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Increasing waste heat recovery from an internal combustion engine by a dual-loop non-organic Rankine Cycle

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ABSTRACT

This research proposes the combination of a dual-loop non-organic Rankine cycle (DNORC) with an internal combustion engine to increase the output power of the recovery system by focusing on the increase in the energy input and system efficiency. In doing so, it investigates the strategy of increasing the mean effective temperature of heat addition in the high-temperature Rankine cycle (HTRC) (to improve the system efficiency and the strategy of increasing the waste heat entering the low-temperature Rankine cycle (LTRC) (to increase the energy input. In this recovery system, by focusing on the recovery of the waste heat from the engine cooling system and exhaust, the radiator can be removed from the engine cooling system, and by mounting fewer parts on the engine, not only can extra power be generated but also the engine can be cooled down faster and more efficiently. By using a thermodynamic analysis, the appropriate matching conditions between the DNORC with the engine are determined. The results showed that as the input energy increased, the recovery rate and system efficiency also increased. The output power of the recovery system exceeded 20kW and the efficiency of the whole engine and the recovery system increased to 33%.

1) Introduction

Typically, the efficiency of the Rankine cycle (RC), used for waste heat recovery, is much lower than that of a conventional RC. One method to improve the efficiency of a Rankine recovery cycle is preheating the engine cooling system [1,2]. Waste heat sources in ICEs can be categorized into high-temperature and low-temperature classes. Cooling fluids are usually considered as a low-temperature waste heat source and exhausts are considered as a high-temperature waste heat source [3].

Research shows that in an ICE, about 20-30% of the fuel combustion energy with a low exergy is released into the ambient air through the cooling fluid and about 30-35% of the combustion energy with a high exergy is wasted through the exhaust [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 [9, 10]. The waste heat of the cooling system has received less attraction.

The waste heat recovery from the cooling fluid is investigated by using direct or 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 Rankine cycle (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, hence, utilization of more heat from the source is possible. In some studies, it has been reported that the waste heat from the engine block is directly used through ducts other than cooling ducts [6]. Another application of the method is in the combination of the organic Rankine cycle (ORC) with the evaporative engine cooling system. A 3% recovery of the engine output energy is reported for such a system [7].

However, in the indirect heat recovery method, the heat from the cooling fluid is transferred to the working fluid of an RC through a heat exchanger. Peris et al. [11] simulated 6 different ORC structures to recover the dissipation heat of the cooling fluid. One application of this method is the utilization of the heat from the cooling fluid in the preheater. Heidrich et al. [1] and Vaja et al. [2] used the waste heat from the cooling fluid in the preheater of the HTRC, while Ge et al. [12] and Shu et al. [13] used it in the preheater of the LTRC. Liu et al. [14] used the dissipation heat of the cooling fluid in the preheater of the interheating ORC with a two-stage expansion. In a supercritical CO_2 cycle, Song et al. [15] used the waste heat from the cooling fluid in the low-temperature preheater. Another common application of this method is the utilization of the heat from the cooling fluid in the evaporator in an LTRC along with an HTRC [16]. This double RC can achieve a higher system efficiency than the ORC [17]; however, the systems is large, complex, heavy and economically unsound [18]. Chintala et al. [8] and Wang et al. [19] used the dissipation heat of the cooling fluid in the evaporator of the LTRC, whereas Negash et al. [20] and Yang et al. [21] used it in the evaporator of the HTRC. Yang et al. [22] utilized the dissipation heat of the cooling fluid in the evaporator of a single RC. Shu et al. [23] also recovered this heat from the evaporator of the low-temperature trans critical RC. The heat from the cooling fluid is also used in the confluent cascade expansion ORC (CCE-ORC) system [8]. He et al. [24] used a Kalina cycle to recover the waste heat from the cooling fluid indirectly, along with a high-temperature ORC intended to recover the exhaust waste heat. Panesar [25] evaluated two recovery structures in studying the system for direct heat recovery from the cooling fluid of a heavy-duty diesel engine. In summary, in the indirect heat recovery method, the dissipation heat of the cooling fluid in the preheaters [26] and in the evaporators are used. Yu et al. [27] showed that the absorbed energy of the cooling fluid in the recovery system is small and, consequently, its influence on the output power of the recovery system is insignificant.

Mashadi et al. [28] showed that the total heat dissipated through the radiator can be re-introduced into the recovery system by determining the matching conditions between a low-temperature RC and an engine cooling system, and by choosing a suitable working fluid and producing more work and decreasing the size of the recovery system. The radiator would also be no longer needed for the cooling system. The main idea behind this work is to increase the amount of input energy into the system.

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, it 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 the water of the engine cooling system entrees 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 an HTRC. Therefore, by creating two different pressures and mass flow rates in an HTRC with a two-stage pumping, it would be possible to recover heat directly from the engine block using water as the working fluid.

One strategy to improve the performance of an HTRC is increasing the mean effective temperature of the heat addition of the cycle, ultimately resulting in an improved efficiency and output. In this study, by directly heating the working fluid of an 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 through increasing the mean effective temperature of the heat addition of the cycle.

To evaluate the proposed structure, the thermodynamic model of the DNORC was created so that the first-law and the second-law efficiencies, the cycle output work, and heat transfers in exchangers could be investigated. The matching conditions between the RC with the engine cooling system was investigated and the working fluid characteristics required for absorbing the entire waste heat from the engine cooling system was specified. Then, the most suitable working fluid was determined and the results derived from the DNORC were validated.

2) Materials and methods

2-1) Engine

The engine considered in this study is M15GS-EU IV, with characteristics listed in table 1.

In this study, the engine is assumed to operate at its maximum power and with a constant rpm.

Table 1: Engine specifications[29].

Engine name	M15GS-EU IV
Engine type	SOHC-MPFI-inline-4 cylinder-8 valve
Fuel	Gasoline EURO IV
Bore (mm) ⁎ Stroke (mm)	75.5*83.6
Compression ratio	9.7
Maximum torque	127N.M/4000 rpm
Maximum power	62.5kW/5200 rpm
Manufacturer	Megamotor
Cooling fluid	egl-5050
Cooling fluid Specific Heat	3.56 kJ/(kg. K)
Cooling fluid Mass Flow Rate	1.003 kg/s
Cooling fluid Inlet Temperature	85 °C
Cooling fluid Outlet Temperature	99 °C
Cooling fluid Inlet Pressure	1.5 bar
Exhaust Gas Mass Flow Rate	0.08635 kg/s
Exhaust Gas Inlet Temperature	827 °C

2-2) Dual-loop non-organic Rankine cycle

In Fig. 1, a DNORC that includes an LTRC and an HTRC with a two-stage pumping is observed.

In the 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 an HTRC. Therefore, in the present work, to fix the problem, two different pressures and mass flow rates are created in an RC with a two-stage pumping.

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 an HTRC, an initial pumping and an initial heating are added to a conventional RC, as seen in figure 1 (Green lines and dots). In view of this and assuming 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 in the mean effective temperature of heat addition and the efficiency of the RC will be investigated.

Having passed through the water pump and the engine block, the mass flow rate of the working fluid is divided into two parts. The first part enters the evaporator after its pressure increases in the main pump through the main route of the cycle, which is at a high pressure (HP), to be evaporated using the heat of the upstream exhaust and be converted into superheated vapor. Having passed through the expander and the condenser, the vapor enters the water pump chamber again.

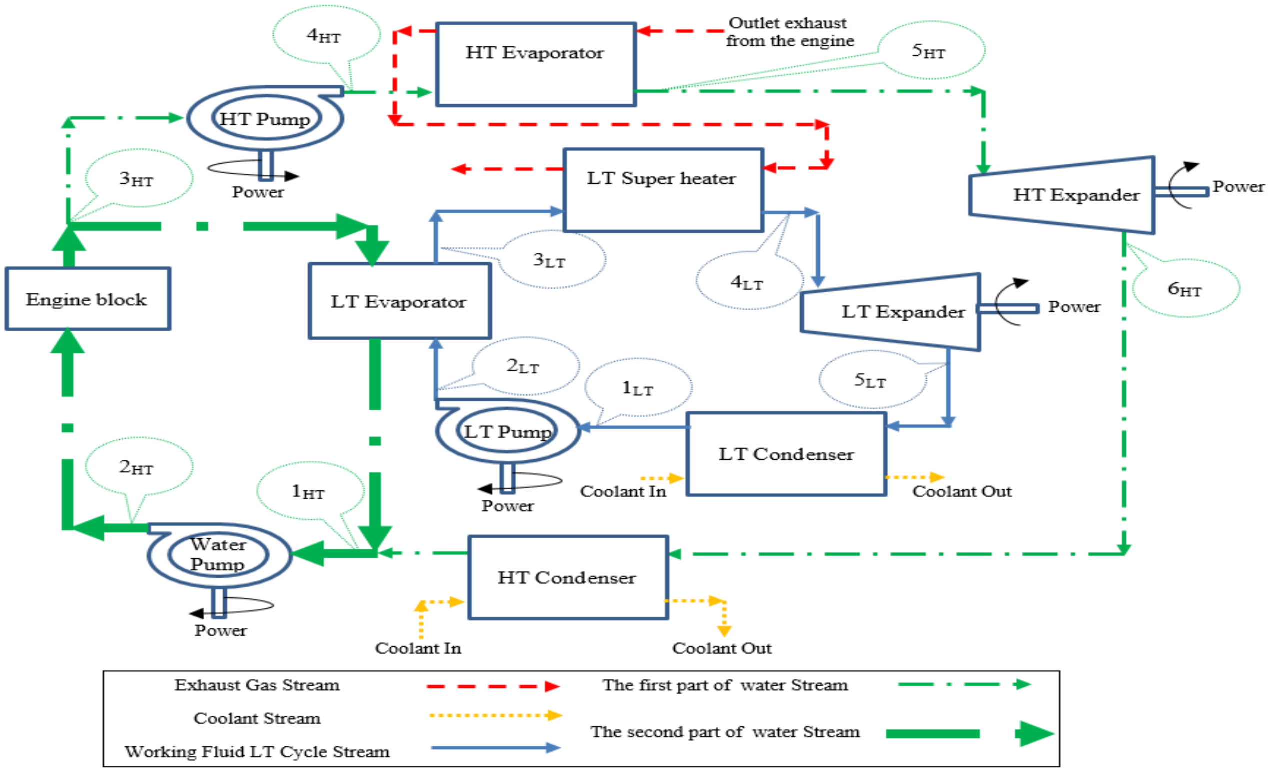

Figure 1: The structure of the DNORC.

By passing through the low-temperature evaporator of the low-temperature cycle, the temperature of the second part of the water through the side route, which is at a low pressure (LP), decreases and enters the water pump chamber once again.

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

For a low-temperature cycle, the LTRC was used by Mashadi et al. [28] (figure 1, Blue lines). To examine the DNORC, the isentropic efficiencies of the pumps and the expanders are assumed to be 80%.

2-3) Matching conditions

The first key point in the proposed design is the matching conditions between a DNORC with the engine, which are examined in two parts of the HTRC and the LTRC as follows.

HTRC

To investigate the proposed solution, to begin with, the boundary conditions of the cycle, including the condensation, the heating, and the evaporation conditions (table 2) must be determined. 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 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 inasmuch as the quality of the working fluid at the end of the expander becomes higher than 95%.
2. In order to use the heat of the downstream exhaust in the LTRC, the minimum temperature of the exhaust gas in the HTRC must not become lower than a given limit.
3. The RC with a two-stage pumping has three pressure levels in the cycle, which are listed in table 2, 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 inasmuch as the temperature of the exhaust gas exiting from the evaporator would not become lower than the limit specified due to the heat

transfer. By passing through the side route, which is working at a low pressure (LP), the remaining mass flow rate of the fluid, transfers the heating to the LTRC.

Table 2: The operational conditions of the HTRC.

Mass Flow Rate of Working fluid LP	0.8299	kg/s
Mass Flow Rate of Working fluid HP	0.0204	kg/s
High Side Pressure (Evaporation Pressure (HP))	45	bar
Mid Side Pressure	1.5	bar
Low Side Pressure (Condensation Pressure (LP))	0.5787	bar
Evaporator Outlet Superheat	185	˚C

LTRC

The current work primarily aims to study the increase in the waste heat recovery from the cooling fluid of an engine by increasing the absorption of the energy input into the RC.

The matching conditions between an LTRC with the cooling system of an ICE must be such that the LTRC can absorb the entire waste heat from the cooling fluid. To this end, the mass flow rate of the working fluid of the LTRC is determined at an amount that can absorb the entire waste heat from the cooling fluid into the evaporator (table3). The evaporator and the condenser pressures are determined based on the evaporation and the condensation temperatures in the saturated region.

Table 3: The operational conditions of the LTRC.

Refrigerant Mass Flow Rate	0.0435	kg/s
High Side Pressure	45	bar
Low Side Pressure	11.5	bar
SuperHeater Outlet Superheat	60.88	˚C

2-4) working fluid

In the HTRC, since recovery from the engine cooling system is carried out directly, water was selected as the working fluid of this cycle.

Mashadi et al. [28], having reviewed the 19 working fluids for the low-temperature cycle, introduced ammonia as the best agent fluid to absorb the total waste heat of the cooling system. In this work, the same working fluid for the low-temperature cycle has been used.

2-5) The thermodynamic model of the DNORC

The processes of the DNORC, based on figure 1, are listed in table 4 and it is assumed that the system is in a steady state. The pump and the expander of the RC are assumed adiabatic, the changes in potential and kinetic energies are ignored.

Table 4: Processes of the DNORC.

Cycle	The process number	The process type	Descriptions
LTRC	1_{LT}-2_{LT}	Pumping	The compression process from the condensation pressure to the evaporation pressure
	2_{LT}-3_{LT}	Evaporation	Heat transfer from the cooling system to the working fluid at the fixed evaporation pressure
	3_{LT}-4_{LT}	Superheating	Heat transfer from the downstream exhaust to the working fluid at the fixed evaporation pressure
	4_{LT}-5_{LT}	Expansion	The working fluid expansion from the evaporation pressure to the condensation pressure
	5_{LT}-1_{LT}	condensation	Heat transfer to the environment at the fixed condensation pressure
HTRC	1_{HT}-2_{HT}	Initial pumping	The compression process from the condensation pressure and temperature to the pressure of the engine cooling system in the water pump

HTRC	2_{HT}-3_{HT}	Heating	Heat transfer from the cooling system to the working fluid at the fixed evaporation pressure in the engine block;
	3_{HT}-4_{HT}	Maine pumping	The compression process from the pressure of the engine cooling system to the evaporation pressure in the main pump
	4_{HT}-5_{HT}	Evaporation	Heat transfer from the exhaust upstream to the working fluid at the fixed evaporation pressure
	5_{HT}-6_{HT}	Expansion	The working fluid expansion from the evaporation pressure to the condensation pressure
	6_{HT}-1_{HT}	condensation	Heat transfer to the environment at the fixed condensation pressure

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 of the exchangers. From the viewpoint of the second law of thermodynamics, by examining the exergy destruction in the recovery system, it can be determined how much of the available energy has been recovered.

To this end, according to Eq. (1), the exergy destruction rate ($\dot{I}$) for every control volume is determined 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 the heat source (T_k) and at the ambient temperature (T_{amb}).

$$\dot{I} = T_{amb}\frac{dS_{gen}}{dt} = \dot{m}\,T_{amb}\left[\sum_{out} s - \sum_{in} s - \sum_{k}\frac{q_k}{T_k}\right] \quad (1)$$

By balancing the energies in the exchangers using Eq. (2), the mass flow rate or the unknown input and output temperatures is obtained.

$$\dot{Q} = \dot{m}\,c_P(\Delta T) \quad (2)$$

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 5.

Table 5: The thermodynamic equations governing the DNORC [30].

Equipment	Modelling overview
Pump	$W_{P,rev} = \dot{m}_{RC}(h_{o,s} - h_i)$ $W_{P,ac} = \dot{m}_{RC}(h_o - h_i)$ $\eta_P = \frac{W_{P,rev}}{W_{P,ac}}$ $\dot{I}_P = \dot{m}_{RC}\,T_{amb}(s_o - s_i)$
Evaporator	$Q_{Eva} = \dot{m}_{RC}(h_o - h_i)$ $\dot{m}_{RC} = \frac{\dot{m}_{cf}c_{cf}(T_{cf,i}-T_{cf,o})}{q_{Eva}}$ $\dot{I}_{Eva} = T_{amb}\dot{m}_{RC}\left[(s_o - s_i) - \frac{(h_o - h_i)}{\bar{T}_w}\right]$
Superheater	$Q_{Sup} = \dot{m}_{RC}(h_3 - h_i)$ $\dot{I}_{Sup} = T_{amb}\dot{m}_{RC}\left[(s_o - s_i) - \frac{h_o - h_i}{\bar{T}_{ex}}\right]$ $\dot{I}_{ex} = \dot{m}_{ex}\,c_{P_{ex}}\left[(T_{ex,o} - T_{amb}) - T_{amb}Ln\left(\frac{T_{ex,o}}{T_{amb}}\right)\right]$
Expander	$x = \frac{s - s_f}{s_{fg}}$ $W_{Exp,rev} = \dot{m}_{RC}(h_i - h_{o,s})$ $W_{Exp,ac} = \dot{m}_{RC}(h_i - h_o)$ $\eta_{Exp} = \frac{W_{Exp,ac}}{W_{Exp,rev}}$ $\dot{I}_{Exp} = T_{amb}\dot{m}_{RC}\,(s_o - s_i)$

Condenser	$\dot{I}_{Cond} = T_{amb}\dot{m}_{RC}\left[(s_o - s_i) - \frac{h_o - h_i}{T_{amb}}\right]$
$\eta_{1stl} = \frac{W_{Exp,ac} - W_{P,ac}}{Q_{Eva} + Q_{Sup}}$	
$\dot{I}_t = \dot{I}_P + \dot{I}_{Eva} + \dot{I}_{Sup} + \dot{I}_{ex} + \dot{I}_{Exp} + \dot{I}_{Cond}$	
$\eta_{2ndl} = \frac{W_{Exp,ac}}{W_{Exp,ac} + \dot{I}_t}$	

3) Results and discussion

3-1) System performance

In the study of the HTRC, with pre-heating by direct recovery from the cooling system waste heat, the mean effective temperature of heat addition increased by 8%, and consequently, the efficiency of the cycle increased by 4.6%, but only 0.27 kW of work was obtained from this The results derived from the investigation on the LTRC with the working fluid of ammonia and the investigation on the HTRC with the working fluid of water are shown in table 6.

direct recovery. This amount is very low as compared to the 12.93 kw work achieved by the recovery of upstream exhaust waste heat. Another negative point in the HTRC with a two-stage pumping is that the condensation pressure of this cycle is less than the atmospheric pressure, and, therefore, negative pressure generation will be difficult. In addition, if higher pressure is used, due to the higher temperature of the fluid outlet than that of the condenser relative to the engine cooling temperature, there is no longer any possibility of the retrieval of the cooling system in this cycle unless the condenser output is colder than the saturated fluid. In the LTRC, due to the increased energy input to the cycle, it is observed that the efficiency of the second law of the cycle is high, and, therefore, the amount of work produced is also significant. In the DNORC, all of the cooling system waste heat is absorbed some of which is recovered. The remainder is transferred to the environment through the condensers of DNORC. Moreover, besides producing more work and decreasing the size of the recovery system, the radiator would be no longer needed for the cooling system. The output power of the system was obtained 20.13 kW. The first law efficiency and the second law efficiency were obtained 18.3% and 38%, respectively.

Table 6: Results of the DNORC.

cycle	Subject	Value
HTRC	HT heater heat rate	1.20 kW
	HT evaporator heat rate	57.61 kW
	HT condenser heat rate	45.52 kW
	HT initial pump power input	0.10 kW
	HT main pump power input	1.45 kW
	HT expander power output	14.75 kW
	Quality	97%
	first law efficiency	22.44%
	second law efficiency	29.22%
	HT expander power from cooling system waste heat	0.27 kW
	HT expander power from upstream exhaust waste heat	12.93 kW
LTRC	LT evaporator heat rate	48.80 kW
	LT superheater heat rate	10.31 kW
	LT condenser heat rate	52.18 kW
	LT pump power input	0.30 kW
	LT expander power output	7.23 kW
	Quality	Superheated vapor
	first law efficiency	11.72%
	second law efficiency	57%
	LT expander Power cooling system waste heat	5.72 kW
	LT expander Power from downstream exhaust waste heat	1.21 kW
DNORC	Net power generation	20.13 kW
	1st law efficiency T	18.3%
	2nd law efficiency T	38%

The efficiency of the whole engine and the recovery system increased from 25% to 33% (Figure 2).

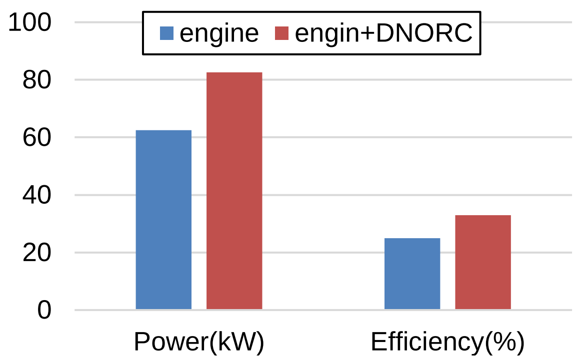

Figure 2: Chart for increasing the efficiency and engine power of the combination of the engine and Rankine

3-2) Validation

To validate the thermodynamic model, on the one hand, the sum of the work and the heat input into the system must be equal to the sum of the work and the heat output from the system according to the first law of thermodynamics, and, on the other hand, the network output from the system must not be more than the reversible work of the Carnot cycle with the same heat resources according to the second law of thermodynamics. In this work, it was observed that the sum of the energy input was equal to the sum of the energy output from the LTRC and HTRC. If the waste heat from the cooling fluid was considered as a high-temperature source at the average temperature of 92$^{\circ}C$ and the ambient air was considered as a low-temperature source at 25$^{\circ}C$, the highest thermal efficiency and the largest available work in the Carnot reversible cycle would be 18.3% and 9.17 kW, respectively. Regarding the exhaust, if the waste heat was considered as a high-temperature at the average temperature of 463.5$^{\circ}C$ with respect to a low-temperature source at 100$^{\circ}C$, the highest thermal efficiency and the largest available work from the downstream exhaust in the Carnot reversible cycle would be 49.3% and 33.55 kW, respectively (Figure 3).
The results obtained in the present work are lower than these 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.]31[, the maximum ideal and real RC efficiency value of 19% and 6% are achieved respectively. Liu et al. [32] reported that they obtained the highest thermal and exergy efficiency of 11.84% and 54.24%, respectively in their ORC system. Wenzhi et al. [33] reported a maximum 12% increase in power output in the system they provided. In performance analysis of combining a dual loop ORC with a gasoline engine, for validation purposes, the specification provided by Wang et al. [34] presented in Table 7, was examined in the thermodynamic model presented in this paper, and the results obtained in their study were compared in table 8 and 9. The margin of error with the results was negligible.

Table 7: System specifications provided by Wang et al. [34].

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

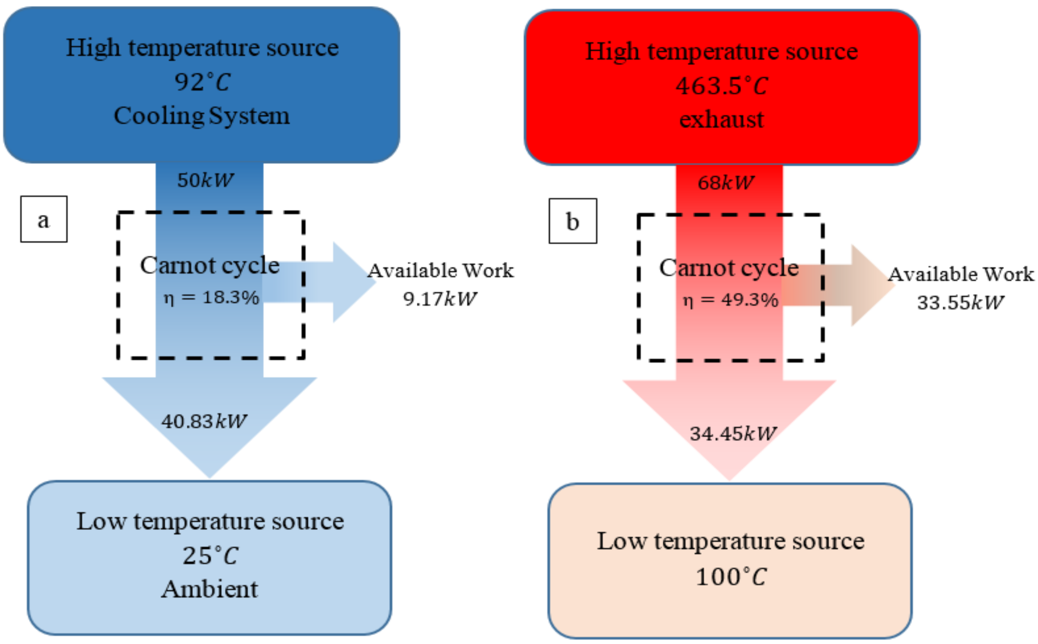

Figure 3: The highest efficiency and the largest available work in the Carnot reversible cycle for the waste heat source from a) the cooling system and b) the exhaust.

Table 8: Comparison of the results of the LTRC provided by Wang et al. [34].

Parameter	Wang et al.	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	1.10
$Q_{out}(kW)$	338.86	338.65	0.06

Table 9: Comparison of the results of the HTRC provided by Wang et al. [34].

Parameter	Wang et al.	present work	Relative error/%
$W_{Exp}(kW)$	10.85	10.70	1.30
$W_P(kW)$	1.277	1.28	0.23
$Q_{in}(kW)$	133.58	131.85	1.30
$Q_{out}(kW)$	124.00	122.43	1.20

4) Conclusion

In the present work, a system was proposed with the purpose of increasing the waste heat recovery from an ICE. In this work, an HTRC with a two-stage pumping along with an LTRC was presented in a DNORC. In addition to increasing the efficiency of the recovery system due to an increase in the mean effective temperature of heat addition, the HTRC allows direct recovery from the cooling system by adding fewer components than other recovery systems to the engine system.

The results showed that the use of this cycle increases the mean effective temperature of heat addition by 8% and increases the efficiency of the cycle by 4.6%. Although the direct power generation from the cooling system is low (0.27 kW) in the HTRC, this direct recovery helps to increase efficiency. In addition, the changes made to the engine system are relatively small.

By determining the suitable conditions for the combination of a DNORC and an engine and by choosing an appropriate working fluid, it is possible that by increasing system efficiency and the capability of absorbing the entire waste heat of the engine cooling system and exhaust with a low mass flow rate of the working fluid, in addition to increasing the waste heat recovery, one can avoid the increase of the system size, by removing the radiator of the engine cooling system and mounting fewer additional parts, the idea could be both extra power generation and engine cooling.

In comparison with other systems, a higher amount of the available energy in the cooling system can be obtained in the proposed system with a working fluid of a much lower mass flow rate.

Nomenclature

Latin symbols		Subscripts and superscripts	
c_P	specific heat ratio at constant pressure $\left({}^{kJ}/_{(k.K)}\right)$	ac	actual
c_{cf}	specific heat of the cooling fluid $\left({}^{kJ}/_{(kg.K)}\right)$	amb	ambient state
h	specific enthalpy $\left({}^{kJ}/_{kg}\right)$	f	fluid saturation point
$\dot{I}$	exergy destruction rate (kW)	fg	vaporization region
L	Latent heat $\left({}^{kJ}/_{kg}\right)$	cf	cooling fluid
m	mass flow rate $\left({}^{kg}/_{s}\right)$	$Cond$	condenser
P	pressure (kPa)	Eva	evaporative
Q	heat transfer (kJ)	Exp	expander
q	Q/m$\left({}^{kJ}/_{kg}\right)$	i	inlet
S_{gen}	entropy production $\left({}^{kJ}/_{K}\right)$	k	heat source
s	specific entropy $\left({}^{kJ}/_{(kg.K)}\right)$	o	outlet
T	temperature (K)	P	pump
$\bar{T}$	average Temperature (K)	RC	Rankine cycle
x	quality of fluid exiting from the turbine	rev	reversible
w	work per mass unit $\left({}^{kJ}/_{kg}\right)$	s	Isentropic
W	output power of turbine (kW)	t	total

Acronyms

$1stl$	first-law
$2ndl$	second-law
ICE	Internal Combustion Engine
$LTRC$	Low–Temperature Rankine Cycle
$HTRC$	High–Temperature Rankine Cycle
HT	High–Temperature
LT	Low– Temperature

Greek symbols

RC	Rankine Cycle
$DNORC$	dual-loop non-organic Organic Rankine Cycle
η	efficiency(%)

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افزایش بازیابی گرمای از دست رفته از موتوری احتراق داخلی با چرخهٔ رنکین غیرآلی دوگانه

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چکیده

این پژوهش با هدف افزایش توان تولیدی سامانهٔ بازیابی و با تمرکز بر افزایش انرژی ورودی به سامانهٔ بازیابی و بازده آن، ترکیب چرخهٔ رنکین غیرآلی دوگانه با یک موتور احتراق داخلی را بررسی می‌کند. راهبرد افزایش دمای میانگین موثر گرماگیری در چرخهٔ رنکین داغ (برای بهبود بازده سامانه) و راهبرد افزایش جذب گرمای اتلافی در چرخهٔ رنکین خنک (برای افزایش انرژی ورودی به سامانه) در چرخهٔ رنکین دوگانه بررسی می‌شود. در این سامانهٔ بازیابی، با تمرکز بر بازیابی گرمای اتلافی سامانهٔ خنک‌کاری موتور، می‌توان مبدل گرمائی را از سامانهٔ خنک‌کاری موتور حذف نمود. این طرح می‌تواند با افزودن قطعات کمتری به موتور، هم کار خنک‌کاری موتور را انجام دهد و هم توان اضافی تولید نماید. با استفاده از تحلیل ترمودینامیکی، شرایط ترکیب چرخهٔ رنکین غیرآلی دوگانه با موتور تعیین شد. نتایج این کار نشان داد که نرخ بازیابی با افزایش بازده سامانهٔ بازیابی و انرژی ورودی به سامانه، افزایش یافت. توان خروجی سامانه بیش از ۲۰ کیلووات حاصل شد و بازده کل سامانه به ۳۳ درصد رسید.

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موتور احتراق داخلی
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