Fuel 322 (2022) 124036

Contents lists available at ScienceDirect

Fuel

journal homepage: www.elsevier.com/locate/fuel

Full Length Article

Power enhancement of a turbo-charged industrial diesel engine by using of a waste heat recovery system based on inverted Brayton and organic Rankine cycles

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ARTICLE INFO

Keywords:
Waste heat recovery
Inverted Brayton cycle
Organic Rankine cycle
Diesel engine
Engine backpressure

ABSTRACT

In this study, energy assessment is performed for an industrial turbocharged diesel engine integrated with a novel waste heat recovery (WHR) system. The exhaust energy is used in inverted Brayton cycle (IBC) for waste heat recovery purpose. Also, the heat energy from IBC heat exchanger is used as the heat source for the organic Rankine cycle (ORC) to produce extra power. The case study engine is modelled in AVL BOOST software and the model is validated against real engine performance data. For studying the performance of the proposed waste energy recovery system, IBC is added to the engine model in AVL BOOST software and the thermodynamic model of the ORC is developed in MATLAB and it is linked to AVL BOOST. Then, the model is solved, and the main engine output parameters are studied at 1800 RPM and various engine loads. The results show that employment of the proposed WHR system leads to enhancement of the system power by about 18%. However, the backpressure produced by installing the WHR system can result in increase of the BSFC up to 3% and reduction of the total thermal efficiency by almost 1% at engine full load condition. The results of this work contribute to determine the interactions between the proposed novel waste heat recovery system (IBC-ORC) and the engine. The proposed bottoming cycle based on IBC-ORC can be installed on existing industrial stationary engines for enhancement of power generation without imposing a new source of power generation.

1. Introduction

Today, governments around the world are applying stringent legislations to promote less-polluting and more efficient internal combustion (IC) engines for different power production applications [1,2]. Therefore, it is important to study the new methods for improving the IC engines efficiency and performance which ultimately leads to the reduction of fossil fuel consumption rate and emissions [3]. In this concept, industry recently started to move towards electrification, however, it will be very challenging for heavy-duty applications at least in near future. On the other hand, the large heavy-duty industrial engines used in trucks, mining and other applications are responsible for a significant share of emissions [4]. They are using diesel or other heavy fuels to generate mechanical or electrical power in different industries [5,6]. The high amount of power generated by such heavy-duty engine requires high fuel consumption which leads to high quantity of emissions. Therefore, they are responsible for a large amount of emissions by burning high amount of fossil fuels [7].

Using waste heat recovery (WHR) systems can help in providing the extra power and improving the performance of heavy-duty industrial engines in different energy systems [8]. For example, the WHR is widely used in multi-generation systems in different buildings and industries [9,10]. WHR especially is beneficial for industrial diesel engines [11] because the exhaust mass flow rate and temperature are both very high. Different thermodynamic power generation cycles are used in WHR systems for converting engine waste heat to mechanical power while the mechanical power can be converted to electrical power by an electric motor. Among various thermodynamic cycles, inverted Brayton cycle (IBC) [12,13] and organic Rankine cycle (ORC) [14] are among the most attractive options for WHR of the heavy-duty engines due to their high thermal efficiency.

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https://doi.org/10.1016/j.fuel.2022.124036
Received 20 January 2022; Received in revised form 13 March 2022; Accepted 28 March 2022
Available online 12 April 2022

Nomenclature

IBC	Inverted Brayton cycle
ORC	Organic Rankine cycle
WHR	Waste heat recovery
BSFC	Brake specific fuel consumption/g $kW^{-1}h^{-1}$
$\dot{m}$	Mass flow rate/kg s^{-1}
h	Enthalpy/kJ kg^{-1}
$\dot{Q}$	Heat transfer rate/kW
$\dot{W}$	Power/kW

Subscripts

t	Turbine
p	Pump
c	Compressor
HX	Heat exchanger
acc	Air cooled condenser
th	Thermal

The ORC was widely considered for recovering engine waste heat in mobile and industrial applications for many years [14]. However, it has its own challenges such as the sensitivity of the ORC fluid to the grade of the input energy and its efficiency. Shi et al., [15] recommended an ORC system for waste heat recovery from a heavy-duty truck engine. The waste heat from engine water jacket and the exhaust was transferred to ORC using two heat exchangers. They also evaluated the system performance at various operating conditions for finding the optimal design point and they provided some specific optimal conditions for engine off-design mode. Liu et al., [16] employed pinch analysis to find the best ORC operating condition for the waste heat recovery from multiple sources in a heavy-duty engine. The results of their work showed that by employment of ORC with preheater, the thermal efficiency can be changed from 9.28% to 14.23%. In another similar study about coupling of the dual-loop ORC to a heavy-duty engine for WHR from the engine exhaust and water jacket heat [14], extra power of about 25% was recovered by the ORC system. The operating fluid of the ORC cycle was R143a in that study. Additionally, based on the review of the recent works on WHR by Lion et al., [17], ORC was the most promising WHR system for the internal combustion engines which is capable of fuel consumption enhancement of up to 10%.

ORC was also used in different levels of WHR systems coupled to the engine. For example, Wang et al., [18] studied the integration of solid oxide electrolyzer (SOE) with an internal combustion engine while the heat from the engine exhaust was used in SOE as an extra input energy source. Additionally, in this study, the ORC was employed for waste heat recovery of the low-grade energy from the SOE system. The efficiency of the employed ORC was reported to be 12.12% which enhanced the total efficiency to 44.13%. Furthermore, Di Battista et al., [19] worked on integration of ORC for waste heat recovery from a CO_2 based Rankine cycle which is coupled with an engine. The low-grade heat available through the condenser of the CO_2 Rankine cycle is transferred to ORC for extra power generation resulting in enhancement of the system total efficiency between 3% and 4%. In another research performed by Lion et al., [20] the impact of ORC employment for low-grade waste heat recovery of the EGR cooling system in a diesel engine was presented. In this WHR system, the waste heat from the engine exhaust and EGR system was recovered simultaneously by using two evaporators in the ORC WHR system resulting in reduction of the total system fuel consumption by 10.6%.

IBC is another thermodynamic cycle which can be used for WHR application in the heavy-duty engines [21,22]. It can be employed as the waste heat recovery system of gas turbine or high temperature fuel cell, while, it can be also used for recovering the low-grade waste energy from low grade heat sources [23]. The working fluid of IBC is gas, therefore, the engine exhaust can be used directly in IBC leading to removal of intermediate working fluid for WHR and reducing the components of WHR system [24]. Relevant studies about IBC application in WHR of the internal combustion engines is limited compared to ORC cycle. It was recently considered as an option for WHR in mobile applications. For example, the impact of adding IBC on the performance of the vehicle propulsion system in light-duty engine applications was studied by Chen et al., [25]. In this work, IBC was used as the bottoming cycle of a naturally aspirated engine. This engine with IBC was studied under WLTP driving cycle for evaluating its performance. It was reported that the engine BSFC was improved by nearly 3.02% with employment of IBC as the WHR system. One of the main important parameters affecting IBC performance is the operating pressure of IBC components. In a parametric analysis done by Copeland et al., [22] the impact of sub-atmospheric pressure produced by IBC compressor on IBC total power output was studied. Based on their results, the pressure reduction of 0.2–0.3 bar in IBC can result in significant increment of IBC thermal efficiency and power generation rate. In another study performed by Copeland et al., [24] the impact of employing IBC as the waste heat recovery system of a vehicle engine in various conditions was highlighted. In this study, IBC is numerically modeled, and it was optimized to achieve the maximum power output. The optimization was performed by considering different range of temperatures in IBC system.

An important component of the IBC is the heat exchanger which is responsible for reducing the temperature of air before entering the IBC compressor. A low to medium grade energy is wasted through IBC heat exchanger and there is a potential for recovering this waste heat, for example by using an ORC system. Abrosimov et al., [12] studied the coupling of IBC and ORC to provide an efficient WHR system for high temperature waste heat harvesting. In this study, an arbitrary exhaust gas at temperature of 470–570 °C was introduced into a combined IBC-ORC system as the typical exhaust flow of a heat engine, and the performance of WHR system at various conditions was evaluated. It is reported that increment of exhaust flow temperature resulted in increase of IBC power generation rate and decrease of ORC power production rate. However, the WHR system was studied separately from the engine and the impacts on engine performance parameters such as back pressure and BSFC were not considered.

From the literature, ORC can be used in a compound WHR systems for extra power generation from low-grade energy wasted in the primary WHR system. So, combining ORC and IBC systems to provide a novel waste heat recovery would be beneficial. However, the impacts of adding such waste heat recovery system on the engine performance has not been addressed in the literature. IBC components such as turbine and compressor directly affect the performance of the engine by applying extra backpressure. The increase of backpressure leads to reduction of engine volumetric efficiency and increment of the fuel consumption. A comprehensive study is needed to evaluate advantages and disadvantages of such WHR system (IBC-ORC) in combination with the engine. In this study, a novel IBC-ORC waste heat recovery system is recommended for a turbocharged diesel engine while the effects on engine performance are also included. The low-grade waste heat from the exhaust in IBC condenser is transferred to ORC for generating extra electrical power which is otherwise wasted. To the best of our knowledge, this is the first study that is evaluating the combined IBC-ORC incorporated within an engine where the interactions between the proposed IBC-ORC waste heat recovery system and the engine is considered.

To achieve the objectives of this work, a heavy-duty turbocharged diesel engine was modelled in AVL Boost software. First, the model was validated against the available real performance data. The IBC was modelled in AVL BOOST while the ORC was modelled in MATLAB software. The developed MATLAB code was coupled to AVL BOOST for running the models simultaneously. The completed mathematical model is used for studying the effect of different parameters such as engine load and backpressure on the performance of the proposed system.

Table 1
Specifications of SCANIA industrial turbocharged diesel engine.

Parameter	Unit	Value
Model number	–	DC13 087A
Bore	mm	130
Stroke	mm	160
Number of Cylinders		6
Working principle		4-stroke
Maximum Power	kW	257
Rated RPM	RPM	1800
Compression Ratio	–	17.3

Furthermore, the impacts of the proposed novel waste heat recovery system on fuel consumption, thermal efficiency, and other engine functional parameters at various engine loads and constant RPM are studied and presented in this paper.

2. System description

The proposed WHR system for this study is the combination of both ORC and IBC integrated with a turbocharged 6-cylinder diesel engine. The technical specification of the engine case study is provided in Table 1. Furthermore, the block diagram of the whole system is shown in Fig. 1. As it can be seen in Fig. 1, the exhaust from the engine turbocharger is connected to the IBC system and the engine exhaust gas (stream 5) flows through the IBC turbine. The pressure of stream 6 (which is located at the IBC turbine outlet), is sub-atmospheric, it means that the engine exhaust gas is expanded to sub-atmospheric pressure in IBC turbine. The inlet of compressor is at sub-atmospheric pressure while the pressure increases at IBC compressor to approximately the atmospheric level.

The exhaust gas flows through IBC shell and tube heat exchanger after leaving the IBC turbine. The heat rejected from the IBC heat exchanger is used as the heat source for the ORC working fluid (stream 9 and 10). The heat transferred from the exhaust flow to ORC working fluid is used for vaporization of ORC working fluid. Then, the vaporized ORC organic fluid enters ORC turbine and produce extra power. An air-cooled condenser is used for ORC system to cool down the ORC working fluid. It is a finned-tube heat exchanger in which the ORC working fluid heat is transferred to the ambient air. This condenser fan can be installed in front of the engine main radiator equipped with the WHR system.

3. Mathematical modelling

For mathematical modelling of the system, the control volume analysis was used for each component of the proposed system presented in Fig. 1 [26]. Energy, mass and concentration conservation equations were applied to each control volume as following [27]:

- Mass equation:

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{1}$$

- Energy equation:

$$\sum \dot{Q} + \sum \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \tag{2}$$

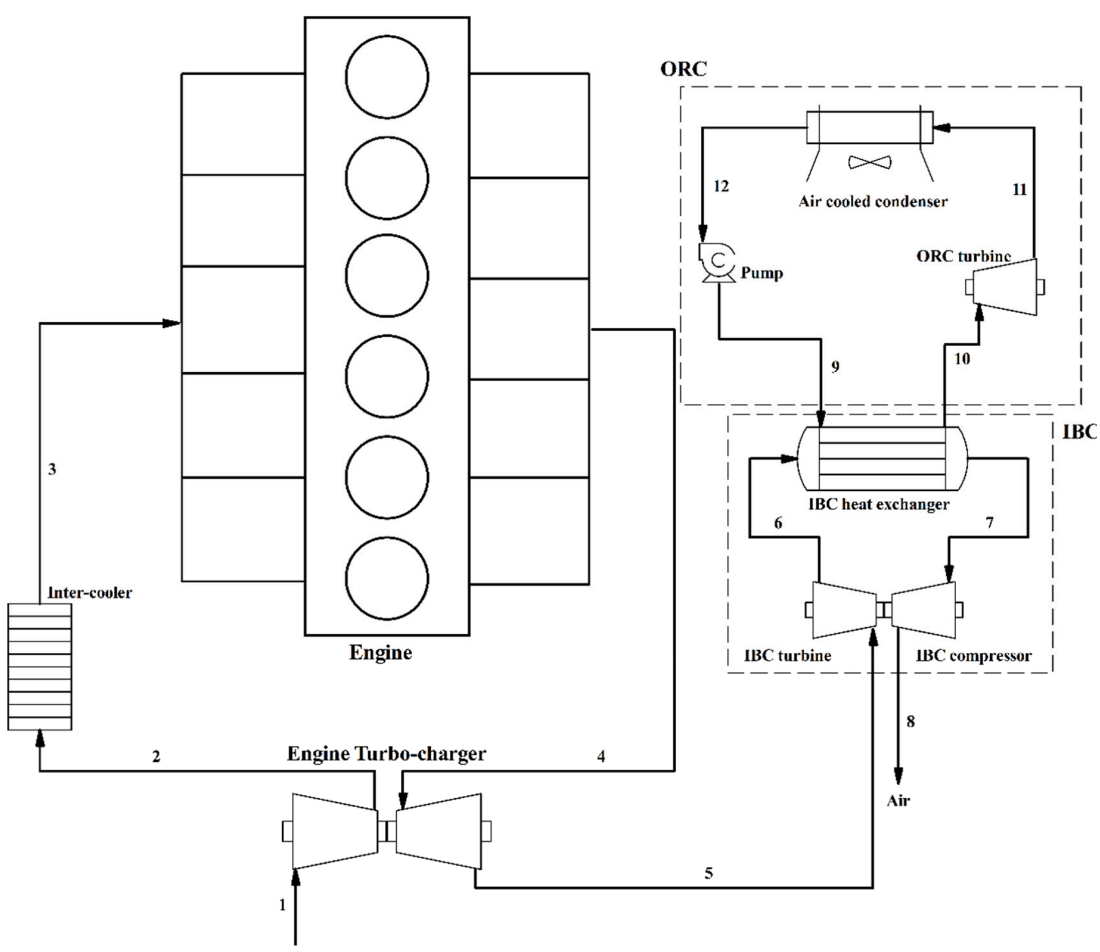

Fig. 1. The block diagram of the proposed system.

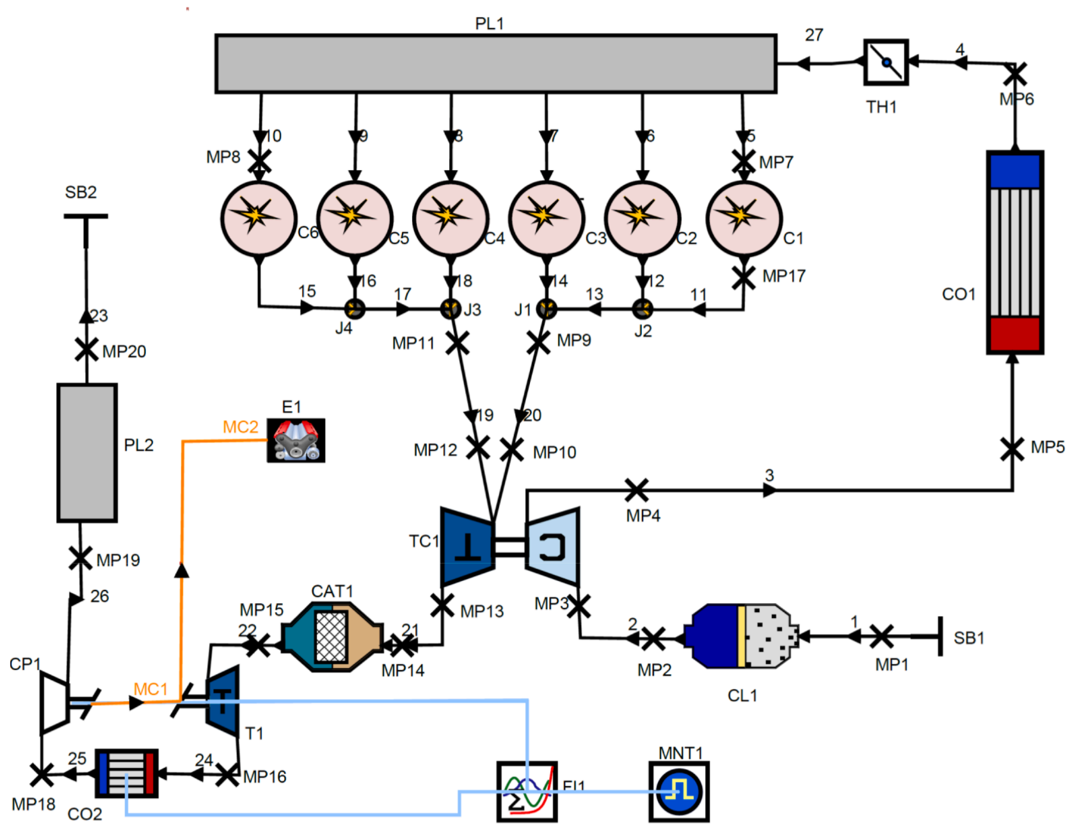

Fig. 2. Block diagram of the engine integrated with IBC in AVL BOOST.

- Concentration equation:

$$\sum \dot{m}_i x_i = \sum \dot{m}_e x_e \tag{3}$$

For development of the mathematical model, the following assumptions were made in this study [26]:

- Thermodynamic equilibrium is assumed for each control volume.
- Steady state condition is considered for each process.
- Pressure-drop due to friction in pipes is added.
- Backpressure due to the installation of WHR components is considered.
- For calculation of heat transfer rate in heat exchangers, the LMTD method is used.

3.1. Engine and IBC

AVL BOOST software was used for numerical simulation of the turbocharged diesel engine integrated with IBC in this study. The employment of such software provides the opportunity of the precise numerical analysis of the whole system [28]. The block diagram of the engine integrated with IBC is shown in Fig. 2. As shown in this figure, the IBC is added after catalyst and turbocharger in engine exhaust pipe. The TC1 and CAT1 components indicates the turbocharger and catalytic converter components, respectively. The CO1, CO2, TCP1, T1 and TH1 are intercooler, IBC heat exchanger, IBC compressor, IBC turbine and throttle valve, consecutively, and C1 to C6 are the engine cylinders. Other abbreviations are described in a table in Appendix A.

The table single zone combustion model [29] and AVL 2000 heat transfer model [30,31] were employed for numerical modelling of the in-cylinder combustion and heat transfer. In the table single zone combustion model [29], the predefined heat release curve were used by AVL

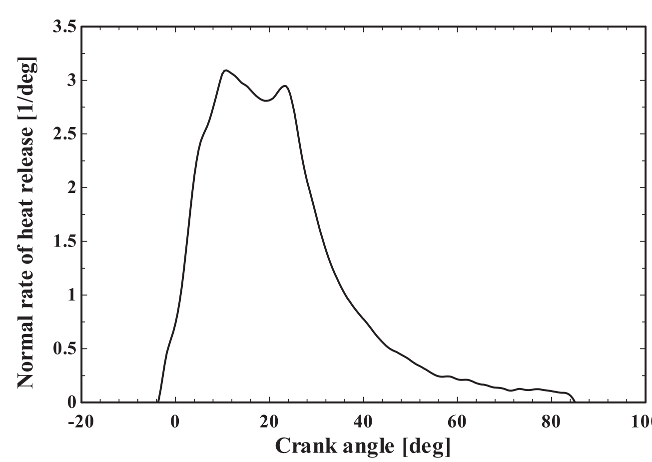

Fig. 3. The normal rate of heat release in various engine crank angles.

Table 2
The assumptions used in IBC [21,25].

Parameter	Unit	Value
Turbine isentropic efficiency	%	80
Turbine mechanical efficiency	%	98
Turbine reference area	mm^2	17,600
Compressor isentropic efficiency	%	80
Compressor mechanical efficiency	%	98
Compressor pressure ratio	–	3
Maximum pressure-drop in heat exchanger	kPa	10
Heat exchanger heat transfer efficiency	%	70–90

software to predict the amount of heat release in various engine conditions. The normal rate of heat release curve used for calculating the heat release rate in the cylinders of the engine, is presented in Fig. 3.

In our model, the engine load was controlled by changing the position of the throttle gate. Therefore, the engine output parameters at various loads and rated RPM (1800 RPM) were obtained using this model. The values used for other parameters of the IBC model in AVL are presented in Table 2. In this paper, the impacts of installing WHR system on volumetric efficiency of the engine is also studied which can be defined as [32]:

$$\eta_v = \frac{V_a}{V_d}, \tag{4}$$

where V_d and V_a are the theoretical volume of engine and actual volume of intake air, respectively.

3.2. ORC

For mathematical modelling of ORC, the control volume analysis was applied for each component and first law of thermodynamics with mass conservation equations was used for each control volume [26]. ORC mathematical model was developed in MATLAB software, and it was coupled to AVL BOOST. R245fa organic fluid was chosen as the working fluid of ORC. The output parameters such as IBC heat exchanger heat load, engine exhaust mass flow rate and the enthalpy of exhaust at stream number 6 (Fig. 1) were transferred to ORC model as input parameters.

Applying the first law of thermodynamics for heat exchanger [26]:

$$\dot{Q}_{HX} = \dot{m}_{ORC}(h_{10} - h_9) = \dot{m}_{ex}(h_6 - h_7) \tag{5}$$

where $\dot{Q}_{HX}$, $\dot{m}_{ORC}$ and $\dot{m}_{ex}$ are heat exchanger heat transfer rate, ORC working fluid mass flow rate and engine exhaust gas mass flow rate, respectively.

For ORC turbine [26]:

$$\dot{W}_{t,ORC} = \dot{m}_{ORC}(h_{10} - h_{11}) \tag{6}$$

$$\eta_{ise,t} = \frac{(h_{10} - h_{11})}{(h_{10} - h_{ise,11})} \tag{7}$$

$\dot{W}_{t,ORC}$ and $\eta_{ise,t}$ indicates the rate of power generated by ORC turbine and isentropic efficiency of turbine, consecutively.

For ORC condenser [26]:

$$\dot{Q}_{acc} = \dot{m}_{ORC}(h_{11} - h_{12}) \tag{8}$$

Applying the first law for ORC pump:

$$\dot{W}_P = \dot{m}_{ORC}(h_9 - h_{12}) \tag{9}$$

$$\eta_{ise,p} = \frac{(h_{ise,9} - h_{12})}{(h_9 - h_{12})} \tag{10}$$

$\dot{W}_p$ and $\eta_{ise,p}$ indicates the rate of power consumed by ORC pump and isentropic efficiency of pump, respectively.

Finally, the overall thermal efficiency of integrated systems can be expressed as [12,21]:

$$\eta_{th} = \frac{\dot{W}_e + \dot{W}_{ORC} + \dot{W}_{IBC}}{\dot{m}_f LHV_f} \tag{11}$$

$$\dot{W}_{IBC} = \dot{W}_{t,IBC} - \dot{W}_c \tag{12}$$

$$\dot{W}_{ORC} = \dot{W}_{t,ORC} - \dot{W}_p \tag{13}$$

where η_{th}, $\dot{W}_e$, $\dot{W}_{IBC}$, $\dot{W}_{ORC}$ are overall thermal efficiency of integrated systems, engine, IBC and ORC power generation rates, respectively. $\dot{m}_f$ and LHV_f are mass flow rate and lower heating value of the diesel fuel injected to engine. The main assumptions for developing the mathematical model of ORC are summarized in Table 3. In the proposed ORC model, the mass flow rate and pressure ratio of ORC turbine and pump were assumed to be constant. Therefore, the turbine inlet temperature was affected by variations of engine exhaust temperature as the result of the change in engine load.

Table 3
The assumptions used in ORC [26].

Parameter	Unit	Value
Mass flow rate	kg/s	0.43
Turbine inlet pressure	bar	12
Turbine pressure ratio	–	3
Turbine isentropic efficiency	%	85
Pump inlet pressure	bar	4
Pump isentropic efficiency	%	90

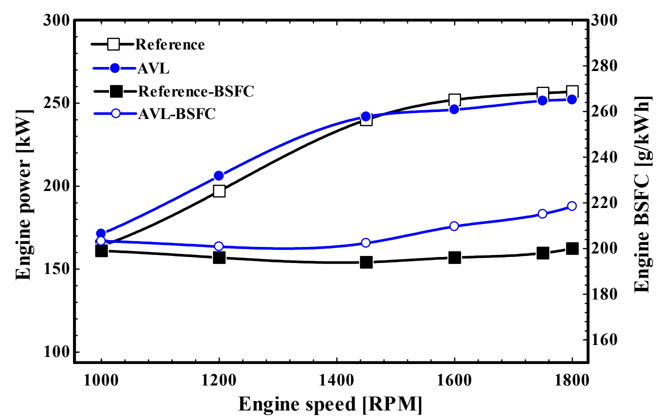

Fig. 4. The rate of engine power generation and BSFC in various RPMs in AVL model and reference [33].

4. Validation

For validation of SCANIA turbocharged industrial diesel engine model, the main output parameters of engine i.e., BSFC and power production rate were compared to the real data obtained from the engine technical catalogue [33]. The BSFC and power were calculated at steady state condition for various RPMs and full load in AVL model. The validation was an iterative process where the data from the AVL simulation were traced and relevant parameters from the model were changed in appropriate range until the data from the model was aligned with the experimental data. The relevant parameters such as combustion shift and air to fuel ratio was varied in AVL model until the minimum fitting error was achieved [34,35].

The results of comparison between calculated parameters from the validated model and real data are presented in Fig. 4. As it can be seen in this figure, there is a good agreement between simulation and real data for all operating conditions and the maximum error of the AVL model is below 10% which shows the high reliability and accuracy of the developed model compared to the real engine performance [33].

5. Result and discussion

To assess the proposed WHR system performance at various engine conditions, the engine model was run at 1800 RPM for different loads in AVL BOOST software. The impacts of the variations of the engine loads on the proposed WHR system output parameters and whole system performance were studied. The engine performance parameters such as engine power and BSFC variations as well as the effects of backpressure

Table 4
The pressure and temperature of engine exhaust gas at outlet of catalyst converter.

Parameter	Without WHR	With WHR
Exhaust pressure [bar]	1.053	1.788
Exhaust temperature [K]	819.8	1056.9
Volumetric efficiency [%]	91.03	85.83
Fuel injection rate [g/s]	15.3	18.36

Table 5
The thermophysical properties of the exhaust air in various streams of IBC in different engine loads.

Parameter	25%	50%	75%	100%
Exhaust mass flow rate [kg/s]	0.131	0.214	0.28	0.346
Turbine inlet temperature [K]	1071.5	1021.1	1044.5	1056.9
Turbine inlet pressure [bar]	0.86	1.09	1.44	1.79
Turbine inlet enthalpy [kJ/kg]	899.68	831.04	861.05	876.65
Turbine outlet temperature [K]	900.8	814.9	789	771.5
Turbine outlet pressure [bar]	0.335	0.339	0.342	0.364
Turbine outlet enthalpy [kJ/kg]	693.163	594.68	574.37	561.83
Compressor inlet temperature [K]	445.9	439.8	440	505.4
Compressor inlet pressure [bar]	0.33	0.324	0.316	0.3
Compressor inlet enthalpy [kJ/kg]	160.66	156.82	161.56	245.15
Compressor outlet temperature [K]	625.2	622.4	627.2	726.7
Compressor outlet pressure [bar]	1	1	1.014	1.026
Compressor outlet enthalpy [kJ/kg]	359	354.29	360.84	476.96

Table 6
The thermophysical properties of the R245fa working fluid at different locations for different engine loads.

Parameter	25%	50%	75%	100%
Mass flow rate [kg/s]	0.43	0.43	0.43	0.43
Turbine inlet temperature [K]	370.7	385.2	427.1	415.1
Turbine inlet enthalpy [kJ/kg]	435.9	491.3	541.5	527.3
Turbine outlet temperature [K]	328.2	354.9	399.2	386.8
Turbine outlet enthalpy [kJ/kg]	422.3	472.8	519.6	506.3
Pump inlet temperature [K]	327.2	327.2	327.2	327.2
Pump inlet enthalpy [kJ/kg]	272	272	272	272
Pump outlet temperature [K]	327.6	327.6	327.6	327.6
Pump outlet enthalpy [kJ/kg]	272.7	272.7	272.7	272.7

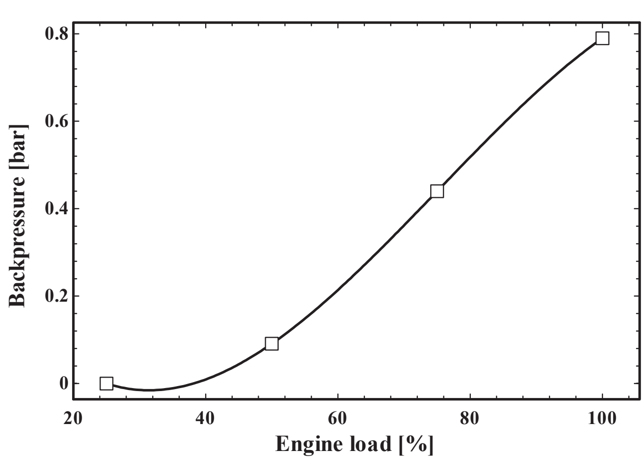

Fig. 5. The amount of backpressure produced by installing the WHR system for various engine loads [37].

produced by the IBC-ORC WHR on engine performance were analyzed. Furthermore, the main performance parameters of the IBC and ORC were investigated.

The engine exhaust pressure and temperature at catalyst converter

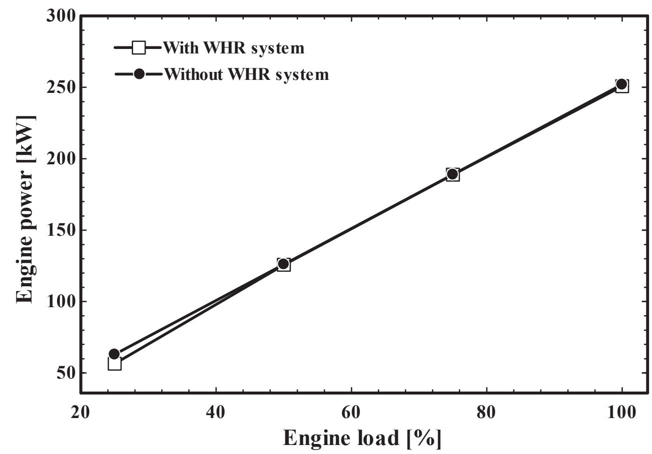

Fig. 6. The variation of engine power in various loads with and without the proposed WHR system.

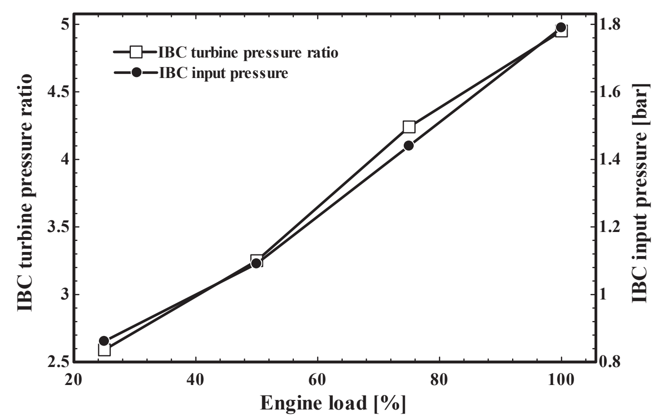

Fig. 7. IBC turbine pressure ratio and input pressure in various engine loads.

outlet (MP15 in Fig. 2) at 1800 RPM and full load with and without IBC system are presented in Table 4. As can be seen, the engine exhaust pressure at the outlet of catalyst converter increased by nearly 0.735 bar with WHR system installed on engine tail pipe resulting in reduction of engine volumetric efficiency by up to 30%. Due to the imposed backpressure and the reduction of engine volumetric efficiency, more fuel should be burnt by the engine to compensate the power loss [13]. This increase in the rate of fuel injection also leads to increase of in-cylinder temperature resulting in increment of exhaust gas temperature as presented in Table 4 [36].

Tables 5 and 6 present the thermophysical properties of the working fluids of IBC and ORC, respectively. As it is demonstrated in Table 6, the mass flow rate of the ORC working fluid was constant in various engine loads. Therefore, the pressure ratio of the turbine was fixed for various engine loads as the ORC working fluid mass flow rate remained constant.

The amount of backpressure produced by installing the WHR system on engine tail pipe at various engine loads and 1800 rpm is presented in Fig. 5. The engine backpressure increased up to 0.8 bar at full load as it can be seen in this figure. This finding confirms the importance of analyzing the WHR systems in combination with the engine to reveal the possible side effects alongside the benefits.

Fig. 6 evaluates the impacts of WHR system on engine power output at various engine loads. According to this figure, the engine power varied between 50 kW and 257 kW for different loads at 1800 RPM. Considering the AF ratio as a constant parameter, the fuel injection increases by increase of the air mass flow rate as the engine load increases. This increase results in the increment in engine input energy,

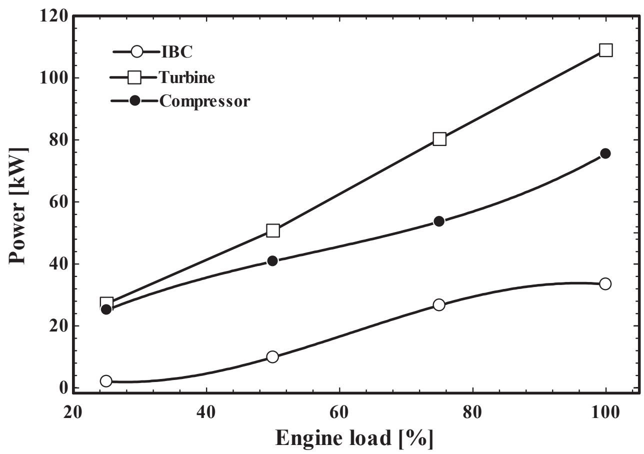

Fig. 8. IBC total power, turbine and compressor powers in various engine loads.

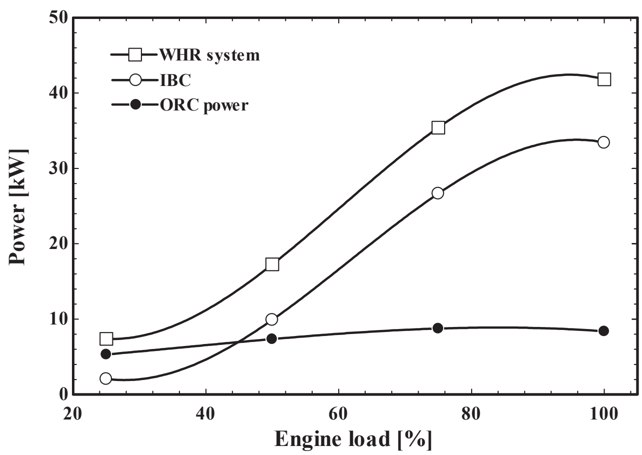

Fig. 9. The rate of mechanical power generated by ORC, IBC and WHR system (combined IBC-ORC) in various engine loads.

consequently in engine power generation.

Furthermore, the engine was delivering almost the same power level before and after adding the WHR system for all operating condition in this simulation and engine power loss was avoided as demonstrated in Fig. 6. Due to the existence of the compressor in IBC system which compensates the backpressure, the engine power remained almost constant when WHR was added.

The variation of IBC turbine pressure ratio and input pressure for various engine loads are shown in Fig. 7. As shown, by delivering the high-pressure exhaust at high engine loads, the pressure ratio of turbine can increased from nearly 2.5 to 4.95 due to the increase in exhaust mass flow rate and exhaust gas pressure (from 0.86 bar to 1.79 bar). This graph confirms the availability of the high energy exhaust even for the turbocharged case study engine which can effectively be used in IBC for WHR.

The IBC turbine power generation and its compressor power consumption rates as well as the IBC total power generation (turbine less compressor power) at different engine loads are shown in Fig. 8. The IBC total power increased significantly from 2 kW in 25% load to 33.5 kW in 100% load. Furthermore, the IBC compressor power consumption increased from 25 kW to 78 kW by increase of the engine load due to the rise in exhaust mass flow rate. The IBC power generation was very low at 25% engine load due to the low exhaust mass flow rate and pressure. This increased by engine load when the turbine power surplused the compressor power. In total, IBC could recover the power up to about 33.5 kW (over 10% of the engine power).

Fig. 9 shows the power generation rates of ORC, IBC and total WHR system at different engine loads. As it is shown in this figure, the IBC power generation rate for the loads smaller than 45%, was lower than ORC power generation rate. The reason for this is the dependency of the IBC power generation in the turbine to the exhaust pressure and mass flow rate while these parameters are lower for the loads below 45% compared to the higher load. By boosting the exhaust pressure and flow rate, the rate of power produced by IBC increased far higher than ORC at high engine loads as presented in Fig. 10. The extra power generation of WHR system was changing between 7.4 kW and 42 kW by variation of the engine load from 25% to 100%. It confirms that the employment of proposed WHR system can result in increase of system total power generation rate up to about 18% of the original engine power.

Fig. 10 compares the BSFC and thermal efficiency of the engine with and without WHR system at full load and 1800 rpm. Adding the WHR system into the engine, resulted in BSFC increase of approximately 3% (increased from approximately 218.25 g/kWh to 225 g/kWh). The main reason for BSFC increment is the extra backpressure produced by the proposed WHR system which leads to increase of engine fuel consumption for overcoming this backpressure and maintaining the same engine output power.

Furthermore, the total thermal efficiency of the system decreased slightly from about 38 to 37 (~1%) from Fig. 10. While recovering waste heat and producing extra power had a positive impact on total power produced by the combined WHR system and the engine, the extra fuel consumption slightly overweighed this benefit according to the thermal efficiency.

6. Conclusion

In this paper, a novel WHR solution with IBC and ORC systems was presented and its impacts on the performance of an industrial turbocharged diesel engine were investigated. For the waste heat recovery systems based on IBC, a large proportion of engine exhaust flow heat was wasted through IBC heat exchanger where the exhaust flow cools down. Therefore, the organic Rankine cycle was employed and integrated with IBC in this study for generating extra electrical power in IBC-ORC waste hear recovery system. In addition, the interactions between the diesel engine and the proposed WHR system was assessed for the first time in this paper.

A turbocharged industrial diesel engine was modelled in AVL BOOST software numerically. Then, the proposed WHR system consisting of IBC and ORC were added to the engine model. While a significant power was recovered by adding the WHR system, the results confirmed the negative effects on BSFC and thermal efficiency. Therefore, it is important to analyse such systems in combination with the engine to include the pressure drop effects, otherwise isolating the waste heat recovery system is not going to provide the full picture.

The main findings of this study can be listed as follow:

- The IBC system can recover up to 33.5 kW power on its own from engine waste energy which is over 10% of the engine original power.
- By adding the ORC, the total power generation by the WHR system increased up to 18% of the engine original power.
- The employment of the WHR system leads to increase in backpressure (up to 0.8 bar) and reduction of engine volumetric efficiency (by approximately 5.2%).
- The IBC power production was lower than ORC at low engine loads and it became higher than ORC at high loads when the engine exhaust energy boosted up.
- The proposed pressure drop from the combined WHR system increased the BSFC by up to 3% and reduced the total thermal efficiency by almost 1% at rated RPM and full load.

Since the proposed system can recover considerable amount of

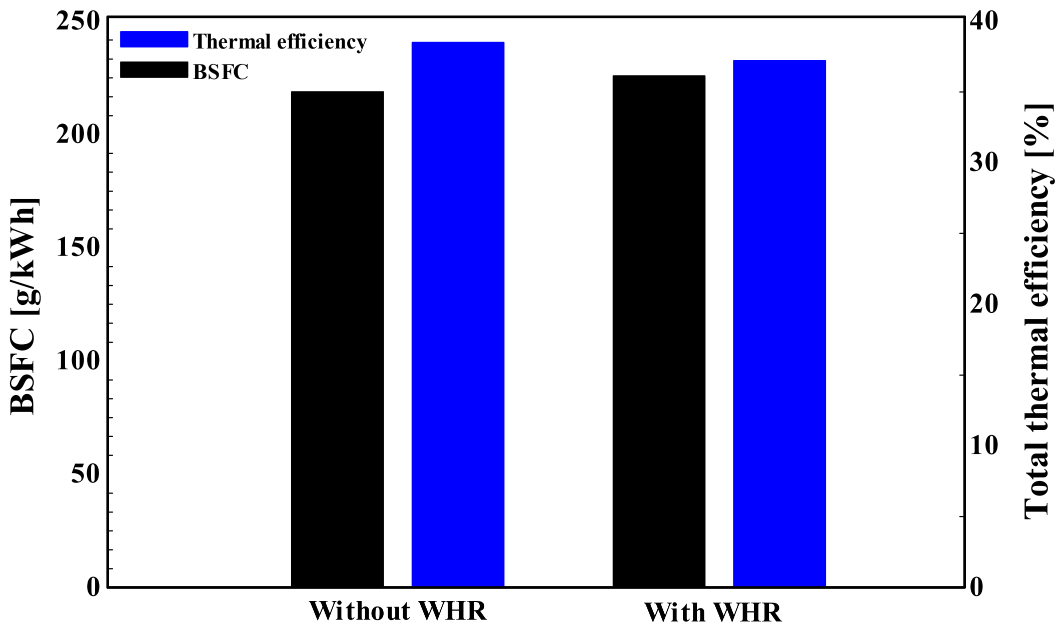

Fig. 10. The engine BSFC and total thermal efficiency with and without proposed WHR system at engine full load.

Table A1
The description of abbreviations used in Fig. 1.

Abbreviations	Description
SB	System boundary
MP	Measuring point
CL	Air cleaner
TC	Turbocharger
CO	Cooler
TH	Throttle
PL	Plenum
C	Cylinder
J	Junction
CAT	Catalytic converter
T	Turbine
TCP	Compressor
MNT	Monitor
FI	Formula interpreter
E	Engine

power from the exhaust, it is highly beneficial in some application of heavy-duty diesel engines such as electric power generation when extra demand in network is happening. In such systems, adding new engine is going to be challenging due to the limitation of the space and capital cost. So, the proposed system can help to respond to the extra demand effectively. However, the extra backpressure imposes increase in fuel consumption to some extend which consequently results in reduction of thermal efficiency. For future research, optimization of the proposed system can be implemented. Also, for the waste heat recovery, other cycles and methods such as transcritical CO2 (tCO2) cycle, supercritical carbon dioxide (sCO2) and thermoelectric generator (TEG) can be investigated.

CRediT authorship contribution statement

Farhad Salek: Conceptualization, Methodology, Software, Validation, Investigation, Writing – original draft. **Meisam Babaie:** Conceptualization, Methodology, Investigation, Supervision. **Mohammad Mahdi Naserian:** Supervision. **Mohammad Hossein Ahmadi:** Supervision.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgement

AVL List GmbH support for proving the simulation tools for University of Salford through their University Partnership Program is greatly appreciated.

Appendix

Table A1

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