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Liwei Wang^b, Anthony Paul Roskilly^a,

^b*Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University, Shanghai, 200240, China*

- A novel integrated cogeneration is proposed to produce power and refrigeration.
- The coolant and exhaust energy from the engine are both recovered.
- The characteristics performance of the cogeneration is investigated.
- The improvement of energy efficiency and fuel consumption are evaluated.

Combined power and refrigeration system; Waste heat recovery; Organic Rankine cycle;
Resorption; Performance analysis

E-mail address: y.j.lu@ncl.ac.uk

19 **Abstract**

20 This paper presents a novel idea of integrating a refrigeration and power system using
21 Organic Rankine cycle (ORC) and solid sorption technology into Internal Combustion Engine
22 (ICE). A 1-D engine model has been coded to evaluate the waste heat quantity of a medium-
23 duty diesel engine. The recoverable waste heat from the coolant and exhaust system has
24 been analysed under engine overall operational region. Based on these results, the working
25 conditions of a cogeneration are designed and the performance of the cogeneration is
26 evaluated throughout the engine operating region. Results indicate that the maximum
27 recoverable from the coolant and exhaust system is 67.9 kW and 82.7 kW, respectively.
28 Meanwhile, the cogeneration can provide the power of 13.88 kW and refrigeration of 16.58
29 kW at the engine rated point. This system has the potential to improve the overall energy
30 efficiency of the ICE from 40 % to 47 % and reduce the average brake specific fuel
31 consumption (BSFC) from 205 g/kWh to 180 g/kWh at full load conditions.

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$BSFC$	brake specific fuel consumption (g/kWh)
C_p	specific heat at constant pressure (kJ/kg ·K)
COP	coefficient of performance
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
M	molar mass (g/mol)
$M_r(i)$	molar mass of different gas (g/mol)
$n(i)$	quantity of different gas (mol)
P	pressure (kPa)
P_0	reference pressure (1×10^5 Pa)
$\dot{Q}$	heat (kW)
R_0	universal gas constant [kJ/(mol· K)]
s	specific entropy [kJ/(kg· K)]
T	temperature (°C)
u	specific internal energy (kJ/kg)
$\dot{W}$	power (kW)
x_i	mass fraction of different gas
Greek letters	
η	efficiency
ΔH	reaction enthalpy (J/mol)
ΔS	reaction entropy [J/(mol · K)]
Δx	conversion ratio
Subscripts	
am	ammonia
b	brake
cog	cogeneration
eq	equilibrium point
ex	exhaust

<i>exp</i>	expander
<i>h</i>	heat of high temperature salt
<i>he1</i>	heater 1
<i>he2</i>	heater 2
<i>i</i>	indicated
<i>in</i>	inlet
<i>out</i>	outlet
<i>p</i>	pump
<i>r</i>	relative
<i>re</i>	regenerator
<i>ref</i>	refrigeration
Acronyms	
<i>ATS</i>	after-treatment system
<i>COP</i>	coefficient of performance
<i>DPF</i>	diesel particulate filter
<i>HTS</i>	high temperature salt
<i>ICE</i>	internal combustion engine
<i>LTS</i>	low temperature salt
<i>ORC</i>	Organic Rankine cycle
<i>SCR</i>	selective catalytic reduction
<i>WHR</i>	waste heat recovery

38 **1. Introduction**

39 With the growing concerns about energy crisis, more attentions have been attracted to
40 improve the efficiency of the existing energy systems to reduce the energy consumption.
41 Internal combustion engine (ICE) consumes huge amount of energy and about 60 % energy
42 is wasted from the exhaust system and cooling system of ICE [1, 2]. Recovering the wasted
43 heat from the ICE into useful energy can effectively increase the overall energy efficiency,
44 decrease the fuel consumption and reduce the emissions. Several thermodynamic cycles
45 were proposed and investigated to integrate with the ICE such as Organic Rankine cycle
46 (ORC), Brayton cycle, adsorption cycle, Kalina cycle and Stirling cycle[3, 4]. Among all of the
47 proposed waste heat recovery (WHR) system, ORC has been recognized as one of the most
48 promising technologies and affordable power generation systems for the WHR of ICE [5]. On
49 the other hand, thermally activated refrigeration has increasingly attracted the attentions of
50 researchers such as adsorption, absorption and desiccant system [6-8]. Adsorption
51 technology has been identified as one of the most practical technologies producing
52 refrigeration from the waste heat of ICE [6, 9]

53 Exhaust gas contains around 30% energy of the total combustion energy from the engine.
54 Many researchers integrated ORC system with ICE to recovery the exhaust energy. Zhang et
55 al. [3] developed an ORC system using R123 as the working fluid and the maximum power
56 output of the ORC is 10.38 kW under the engine brake power at 250 kW. Srinivasan et al. [10]
57 evaluated the potential improvements of a dual-fuel ICE using a bottoming ORC system and
58 the average emissions reductions was achieved by 18% while the fuel conversion efficiency
59 was improved by 7%. Shu et al. [11] conducted an investigation on the performance of
60 Alkane-based ORC system to recovery the exhaust energy from diesel engine. Based on the

61 energy and exergy analysis, the cyclic Alkanes Cyclohexane and Cyclopentane were
62 recognized as the most suitable working fluid in the WHR system from ICE.

63 Applying heat-driven refrigeration system such as adsorption, absorption and desiccant
64 technologies [7, 8, 12-19] into ICE is the other effectively approach of recovering the waste
65 heat from the ICE. Wang et al.[4] tested an adsorption refrigeration system integrated with a
66 40 hp diesel engine and the results indicated this system could achieve the COP at 0.38 using
67 zeolite-water as the adsorption working pair. Jiangzhou et al. [16, 20] experimentally
68 investigated the performance of an adsorption air conditioner with locomotive, which could
69 generate 3 kW to 5 kW cooling capacity with the COP achieved at 0.21. Wang et al. [7]
70 conducted an investigation of a mixed effect absorption refrigeration system integrated with
71 ICE using both jacket water and exhaust gas as the heat source, which could theoretically
72 produce 34.4kw refrigeration from a 16kw ICE. A micro tri-generation system was
73 developed and investigated by Henning et al.[8]. This tri-generation system applies the
74 desiccant system for the refrigeration using the waste heat from the ICE, which can
75 promisingly save more than 30% of electricity energy compared with a conventional air
76 handling unit. Nevertheless, the demand of improving the overall energy efficiency of the
77 ICE forces the scientist to find new approaches to recovery the waste heat.

78 Several advanced heat-driven energy systems were proposed and studied to improve the
79 energy efficiency of existing systems through recovering the waste heat, such as dual-loop
80 ORC system[21], combined ORC/Kalina system[22], trilateral cycle[23] and transcritical ORC
81 system[24], etc. Wang et al. [2]conducted a characteristics study of a novel dual-loop ORC
82 system recovering the coolant and exhaust heat from a gasoline engine. The effective
83 improvement of the thermal efficiency of the ICE was achieved by 3-6% under engine
84 various operational region. A performance analysis of a dual-loop ORC system recovering the
85 waste heat from engine intake air, coolant and exhaust was reported by Zhang et al. [25],
86 which has indicated a promising application of this system with light-duty diesel engine.

Boretti reported a study on applying two ORC systems to recovery the coolant and exhaust energy using R245fa as working fluid. The contribution from the exhaust ORC for the fuel consumption was around 3.4% while the coolant ORC increased the fuel efficiency by 1.7% [26]. A combined thermodynamic cycle including a ORC and Kalina cycle was proposed by He et al. [22], which could simultaneously recover the heat from exhaust, coolant and lubrication system. Extensive studies on combined power/cooling cycle using low-grade heat as the heat source were conducted by several research groups since 2000s[27-32]. Liang et al. [27] studied a cascade cycle including a Rakine cycle and an absorption refrigeration system for electricity and cooling production from a marine engine. The results indicated a promising exergy efficiency increased by 84% when the condensing temperature is 323 K and the superheat is 100 K. However, the investigation of the combination of ORC/resorption technology for power/refrigeration generation using waste heat from ICE has never been reported and evaluated under engine overall operating conditions.

In this study, a novel combined power and refrigeration system using ORC and resorption system was proposed and assessed to recovery both the coolant energy and exhaust energy from a medium-duty diesel engine under various operating conditions. A 1D fluid-dynamic engine including the after-treatment system (ATS) was coded in WAVE to evaluate the operating characteristics and recoverable waste heat of a diesel engine. The after-treatment unit of the recently developed engine consumes some of the exhaust heat from the engine to react with the emission gases. Therefore, the evaluation of recoverable heat from the exhaust system after the ATS under engine overall operational region, which has hardly been considered and reported from the previous studies, is valuable for the development of waste heat recovery system into ICE. This cogeneration applies resorption system for the refrigeration generation, which contains no liquid phase working fluid in the system and is therefore favourable to be used under vibration condition[29]. R245fa was selected as the working fluid of the power generation part and $\text{MnCl}_2\text{-SrCl}_2$ was chosen as the working pair

of the refrigeration part of the cogeneration system. The mathematical model of the cogeneration was coded in Engineering Equation solver (EES)[33]. The performance map of the cogeneration system for power and refrigeration generation was analysed under engine total operational region.

2. Design of the system

(a)

Fig. 1. Schematic diagram of the integrated ORC and resorption system for power and refrigeration (a) HTS₁ and LTS₁, regeneration process, HTS₂ and LTS₂, refrigeration process; (b) HTS₁ and LTS₁, refrigeration process, HTS₂ and LTS₂, regeneration process

The system design is shown in Fig. 1, which is a combination of ORC and resorption refrigeration system. This system is designed to recovery both the coolant energy and exhaust energy from engine into power and refrigeration.

The ORC contains a pump, a regenerator, two evaporators, an expander and a condenser. The heater 1 recovers the waste heat from the coolant while exhaust energy is recovered by the heater 2. The working principle of this ORC is illustrated in Fig. 2. The inlet and outlet conditions of the expander are both designed at superheated region of the R245fa, which are represented as point 5 and point 6 in Fig. 2. The resorption system locates after the outlet of the expander to recover the heat from the gas phase R245fa. This cogeneration applies two sets of resorption systems to generate continuous refrigeration. One set of the resorption system is in refrigeration process while the other set is in regeneration process.

The principle of the system switch between two sets of resorption system is illustrated in Fig. 1 (a) and Fig. 1 (b). The operation process of resorption system is shown in Fig. 3 and the theoretical Clapeyron diagram is based on the following equations:

Resorption system is an advanced adsorption cycle replacing the evaporator/condenser of conventional adsorption system with a second adsorbent bed, which eliminates the liquid phase fluid inside the system. Under the same equilibrium pressure, the adsorbent who has a relative higher equilibrium temperature is called high temperature salt (HTS) and the other adsorbent is named as low temperature salt (LTS). The resorption system utilises desorption heat of LTS as refrigeration, which is almost twice of the latent heat of ammonia. Therefore the cooling capacity of resorption system is almost twice of that from conventional adsorption system [34, 35].

As shown in Fig. 3, line A-B is on the equilibrium line of the HTS and illustrates the heat required for the HTS before reaction. Line B-C is the regeneration process when the ammonia desorbs from the HTS and adsorbs by the LTS. During this process, the LTS rejects the adsorption heat into the environment by the air and the HTS is heated by the heat source. Line C-D is on the equilibrium line of the LTS and illustrates the heat removal process of the LTS from the environmental temperature T_A to refrigeration temperature T_D . The refrigeration process is indicated as line D-A. The ammonia desorbs from the LTS and adsorbs by the HTS during this process. The adsorption heat from the HTS is removed by the air and the cooling effect is provided by the desorption heat of LTS. The heat provided from the ORC to the resorption system has been illustrated from Point 6 to Point 6' in Fig. 2. Line 6'-7 and line 2-2' represent the heat regeneration process in the regenerator of the ORC.

Fig. 2. Schematic T-s diagram of the Organic Rankine cycle

Fig. 3. Theoretical Clapeyron diagram of the resorption system

3. Mathematical model

The required power for the pump is described as

$$\dot{W}_p = \dot{m} \times (h_2 - h_1) / \eta_p \quad (2)$$

157 Where $\dot{W}_p$ is the power consumed by the pump, $\dot{m}$ is the mass flow rate of the ORC system
 158 and η_p is the isentropic efficiency of the pump.

159 The total heat provided to the cogeneration from the ICE includes the heat from coolant
 160 water and the heat from exhaust. The heat transfer efficiency of heater 1 and heater 2 are
 161 η_{he1} and η_{he2} , respectively.

$$\begin{aligned}\dot{Q}_{he1} + \dot{Q}_{he2} &= \dot{m} \times (h_5 - h_2) \\ &= \dot{Q}_{coolant} \times \eta_{he1} + \dot{Q}_{exhaust} \times \eta_{he2} \\ &= \dot{Q}_{coolant} \times \eta_{he1} + \dot{m}_{ex} \times (h_{exin} - h_{exout}) \times \eta_{he2}\end{aligned}\quad (3)$$

162 The regeneration process happens in the regenerator is calculated as

$$\dot{Q}_{re} = \dot{m} \times (h_2 - h_7) = \dot{m} \times (h_6 - h_7) \times \eta_{re} \quad (4)$$

163 The power generated by the expansion machine is

$$\dot{W}_{ORC} = \dot{m} \times (h_5 - h_6) \times \eta_{exp} \quad (5)$$

164 The total amount of heat consumed by the resorption system for the regeneration process is
 165 expressed by this equation, which includes the reaction heat and the sensible heat of the
 166 adsorbent. $\dot{Q}_h$ is the total heat consumed by the HTS and T_h is the heat source temperature.

$$\begin{aligned}\dot{Q}_h &= \dot{m}_{am} \times \Delta x \times \Delta h_{HTS} + \dot{m}_{am} \times \Delta u_{HTS} \\ &= \dot{m}_{am} \times \Delta x \times \Delta h_{HTS} + \dot{m}_{HTS} \times C_{pHTS} \times (T_h - T_{air}) + \dot{m}_{am} \times C_{pam} \times (T_h - T_{air}) \\ &= \dot{m} \times (h_6 - h_7) \times \eta_{HTS}\end{aligned}\quad (6)$$

167 The refrigeration provided from the LTS is then calculated by Eq. (7). The first part of this
 168 equation is the reaction heat of the LTS and the second part is the sensible heat required for
 169 the LTS to cool down at refrigeration temperature.

$$\begin{aligned}\dot{Q}_{ref} &= \dot{m}_{am} \times \Delta x \times \Delta h_{LTS} + \dot{m}_{am} \times \Delta u_{LTS} \\ &= \dot{m}_{am} \times \Delta x \times \Delta h_{LTS} - \dot{m}_{LTS} \times C_{pLTS} \times (T_{air} - T_{ref}) - \dot{m}_{am} \times C_{pam} \times (T_{air} - T_{ref})\end{aligned}\quad (7)$$

The thermodynamic parameters used for the calculation of the resorption system are list in Table 1. The equilibrium lines of SrCl_2 and MnCl_2 plotted in Fig. 3 were obtained from the Clausius-Clapeyron equation[36].

$$\ln\left(\frac{P_{eq}}{P_0}\right) = -\frac{\Delta H}{R_0 T_{eq}} + \frac{\Delta S}{R_0} \quad (8)$$

Table 1

Thermodynamic properties of the salts and ammonia[36]

Molecular formula	ΔH (J/mol)	ΔS [J/(mol·K)]	C_p [J/(mol·K)]	M (g/mol)	Reaction salt with per mol ammonia (g)
NH_3	22839	191.39	80.27(liquid)	17.03	
$\text{SrCl}_2(8-1)$	41431	228.8	75.53	158.53	22.65
$\text{MnCl}_2(6-2)$	47416	228.07	72.86	125.84	31.46

The effective energy efficiency of the ICE, ICE integrated with ORC only and ICE integrated with the cogeneration are defined as following equations. $\dot{m}_{fuel}$ is the mass flow rate of the fuel consumed by the ICE and h_{fuel} is the heating value of the diesel, which is set at 43400 kJ/kg in this study.

$$\eta_{ICE} = \frac{\dot{W}_b}{\dot{m}_{fuel} \times h_{fuel}} \quad (9)$$

$$\eta_{ICE+ORC} = \frac{\dot{W}_b + \dot{W}_{ORC} - \dot{W}_p}{\dot{m}_{fuel} \times h_{fuel}} \quad (10)$$

$$\eta_{ICE+cog} = \frac{\dot{Q}_{ref} + \dot{W}_b + \dot{W}_{ORC} - \dot{W}_p}{\dot{m}_{fuel} \times h_{fuel}} \quad (11)$$

The brake specific fuel consumption (BSFC) of the engine is calculated as

$$BSFC_{ICE} = \frac{\dot{m}_{fuel}}{\dot{W}_b} \quad (12)$$

The BSFC of the ICE with the ORC system is

$$BSFC_{ICE+ORC} = \frac{\dot{m}_{fuel}}{\dot{W}_b + \dot{W}_{ORC} - \dot{W}_p} \quad (13)$$

181 To evaluate the contribution of the refrigeration for the fuel consumption, relative brake
 182 power is defined from the following equation. COP_r is the relative coefficient of
 183 performance from conventional refrigeration system in vehicle application.

$$\dot{W}_r = \frac{\dot{Q}_{ref}}{COP_r} \quad (14)$$

184 Therefore, the BSFC of the cogeneration system can be calculated as

$$BSFC_{ICE+cog} = \frac{\dot{m}_{fuel}}{\dot{W}_b + \dot{W}_{ORC} + \dot{W}_r - \dot{W}_p} \quad (15)$$

185 To analyse the operational characteristics of the cogeneration with the ICE under various
 186 operational conditions, several assumptions are applied in this study.

- 187 1. The operating characteristics of the ICE are all in steady state conditions and the
 188 heat loss from the pipe lines/engine block are ignored.
- 189 2. The pressure at the inlet of the expander is set at 790.8 kPa, where the evaporating
 190 temperature of R245fa is 80 °C. The coolant energy could be comprehensively
 191 utilized for the cogeneration.
- 192 3. The condensing temperature of the ORC and the heat sink temperature T_A of the
 193 resorption system are set at 30 °C.
- 194 4. The isentropic efficiency of the pump η_p and the expander η_{exp} are both set at 0.8.
- 195 5. The heat transfer efficiency of all the heat exchangers are all set at 0.95 including
 196 η_{he1} , η_{he2} , η_{re} , and η_{HTS} .
- 197 6. Considering about the harsh working conditions of the ICE, the conversion ratio of
 198 the resorption system Δx is assumed at 50%.

7. The COP of the conventional transportation refrigeration system is generally ranging from 0.5 to 1.5 [37]. In this study, the relative COP is set at 1.5, which has been used in Eq. (14).

4. Results and discussions

4.1 Waste heat evaluation

To investigate the operating characteristics and evaluate the waste heat quantity of ICE, a 1D engine dynamics model has been coded in Ricardo WAVE, which is a widely used engine simulation tool in industry sectors including passenger car, motorcycle, truck and locomotive, etc [38, 39]. The specifications of the diesel engine are list in Table 2. The simulation model mainly consists of intake system, an intake manifold, cylinders/injectors, exhaust manifold and the ATS. The ATS includes a diesel particulate filter (DPF) and selective catalytic reduction (SCR) system. This model was calibrated and the simulation results were validated in previous study by the authors [40].

Table 2

Engine specifications

Characteristic	Value
Engine model	ISB 4.5
Engine Displacement (L)	4.5
Number of cylinders	4
Cylinder bore (mm)	107
Cylinder stroke (mm)	124
Compression ratio	17.3
Fuel injection system	High pressure Common Rail Direct Injection
Injection pressure (bar)	1800

Turbocharger	Waste Gate Turbocharger
Max Torque (N.m)	760 at 1400-1800 RPM
Max Power (kW)	152 at 2300 RPM
Emission	Euro V

214 The energy conservation for the combustion process can be described by

$$\dot{m}_{fuel} \times h_{fuel} + \dot{m}_{air} \times h_{air} = \dot{W}_i + \dot{Q}_{coolant} + \dot{m}_{ex} \times h_{exin} - \dot{Q}_{loss} \quad (16)$$

215

216 $\dot{m}_{air}$ and h_{air} are the air mass flow rate and enthalpy of the intake air during the combustion
217 process. $\dot{W}_i$ is the indicated power from the diesel engine. $\dot{Q}_{loss}$ is the heat loss from the
218 pipelines and engine block, which has been ignored in this study. The engine indicated
219 power, exhaust mass flow rate and the temperature of the exhaust air after the SCR unit
220 were collected from the simulation model and the results were plotted in the Fig. 4 (a), (b)
221 and (c), respectively. The total input energy for the combustion process including the fuel
222 energy and the energy of intake air has been calculated based on the Eq. (16) and the results
223 are presented in Fig. 4 (d). The highest exhaust mass flow rate achieved from the 4.5 L diesel
224 engine was around 0.19 kg/s, which was on the performance curve under the operational
225 speed ranging from 2100 rpm to 2400 rpm. The temperature of the exhaust air after SCR
226 almost increases linearly with the torque. The highest exhaust temperature was achieved at
227 700 °C around the engine rated point. The combustion energy of the fuel and intake air is
228 calculated from the Eq. (16) and the performance map of the combustion energy is shown in
229 Fig. 4 (d).

(a)

(b)

(c)

(d)

Fig. 4. Engine performance evaluation of a medium-duty diesel engine (a) brake power, (b) mass flow rate of the exhaust air, (c) temperature of the exhaust air after SCR, (d) combustion energy

Petroleum diesel contains around 75% saturated hydrocarbons and 25% aromatic

hydrocarbons. The average chemical formula of petroleum diesel can be presented by

$C_{12}H_{23}$ [41]. The exhaust heat exchanger is located after the ATS. Therefore, the chemical

reaction equation for the combustion process can be written as

The mass fraction of the CO_2 , H_2O , N_2 and O_2 from the exhaust gas can be calculated from

$$x_i = n(i) \times M_r(i) / \sum_{i=1}^N [n(i) \times M_r(i)] \quad (18)$$

The specific enthalpy of the exhaust gas under different temperature is defined as

$$h_{ex}(T) = x_{CO_2} \times h_{CO_2}(T) + x_{H_2O} \times h_{H_2O}(T) + x_{N_2} \times h_{N_2}(T) + x_{O_2} \times h_{O_2}(T) \quad (19)$$

239 The exhaust temperature is ranging from 200 °C to 700 °C and the exhaust pressure is very
 240 close to environmental pressure. Therefore, the exhaust gas can be treated as a mixture of
 241 ideal gases. The enthalpy of the exhaust gas is drawn in Fig. 5 (a), which has a similar
 242 tendency of the exhaust mass flow rate in Fig. 4 (b). This indicates the mass flow rate has
 243 more influence on the exhaust enthalpy than the exhaust temperature. The waste heat from
 244 the coolant water is calculated by the Eq. (12) and the calculated results are plotted in Fig. 5
 245 (b). The waste heat from the engine coolant is almost linearly related with the engine torque.
 246 The recoverable heat from the exhaust system has been defined in Eq. (3). The temperature
 247 of the exhaust gas after the ATS are collected from the model and drawn in Fig. 5 (c). The
 248 waste heat produced from the exhaust system is then plotted in Fig. 5 (d). The recoverable
 249 heat achieved at engine rated point from the exhaust system and coolant system is 82.7 kW
 250 and 67.9 kW, respectively.

(a)

(b)

(c)

(d)

Fig. 5. Evaluation of the waste heat of a medium-duty diesel engine (a) Enthalpy of the exhaust air, (b) waste heat from coolant water, (c) recoverable heat from exhaust air

4.2 Performance of power and refrigeration

To comprehensively utilize the waste heat from two sources of the engine, the evaporation temperature of the ORC is designed at 80 °C, where the coolant heat is sufficient to produce the required energy for the working fluid of the ORC to reach the two phase region. The designed working conditions of the ORC system can be found in Table 3. The mass flow rate of the ORC system is calculated from the Eq. (3) and plotted in Fig. 6 (a). The highest mass flow rate is 0.45 kg/s at the engine rated point. Based on all the data collected above, the performance of the cogeneration system can be calculated by the Eq. (2)-(8). The performance maps for power and refrigeration of the cogeneration system are shown in Fig. 6 (b) and (c), respectively. The maximum power provided from the system is 13.88 kW and the maximum refrigeration generated from the cogeneration is 16.58 kW at engine rated point.

Table 3

Thermodynamic parameters of ORC system

Point	Temperature	Pressure	Specific	Specific
	(°C)	(kPa)	Enthalpy	Entropy
			(kJ/kg)	(kJ/kg.K)

1	30	177.2	239.1	1.135
2	30.2	790.8	239.6	1.135
2'	50.6	790.8	267.2	1.228
3	80	790.8	309.2	1.348
4	80	790.8	461.7	1.779
5	160	790.8	553	2.013
6	120.4	177.2	516.3	2.013
6'	60	177.2	455.5	1.845
7	30	177.2	426.5	1.754

(a)

(b)

(c)

Fig. 6. Evaluation for power and refrigeration performance of the cogeneration under engine' various operational conditions (a) ORC mass flow rate, (b) Power generated from the cogeneration, (c) Refrigeration obtained from the cogeneration

The overall energy efficiency of the ICE is around 40%, when the engine is running at full load conditions. The cogeneration system is able to improve the energy efficiency of the ICE by 7% and the ORC only system can improve the energy efficiency by 3%. The comparison of the energy efficiency of the ICE, ICE + ORC and ICE + Cogeneration at full load conditions is shown in Fig. 7. Furthermore, the relative COP is introduced here to evaluate the contribution of the refrigeration part of the cogeneration for the fuel consumption. The calculation methods can be found in Eq (12)-(15). The effect of the cogeneration and ORC for the fuel consumption of the ICE is plotted in Fig. 8. Results indicate the average BSFC of the ICE has been reduced from 205 g/kWh to 190 g/kWh with the application of ORC system and the average BSFC of the ICE + cogeneration system is around 180 g/kWh, when the engine is running at full load conditions.

Fig. 7. Improvement of energy efficiency

Fig. 8. Improvement of BSFC

5. Conclusion

In this paper, a novel power and refrigeration cycle was proposed for the integration with internal combustion engine. An engine model was coded in WAVE to evaluate the

recoverable waste heat from a 4.5 L diesel engine. The recoverable wasted heat from the coolant and exhaust system were analysed under the engine operational region. The design and the working conditions of the cogeneration were then proposed to comprehensively recovery both of the heat sources from the ICE. The performance characteristics of the cogeneration were investigated under various operating conditions of the ICE. The conclusions of this study can be summarised as

1. The exhaust enthalpy occupies about 40% of the total input energy of the ICE and the exhaust enthalpy is slightly greater than the indicated power throughout the engine operational conditions. However, the recoverable heat from the exhaust system after the after-treatment unit only occupies 25% to 40% of the exhaust enthalpy under different engine loads and speed. The highest recoverable heat from the exhaust system is 82.7 kW achieved at engine rated point
2. The waste heat from the coolant system shows linear increase with the torque. The highest coolant heat obtained at engine rated point is 67.9 kW, which is lower than that from the exhaust system. Results also indicate coolant system can provide more energy than exhaust system under engine low operational region.
3. An integrated cogeneration system is then proposed and analysed to produce continuously power and refrigeration from the waste heat of a medium duty diesel engine. To comprehensively utilise the waste heat from the coolant water, which has been proven as a valuable heat source under engine low operational region, the evaporation temperature of the ORC system is set at 80 °C. The cogeneration applies R245fa as the working fluid in the ORC system and $\text{SrCl}_2\text{-MnCl}_2$ as the working pair in the resorption system. The cogeneration achieved 13.88 kW power and 16.58 kW refrigeration at the engine rated point.
4. The cogeneration system is able to improve the energy efficiency of the ICE by 7% at full load conditions. The average BSFC can also be reduced from 205 g/kWh to 180

g/kWh after the integration of this system. This cogeneration system has the advantages of reducing fuel consumption of ICE, producing power/refrigeration simultaneously and comprehensively utilising both the coolant and exhaust energy from the ICE.

In summary, this novel integrated system has the potential applications for food transport refrigeration where the reduction of fuel consumption is preferable and the refrigeration is required.

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