Energy and Exergy Analyses for Flue Gas Assisted Organic Rankine Cycle

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Effective use of industrial heat waste at low and medium temperatures is seen as one of the solutions that can be used to increase energy efficiency and reduce the problem of environmental pollution. Within the understanding of this framework, the organic Rankine cycle (ORC) maintains to gain attention and further development by researchers and/or manufacturers due to its technical and economical use and credibility. This study presents thermodynamic and economic analyses on flue gas assisted organic Rankine cycle (FGA-ORC) based on both concepts of energy and exergy. The heat source for the FGA-ORC system is the exhaust flue gas of the stenter machine, which is highly used in the textile finishing process. In this study, an optimization investigation has been carried out for a cycle architecture, which converts thermal energy into electrical and/or mechanical energy. The effect of the working parameters of the stenter frame on the performance indicators such as efficiency, performance ratio and economic profit was parametrically analyzed, and the net-work, exergy destruction and efficiency values were determined. The results of these analyses showed that the optimum working parameters of the FGA-ORC system were P_1*=1311 kPa,* P_{mid}*=970 kPa,* $T_{gas,in}$*=140°C, PC=35%,* T_o*=25°C for an exergetic efficiency of 68.86%.*

Keywords: Waste-heat recovery, organic rankine cycle, exergy analyses, flue-gas, energy analyses.

1 Introduction

It is well known that because of the gradual increase in industrialization and as a result an increase in energy consumption, mankind is faced with economic and environmental issues. In order to counter these adverse effects, new energy policies, and environmental regulations have been put forth which pushed countries and businesses to use alternative forms of energy and investigate re-using techniques. To do so, detailed performance evaluations have become a must for energy conversion systems.

One of the toughest challenges engineers face in today's world is to design a system that is cost-effective with low initial capital investment and also has a low running cost while also meeting government requirements on environmental conservation. While energy demands are increasing, our natural resources are dwindling, and it is becoming increasingly crucial to use our recurse more efficiently day by day [1]. A great example of wasted energy would be most industrial heating processes. Generally, the waste energy generated from these processes cannot be recovered very efficiently and low-grade waste energy results in thermal pollution because it is thrown out completely. So, we can safely say that using these wasted low-grade energy sources would benefit the world both economically and environmentally.

It is known that using traditional steam power cycles with low-grade waste heat as an energy source is not feasible because of their low efficiency with low-grade heat sources and high cost. But the organic Rankine cycle is said to be a good way to utilize the wasted low-grade heat energy [2]. Studies were done to investigate the effects of different working fluids on the waste recovery organic Rankine cycle's performance by Wei et al [3], Liu et al. [4] and Hung [5]. The organic Rankine cycle converts waste heat into power by using organic materials, which have low boiling temperatures, as working fluids and because of its simple construction, ability to be easily implemented to different working conditions, its size and its power generating capability makes the organic Rankine cycle a great choice for producing heat and power using low-grade waste heat sources [6]. There is a great deal of research in the literature when it comes to the organic Rankine Cycle and yet this area is still very active due to its inclusion of better energy use. This is supported by the studies of Velez et al. [7], Tchanche et al. [8], Wang et al. [9], Chen et al. [10], Papadopoulos et al. [11], Desai and Bandyopadhyay [12] and Saleh et al. [13].

A number of review articles have been published to help navigate future endeavors on the subject of the organic Rankine cycle [14,15]. Yamamoto et al. examined the closed type organic Rankine cycles performance and characteristics by using HCFC-123 and water as working fluids [16]. From this study it can be seen that the organic Rankine cycle could be implemented with low-grade heat sources effectively and that HCFC-123 significantly increases the organic Rankine cycles performance. In a study by Hung et al., cryogens such as benzene, ammonia, R11, R12, R134a and R113 were used as working fluids for the organic Rankine cycle and the efficiencies were parametrically analysed and compared [17]. They concluded that the beast chois for recovering low-temperature heat was isentropic fluids. Etemoglu proved that the operation conditions and the working fluid could change the systems performance drastically [18]. Kermani et al. presented an optimization technique to obtain the optimum thermal architecture by changing the operating conditions and the working fluid [19]. A thermal analyses was done to investigate the effects of different fluid on the ORC and steam turbines using geothermal energy as the heat source by Akbay et al. [20]. Yuksel presented a thermal and performance evaluation of an integrated geothermal energy based multigeneration plant which includes hydrogen, electricity, hot water, drying, cooling and heating [21].

A heat recovery and working fluid testing system was constructed by Panesar et al. [22] to investigate a wide range of realistic gaseous sources such as the direct utilization of High Temperature (HT) exhaust gases.

To obtain a more optimum and cost-effective ORC system, the continuation of these researches is essential. This study presents thermodynamic and economic analyses on flue gas assisted organic Rankine cycle (FGA-ORC) based on both concepts of energy and exergy. The heat source for the FGA-ORC system is the exhaust flue gas of the stenter frame, which is highly used in the textile finishing process. In this study, an optimization investigation has been carried out to find the best cycle conditions for a system, which converts thermal energy into electrical and/or mechanical energy. This study shows a straightforward method to calculate the energy and exergy analyses of the FGA-ORC system and assess the economic contribution of the system and the exergy destruction of each component of the system using this method, which is based on the first and second law of thermodynamics, thus showing where the most effective improvements can be applied. ORC systems should provide a more efficient energy utilization for the needs of specific operating conditions of the industrial processes. Therefore, the new thermal architecture of the cycle is designed to improve the work-output of the ORC system (see Figure 1). In this new thermal architecture, the effects of the turbine-to-heater mass flow rate and the internal heat exchanger on the system performance parameters are investigated. In this study, an optimization technique is presented to get the optimal operating conditions for a new thermal architecture with a specific working fluid.

2 Material and Methods

2.1 Theoretical Analyses

The stenter-frame, which is a continuous dryer regularly used in the textile industry, is an apt choice as a new energy source for recovery because, at the end of the stenter-frames drying process, a considerable amount of low-temperature flue-gas is released to the atmosphere.

The maximum possible amount of useful work that can be obtained from a system is the work potential of the energy contained in that very system at a specified state. The term that best describes this is exergy. Exergy is generated from a wide variety of flows (mass, heat, work, etc.) flowing through the system.) it quantitatively represents useful energy or the ability to do or receive work - work content [23,24]. Thus, making it a very useful tool for designing, analyzing and classifying thermal systems. The specific flow exergy of a fluid at any state, e, ignoring the changes in kinetic and potential energy, can be calculated with Eq. (1).

$$e = h - h_o - T_o(s - s_o) \tag{1}$$

where h is the specific enthalpy (kJ/kg), s is the specific entropy (kJ/kgK), T is the temperature (K) and o is the dead state conditions. The exergy rate, $\dot{E}$, is calculated by multiplying the specific exergy, e, with the fluids mass flow rate, $\dot{m}$.

$$\dot{E} = \dot{m}e \tag{2}$$

The entropy generation rate can be calculated with Eq. (3).

$$\underbrace{\dot{S_{in}} - \dot{S_{out}}}_{\text{Net entropy transfer rate}} = \underbrace{\dot{S_{gen}}}_{\text{Entropy generation rate}} \tag{3}$$

and for a steady-flow system, the entropy generation rate, $\dot{s}_{gen}$, can be calculated from Eq. (4).

$$\dot{s}_{gen} = \sum \dot{m}_{out}s_{out} - \sum \dot{m}_{in}s_{out} - \sum \frac{\dot{Q}}{T} \tag{4}$$

where $\dot{q}$ is the heat transfer rate. The exergy destruction rate (i.e., rate of irreversibility), $\dot{I}$ for the steady state can be expressed by the following equation based on the general exergy ratio equilibrium for the open system:

$$\dot{I} = T_o\dot{S}_{gen} \tag{5}$$

where T_o is the dead state temperature. The performance ratio of the ith component of the system, PR_i, is calculated by comparing the exergy destruction rate of that component to the total exergy destruction, $\dot{I}_{TOTAL}$, which is calculated by adding up the exergy destruction rate of each device.

$$PR_i = \frac{\dot{I}_i}{\dot{I}_{TOTAL}} \tag{6}$$

The first law efficiency by itself is not enough to measure the performance of thermal devises. To compensate for this deficiency the second law efficiency is used. The second law efficiency, η_{II}, also known as the exergetic efficiency, is the ratio of the device's performance to that same device's performance under reversible conditions for the exact final state [23].

$$\eta_{II} = \frac{\eta}{\eta_{rev}} \tag{7}$$

whereη is the thermal efficiency of the FGA-ORG with the irreversibilities taken into account and η_{rev} is the highest possible thermal efficiency under the same conditions. The work-done and/or the heat transfer rate can be calculated based on the first law of thermodynamics for the turbine, pumps, evaporator, condenser, heat exchanger and the heater which are steady-flow engineering devices. For the $\dot{i}_{th}$ apparatus of the system,

$$Q_i - W_i = \sum H_{i,outlet} - \sum H_{i,inlet} \tag{8}$$

and

$$\sum \dot{m}_{i,inlet} = \sum \dot{m}_{i,outlet} \tag{9}$$

The economical profit of the FGA-ORC for one hour operation can be expressed as the total price of the net electricity generated in the system. The assumptions made for the FGA-ORC system's first and second law analyses are as follows [25]:

1. All systems are steady-state and steady flow with negligible effects of kinetic and potential energy with no chemical or nuclear reactions.
2. The positive directions for the heat and work transfers for the system are, to the system and from the system respectively.
3. The turbine operations adiabatic efficiency is 80%.
4. The circulation pumps adiabatic efficiencies are 85%.
5. Pressure drops are neglected.
6. The condenser's output temperature is $25°C$ and the quality is 0.
7. The dead state temperature is $T_o = 20°C$ and the dead state pressure is P_o=100 kPa.

The energy of the stenter-frame's flue gas, which is the waste heat source, is transferred to the cycle's working fluid at the evaporator. Thus, the required heat energy is provided to the system. At the turbine outlet, some of the working fluid is dispatched to the heater and some to the heat exchanger. This approach makes it possible to examine the effect of the amount of fluid separated from the turbine to the heater on the efficiency. As a result, system efficiency is increased by a heat exchanger and a heater located on two separate channels at the turbine outlet. For this cycle architecture, the goal of highest efficiency is achieved, depending on equipment layout and parameters (Figure 1).

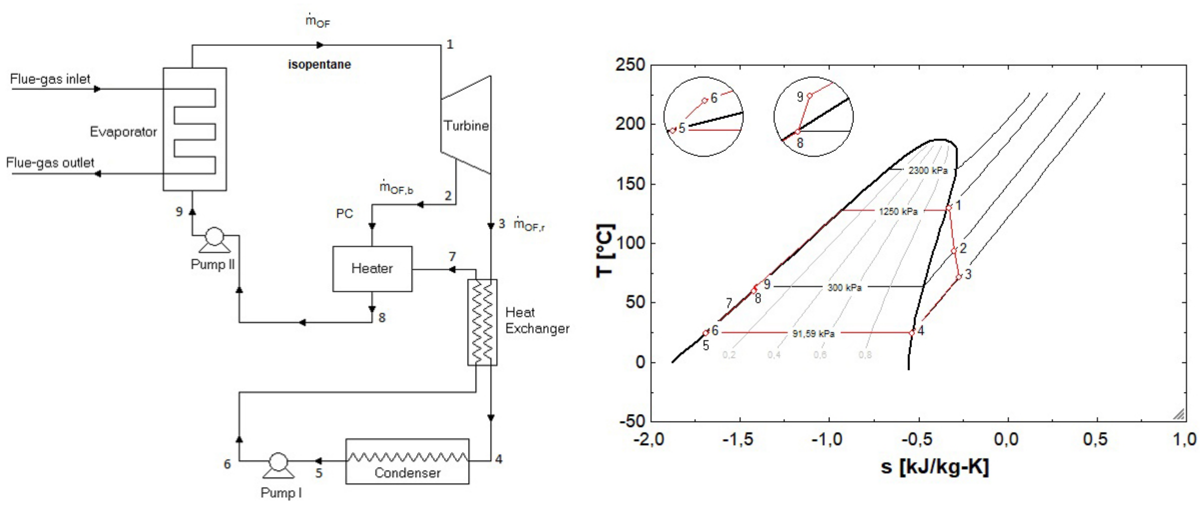


Figure 1: *Schematic illustration of the FGA-ORC system.*

3 Results and Discussion

The optimization for this thermal architecture is based on our codes on Engineering Equation Solver, EES. The present study can be used to predict the effects of many parameters such as temperature, pressure, mass flow rate, and dead state conditions on the performance of this FGA-ORC system. Since the irreversibility value of the heat exchanger is negative at 1312 kPa turbine inlet pressure under the conditions discussed, the thermodynamic limit is exceeded and the system cannot operate. In other words, the highest inlet pressure that can be reached is 1311 kPa. The evaporator capacity is constant so we can calculate the flow rate of the organic fluid used in the system for every different pressure value from the energy equilibrium of the evaporator. Thus, allowing us to investigate each pressure condition for the constant evaporator capacity. In these analyses, the highest irreversibility calculated is at the evaporator (37,58%) for P=1311 kPa. The effect of the evaporator, heater and turbine, which are the three equipment with the highest irreversibility, on the total irreversibility was calculated as 91%. These equipment constitute the greatest irreversibility. As seen in these analyses, as the turbine inlet pressure increases from 800 kPa to 1311 kPa, the first law efficiency increases, and the profit from electricity generation increases by approximately 23%. At the same time, as the turbine inlet pressure increases from 800 kPa to 1311 kPa, as expected the second law efficiency of the system increases because of the decrease in the total irreversibility value of the system. The results are consistent with each other (see Figure 2).

This is the first of the graphs prepared for the effect of the cycle mid-pressure. When 300, 400, 500 kPa are used as the mid pressure, the 1st law efficiency, the 2nd law efficiency, profit increases, and the total irreversibility decreases in all cases when the mid pressure increases from 300 to 500 (Figure 3a).

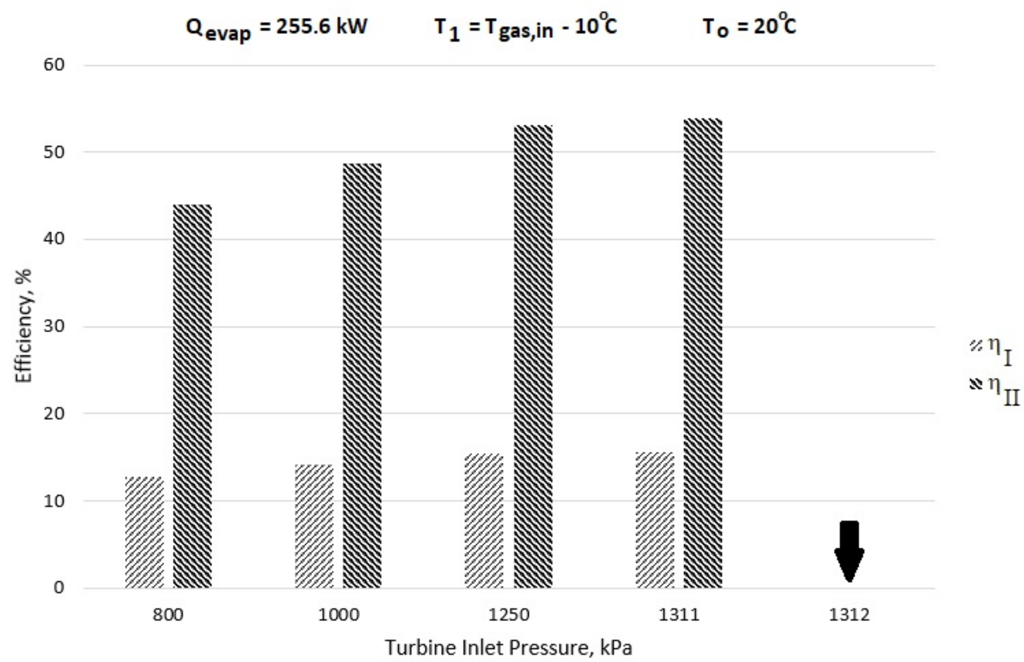
Figure 3b was prepared to find the best possible "cycle mid pressure" for a turbine inlet pressure of 1311 kPa. The mid pressure can be increased up to a maximum of 970 kPa. At 971 kPa, the Heater can operate under current thermodynamic conditions only if it receives extra energy from the outside. In other words, the system will not operate at 971 kPa mid pressure. Again, it is seen from the graph that for P_1=1311 kPa, as the cycle mid pressure increases, the efficiency of the first law of thermodynamics increases, and the profit that can be obtained increases accordingly. Similarly, as the cycle mid pressure increases, the total irreversibility decreases and accordingly the second law efficiency increases. These increases are limited by the thermodynamic operating condition of the heater.

PC represents the percentage of the working fluid sent from the turbine to the heater. When it is reduced from 50%, the efficiency increases up to 35%. The heater provides heat transfer to the outside (i.e., negative value) as it should be. However, at 34%, it becomes necessary to give additional heat from outside for the heater to work, which is against the working conditions of the cycle. Thus, PC reaches the limit value of 35% (Figure 4).

The maximum possible operation of a system in a given situation depends on the conditions of the dead state, as well as on the thermo-physical properties of the system. In other words, exergy is a property of the combination of the system and its environment. Thus, the results of the energy and exergy analyses of the calculated performance indicators of the FGA-ORC are sensitive to changes in the above-mentioned characteristics. The results obtained from the analyses with an increase in the dead state temperature show that the exergy efficiency increases (see Fig. 5). But thermodynamic limits should always be considered because FGA-ORC cannot be operated over the dead-state temperature of 26^{o}C due to I-Condenser<0. Figure 5 also shows the irreversibility values in some equipment of the cycle.

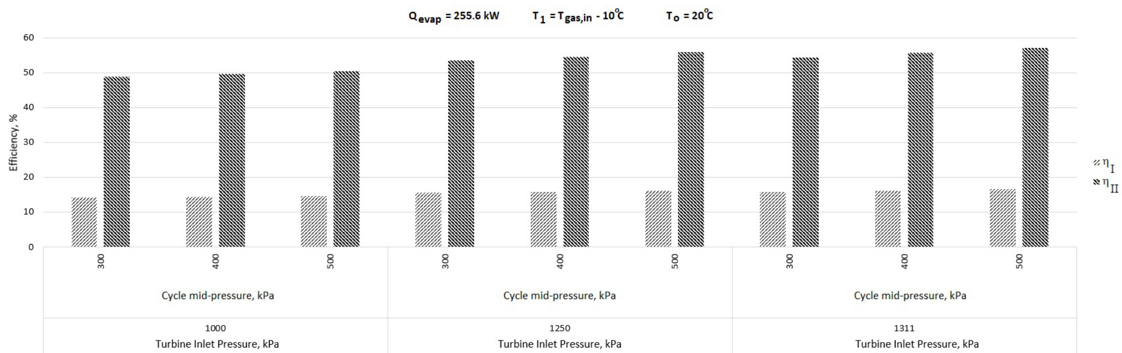
Figure 6 shows the PR values of the FGA-ORC equipment for the optimum condition. In the optimum case, the total irreversibility value of the system is 22.31 kW, and the calculations show that the turbine, evaporator, and heater have the highest PR ratios.

Under the optimum conditions, which are P_1=1311 kPa, P_{mid}=970 kPa, T_{gas},in=140°C, PC=35%, T_o=25°C, the cycles efficiencies are η_I=0.1911 and η_{II}=0.6888 and the exergy destruction rates (I) for each component under these optimum conditions are shown in Table 1.



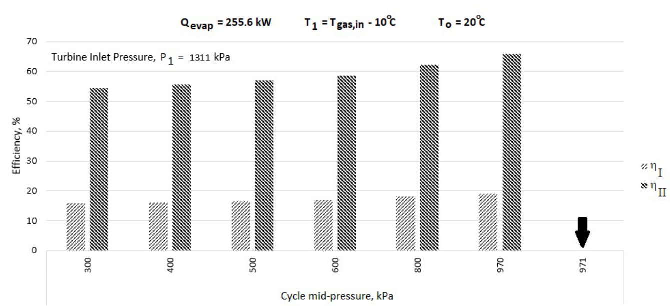
P, kPa	800	1000	1250	1311	1312
m_OF, kg/s	0.5671	0.577	0.5928	0.5975	-
Profit_E, $/h	3	3.318	3.617	3.677	-
I-Evaporator, kW	19.71	16.48	13.49	12.89	-
I-Turbine, kW	6.797	7.692	8.633	8.845	-
I-Heat Exchanger, kW	1.076	0.789	0.5265	0.4762	I < 0
I-Condenser, kW	2.131	2.168	2.228	2.245	-
I-Pump I, kW	0.01647	0.01676	0.01722	0.01736	-
I-Heater, kW	11.85	10.95	9.917	9.661	-
I-Pump II, kW	0.08404	0.1166	0.1595	0.1706	-
PR-Evaporator, %	47.3	43.12	38.57	37.58	-
PR-Turbine, %	16.31	20.13	24.69	25.78	-
PR-Heat Exchanger, %	2.583	2.065	1.506	1.388	-
PR-Condenser, %	5.115	5.675	6.371	6.545	-
PR-Pump I, %	0.03953	0.04386	0.04924	0.05059	-
PR-Heater, %	28.45	28.67	28.36	28.16	-
PR-Pump II, %	0.2017	0.305	0.4562	0.4972	-

Figure 2: *Effect of turbine inlet pressure on the FGA-ORC system.*



Turbine Inlet Pressure, kPa	1000			1250			1311		
Cycle mid-pressure, kPa	300	400	500	300	400	500	300	400	500
η_I	14.16	14.36	14.62	15.5	15.82	16.2	15.77	16.12	16.53
η_{II}	48.91	49,58	50.47	53.51	54.62	55.93	54.45	55.66	57.07
m_OF, kg/s	0.5991	0.6409	0.6812	0.6161	0.6605	0.7034	0.6212	0.6663	0.7101
Profit_E, $/h	3.334	3.38	3.441	3.648	3.723	3,813	3.712	3.794	3,89
I-TOTAL, kW	38.04	37.54	36.88	34.63	33.81	32.84	33.94	33.04	32

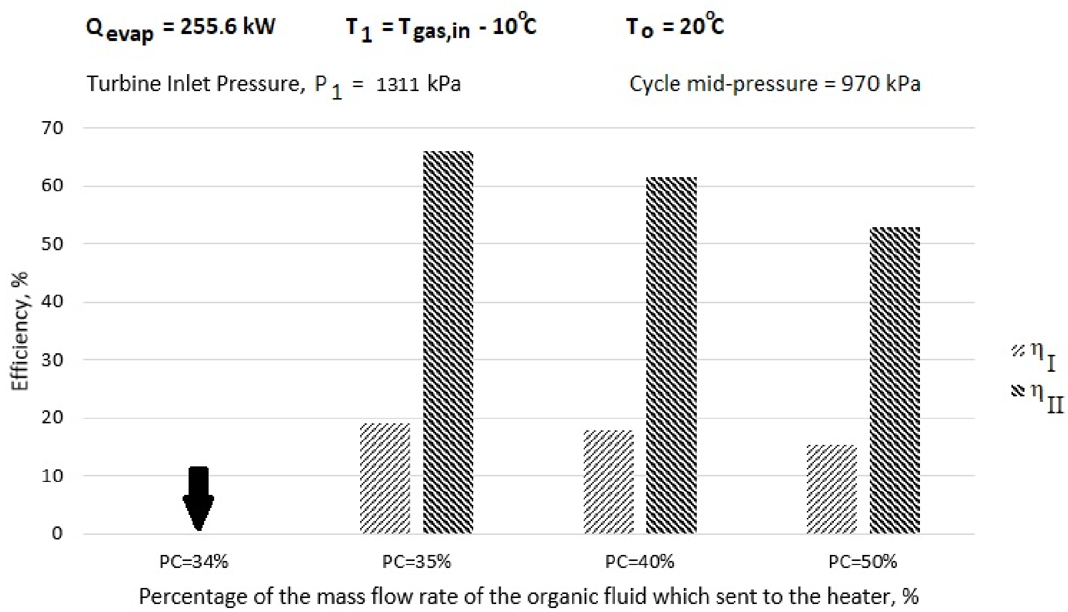
(a)



Turbine Inlet Pressure, kPa	1311						
Cycle mid-pressure, kPa	300	400	500	600	800	970	971
η_I	15.77	16.12	16.53	16.99	18.05	19.11	-
η_{II}	54.45	55.66	57.07	58.66	62.32	65.98	-
m_OF, kg/s	0.6212	0.6663	0.7101	0.7535	0.8424	0.9225	-
Profit_E, $/h	3.712	3.794	3.89	3.998	4.248	4.498	-
I-TOTAL, kW	33.94	33.04	32	30.83	28.11	25.4	-

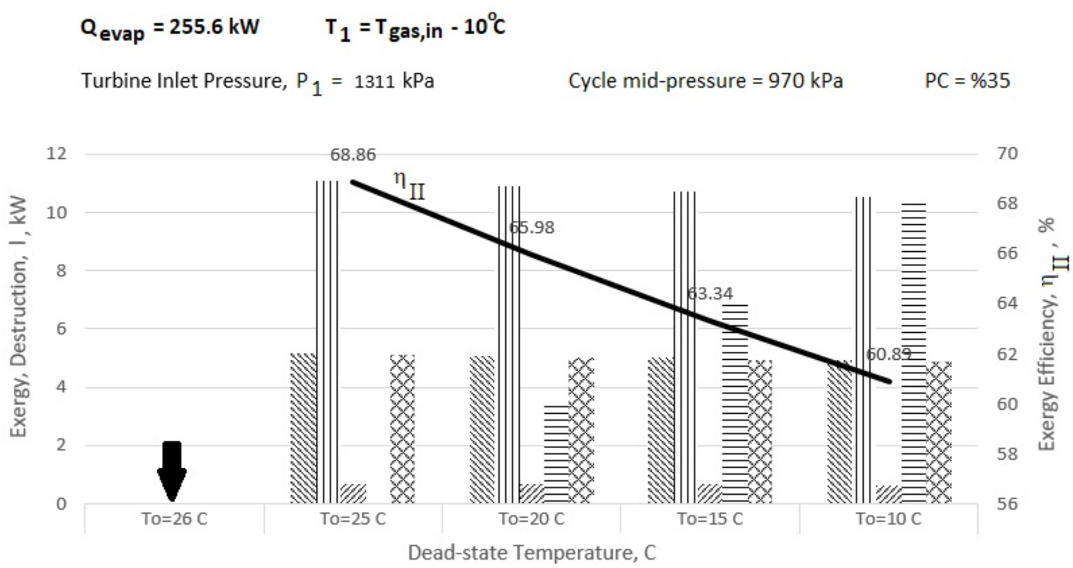
(b)

Figure 3: *(a) Effect of the cycle mid pressure, (b) Effect of the cycle mid pressure on the FGA-ORC system.*



	PC=34%	PC=35%	PC=40%	PC=50%
η_I	-	19,11	17,85	15,34
η_{II}	-	65,98	61,64	52,95
m_OF, kg/s	0,9225			
Profit_E, $/h	-	4,498	4,202	3,61
Q-heater, kW	3,72	-0,08104	-19,2	-57,43
PR-heater, %	-	19,83	32,53	50,94

Figure 4: *Effect of PC on the FGA-ORC system.*



I-Evaporator, kW		5.181	5,094	5.008	4.921
I-Turbine, kW		11.08	10.9	10.71	10.53
I-Heat Exchanger, kW		0.6847	0.6732	0.6617	0.6503
I-Condenser, kW	< 0	0.00004	3.467	6.933	10.4
I-Heater, kW		5.121	5.036	4.952	4.867
η_{II}		68.86	65.98	63.34	60.89

Figure 5: *Effect of the dead state temperature (To) on the FGA-ORC system.*

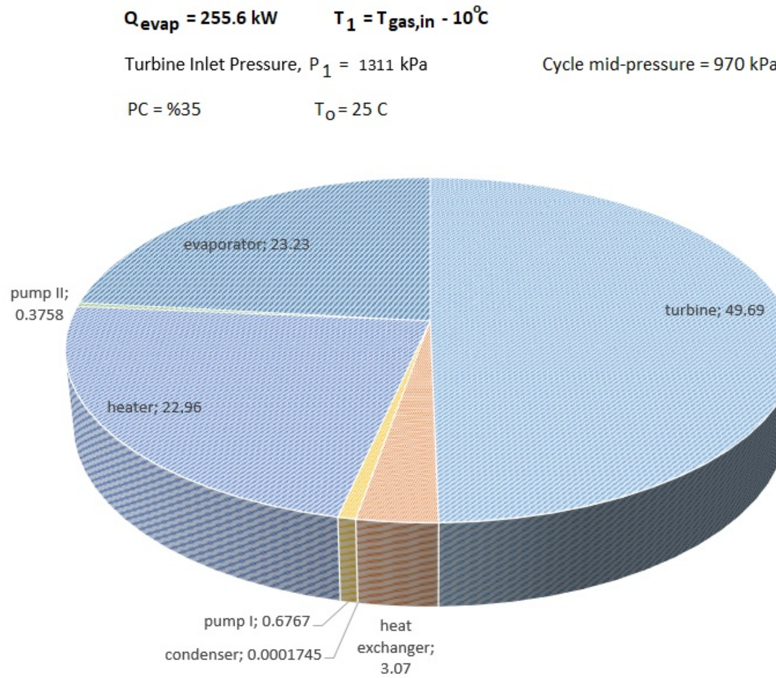


Figure 6: *Pie chart for the performance ratios (PR) of the equipment under optimum cycle conditions.*

Table 1: *Exergy destruction rates of each component under optimum cycle conditions*

Components	**Exergy destruction rate (I), kW**
Evaporator	5.181
Turbine	11.08
Heat Exchanger	0.6847
Condenser	0.00003893
Pump I	0.151
Heater	5.121
Pump II	0.08383
TOTAL	22.31

4 Conclusions

ORCs are a promising feasible, and economical technology with minimizes the risk to human health and the environment. Energy and exergy-based thermodynamics analyses were carried out for FGA-ORC to provide better guidance for system improvement. In this study, a thermal architecture for FGA-ORC was investigated using isopentane as an organic fluid to provide high flexibility and efficiency for waste heat recovery from low to moderate heat sources.

Exergetic optimization is a valuable method for determining the optimal design of thermal systems under thermodynamic constraints. Optimum pressure, temperature, and mass flow rate values for FGA-ORC were obtained from the analysis results. From the results of the analyses for FGA-ORC, it is found that the maximum value of exergetic efficiency is 68.86% at P_1=1311 kPa, P_{mid}=970 kPa, T_{gas},in=140°C, PC=35%, T_o=25°C.

As mentioned before, exergy is the maximum useful work that could be obtained from the system at a given state in a specified environment. Exergy represents the upper limit of the amount of work a device can deliver without violating any thermodynamic laws. Exergy destruction is the wasted-work-potential during a process as a result of irreversibilities. The smaller the exergy destruction associated with a process, the greater the work that is produced [23]. As the amount of work produced by the turbine increases, so will electricity production and economic profit. As expected, the turbine work output and economic profit increase while the total exergy destruction decreases.

Abbreviations

The following abbreviations are used in this manuscript:

e	specific exergy (kJ/kg)
$\dot{E}$	exergy rate (kW)
h	specific enthalpy (kJ/kg)
H	enthalpy (kJ)
$\dot{I}$	exergy destruction (kW)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
PC	percentage ratio (%)
PR	performance ratio (%)
Q	the heat transfer rate (kW)
s	specific entropy (kJ/kgK)
T	temperature (°C or K)
W	work (kW)
η	efficiency (%)

Subcripts

o	dead state conditions
II	second law
$evap$	evaporator
in	inlet
out	outlet
OF	organic fluid

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