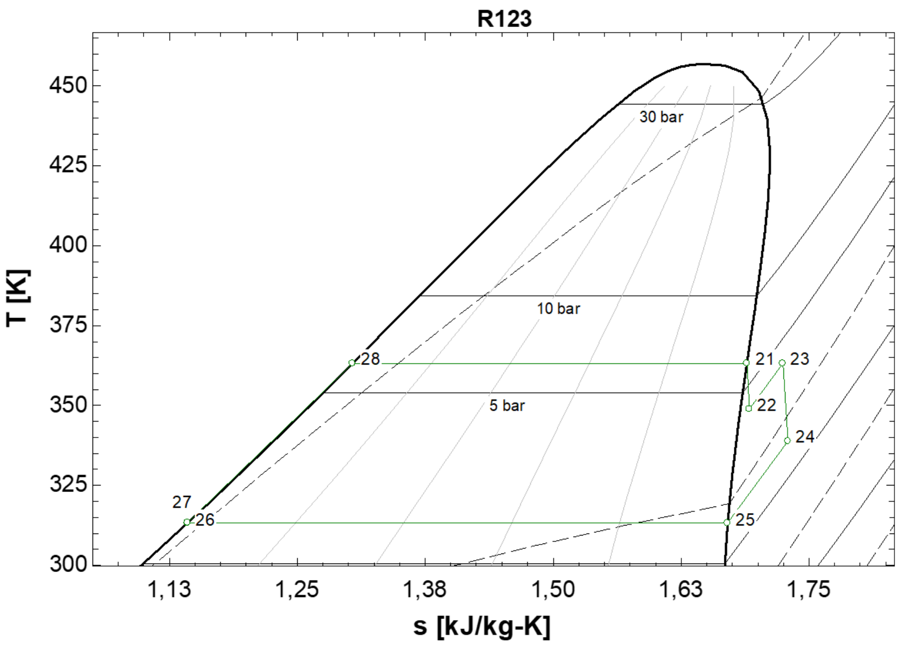

Figure 5. Temperature entropy diagram for ORC.

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calculated as 60.7%. fei shows exergoenvironmental impact factor (0.138) is total exergy destruction rate, (37.049 kW) is input exergy rate(268.02 kW), Cei is exergoenvironmental impact coefficient (1.64), ψ_ex represents exergy efficiency of the system (60.7), Φei is exergoenvironmental impact index (0.227), Φeii represents exergoenvironmental impact improvement (4.39), fes is the exergy stability factor (60.7), Φest represents exergetic sustainability index (2.66). When Exergo Environment Analysis was made, exergy stability factor was found to be %60.7, exergetic sustainability index was found to be 2.66 . There is also 370 K waste heat. It is recommended to use a temperature of 370 K for drying units. Since carbon dioxide has a higher heat conduction coefficient than water, it accelerates heat transfer. At the same time, carbon dioxide has a lower viscosity than water, which allows the fluid to move more easily within the heat exchanger and reduces energy losses. The study results reveal compatible results when compared to other literature studies. The focus of this study will make a significant contribution towards increasing the usability of gradual expansion.

CONCLUSION

The rapid increase in consumer societies means an increase in waste facilities. Processing these waste products in facilities brings with it high energy costs. The gas turbines used in these facilities have serious waste heat. In this study, a thermodynamic proposal has been

Table 5. Thermodynamic results of all subcomponents.

location	H_2O Cycle	location	R123 Cycle	location	R744 Cycle
Q_{in}^{heat} [9-20]	408.1 kW	Q_{in}^{heat} [15-17]	358.9 kW	Q_{in}^{heat} [9-10]	124,5 kW
ψ H2Oexergy	4.3 %	ψ R123exergy	7.3 %	ψ R744exergy	78.96 %
η H2Oenergy	12 %	η R123energy	10.73 %	η R744energy	24.9 %
Ex pump.1	0.01 kW	Ex pump.2	0.06 kW	Ex comp1	1.71 kW
Ex Turb.3	1.15 kW	Ex Turb.5	1.17 kW	Ex Turb.1	1.08 kW
Ex Turb.4	2.78 kW	Ex Turb.6	2.49 kW	Ex Turb.2	1.76 kW
Ex HX3	90.6 kW	Ex HX4	21.52 kW	Ex HX1	1.889 kW
Ex HX4	21.52 kW	Ex HX5	26.67 kW	Ex HX2	20.12 kW
ϕ HX3	60.39 %	ϕ HX4	74.7 %	ϕ HX1	95.19 %
ϕ HX4	74.7 %	ϕ HX5	25.1 %	ϕ HX2	10.08 %
ϕ pump.1 [17-18]	92.37 %	ϕ pump.2 [26-27]	90.68 %	ϕ comp1 [1-2]	92.17 %
ϕ Turbine.3 [12-13]	93.25 %	ϕ Turbine.5 [21-22]	91.5 %	ϕ Turbine.1 [3-4]	92.92 %
ϕ Turbine.4 [14-15]	92.29 %	ϕ Turbine.6 [23-24]	91.26 %	ϕ Turbine.2 [5-6]	92.66 %
m water	0.1595 kg/s	m R123	1.739 kg/s	m R744	0.6608 kg/s
Wpump.1 [17-18]	0.1213 kW	Wpump.2 [26-27]	0.6402 kW	Wcompressor.1 [1-2]	21.88 kW
WTurbinr.3 [12-13]	15.93 kW	WTurbinr.5 [21-22]	12.61 kW	WTurbinr.1 [3-4]	14.19 kW
WTurbinr.4 [14-15]	33.31 kW	WTurbinr.6 [23-24]	25.98 kW	WTurbinr.2 [5-6]	22.25 kW
Q_{in}^{heat} [15-17]	358.9 kW	Q_{in}^{heat} [24-26]	321 kW	Q_{in}^{heat} [1-6]	109.9 kW
m air [20-10]	1 kg/s	m water [29-30]	7.676 kg/s	m water [7-8]	2.616 kg/s

put forward on how to reduce this waste heat into useful energy. Carbon dioxide's higher heat conduction coefficient accelerates heat transfer, while its lower viscosity allows for easier fluid movement within the heat exchanger, reducing energy losses. In terms of energy efficiency, it is seen that the R744 transcritical gradual expansion has the highest energy and exergy efficiency. Water cooling in heat exchangers reduces exergy efficiency. Since the cooling water has a low temperature, the temperature difference is high. This can increase heat transfer efficiency and improve exergy efficiency. The mass flow requirement is highest in ORC, where R123 fluid is used. The energy efficiency for the entire system was calculated as 22.4% and the exergy efficiency for the entire system was calculated as 60.7%. When Exergo Environment Analysis was made, exergy stability factor was found to be %60.7, exergetic sustainability index was found to be 2.66. It is recommended to use a temperature of 370 K for drying units. The study results reveal compatible results when compared to other literature studies.

As a result, reheating and increasing the temperature of the fluid using reheat or intermediate stage heating in gas turbines will provide a number of benefits such as increasing thermal efficiency, improving performance, having a wide operating range and optimizing control. This strategy will be an important design method used by engineers to ensure efficient operation of the facility with systems integrated into waste facilities.

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CONFLICT OF INTEREST

Authors approve that to the best of their knowledge, there is not any conflict of interest or common interest with an institution/organization or a person that may affect the review process of the paper.

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