



A steam Rankine cycle with two-stage pumping to enhance the waste heat recovery from internal combustion engines

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Article history: Received : Accepted: Published:	In this research, a high-temperature Rankin cycle (HTRC) with two-stage pumping is presented and investigated. In this cycle, two different pressures and mass flow rates in the HTRC result in two advantages. First, the possibility of direct recovery from the engine block by working fluid of water, which is a low quality waste heat source, is created in a HTRC. Secondly, by doing this, the mean effective temperature of heat addition increases, and hence the efficiency of the Rankin cycle also improves.
Keywords: Waste heat recovery Rankine cycle Internal combustion engine cooling system Two-stage pumping Mean effective temperature of heat addition.	The proposed cycle was examined with the thermodynamic model. The results showed that in a HTRC with a two-stage pumping with an increase of 8% in the mean effective temperature of heat addition, the cycle efficiency is slightly improved. Although the operational work obtained from the waste heat recovery from the engine cooling system was insignificant, the effect of the innovation on the recovery from the exhaust was significant. The innovation seems not economical for this low produced energy. However, it should be said that although the effect of the innovation on the increase of the recovery cycle efficiency is low, the changes that must be implemented in the system are also low.

1. Introduction

Typically, the efficiency of the RC that is used for waste heat recovery is very lower than that of a conventional RC. A method to improve the efficiency of a Rankine recovery cycle is by preheating from the engine cooling system [1, 2]. Waste heat sources in ICEs can be classified in high-temperature and low-temperature classes. cooling fluids are usually considered as a low-temperature waste heat source and exhaust are considered as a high-temperature waste heat source [3].

Research shows that in an ICE, about 20-30% of the fuel combustion energy is released into the ambient air through the cooling fluid with a low exergy and about 30-35% of the combustion energy is released into the ambient air through the exhaust with a high exergy [4-8].

Since the available work and the quality of the waste heat from the exhaust are higher, the studies in this field have usually focused on the waste heat recovery from the exhaust, and the waste heat of the cooling system is considered along with that. This causes that the utilization of the waste heat from the cooling system remains low, while

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it is possible to utilize a larger part of the heat source. In the following, by reviewing the literature, the subject will be investigated in detail. The waste heat recovery from the cooling fluid is investigated in direct and indirect methods [4]. In the direct recovery method, some parts of the engine cooling system are removed, and the engine block is used as the heat exchanger of the RC recovery system[5]. In this method, the waste heat from the engine cooling fluid can be used at a higher temperature in the recovery system, and hence the utilization of the heat source is more possible. One limitation of this design is that the maximum temperature of the cooling fluid should not exceed the allowed limit. In some studies, it has been reported that the waste heat from the engine block is directly used through ducts other than cooling ducts[4, 6]. This method does not suffer from the limitations of the previous design, but has additional modifications and equipment in the engine. Another application of the method is in the combination of the organic RC with the evaporative engine cooling system, where the engine block works as the evaporator, the engine water pump plays the role of the RC pump, and the engine radiator is replaced with a RC condenser. The working fluid of the RC is HCFC123 (R123). They reported a 3% recovery of the engine output energy [7].

In the indirect heat recovery method, the heat from the cooling fluid is transferred to the working fluid of a RC using a heat exchanger. Usually in the high-temperature Rankine recovery cycle that uses this method, the heat from the cooling fluid is used in the pre-heating exchanger. Another common application of this method is the utilization of the heat from the cooling fluid in the evaporative exchanger in a low-temperature RC (along with a high-temperature RC)[8]. The heat from the cooling fluid is also used in the confluent cascade expansion organic Rankine cycle (ORC) system [9]. He et al. [10] used a Kalina cycle, to recover the waste heat from the cooling fluid indirectly, along with a high-temperature ORC, responsible for recovering the exhaust waste heat, and they reported a more heat recovery from their hybrid cycle compared to that from a conventional Rankine recovery cycle. Panesar [11] evaluated two recovery structures in studying the system of direct heat recovery from the cooling fluid of a

heavy-duty diesel engine. In the first structure, as a cascade structure, the indirect heat recovery from the cooling fluid is used for heating and evaporating the organic fluid. In the second structure, the dual-pressure structure, in a combination of an engine cooling system and a RC recovery system with the same working fluid (water, or a combination of water and another fluid), the heat recovery from the cooling fluid as the RC evaporator is performed directly. For their dual-pressure (direct) system, they reported a potential average improvement of 20% in the system power, a 50% reduction in total exchangers, and a decrease in the system complexity compared to the in-series (indirect) system.

Yu et al. [12] showed that the absorption of energy from the cooling system in the recovery system is severely low, such that its effect on the production of power in the recovery system is insignificant. In the classification of the waste heat recovery from the cooling fluid of an ICE (table 1), the highest efficiency corresponds to the direct recovery using ducts other than the engine cooling ducts and the highest recovery corresponds to the indirect method in a double RC. In most of the research in this field, the cooling fluid waste heat is utilized indirectly or in the high-temperature RC for pre-heating or in the low-temperature ORC, and in both methods, the utilization of the cooling fluid waste heat is low.

In the engine cooling system, the cooling fluid must have a high mass flow rate to control the temperature of the engine block. Regarding the direct recovery from the engine block with water as the working fluid, which is a wet fluid, the water must be superheated to avoid the deterioration of the output steam quality at the end of the expander of the RC. If the entire mass flow rate of water of the engine cooling system were entered the RC, it would never be superheated. Studies show that given the specifications of the waste heat from the exhaust, a very little mass flow rate of water can be superheated in a HTRC. Therefore, by creating two different pressures and mass flow rates in a HTRC with a two-stage pumping, it would be possible to recover directly from the engine block using water as the working fluid.

Table 1. Classification of the research carried out in the field of waste heat recovery from the cooling fluid of ICEs.

Waste heat recovery from the cooling system of an ICE	Direct	As an evaporator	Heat recovery from cooling fluid along with heat recovery from exhaust [5,6,12-14]	+ Requires fewer components. - Requires redesigning the engine and the cooling system.
			Heat recovery from only the cooling fluid [4-6]	+ Requires fewer components and eliminates the engine cooling system - Without recovery from the exhaust waste heat. - Requires redesigning the engine and the cooling system.
		As a heater	Using ducts other than cooling ducts in an engine block [4-6]	+ Waste heat recovery from the engine with more efficiency. - Requires changing the engine block.
	Indirect	As a preheater	Only using an additional liquid-liquid heat exchanger [1,2,4-6,15-22]	+ Uses only a shared working fluid to recover from two waste heat sources with different temperatures. - Recovers only a part of the cooling fluid waste heat.
			As an evaporator	In a low-temperature RC along with a HTRC (Double RC) [5, 6, 8, 11, 13-19]
		In confluent cascade expansion ORC (CCE-ORC) system [9]		+ Better recovery from two heat sources with different temperatures. - Complexity of the system.

A strategy to improve the performance of a RC is by increasing the mean effective temperature of heat addition of the cycle, which results in an improved efficiency and output. In this study, by directly heating the working fluid of a RC from the waste heat of the engine cooling system with a two-stage pumping, in addition to the direct recovery from the waste heat of the cooling system, the cycle efficiency would be improved by increasing the mean effective temperature of heat addition of the cycle.

In this study, in a conventional Rankine recovery cycle, by using a thermodynamic model of first and second laws of thermodynamics, we investigated the effect of the addition of pumping and heating, implemented in the water pump and the engine block, respectively, on the increase of the mean effective temperature of heat addition,

the efficiency of the RC, and the amount of the waste heat recovery from the cooling system and the exhaust of an engine.

In this study, by reviewing the literature, we first determine the conditions to combine a HTRC with an engine. Then, by using first and second laws of thermodynamics, a conventional steam recovery cycle is modelled with and without a two-stage pumping to directly recover the waste heat from the cooling system in the engine block. With this model in a conventional RC, we first investigate the effect of the addition of pumping and heating, implemented in the water pump and the engine block, respectively, on the increase of the mean effective temperature of heat addition, the efficiency of the RC, and the amount of the waste heat recovery from the cooling system and the exhaust of an engine.

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2. Description of the recovery system

2.1. Engine

The engine considered in this study is M15GS-EU IV, with characteristics listed in table 2. In this study, we assume that the engine operates at full load and with a constant rpm.

Table 2. Engine specifications[20].

Engine name	M15GS-EU IV
Engine type	SOHC-MPFI-inline-4 cylinder-8 valve
fuel	Gasoline EURO IV
Bore (mm)*Stroke (mm)	83.6*75.5
Compression ratio	9.7
Maximum torque	127N.M/4000rpm
Maximum power	62.5kW/5200rpm
Manufacturer	Mega motor

In the engine above, about 20% of the fuel energy, equivalent to 50 kW, is dissipated through the cooling system and about 30% of the fuel energy, equivalent to 75 kW, is dissipated through the exhaust gas. If the temperature of the exhaust gas exiting from the engine was 827 °C, by considering the air as the exhaust gas, its average specific heat between 827 °C and the ambient temperature 25 °C was obtained 1.0833 kJ/kg.K from tables of thermodynamic specifications of air. Therefore, by using the equation (1), the mass flow rate of the exhaust gas was obtained 0.0863 kg/s.

The cooling fluid enters the engine at the temperature of 85 °C and exits from it at the temperature of 99.62 °C. The average specific heat of water at the two temperatures and at the pressure of 1.5 bar can be obtained from tables of thermodynamic specifications of air as 4.207 kJ/kg.K . Therefore, the mass flow rate of water, required for transferring 50 kW of the waste heat in the cooling system, can be obtained 0.813 kg/s from equation (1). All the specifications of the cooling fluid and the exhaust gas of the engine are listed in table 3.

$$\dot{Q} = \dot{m} c_p (\Delta T) \quad (1)$$

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According to the data in tables 3, if the waste heat from the cooling fluid was considered as a high-temperature source at the average temperature of 92.31°C and the ambient air was considered as a low-temperature source at 25°C, the highest thermal efficiency and the largest available work in the Carnot reversible cycle would be 18.45% and 9.224 kW, respectively. Regarding the upstream exhaust, from which 57.66 kW heat is dissipated, if the waste heat was considered as a high-temperature source at the average temperature of 522°C with respect to the ambient air, the highest thermal efficiency and the largest available work from the downstream exhaust in the Carnot reversible cycle would be 62.5% and 36 kW, respectively (Figure 1).

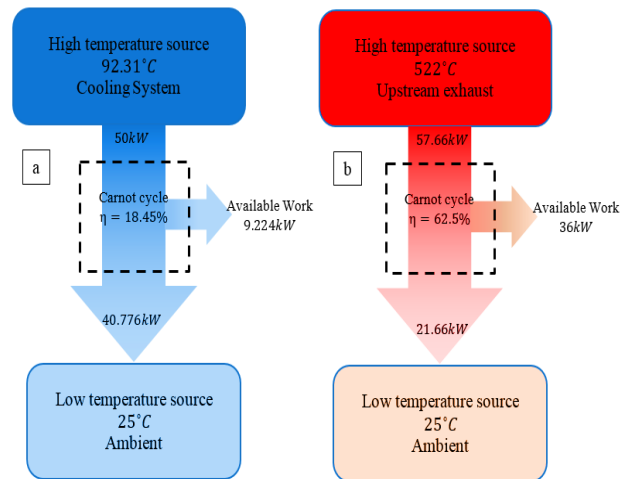


Figure 1. The highest efficiency and the largest available work in the Carnot reversible cycle for a) the cooling system and b) the downstream exhaust

2.2. The steam Rankine cycle with a two-stage pumping (HTRC)

Regarding the direct recovery from the engine block with water as the working fluid, which is a wet fluid, the water must be superheated to avoid the deterioration of the output steam quality at the end of the expander of the RC. If the mass flow rate of the RC was equal to that of the cooling system, water in the RC would never be superheated. Studies show that given the specifications of the waste heat from the exhaust, a very little mass flow rate of water can be superheated in a HTRC. Therefore, in the present work, to fix the problem, two different pressures

and mass flow rates are created in a RC with a two-stage pumping.

Table 3. Specifications of the cooling fluid and the exhaust gas of the engine.

Cooling System	Cooling fluid	WATER	
	Cooling fluid Specific Heat	4.207	$kJ/(kg.K)$
	Cooling fluid Mass Flow Rate	0.813	kg/s
	Cooling fluid Outlet Temperature	99.62	$^{\circ}C$
	Cooling fluid Inlet Temperature	85	$^{\circ}C$
Exhaust	Exhaust fluid	air	
	Exhaust fluid Specific Heat	1.095	$kJ/(kg.K)$
	Exhaust fluid Mass Flow Rate	0.0863	kg/s
	Exhaust fluid Outlet Temperature	827	$^{\circ}C$
	Exhaust fluid Inlet Pressure	1.4	bar

In the proposed solution, by the shared usage of a water pump as the initial pump and the engine block as the heating converter in a HTRC, an initial pumping and an initial heating are added to a conventional RC (figure 2), as seen in figure 3. With this work and with this assumption that the work done by the main pump of the RC as well as the absorbed heat from the exhaust gas are fixed, the increase of the mean effective temperature of

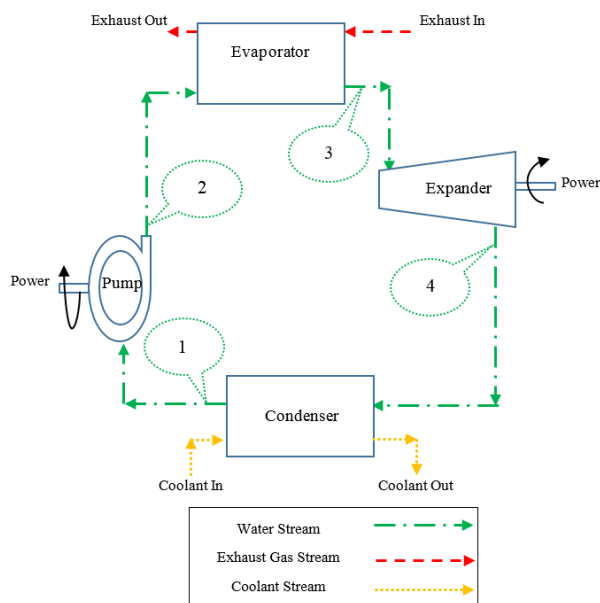


Figure 2. A conventional RC for the waste heat recovery of the engine exhaust.

heat addition and the efficiency of the RC will be investigated.

The mass flow rate of the working fluid, after passing through the water pump and the engine block, is divided into two parts. The first part enters the evaporator after its pressure was increased in the main pump through the main route of the cycle that is at a high pressure (HP), to be evaporated using the heat of the upstream exhaust and be converted to the superheated vapor. After passing through the expander and the condenser, the vapor enters again the water pump chamber. The temperature of the second part of the water through the side route that is at a low pressure (LP), by passing through the radiator or the evaporator exchanger of the low-temperature cycle is decreased and enters again the water pump chamber.

Actually, the pumping operation in this RC is performed in two stages: the first stage in the engine water pump and the second stage in the secondary pump of the RC. In this design, without adding any additional component to the engine system, an initial pumping is performed in the engine water pump and heating is performed within the engine block, which increases the mean effective temperature of heat addition of the RC.

To investigate the proposed solution, we must first determine the boundary conditions of the cycle, including the condensation, the heating, and the evaporation conditions. The specifications of the exhaust gas exiting from the engine determine the evaporation conditions of the cycle, the conditions of the engine cooling fluid determine the heating conditions in the cycle, and the

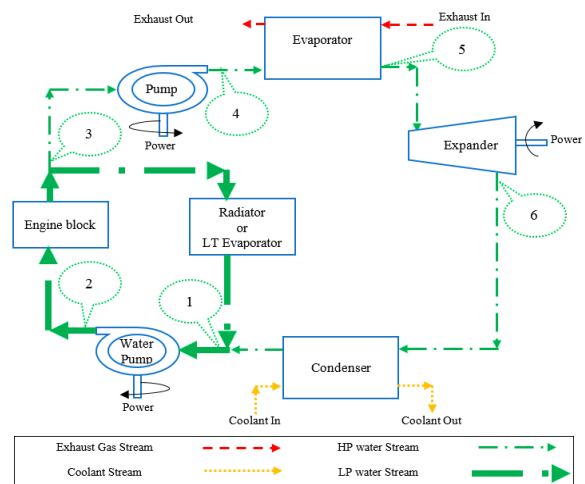


Figure 3. The structure of the HTRC to recover the heat of the cooling water and the exhaust.

environmental conditions determine the condensation conditions of the cycle. The following operational conditions are determined according to the specifications of the engine under study for the HTRC.

1. The maximum superheating temperature of the water must be at least an amount so that the quality of the working fluid at the end of the expander becomes higher than 95%.
2. Usually, in the waste heat recovery from an ICE along with a HTRC, a low-temperature RC is used. Therefore, in order to use the heat of the downstream exhaust in the low-temperature RC, the minimum temperature of the exhaust gas in the high-temperature cycle must not become lower than a limit. The temperature ranges specified for waste heat sources of the engine that have been used in this cycle are listed in table 4.
3. The RC with a two-stage pumping has three pressure levels in the cycle, which are listed in table 4 assuming that the middle level is equal to the pressure of the engine cooling fluid.
4. The mass flow rate of the high-pressure (HP) working fluid of the cycle must be at most an amount so that the temperature of the exhaust gas exiting from the Evaporator, due to the heat transfer, would not become lower than the limit specified in table 4. The remaining mass flow rate of the fluid, by passing through the side route, which is at a low pressure (LP), transfers the heating to the low-temperature cycle or the radiator.

Table 4. The operational specifications of the HTRC.

Working fluid	Working fluid	WATER	
Heater	Mass Flow Rate of Working fluid	0.813	kg/s
	Mass Flow Rate of Working fluid LP	0.793	kg/s
	Mass Flow Rate of Working fluid HP	0.02	kg/s
	Working fluid Inlet Temperature	99.62	°C
	Working fluid Outlet Temperature	85	°C

	Heating Pressure	1.5	bar
Evaporator	Exhaust fluid	air	
	Exhaust fluid Specific Heat	1.095	$\frac{kJ}{(kg.K)}$
	Exhaust fluid Mass Flow Rate	0.0863	kg/s
	Exhaust fluid Inlet Temperature	827	°C
	Exhaust fluid Outlet Temperature	217	°C
	Exhaust fluid Inlet Pressure	1.4	bar
	Evaporation Temperature	88.88	°C
	Evaporation Pressure	45	bar
	Evaporator Outlet Superheat	185	°C
Condenser	Condensation Temperature	85	°C
	Condensation Pressure	0.5787	bar
Water Pump	Water Pump Efficiency	0.8	fraction
Pump	Pump Efficiency	0.8	fraction
Expander	Expander Efficiency	0.8	fraction

2.3. The thermodynamic model of the HTRC

The processes of the HTRC, based on figure 3, are listed in table 5 and their assumptions are listed in table 6.

By examining the energy balance in every part of the recovery system, one can investigate the pump work, the expander work, and the transferred heat in exchangers. From the viewpoint of the second law of thermodynamics, by examining the exergy destruction in the recovery system, we can determine how much of the available energy has been recovered.

Table 5. Processes of the HTRC.

The process number	The process type	Descriptions
6-1	Pumping	The working fluid compression from the condensation pressure to the Heating pressure in water pump.
1-2	Heating	Heat transfer from the engine

		block to the working fluid at the fixed Heating Pressure;
2-3	Pumping	The working fluid compression from the condensation pressure to the evaporation pressure.
3-4	Evaporation	The heat transfer from the downstream exhaust to the working fluid at the fixed evaporation pressure;
4-5	Expansion	The working fluid expansion from the evaporation pressure to the condensation pressure;
5-6	condensation	Heat transfer to the environment at the fixed condensation pressure;

Table 6. The assumptions of the thermodynamic model of the HTRC.

1) The system is in a steady state.
2) The working fluid's properties are assumed, on average, constant during a process.
3) The working fluid is assumed incompressible.
4) The changes in potential and kinetic energies are neglected.
5)The exhaust gas is considered as the air.
6)The drop in the exhaust pressure in the exchangers has been neglected.

For this purpose, according to the equation 2, we can get the exergy destruction rate (\dot{i}) for every control volume based on the mass flow rate (\dot{m}), the input entropy ($\sum_{in} s$), the output entropy ($\sum_{out} s$), the heat transfer (q_k) at the temperature of heat source (T_k) and at the ambient temperature (T_{amb}).

$$\dot{i} = T_{amb} \frac{dS_{gen}}{dt} = \dot{m} T_{amb} \left[\sum_{out} s - \sum_{in} s - \sum_k \frac{q_k}{T_k} \right] \quad (2)$$

By balancing the energies in exchangers using Eq. (1), we can obtain the mass flow rate or the unknown input and output temperatures.

By considering the control volume for every part of the RC, the equations corresponding to the first and second laws of thermodynamics in the proposed thermodynamic model are presented in table 7 (the equations and the characteristics of the working fluid were adopted from [21]).

3. Results and discussion

In this section, the results of the calculations performed for the RC for the cases before and after the addition of the initial pumping and heating with the assumptions mentioned in previous sections are presented. Table 8, a slight increase of the amounts in the case with the two-

stage pumping, shows the improvement of the efficiency of this cycle. As it was expected, by heating the fluid from the waste heat of the cooling system, the mean effective temperature of heat addition of the cycle has increased by more than 8%, resulting in a 4.6% increase in the first-law efficiency, a 6.67% increase in the output operational work, and a 6.7% increase in the exergy efficiency of the cycle.

Regarding the recovered amount, as it was mentioned in the literature review section, it is observed that from 50 kW of the waste heat of the cooling system, only 1.14 kW is absorbed in the RC, and 24.2% of the absorbed energy, equivalent to 0.276 kW, is converted to the operational work, which is insignificant. However, the effect of the innovation on the increase of the recovery from the exhaust must not be neglected. Due to the increased efficiency of the cycle with the two-stage pumping, it is observed that 0.614 kW more power is absorbed from the same waste heat of the exhaust. Indeed, the total effect of the increased output work of the cycle with a two-stage

Table 7. The thermodynamic equations governing the HTRC [21].

Equipment	Modelling overview
Pump	$W_{WP,ac} = \dot{m}_{cf}(h_e - h_i)$ $W_{P,rev} = \dot{m}_{RC}(h_{e,s} - h_i)$ $W_{P,ac} = \dot{m}_{RC}(h_e - h_i)$ $\eta_P = \frac{W_{P,rev}}{W_{P,ac}}$ $\dot{i}_P = \dot{m}_{RC} T_{amb}(s_e - s_i)$
Heater, Evaporator	$Q_{i-e} = \dot{m}_{RC}(h_e - h_i)$ $\dot{i}_{Hx} = T_{amb} \dot{m}_{RC} \left((s_e - s_i) - \frac{(h_e - h_i)}{\bar{T}_w} \right)$
expander	$x = \frac{s - s_f}{s_{fg}}$ $W_{Exp,rev} = \dot{m}_{RC}(h_i - h_{e,s})$ $W_{Exp,ac} = \dot{m}_{RC}(h_i - h_e)$ $\eta_{Exp} = \frac{W_{Exp,ac}}{W_{Exp,rev}}$ $\dot{i}_{Exp} = T_{amb} \dot{m}_{RC} (s_e - s_i)$
Condenser	$\dot{i}_{Cond} = T_{amb} \dot{m}_{RC}(s_e - s_i) - \frac{(h_e - h_i)}{T_{amb}}$
Exhaust	$\dot{i}_{ex} = \dot{m}_{ex} c_{p,ex} \left[(T_{ex,e} - T_{amb}) - T_{amb} \ln \left(\frac{T_{ex,e}}{T_{amb}} \right) \right]$
$\eta_{1stl} = \frac{W_{Exp,ac} - W_{P,ac} - W_{WP,ac}}{Q_{i-e}}$ $\dot{i}_t = \dot{i}_P + \dot{i}_{Hx} + \dot{i}_{Exp} + \dot{i}_{Cond} + \dot{i}_{ex}$ $\eta_{2ndl} = \frac{W_{Exp,ac}}{W_{Exp,ac} + \dot{i}_t}$	

pumping is 0.89 kW. In the T-s plot in figure 4, it is observed that in the condensed liquid region, the effect of the two-stage pumping is slightly negative, it is slightly positive in the saturated region, but it is significantly and positively increased in the superheated region.

The cycle with the two-stage pumping positively affects the efficiency and the utilization of the waste heat sources. However, since the implementation of the idea requires some changes in the RC and in the engine cooling system, the idea seems not economical at first glance for this low energy. However, in the combination of the HTRC with the engine cooling system, no additional component is added to the system and it is possible to provide this improved efficiency for the recovery cycle with slight changes in the same components of a conventional RC and in the engine. Actually, although the effect of the innovation on the increase of the recovery cycle efficiency is low, the changes that must be implemented in the system are also low.

Table 8. Results of the HTRC with and without a two-stage pumping.

	Without the two-stage pumping	With the two-stage pumping	increase(%)
The mean effective temperature of heat addition (°C)	250.77	271.32	8.19
Evaporation temperature(°C)	256.75	257.43	0.26
The maximum superheating temperature(°C)	416.02	442.44	6.35
The condensation temperature(°C)	85	85	0
The quality of the vapor exiting from the expander(%)	96.55	97.08	0.55
The net output work	13.34	14.23	6.67
The heat absorbed from the exhaust(kW)	57.66	57.66	0
The heat absorbed from the cooling system (kW)	0	1.14	100
1st low efficiency(%)	0.2314	0.242	4.62
2nd low efficiency(%)	0.3706	0.3954	6.7

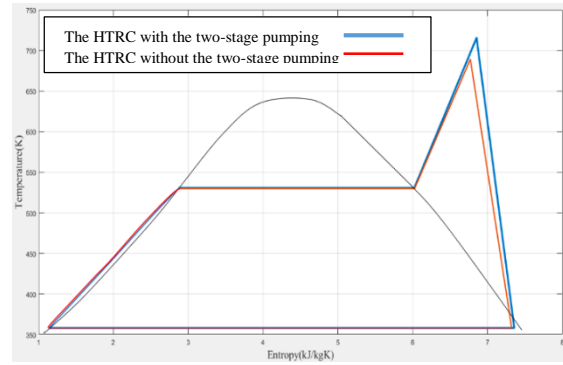


Figure 4. The T-S plot of the HTRC with and without the two-stage pumping.

4. Validation

To validate the thermodynamic model, we examined that according to the first law of thermodynamics the sum of the work and the heat input into the system is equal to the work and the heat output of the system, and also according to the second law of thermodynamics, the net output work of the system must not be higher than the reversible work of the Carnot cycle with the same heat sources. Or in the other words, the Carnot cycle has the highest efficiency and no cycle can have a higher efficiency. For the Carnot cycle with the upstream exhaust and the cooling system as the high-temperature sources and the ambient air as the low-temperature source (with specifications mentioned in previous sections), the thermal efficiency of 61.65% and the available work of 36.25 kW are obtained. The results obtained in the present work are lower than the amounts. Furthermore, the results of the work are confirmed by, and compatible with, the theoretical solutions and experimental results in the literature. In experimental studies conducted by Galindo et al. [22] the maximum ideal and real RC efficiency value of 19% and 6% respectively are achieved, Liu et al. [23] reported that they obtained the highest thermal and exergy efficiency of 11.84% and 54.24%, respectively in their ORC system. Wenzhi et al. [24] reported a maximum 12% increase in power output in the system they provided. Wang *et al.* [25] in Performance analysis of a novel system combining a dual loop organic Rankine cycle (ORC) with a gasoline engine, provided results. For validation, their system, the specification of which is presented in Table 9, was examined in the thermodynamic model presented in this paper, and the results obtained with their results were compared

in table 10 and 11. It was found that the error of the results was negligible.

Table 9. System specifications provided by Wang et al. [25].

cycle	LTRC	HTRC
Working fluid	R134a	R245fa
Mass flow Rate(kg/s)	1.891	0.7448
P_{Cond} (bar)	7.7	7.89
P_{Eva} (bar)	21.17	24
Pump Efficiency(%)	80	80
Expander Efficiency(%)	75	75

Table 10. Compare the results of the LTRC provided by Wang et al. [25].

Parameter	Wang <i>et al</i>	present work	Relative error/%
$W_{Exp}(kW)$	29.064	29.054	0.03
$W_P(kW)$	2.675	2.673	0.07
$Q_{in}(kW)$	365.24	360.95	1.1
$Q_{out}(kW)$	338.86	338.65	0.06

Table 9. Compare the results of the HTRC provided by Wang et al. [25].

Parameter	Wang <i>et al</i>	present work	Relative error/%
$W_{Exp}(kW)$	10.85	10.7	1.3
$W_P(kW)$	1.277	1.28	0.23
$Q_{in}(kW)$	133.58	131.85	1.3
$Q_{out}(kW)$	124	122.43	1.2

5. Conclusions

In this work, aimed at increasing the waste heat recovery from an ICE and by focusing on the improvement of the RC efficiency through preheating, a design was proposed to recover the waste heat of the engine cooling system with a two-stage pumping in the same HTRC, in addition to improvement of the HTRC (resulting from the increased mean effective temperature of heat addition). In this work, using the thermodynamic model of the first and second laws of thermodynamics with the appropriate conditions for the combination of the HTRC with the cooling system and the exhaust of the engine, we

investigated the effect of the addition of an initial pumping and an initial heating in the HTRC on the increase of the mean effective temperature of heat addition and the efficiency of the RC. Furthermore, the effect of the pumping and the heating from the cooling system on the waste heat recovery from the exhaust and the cooling system was also investigated. The followings can be summarized as the conclusions of the study.

1. A slight improvement was observed in the cycle efficiency in the HTRC with the two-stage pumping. As it was expected, by heating the fluid from the waste heat of the cooling system, the mean effective temperature of heat addition in the cycle increased by more than 8%, resulting in a 4.6% increase in the first-law efficiency, a 6.67% increase in the output operational work, and a 6.7% increase in the exergy efficiency of the cycle.
2. The operational work obtained from the waste heat recovery from the engine cooling system was insignificant (0.276 kW), which is due to the low absorption of the energy into the HTRC (1.14 kW from 50 kW).
3. Due to the innovation, the recovery from the exhaust was slightly increased. Due to the increased cycle efficiency with the two-stage pumping, it was observed that 0.614 kW more power was observed from the same waste heat of the exhaust.
4. In total, the cycle with the two-stage pumping produced more operational work by 0.89 kW. This idea seems not economical at first glance for this low energy. However, although the effect of the innovation on the increase of the recovery cycle efficiency is low, the changes that must be implemented in the system are also low.
5. Since a high amount of the mass flow rate of the cooling fluid, which carries a high amount of energy, cannot be absorbed in the HTRC, we can increase the recovery by using a low-temperature RC along with this cycle.

Nomenclature

Latin symbols	Subscripts and superscripts
c_p specific heat ratio at constant pressure ($\frac{kJ}{(kg.K)}$)	ac actual
h specific enthalpy ($\frac{kJ}{kg}$)	amb ambient state
\dot{m} mass flow rate ($\frac{kg}{s}$)	$Cond$ condenser
i exergy destruction rate (kW)	e exergy

P	pressure (kPa)	e	exit
Q	heat transfer (kJ)	ex	exhaust
s	specific entropy ($kJ/(kg.K)$)	Eva	evaporative
S_{gen}	entropy production (kJ/K)	Exp	expander
T	temperature (K)	f	fluid saturation point
\bar{T}	average temperature (K)	fg	vaporization region
x	quality of fluid exiting from the expander	i	inlet
w	work per mass unit (kJ/kg)	$i - e$	process i to e
W	power (kW)	rev	reversible
Acronyms		RC	Rankine cycle
RC	Rankine Cycle	P	pump
ICE	Internal Combustion Engine	t	total
HTRC	High-Temperature Rankine Cycle	wP	water pump
ORC	Organic Rankine Cycle	$1stl$	first-law
		$2ndl$	second-law
		Greek symbols	
		η	efficiency(%)

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