Numerical study of the waste heat recovery potential of the exhaust gases from a tractor engine

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Abstract

The paper presents an analysis of the possibilities of exhaust gas heat recovery for a tractor engine with an output power of 110 kW. On the basis of a literature review, the Rankine cycle seems to be the most effective way to recover the exhaust gas energy. This approach reduces the fuel consumption and allows engines to meet future restrictions on carbon dioxide emissions. A simulation model of the engine by means of a one-dimensional approach and a zero-dimensional approach was built into the simulation code AVL BOOST, and a model of the Rankine cycle was implemented. The experimental values of the effective power of the engine, the mass flow and the exhaust gas temperature were used to validate the engine model. The energy balance of the engine shows that more than 28.9% of the fuel energy is rejected by exhaust gases. Using the engine model, the energy and the exergy of the exhaust gases were studied. An experimental study of the real working cycle of a tractor engine revealed that the engine operates most of the time at a constant speed (n = 1650 r/min) and a constant load (brake mean effective pressure, 10 bar). Finally, Rankine cycle simulations with four working fluids were carried out at the most typical operating point of the engine. The simulation results reveal that the output power of the engine and the efficiency of the engine increase within the range 3.9–7.5%. The highest value was achieved with water as the working fluid while the lowest value was obtained with the organic fluid R134a. The power obtained with water as the working fluid was 6.69 kW, which corresponds to a Rankine cycle efficiency of 15.8%. The results show good prospects for further development of the Rankine cycle.

Keywords

Waste heat recovery, diesel engine, exhaust gas, Rankine cycle, energy, exergy, simulation model

Introduction

Reducing the fuel consumption and the carbon dioxide (CO_2) emissions will be the major challenge for all types of internal-combustion engine in the years to come. Reducing the CO_2 emissions depends on the overall engine efficiency. It is well known that modern engines already have good thermodynamic and mechanical efficiencies, which makes future improvements very difficult. All the current techniques such as high-pressure direct injection, downsizing, variable valve timing, a variable compression ratio, homogeneous charge compression ignition and a variable-geometry turbine will not suffice to meet future restrictions. More sophisticated systems for improving the engine efficiency therefore need to be developed over the next few years.

Despite the high level of development of engine systems and control, a maximum efficiency of only 40% is reached at certain operating points.[1] Most of the time, engines run with an efficiency of 15–35%.[1] This means that more than 60% of the fuel energy is lost.[1–3] The lost energy is in the form of heat in the cooling system and the exhaust system. Research shows that the

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highest potential for recovery is in the heat of the exhaust gas.[4]

State of the art of exhaust heat recovery systems

There are several techniques for waste heat recovery from exhaust gas: turbocompounding, heat recovery based on thermodynamic cycles (the Rankine cycle (RC), the Stirling cycle, etc.), thermoelectric generators (TEGs) and thermo-acoustic systems.

Turbocompounding is currently used in some heavy-duty engines. Mechanical turbocompounding can decrease the brake specific fuel consumption (BSFC) from 1% to 5%.[5] Weerasinghe et al.[6] reported an improvement of 4.1% in the output power of the engine. The problem is the back pressure which affects the gas exchange processes of the engine, especially at lower engine loads. Electrical turbocompounding appears to be a more promising method of energy recovery[7] since research shows the possibility of improving the BSFC by 6–9% when using a highly efficient turbocompressor. TEGs, because of their compactness, have been widely studied for years.[8–11] They do not produce a back pressure but their efficiency rate in the conversion of waste energy into electrical power is low. Their efficiency depends greatly on the heat source and the materials used.[9] Crain and LaGrandeur[10] reported the power output of a TEG of up to 125 W at 600 °C on a hot-air test bench. Dong et al.[12] proposed a modern approach to optimize the TEG parameters. The technology is not yet very well developed, however, which means that real applications are still awaited.

Waste heat recovery based on thermodynamic cycles, such as Stirling engines and Ericsson engines, has also been studied.[13,14] Both technologies show good prospects for stationary applications but they are not suitable for mobile applications because of their weight, cost and size.

A number of studies have revealed that the RC is the most promising means of exhaust heat recovery. The RC can be used in heavy-duty diesel engines[15–17] as well as in automobile applications.[4,18–22] Daccord et al.[23] reported a cycle efficiency of 6.7% for an organic Rankine cycle (ORC) and 10.3% for an RC using water as the working fluid. Hountalas and Mavropoulos[5] presented a numerical study showing an RC efficiency in the range 6–9% and an ORC efficiency of up to 11%. Glavatskaya et al.[3] estimated that an RC efficiency of 12–14% could be achieved. Yang et al.[17] calculated a maximum waste heat energy efficiency of 12.17% by means of an ORC with R416A as the working fluid. Espinosa et al.[24] reported an RC efficiency from 10% to 15%. A numerical study of the ORC using R123 as the working fluid was presented by Chen et al.[25] The study reported a maximum mechanical power of 15 kW and a thermal efficiency of 13.4%. It increased the power of a heavy-duty diesel engine by approximately 6% at full engine load.

The RC by means of a combination of low-temperature heat sources (the cooling system and the intake air) and high-temperature heat sources (the exhaust gases) was also studied.[3,26,27]

The efficiency of the RC is highly dependent on the working fluid. Many working fluids such as water, alcohols, *n*-pentane, toluene, CO_2 and the refrigerator agents R245fa, R236fa and R134a can be used.[28–31] The choice of the working fluid depends on the temperature of the heat source; water is suitable for high-temperature heat sources while organic fluids are used for lower heat source temperatures.

For overall efficiency of the RC, the design of the elements of the system is important. The main components (the heat exchanger, the expander, the condenser and the pump) should be developed to suit the engine application, the temperature of the heat source, the output power of the cycle and the available space. A numerical study of heat transfer in a finned-tube heat exchanger was presented by Zhang et al.[32] in which the effectiveness of the heat transfer was estimated to be within the range 60–70%. Glavatskaya et al.[33] reported an isentropic efficiency of a reciprocating-piston expander to be within the range 55–70%. The value varies depending on the speed and the working pressure. An experimental test[34] conducted with a piston expander revealed a maximum isentropic efficiency of 54%. Radial turbines have also been studied as expander machines. Kang[34] reported turbine efficiencies in the range 76–82.2%. Lemort et al.[35] achieved an isentropic effectiveness of 71% for a hermetic scroll expander.

The main objective of this study

The fuel consumption and the pollutant emissions of tractors have been assessed using the Organisation for Economic Co-operation and Development regulations concerning off-road vehicles.[36] Tractors have many applications but they run most of the time in the field (during ploughing). In this regime the fuel consumption and the CO_2 emissions are high because a higher engine power is needed. An experimental test conducted on a Massey Ferguson 7465 tractor during ploughing revealed a significant fuel consumption[36] of 20.4 l/ha. A reduction in the fuel consumption will reduce the CO_2 emissions of off-road vehicles and will decrease the price of agricultural products. Few studies have been reported, however, on waste heat recovery in tractors.

For this reason, the study focuses on the evaluation of the waste heat recovery potential of the exhaust gases of a tractor engine as a promising candidate for further development of a waste heat recovery system based on the RC.

This study estimates the potential of heat recovery on the exhaust gases of a tractor engine, and its conversion into mechanical work. The estimation was made by means of the energy balance of the engine and calculation of the enthalpy and the exergy of the exhaust

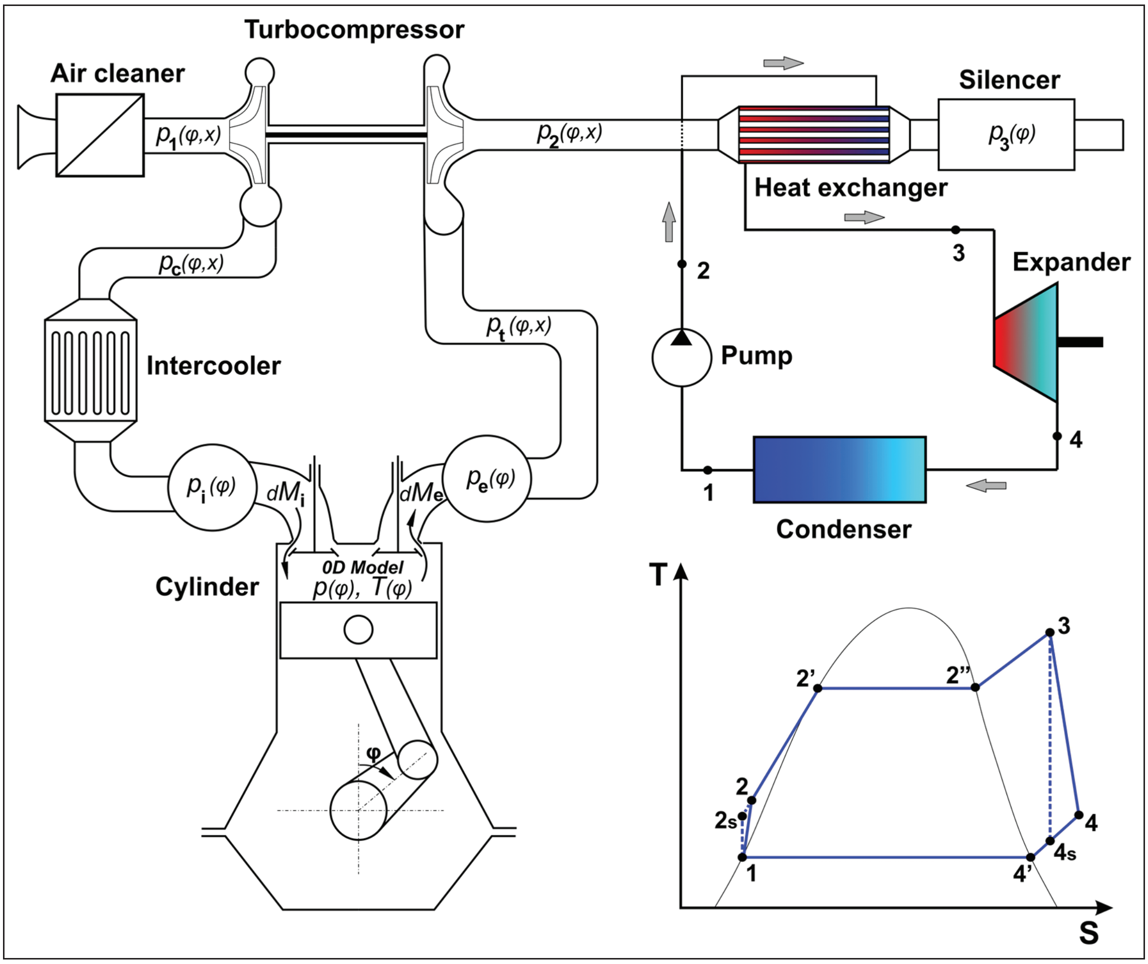

Figure 1. Models of the engine and the RC.

gases as well as the RC output power calculation with different working fluids. For estimation, a mixed zero-dimensional (0D)–one-dimensional (1D) model of the engine and a quasi-steady-state model of the RC was used.

Simulation model

Engine model

A physical model of the engine as well as the RC model are shown in Figure 1. A precise model of the engine studied was necessary for an energy and exergy analysis. The advanced engine simulation code AVL BOOST was chosen because of its capacity to simulate the working cycle by means of both the 1D approach and the 0D approach. The model provides opportunities for analysing the energy balance of the engine as well as the enthalpy of the exhaust gases. The results of the model are based on the calculation of the parameters in the cylinders, taking into account the specifics of the combustion process (the rate of heat release or the rate of injection), the heat exchange with the cylinder wall and the gas exchange processes. The mass flow rate in both the intake port and the exhaust port is modelled by means of a quasi-steady flow calculation. The mass flow rate through the ports depends on the pressure differences between the two sides of the valve and the effective cross-section.

The 1D model in the pipes of the intake system and the exhaust system provides calculation of the pressure in the valve ports, taking into account the unsteady flow. This approach is quite accurate.[37] In our experience, it reduces the calculation time to less than a minute for a cycle simulation.

The model of friction losses available in the software was used to calculate the friction power. The losses depend on the construction of the engine, the oil characteristics and the available accessories which consume the engine power

To estimate the recovered energy, a model of the RC is also presented. The enthalpy of the exhaust gases is used as the heat source in the heat exchanger of the system. With this model of the RC, the performance was simulated using different working fluids.

RC model

The pump in the system consumes external power to increase the pressure of the working fluid, i.e. process 1–2. The power consumed by the pump can be determined by means of the equation

Table 1. Parameters of the working fluids.

Fluid	m_f (kg/s)	T_1 (K)	T_3 (K)	p_2 (bar)	p_1 (bar)
Water	0.016	373	573	20	1
Ethanol	0.033	350	573	20	1
R245fa	0.12	306	473	20	2
R134a	0.12	297	473	20	6

$$W_p = \frac{m_f(h_{2s} - h_1)}{\eta_p} \tag{1}$$

Process 2–3 involves heating the working fluid by the exhaust gases conducted at a constant pressure. At point 3 the fluid is in the form of superheated vapour. The heat exchanger is represented by a simple model that assumes a constant effectiveness of the heat transfer. In this case, the enthalpy of the working fluid at the output of the heat exchanger is calculated by means of the equation

$$h_3 = h_2 + \frac{Q_{he}}{m_f} \tag{2}$$

The heat flow transferred by the heat exchanger is estimated as

$$Q_{he} = m_g\left(h_{g,in} - h_{g,out}\right) \tag{3}$$

Process 3–4 is expansion of the vapour in the expander machine. The output power of the expander is calculated from the equation

$$W_t = m_f(h_3 - h_{4s})\eta_t \tag{4}$$

Fluid condensation is conducted at a constant pressure in the condenser, i.e. process 4–1. The heat flow from the fluid to ambient air can be calculated as

$$Q_c = m_f(h_1 - h_4) \tag{5}$$

The power recovered by the RC can be estimated as the difference between the power produced by the turbine and the power consumed by the pump, according to

$$W_{RC} = W_t - W_p \tag{6}$$

The RC efficiency is determined as the recovered power with respect to the heat flow transferred by the heat exchanger as

$$\eta_{RC} = \frac{W_{RC}}{Q_{he}} \tag{7}$$

The parameters of the working fluid are defined as functions of the type of fluid. The main parameters are listed in Table 1. The parameters of the exhaust gases at the inlet section of the heat exchanger (the mass flow, the temperature and the specific heat capacity) have the same values for all the fluids studied. These values were estimated by means of the engine model. They depend on the operating point of the engine.

Table 2. Main parameters of the engine.

Type of engine	1106D
Number of cylinders	6 in line
Volume	6.6 l
Bore	105 mm
Stroke	127 mm
Compression ratio	16.2
Number of valves per cylinder	4

The parameters of the engine under the study conditions

The diesel engine is a 6.6 l six-cylinder engine equipped with a turbocompressor and an intercooler. The boost pressure is limited to 1.8 bar. The engine is equipped with a direct-injection common-rail system. The pressure in the system is electronically controlled within the range 70–130 MPa by the engine control unit through a solenoid valve in the high-pressure pump. A valvetrain mechanism with four valves per cylinder, driven by a single camshaft and push rods, is fitted. There is no after-treatment system, which makes the exhaust system of the engine suitable for application of an RC heat exchanger. The emission level compliance is US Environmental Protection Agency Tier 3.

The geometries of the intake and the exhaust system were measured on the tractor engine while the geometries of the cylinders and valves were taken from the technical documentation. As some parameters, such as the characteristics of the turbocompressor and the valve lift curves, were missing, a simple model of the turbocompressor was input into the engine model owing to difficulties in obtaining the turbocompressor maps from the manufacturer. The compressor was represented by the constant efficiency and the pressure ratio while the turbine was represented in a more complex way. Valve lift curves were represented by the curves of a similar heavy-duty engine which are available within AVL BOOST tutorials. The main parameters of the engine are listed in Table 2.

Validation of the engine model

Model validation was carried out before further numerical simulations. The effective power of the engine and the mass flow of the exhaust gases were used to calibrate the model. The comparison between the experimental data and the calculated values of the effective

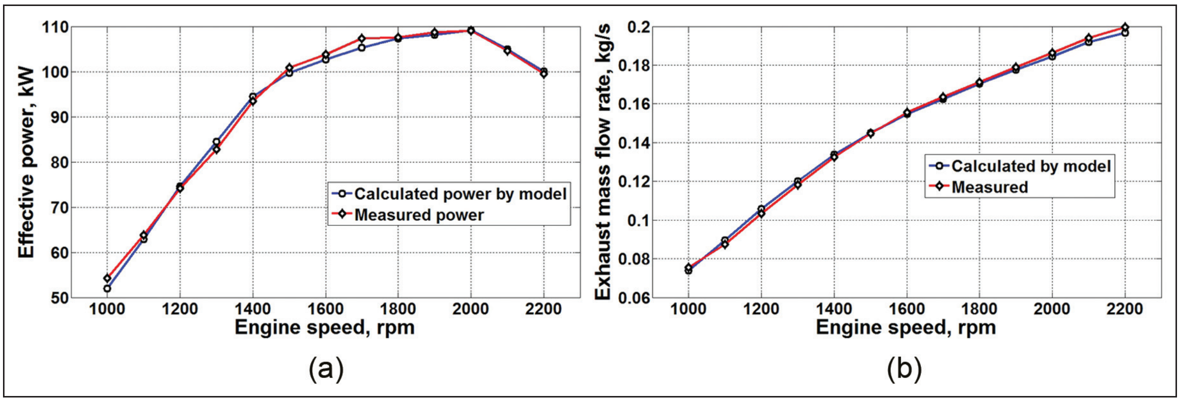

Figure 2. Comparison between (a) the measured and calculated engine effective powers and (b) the measured and calculated exhaust mass flow rates.
rpm: r/min.

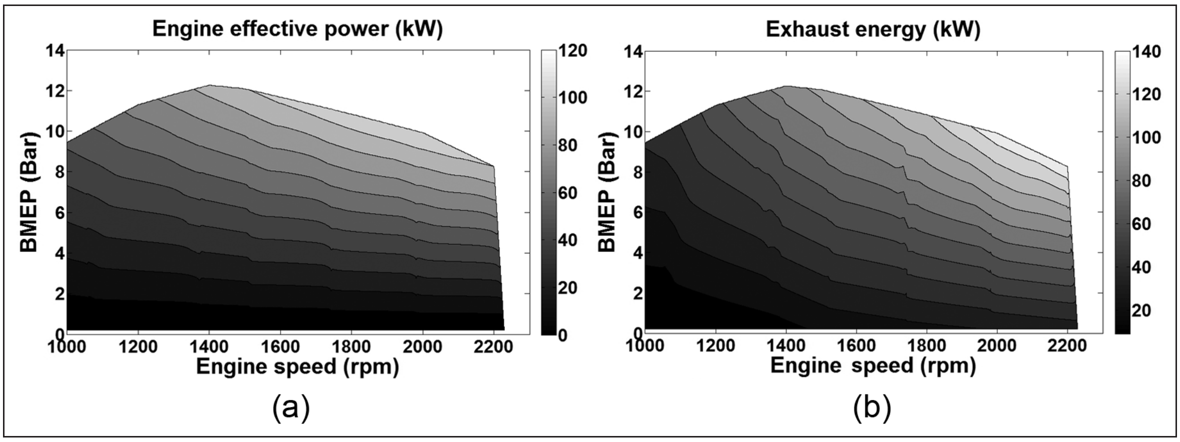

Figure 3. (a) Engine effective power; (b) waste energy through the exhaust gases.
BMEP: brake mean effective pressure; rpm: r/min.

power and the mass flow was made at full engine load. The experimental data for the fuel consumption, the engine load, the engine speed, the torque, the effective power, the inlet temperature, the mass flow, etc., were measured on a hydraulic test bench. The values of the fuel consumption were input into the model. The intake pressure was calculated depending on the engine load.

The comparison for the effective power shows a slight difference between the experimental performance curves and the predicted performance curves. A maximum deviation of 4.2% was observed at 1000 r/min while at other speeds it did not exceed 2%. The results are shown in Figure 2(a).

To evaluate the recovery potential of the engine's exhaust gases, the most important point is an accurate estimation of the enthalpy. For that reason the measured exhaust mass flow rates and the calculated exhaust mass flow rates were compared (see Figure 2(b)). The mass flow rate comparison was conducted at full load. A maximum deviation of 5.9% was observed at 1100 r/min. The results show a maximum mass flow through the engine of 0.2 kg/s at 2200 r/min.

For further simulations the temperature as well as the enthalpy of the exhaust gases were calculated at the location chosen for placing the RC evaporator.

Validation of the engine model enabled us to use the model to assess the energy balance of the engine as well as the energy and exergy contents of the exhaust gases.

Results and discussion

Energy balance of the tractor engine

By means of the engine model constructed in AVL BOOST, the distribution of the fuel energy over the whole operating domain was determined. In general, the fuel energy is distributed as the energy converted into effective power and the lost energy. Energy is lost through the exhaust gases, the cooling system and the friction (auxiliaries). Calculation of the exhaust gas energy is important in order to estimate the possibilities for heat recovery. The results are given in Figures 3 and 4.

The results, which are calculated in energy per time, i.e. power, show that the energy of the exhaust gases

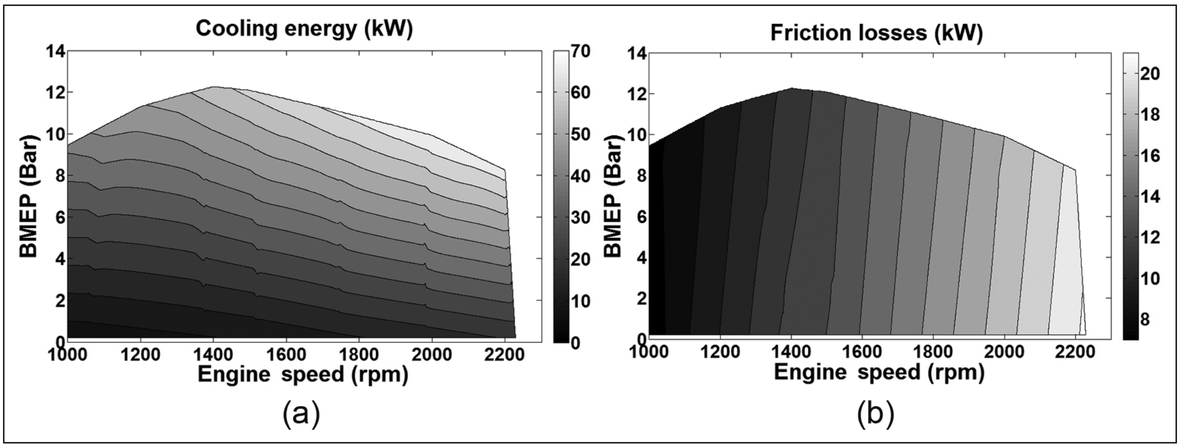

Figure 4. (a) Waste heat through the cooling system; (b) friction losses.
BMEP: brake mean effective pressure; rpm: r/min.

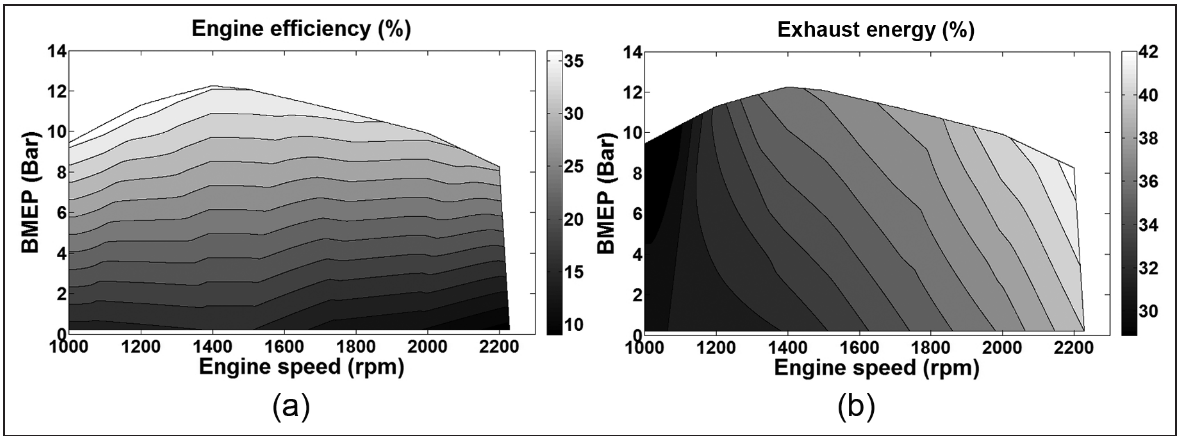

Figure 5. (a) Engine efficiency; (b) exhaust energy.
BMEP: brake mean effective pressure; rpm: r/min.

accounts for a significant part of the fuel energy over the whole operating domain. At engine speeds lower than 1400 r/min, the quantity of waste energy in the exhaust gases is similar to both the energy converted into effective power and the energy lost in the cooling system. At engine speeds above 1400 r/min, this exhaust gas energy forms the largest part of the fuel energy. In absolute values the waste energy in the exhaust gases at full engine load varies from 41 kW to 139.9 kW. The maximum value of waste energy in the exhaust system is at the maximum speed of 2200 r/min. In comparison, the maximum effective power of the engine is 109.2 kW at 2000 r/min (see Figure 3(a) and (b)).

Figure 5 shows the engine efficiency and the waste energy through the exhaust gases calculated as a percentage of the fuel energy. The maximum engine efficiency of 37.2% was obtained at full load and 1200 r/min. In relative values (Figure 5(b)) the exhaust gas energy accounts for more than 28.9% of the fuel energy. This value increases slightly as a function of the engine speed up to 42.5% at full load and 2200 r/min. An increase in the waste energy depending on the engine speed can be attributed to a reduction in the time for the working cycle. This extends the combustion process, and part of the fuel energy is released during the expansion stroke. Usually, it increases the exhaust gas temperature and reduces the engine efficiency.

Energy analysis at the exhaust system

To estimate the recovery potential of the exhaust gases, it is not sufficient to calculate the energy rejected from the cylinders. Part of this energy is converted into mechanical power in the turbine, and another part is lost by heat transfer with the environment in the exhaust pipes. In order to estimate the recovery potential of exhaust gases, estimations of the exhaust gas parameters were conducted 1800 mm downstream of the turbine outlet. After a preliminary analysis of the exhaust system design, this location was chosen as the inlet of the RC heat exchanger.

In order to size the elements of the RC and to choose the working fluid, it is necessary to calculate the mass flow, the temperature, the specific heat capacity and the

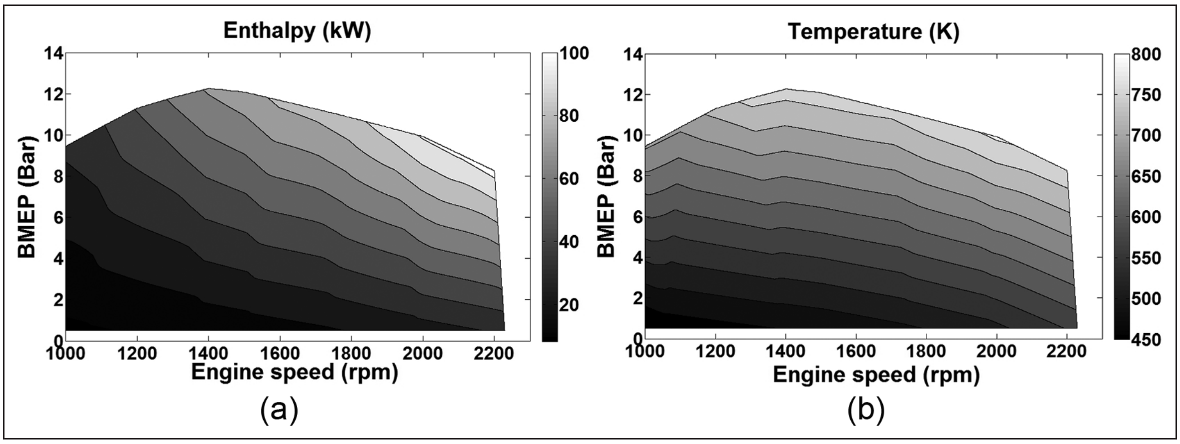

Figure 6. Simulation results at the studied location of the exhaust system: (a) enthalpy; (b) mass flow.
BMEP: brake mean effective pressure; rpm: r/min.

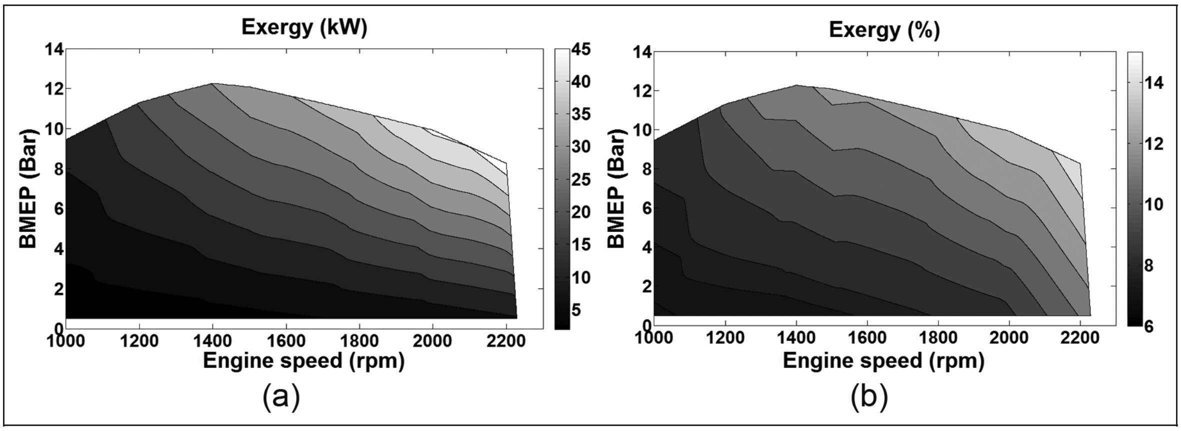

Figure 7. Exergy of the exhaust gases: (a) in kilowatts; (b) in percentages of the fuel energy.
BMEP: brake mean effective pressure; rpm: r/min.

enthalpy of the exhaust gases over the whole operating field of the engine. The parameters of the exhaust gases at the chosen location were therefore studied. The results are presented in Figure 6.

The temperature of the exhaust gases is within the range 450–785 K. The maximum calculated value was obtained at full engine load and 2000 r/min. The temperature decreases slightly with decreasing engine load and decreasing speed. The specific heat capacity c_p is a function of the exhaust gas composition and the temperature. The composition of the gases depends on the air-to-fuel equivalence ratio. The air-to-fuel equivalence ratio value decreases slightly when the engine load increases and remains constant as a function of the engine speed. The estimated value of c_p is within the range 1030–1163 J/kg K. The maximum exhaust mass flow was achieved at full load and 2200 r/min. The value is 0.2 kg/s. The minimum value of 0.6 kg/s was obtained at idle. The enthalpy of the exhaust gases at the studied location (the inlet of the heat exchanger) is lower than that of the gases rejected from the cylinders. Approximately 25% of the exhaust gas enthalpy at full load is lost in the exhaust path from the exhaust valve to the inlet of the heat exchanger. The enthalpy value is within the range 8.3–103.4 kW.

Exergy analysis at the exhaust system

It is well known from thermodynamics that not all the energy of the exhaust gases can be converted to useful work. Thus the waste heat recovery potential must be estimated by means of the available energy, i.e. the exergy. The exergy represents the maximum mechanical work which can be obtained by a closed-loop thermodynamic cycle when it reaches a state of equilibrium. The exergy (J/s) is defined as

$$\text{Exergy} = \left(H_{g,in} - H_{g,0}\right) - T_{g,0}\left(S_{g,in} - S_{g,0}\right) \tag{8}$$

The exergy of the exhaust gases was calculated at the same location as in the energy analysis above. The calculated exergy represents the theoretical potential of waste heat recovery from the exhaust gases by the RC.

The simulation results are shown in Figure 7. They reveal that the exergy is not more than 45% of the enthalpy of the exhaust gases. Theoretically, a

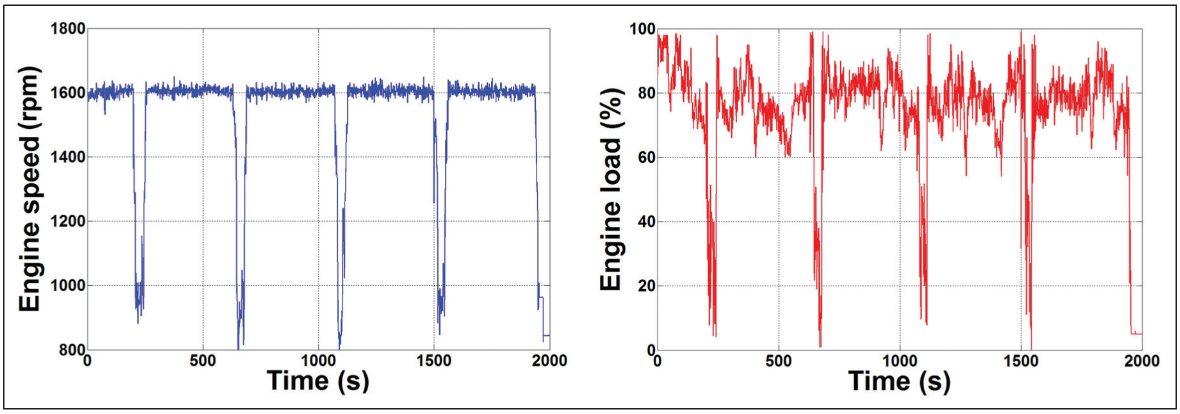

Figure 8. Operating parameters of a tractor engine during ploughing.

maximum of about 45 kW can be converted into mechanical power by the RC. This represents 40.9% of the maximum effective power of the engine. This exergy is available in only a small part of the operating domain at the maximum speed and full load.

The output power received from the RC depends on the available energy of the exhaust gases as well as on the efficiency of the elements comprising the RC. As tractor engines run in a constant regime most of the time, that provides a good opportunity to develop a high-efficiency RC.

Interesting results for the exergy were calculated in percentages of the fuel energy and are shown in Figure 7(b). The values vary from 7% to 15%. Below 1400 r/min and above 2000 r/min, the influence of the engine speed is much greater than that of the engine load, while at medium speeds there is an almost constant relative exergy as a function of the engine speed.

Operation of a tractor engine

Research conducted on a tractor by our team showed that in the field the tractor runs with almost a constant speed from one side of the field to the other; then it takes some time to turn at the end of the field. This cycle is repeated many times. A typical operating cycle during ploughing is shown in Figure 8.

The constant speed of the tractor means that the engine runs at a constant speed and at almost constant load. The fuel consumption, the engine power, the engine efficiency and the enthalpy of the exhaust gases are constant during most of the operating time. Based on the experimental results, it can be assumed that a constant quantity of high-level waste energy is produced during approximately 80% of the time when the tractor works in the field. This makes it relatively easy to predict the energy of the exhaust gases and to develop a highly efficient RC.

The mean value of the engine speed during a high engine load was calculated at 1650 r/min as the engine load mean value was 80%, which corresponds to a brake mean effective pressure (BMEP) of 10 bar.

Table 3. The parameters of the RC.

Exhaust mass flow m_g (kg/s)	0.158
Specific heat capacity c_p (J/kg K)	1125
Exhaust gas inlet temperature $T_{g,in}$ (K)	691
Exhaust gas outlet temperature $T_{g,out}$ (K)	455
Exhaust gas enthalpy (kW)	70.7
Isentropic efficiency η_p of the pump (–)	0.8
Isentropic efficiency η_t of the expander (–)	0.7

RC simulation results

Based on a typical operating cycle of the tractor engine, it was decided to estimate the RC output power and the RC efficiency at the operating point of the engine defined as the point most commonly used (n = 1650 r/min and a BMEP of 10 bar). The calculation was conducted with four different fluids (water, ethanol, R245fa and R134a). The parameters of the exhaust gases at the inlet of the heat exchanger of the RC were calculated by means of the results shown in Figure 6. The exhaust gas temperature at the heat exchanger output was calculated assuming a heat transfer effectiveness of 0.6. The values are listed in Table 3. All the parameters of the exhaust gases as well as the pump efficiency and the expander efficiency have the same values for each of the fluids.

The simulation results for the RC output power and the RC efficiency for each of the fluids are given in Figure 9. The maximum power was obtained with water as the working fluid. The value is 6.69 kW, which corresponds to an RC efficiency of 15.8%. The RC with ethanol as the working fluid performed less well in these simulations. The calculated output power was 6.05 kW (14.2%). Using organic fluids decreases the RC performance and the RC efficiency. The least efficient organic fluid was R134a with an RC output power of 3.45 kW; with R245fa the power obtained was 4.92 kW.

Figure 10 shows the engine output power and the efficiency gain by means of the RC exhaust heat recovery at the studied operating point. The parameters were calculated for each of the fluids.

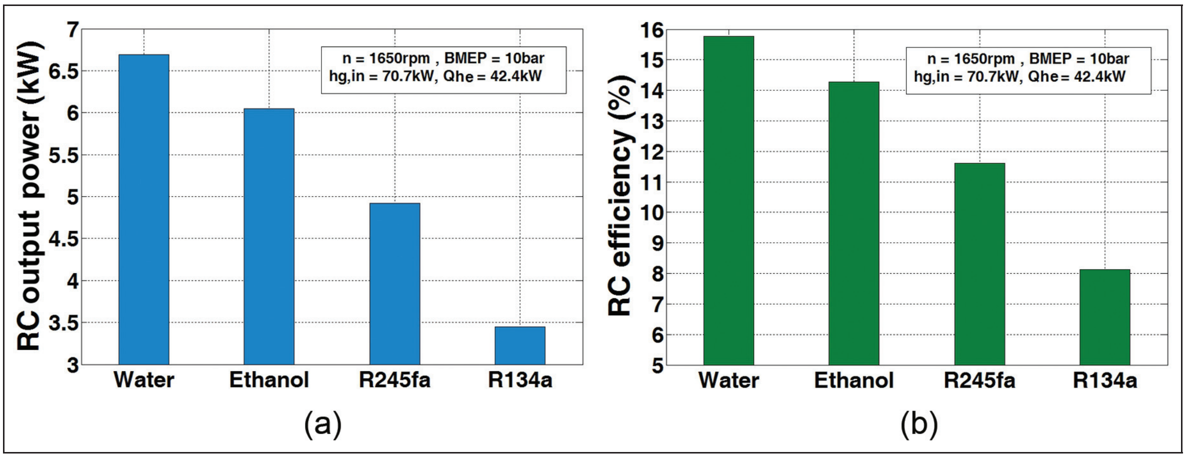

Figure 9. (a) RC output power and (b) RC efficiency as functions of the working fluid.
RC: Rankine cycle; rpm: r/min; BMEP: brake mean effective pressure.

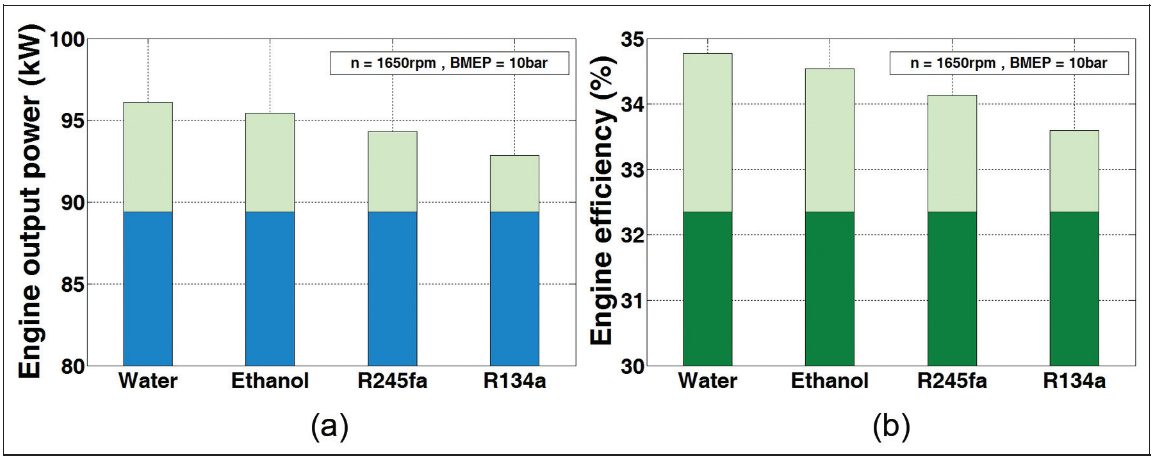

Figure 10. (a) Engine output power and (b) engine efficiency gain as functions of the working fluid.
rpm: r/min; BMEP: brake mean effective pressure.

The simulation results reveal that the output power and the efficiency of the engine increase within the range 3.9–7.5%. The highest value was achieved with water as the working fluid while the lowest value was obtained with R134a. At the studied operating point, the engine efficiency can be increased up to 34.8% by means of an RC exhaust heat recovery system. This value is obtained when water is chosen as a working fluid. For the less efficient organic fluids, the engine efficiency is within the range 33.6–34.1%.

The weight and size of the RC components were not taken into account in the study as the elements are at the prototype stage and their dimensions can be further optimized. However, a comprehensive analysis of the tractor design was made to find sufficient space for each of the components. The most critical element is the heat exchanger which can be mounted vertically just before the silencer in the exhaust system. This enables a wide variety of heat exchanger dimensions to be developed. In all cases it should be a counter-flow exchanger, as this increases the transfer efficiency. To reduce the size of the condenser, it is preferable to use water as the working fluid owing to the high condensation temperature and better efficiency. Additionally, the engine's cooling system can be used as a cold source for the RC. Because of the relatively low mechanical output power of the RC (less than 6.69 kW for the studied point), it is preferable to use a piston machine as the expander as its rotational speed is compatible with the crankshaft speed of the engine. To achieve that, the recovered power can be added directly to the transmission.

Conclusions

The present study reveals that a combination of the 1D approach and the 0D approach used for modelling a diesel engine in the advanced simulation code AVL BOOST is an appropriate way to conduct an energy analysis of the engine and the exhaust system. Validation of the model presents a maximum deviation of 4.2% between the calculated values and the measured values of the effective power of the engine and a maximum deviation of 5.9% in the mass flow comparison.

The energy balance of the tractor engine under study confirms that a significant part of the fuel energy is lost through exhaust gases. At full engine load it varies from 28.9% to 42.5% of the fuel energy. As absolute values in the form of power, it ranges from 41 kW to 139.9 kW.

The energy analysis of the exhaust system was carried out by calculating the temperature, the specific heat capacity, the exhaust mass flow and the enthalpy at the location chosen as the inlet of the RC heat exchanger. The results show that no more than 75% of the energy rejected from the cylinders can be used as a heat source for the RC energy recovery system. The rest of the energy is used in the turbine or is lost owing to heat transfer through the gas path to the chosen location. At full load the calculated exhaust gas enthalpy is within the range from 32.1 kW to 103.4 kW as the corresponding temperature varies from 693 K to 785 K.

The exergy analysis of the exhaust gases at the same location shows that a maximum of 14.8% of the fuel energy can be converted into mechanical work by means of a closed-loop thermodynamic cycle. At full engine load, this value is between 8.36% and 14.8%. However, for the tractor engine studied here, the absolute value of this exergy varies from 11.8 kW to 48.1 kW over the whole operation map.

Experimental research on a tractor engine during ploughing has shown that the most typical operating point is at an engine speed $n = 1650$ r/min and a BMEP of 10 bar. A theoretical study of the RC performance at this operating point of the engine reveals that water as the working fluid provides the highest RC output power and efficiency. In this case, the RC output power is 6.69 kW and the cycle efficiency is 15.8%. The lowest RC efficiency is observed with the organic fluids R245fa and R134a. The minimum RC output power of 3.45 kW was obtained with R134a as the working fluid.

By means of an RC exhaust heat recovery system and water as the working fluid, the output power and efficiency of the engine can be increased by 7.5% at the studied operating point. When R245fa is chosen as the working fluid, the engine efficiency increases by 5.5%.

The novelty of this study lies in the combination of numerical approaches to assess the exhaust gas recovery potential and RC simulations while the most commonly used operating point of the engine was defined by experimentation. The results reveal good prospects for the development of an effective RC for waste heat recovery from the exhaust gases of the engine under study. However, for further development of the elements as well as the parameters of the RC, a more comprehensive model of the cycle needs to be pursued. For an appropriate choice of the working fluids, more studies should be conducted at different operating points of the engine, which correspond to different modes of tractor operation.

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Declaration of conflict of interest

The authors declare that there is no conflict of interest.

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Appendix 1

Notation

c_p	specific heat capacity (J/kg K)
h	specific enthalpy (J/kg)
H	enthalpy flow (J/s)
m	mass flow (kg/s)
n	speed of the engine (r/min)
p	pressure of the working fluid (bar)
Q	heat flow (J/s)
S	entropy flow (J/s K)
T	temperature (K)
W	power (W)
η	efficiency (–)

Abbreviations

BMEP	brake mean effective pressure (bar)
BSFC	brake specific fuel consumption (g/kW h)

CO_2	carbon dioxide
ORC	organic Rankine cycle
RC	Rankine cycle
TEG	thermoelectric generator
0D	zero-dimensional
1D	one-dimensional

Subscripts

c	condenser
f	working fluid
g	exhaust gases
he	heat exchanger
in	inlet of the heat exchanger
out	output of the heat exchanger
p	pump
s	isentropic
t	expander
0	ambient