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A review of waste heat recovery and Organic Rankine Cycles (ORC) in on-off highway vehicle Heavy Duty Diesel Engine applications

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ABSTRACT

Heavy Duty Diesel Engine (HDDE) are between the biggest contributors to CO_2 emission and ambient pollution as they are the most widely used technology for commercial vehicles and ship propulsion applications, as well as, together with reciprocating gas engines, for small medium-size distributed stationary power generation.

New emission legislations in the on and off highway sectors, such as for example EURO VI and Tier 4 final, regarding NO_x and Particulate Matter (PM), are also becoming year by year more stringent.

For these reasons, in the last years, concerns about further engine development and efficiency improvement are of primary importance and several technologies have been studied and implemented.

This review is meant to give an overview of the Organic Rankine Cycle (ORC) technology to recover wasted thermal energy in Heavy Duty Diesel Engines (e.g. exhaust gas, EGR, coolant circuit, charge air cooling, oil circuit) with particular focus on vehicle applications for on and off highway sectors (e.g. long-haul trucks, earth-moving machines, agricultural tractors). In addition, multiple different engine operating profiles in terms of torque and speed are gathered and reported for a variety of typical vehicles, in order to characterize the best system design point for the chosen application.

1. Introduction

Heavy Duty Diesel Engines (HDDE) are widely used in several applications, such as vehicle and ship propulsion, as well as, together with reciprocating gas engines, for small-medium size distributed stationary power generation. However, they are also among the main contributors to CO_2, Green House Gases (GHG) and pollutants emissions. The US EPA [1] reports that the road transport sector, mostly powered by HDDE, has been estimated to contribute for 14% to the world global Green House Gases (GHG) emissions in 2014, while the global carbon emissions from fossil fuels have significantly increased since 1900, with a 1.5 factor in the years between 1990 and 2008. For these reasons, the emission reduction challenge, in order to fulfil new stringent legislations, is pushing engine manufactures and developers in the direction of further increasing energy efficiency.

Several strategies are adopted for this purpose, and can be divided basically in 2 categories: engine-powertrain-applied or engine-bottoming technologies, depending if they are directly applied or retrofitted to the engine-powertrain system, or if they recover wasted engine energy.

Examples of first category technologies are engine downsizing, using advanced turbocharging or boosting technologies (e.g. waste gate, variable geometry turbines, e-boost, two-stage turbocharging) [2,3], coupled with EGR for NO_x reduction [4], Variable Valve Timing (VVT) and advanced Miller timing strategies [5]. Other possibilities are related to combustion improvement using particularly shaped combustion chambers (optimized using Computational Fluid Dynamics and improved chemical kinetics combustion modelling) together with high pressure injection (up to more than 2500 bar) and advanced injection strategies [6]. Moreover, engine friction reduction is also under study [7], using improved coatings for the cylinder liners, better surfaces finish, new piston rings and bearings designs, lubricants and seals as well as variable speed electrically driven lubricating oil and coolant water pumps and cooling fans.

Furthermore, engine-tailpipe or bottoming technologies are under development, such as advanced aftertreatment strategies using Diesel Oxidation Catalysts (DOC), Diesel Particulate Filters (DPF) and Selective Catalytic Reduction (SCR) with urea injection for emissions reduction [8], or waste heat recovery technologies such as Organic Rankine Cycles (ORC) [9,10], turbo-compounding [11] and Thermo-Electric Generators (TEG) [12,13].

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Nomenclature

AC	Air Conditioning
AT	Aftertreatment
BSFC	Brake Specific Fuel Consumption, g/kWh
C	Compressor
CAC	Charge Air Cooler
CFC	Chloro-Fluoro-Carbon
CFD	Computational Fluid Dynamics
CHP	Combined Heat and Power
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COP	Coefficient Of Performance
DI	Direct Injection
DOC	Diesel Oxidation Catalyst
DOE	Department of Energy (United States)
DORC	Dual-Loop Organic Rankine Cycle
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
EPA	Environmental Protection Agency (United States)
ESC	European Stationary Cycle
ETC	European Transient Cycle
EXP	Expander
FTP	Federal Test Procedure
GHG	Green House Gases
GWP	Global Warming Potential
HC	Hydro-Carbon
HCFC	Hydro-Chloro-Fluoro-Carbon
HDDE	Heavy Duty Diesel Engines
HFC	Hydro-Fluoro-Carbon
HP	High Pressure
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HT	High Temperature
LNG	Liquefied Natural Gas
LP	Low Pressure
LPC	Low Pressure Compressor
LPT	Low Pressure Turbine
LT	Low Temperature
NEDC	New European Driving Cycle
NFPA	National Fire Protections Association
NIST	National Institute of Standards and Technology
NO_x	Nitric Oxides
NPSH	Net Positive Suction Head
NRSC	Non-Road Steady Cycle
NRTC	Non-Road Transient Cycle
ODP	Ozone Depletion Potential
OEM	Original Equipment Manufacturer
ORC	Organic Rankine Cycle
p	Pressure, bar
P	Power, kW or Pump
PM	Particulate Matter
PN	Particle Number
RC	Rankine Cycle (steam)
rpm	Rounds-per-minute
SCR	Selective Catalytic Reduction
SET	Supplemental Emissions Test (heavy duty)
SI	Spark Ignited
T	Turbine
TEG	Thermo-Electric-Generator
TERS	Thermal Energy Recovery System
UDDS	Urban Dynamometer Driving Schedule
US	United States
VGT	Variable Geometry Turbocharger
VVT	Variable Valve Timing
WHR	Waste Heat Recovery
WHSC	World Harmonized Stationary Cycle
WHTC	World Harmonized Transient Cycle

In the last years, great importance has been given also to the study of alternative powertrain possibilities, such as, for example, hybrid-electric [14,15] and fuel cell powered vehicles [16] architectures.

New fuels, previously not considered for engine or vehicle applications, such as LNG (Liquefied Natural Gas) [17,18], biofuels and biodiesel (or diesel additives) [19–24] are also currently investigated and developed in order to reduce engine emissions. In particular, as reported by Chauhan *et al.* [19] and Shahir *et al.*et al. [23], the use of biodiesel blends in traditional compression ignition engines tends to reduce particular matter (PM), unburned hydrocarbons (HC) and carbon monoxide (CO) emissions, at the price of a slight increase in fuel consumption and nitric oxide (NO_x) emissions.

Kinetic energy recovery systems are also under development such as brake energy recovery or flywheels [25].

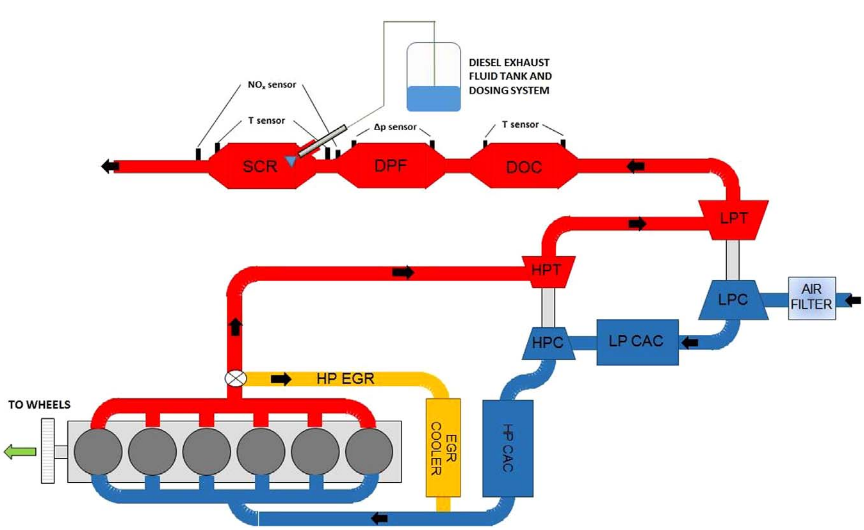

Fig. 1. Possible advanced HDDE scheme for Tier 4f and Euro VI emission regulation compliancy.

This review work focuses on Organic Rankine Cycle (ORC) waste heat recovery technology applied to Heavy Duty Diesel Engines (HDDE), with particular interest for the on and off highway vehicle sectors.

Multiple review articles, publications and reports are available about waste heat recovery for different applications [26–32] but none of them is reporting an overview of the considered technology with particular focus on commercial vehicles, as well as reporting an overview of typical engine operational profiles.

The review has been organized considering, at the beginning, the engine size target, the typical operating profiles for this type of applications, and the heat sources available to be recovered in order to assess the possible efficiency gain when retrofitting the engine.

In a second step an overview of the considered ORC waste heat recovery technology has been reported, with particular interest on cases studies, process modelling considerations, working fluid choice, typical architectures and main components.

2. Heavy Duty Diesel Engines (HDDE)

Most of nowadays commercial vehicles are powered by Heavy Duty Diesel Engines (HDDE) with a brake power output usually up to 600 kW for on-highway applications, and even more for off-highway (e.g. heavy haul mining trucks). The last generation engines are commonly high pressure common rail direct injection (DI) Diesel engines with in-line or V configurations and 4–12 cylinders, with displacements up to around 12 l.

Common implemented technologies are high pressure direct injection (up to more than 2500 bars), waste gate or variable geometry turbochargers, cooled exhaust gas recirculation (EGR, for NO_x and PM emissions reduction), intercooling and/or aftercooling, and aftertreatment devices such as Diesel Oxidation Catalysts (DOC), Diesel Particulate Filters (DPF) and, more recently, Selective Catalytic Reduction (SCR) with urea injection (e.g. DEF, Diesel Exhaust Fluid, or the commercial AdBlue®) for further NO_x reduction.

Regarding after treatment, engine OEMs usually adopt different strategies, with or without EGR or SCR, to meet the stringent emission legislations (e.g. Tier 4 final and EURO VI, as well as future Stage V). An advanced HDDE two-stage turbocharged configuration has been reported in Fig. 1, with EGR cooler, DOC, DPF and SCR after treatment, as well as aftercooler (HP CAC, High Pressure Charge Air Cooler), intercooler (LP CAC, Low Pressure Charge Air Cooler), HP turbine-compressor (HPT, HPC), LP turbine-compressor (LPT, LPC).

2.1. Emissions legislations for on and off-highway vehicles engines

In Table 1, the last emissions legislations for heavy duty Diesel on-highway vehicles have been reported, considering in particular the US Federal EPA'10 and the European standards Euro V and Euro VI. The emission limits for NO_x, PM (Particulate Matter), CO and PN (Particle Number, introduced with Euro VI) have been presented, together with the typical test cycles used for certification purpose and the date of introduction. The data have been obtained from Ricardo plc EMLEG database [33].

In Table 2, the last emission legislations for off-highway (non-road) HDDE vehicles have been reported, divided for engine brake power category (data are presented only for engines with more than 50–60 kW brake power and divided by power range). The US Federal Tier 4 interim and Tier 4 final, as well as the European Stage IV and future Stage V have been considered.

2.2. Typical vehicle operating profiles

In the first development stage of waste heat recovery (WHR) systems for vehicles applications, it is very important to study the real-life operating profiles, so to have a correct idea of which is the engine operating point (torque and speed) at which the WHR system must be designed and optimized. This optimum point is usually the point at which the engine spends most of his time during his working cycle. Some optimum points are then more suitable than others regarding, for examples, exhaust gas mass flows and temperature levels.

Some examples have been reported in Figs. 2 and 3 about typical on and off-highway vehicles. The data have been obtained from Ricardo plc experience and EPA (United States Environmental Protection Agency, [34]), and have been reported based on engine torque and speed time percentage histograms.

Other typical vehicles operating profiles and activities data can be found also in [35].

Some considerations can be drawn from the histograms reported above.

A truck engine, in a typical city cycle, spends most of the time at medium-low speed levels, due to idle periods at traffic lights stops and in the traffic jam. In case of highway operating profile, the truck engine spends more time at medium-high speed levels (the vehicle is commonly at cruise speed), and the speed and torque profiles are quite constant over the time, being thus very suitable for waste heat recovery.

In the case of the city bus, the engine runs most of the time at low-medium torque and speed levels with large amount of time spent at idle conditions (around 600 rpm) at the bus stops, traffic lights and in the city traffic jam. Moreover, it is possible to observe, especially in the case of the Euro V Diesel-hybrid bus, how the speed and torque histograms columns are more concentrated towards low values regions (0–20%). Indeed, for this application, the combustion engine is switched-off during a part of the operating profile, and the propulsion is supplied by the electrical engines. For this reasons, waste heat recovery bottoming cycles are more difficult to be developed in this case, because of the lower availability of exhaust gas mass flow at medium-high temperature (the engine is also smaller compared to the non-hybrid-Diesel bus). A possible benefit, in a hybrid application, could be related to the more stable and constant operating profile of the engine acting as power generator to supply energy for the batteries (transient behaviour is usually avoided and steady state conditions are more common).

In the cases of off-highway vehicles, it is possible to observe stable high speed and torque profiles during operations for agricultural tractors and excavators, thus leading to the conclusion of a good potential for waste heat recovery systems implementation. The other vehicles show a wider distribution of speed and torque between different intermediate category, suggesting more transient and variable profiles over the time.

However, in case of off-highway applications, the practical absence of ram air effect during vehicle operations, leads to higher cooling fan load and parasitic power consumption demands. In this case, indeed, the recovery of exhaust gas waste heat could be challenging since

Table 1
Emission legislation for HDDE for on-Highway applications.

	EPA '10	EU V	EU VI
NOx (mg/kWh)	270	2000	400 (WHSC) 460 (WHTC)
PM (mg/kWh)	13	20	10
CO (mg/kWh)	20786	1500	1500 (WHSC) 4000 (WHTC)
PN (#/kWh)	/	/	8.0×10^{11} (WHSC) 6.0×10^{11} (WHTC)
Test Cycles	FTP & SET	ESC & ETC	ESC & ETC (Future WHSC & WHTC)
Introduction	1/1/2010	10/2008	31/12/2013

Table 2
Emission legislation for HDDE for off-highway (Non-Road) applications.

	Tier 4i	Tier 4f	Stage IV	Stage V
NOx (mg/kWh)	3500 (kW > 900, all others) 670 (kW > 900, gensets) 3500 (560–900 kW) 400 (56–560 kW)	400 (56–560 kW)	400 (56–560 kW)	3500 (kW > 560) 400 (56–560 kW)
PM (mg/kWh)	100 (kW > 560) 20 (56–560 kW)	40 (kW > 560, all others) 30 (kW > 560, gensets) 20 (56–560 kW) 30 (19–56 kW)	25 (56–560 kW)	45 (kW > 560) 15 (56–560 kW)
PN (#/kWh)	–	–	–	1×10^{12} (56–560 kW)
CO (mg/kWh)	3500 (kW > 130) 5000 (37–130 kW)	3500 (kW > 130) 5000 (19–130 kW)	3500 (130–560 kW) 5000 (56–130 kW)	3500 (130–560 kW) 5000 (56–130 kW)
Test Cycles	NRSC & NRTC	NRSC & NRTC	NRSC & NRTC	NRSC & NRTC
Introduction (approval)	2011 (kW > 130) 2012 (56–130 kW)	1/1/2014 (130–560 kW) 1/1/2015 (56–130 kW)	31/12/2012 (130–560 kW) 30/09/2013 (56–130 kW)	Proposed 09/2014 Planned 2018/2019

additional heat must be dissipated in the cooling system and in the cooling pack of the vehicle. EGR heat recovery, on the contrary, can be beneficial regarding vehicle thermal management, decreasing the amount of heat rejected from the cooler, that otherwise should be dissipated in the engine radiator.

2.3. Heat sources

In recent commercial heavy duty Diesel engines, around maximum 40–45% of the fuel energy is converted into brake power for propulsion use. The remaining energy is wasted because of friction losses, heat transfer losses and unused exhaust gas discharged into the ambient.

The main heat sources available for heat recovery in HDDE can be categorized in two classes, with relative common temperature range, depending on the engine operating point [36]:

High-temperature heat sources:

Exhaust gas (200–600 °C)
Exhaust Gas Recirculation (EGR, 200–750 °C)
Low-temperature heat sources:
Coolant (80–100 °C)
Lube oil (80–120 °C)
Charge Air Cooling (CAC, 50–70 °C, indirect cooling using a low temperature cooling circuit)

For engine bottoming cycle performance evaluation, data about these heat sources are necessary, and usually reported based on engine load and speed steady state points from real tests or simulations, or based on torque-speed maps.

In engine waste heat recovery systems studies and prototypes, exhaust gas and EGR heat sources are commonly exploited due to their high heat content and high temperatures, while only a few references about engine coolant, CAC and lube oil recovery are available in literature because of their lower temperature and potential [37] (even if the heat recovery could be beneficial for the whole vehicle thermal management and cooling circuit impact).

Recovery of several different heat sources is also not really practical in vehicle applications, unless complicated layouts are used (dual-loop or cascaded cycles, pre-heating, re-superheating), thus leading to increased weight, costs and space requirements, and more complicated control strategies implementations.

Recovering EGR heat (as well as engine cooling jacket water heat), however, is beneficial in particular for the vehicle cooling circuit, since the heat that should be rejected to the coolant, and then to the ambient through the cooling pack, is used to produce additional useful power.

In the following graphs, some examples of available exhaust gas and EGR thermal power at full load conditions, have been collected for typical heavy duty Diesel engines in the 300–400 kW brake power range, using different boosting and EGR strategies. Regarding the exhaust thermal power, in the calculations it has been considered to cool down the gases to ambient temperature, even though, especially in case of high sulphur content fuels, a temperature of 90–120 °C should be considered in order to avoid acid condensation.

In particular, in Fig. 4, a comparison of thermal power levels between four different engines has been reported: a single stage turbocharged engine with no EGR, a single stage turbocharged engine with HP (high pressure) EGR, a two stage turbocharged engine with HP EGR, and another two stage turbocharged engine with HP EGR.

In Fig. 5, a comparison of thermal power levels for the same engine (two-stage turbocharged Euro VI compliant, with HP EGR) has been reported, considering some different speed-load points, following the ESC cycle test modes guidance [38]. Data are obtained from Ricardo plc testing campaigns.

3. Organic Rankine Cycles (ORC)

The Organic Rankine Cycle (ORC) is a Rankine cycle in which the working medium is an organic fluid with higher molecular mass and lower boiling point compared to water-steam. This waste heat recovery technology has the potential to recover low and medium temperature heat in several applications: internal combustion engines, geothermal plants, solar thermal systems, biomass plants and industrial processes are some examples.

Regarding internal combustion engines waste heat recovery, ORC systems are currently mostly developed and commercialized for stationary power generation applications. Marine applications are also in a promising development phase due to their favourable stable operating profiles, and some commercial products are already on the market [39]. On and off-highway vehicles applications are currently in a research and testing phase and are expected to enter the market around 2020, with particular focus on long-haul trucks engines heat recovery. Vehicle applications are more challenging due to their transient and highly variable operating profiles, leading to the need of implementing accurate control strategies to achieve performance, reliability and durability targets. Safety issues, for example, in case of handling flammable working fluids, must also be considered.

3.1. On-off highway and vehicles ORC case studies, developments and implementations review

Several publications and studies about internal combustion engines waste heat recovery with Organic Rankine Cycles are available in literature (some examples can be found in [40–42]), regarding the choice of the most appropriate ORC layout and working fluid. A more limited number of publications is however related to ORC engine waste heat recovery applied to internal combustion engines for vehicle

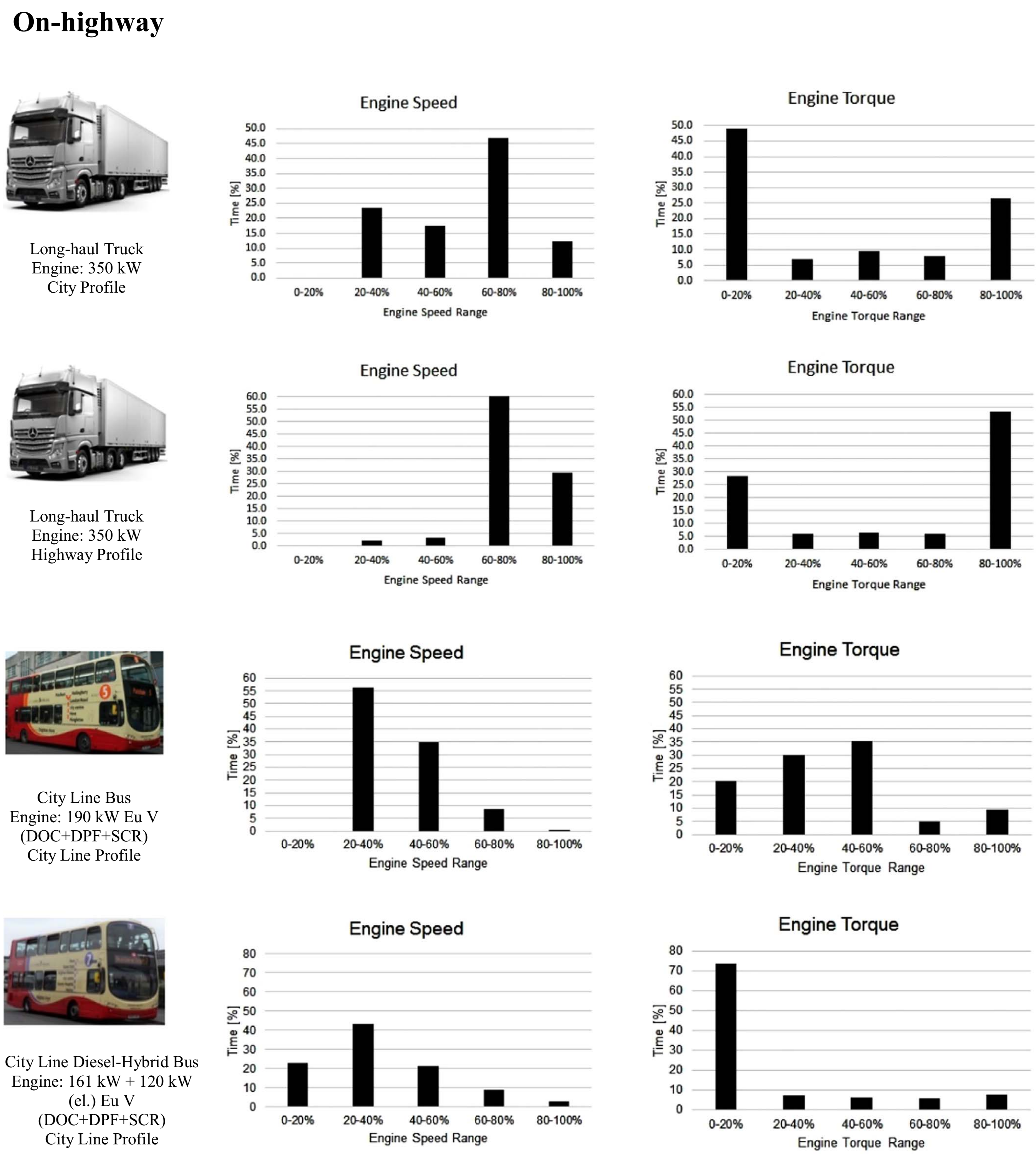

Fig. 2. Typical engine speed and torque time percentage distribution for on-highway vehicles.

applications, investigating also other topics such as vehicle thermal management and powertrain integration. Mostly, simulation studies are available, while experimental and practical implementations and case studies are less common in literature due also the cost of equipment development or purchasing.

Overviews of waste heat recovery for internal combustion engines using ORC have been reported by Sprouse and Depcik [43] and by Wang et al. [32].

The first application of an ORC system used to recover engine wasted heat in a vehicle has been reported by Patel and Doyle [44] in 1976. They recovered energy from the exhaust gas of a Mack 676 Diesel engine mounted on a long-haul truck, obtaining a gain of 13% in power without additional fuel at peak power conditions, using Fluorinol-50 as working fluid. Some additional testing results on the same system have been reported by Doyle et al. [45], together with a complete description of the hardware, considering also the system thermal management (using a compound radiator for the engine and ORC). A 15% improvement in fuel efficiency is suggested to be possible for the

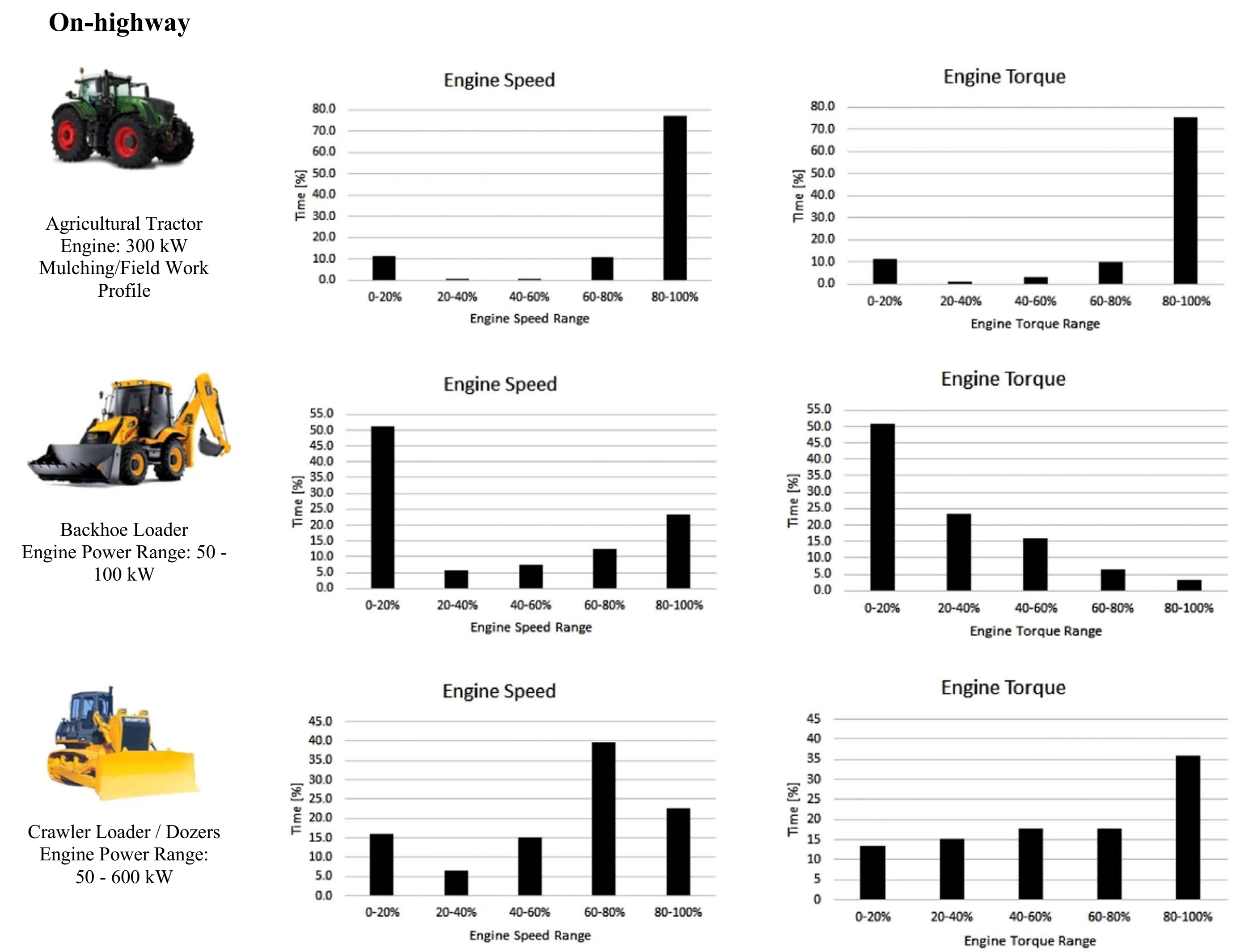

Fig. 3. Typical engine speed and torque time percentage distribution for off-highway vehicles.

combined system. DiBella et al. [46] reported additional results about the same system, regarding laboratory tests with improved components and implementation of control strategies. The authors reported an overall fuel consumption saving of 12.5% for a long-haul truck, with mechanical utilization of the additional recovered power, feeding it to the engine crankshaft through the use of a gearbox.

Chammas and Clodic [47] in 2005 reported a concept study about the possibility of recovering exhaust and cooling circuit heat of an hybrid electric vehicle (HEV) powered by a gasoline 1.4 L engine. The recovered exhaust heat is transformed in electrical power through a turbine/generator and used for auxiliaries. Water-steam and other organic fluids are evaluated through simulations. Water shows favourable performance (between 12% and 27% fuel economy declared) but also some issues, especially regarding complicated expanders designs. Favourable performance is obtained also using iso-pentane or R-245ca (17–32% fuel economy declared) but with more marked environmental and safety issues.

In 2006, Arias et al. [48] proposed different simulated ORC configurations to recover heat from exhaust, coolant and combined exhaust-engine coolant of a SI (Spark Ignition) engine installed on a hybrid passenger car. The configuration with working fluid pre-heating in the engine block and superheating with the exhaust gas, is found to be the most promising showing 8.1% cycle thermal efficiency.

The engine manufacturer Cummins started, in 2005, to study an ORC system to recover heat from an ISX HDDE model. Nelson [49], in 2009, reported a presentation regarding Cummins ORC activity, recovering mainly EGR and exhaust heat, stating that the development of efficient SCR after-treatment systems is supposed to decrease the benefit of an ORC fitted on the EGR. Cummins claimed a potential improvement in engine total efficiency between 5% and 8%.

Endo et al. (Honda), in 2007 [50], reported an implementation of a water-steam Rankine cycle to recover exhaust heat from a passenger car 2.0 L gasoline engine installed in a hybrid vehicle. The evaporator has been integrated in the catalytic converter to reduce the overall dimensions. The expander used is a volumetric swash plate axial piston-type, integrated with the generator. In the vehicle test, at constant speed of 100 km/h, an improvement of 3.8% of thermal efficiency has been claimed for the combined system compared to the baseline engine. Transient analysis and test bench results have been reported for the same system by Kadota et al. [51].

BMW, in 2008 and 2009 [52,53], reported the implementation of a Rankine cycle system, called “Turbosteamer”, for the recovery of high temperature exhaust gas and lower temperature coolant for passenger cars applications. Water-steam was used in the high temperature loop, while ethanol in the low temperature loop. Vane expanders have been used for both the circuits. An increased power output of 15% with no additional fuel consumption has been obtained from tests, and, in general, a 10% value has been considered feasible under relevant stationary realistic conditions. Also some simulations have been carried out using Dymola to assess different heat recovery systems based on

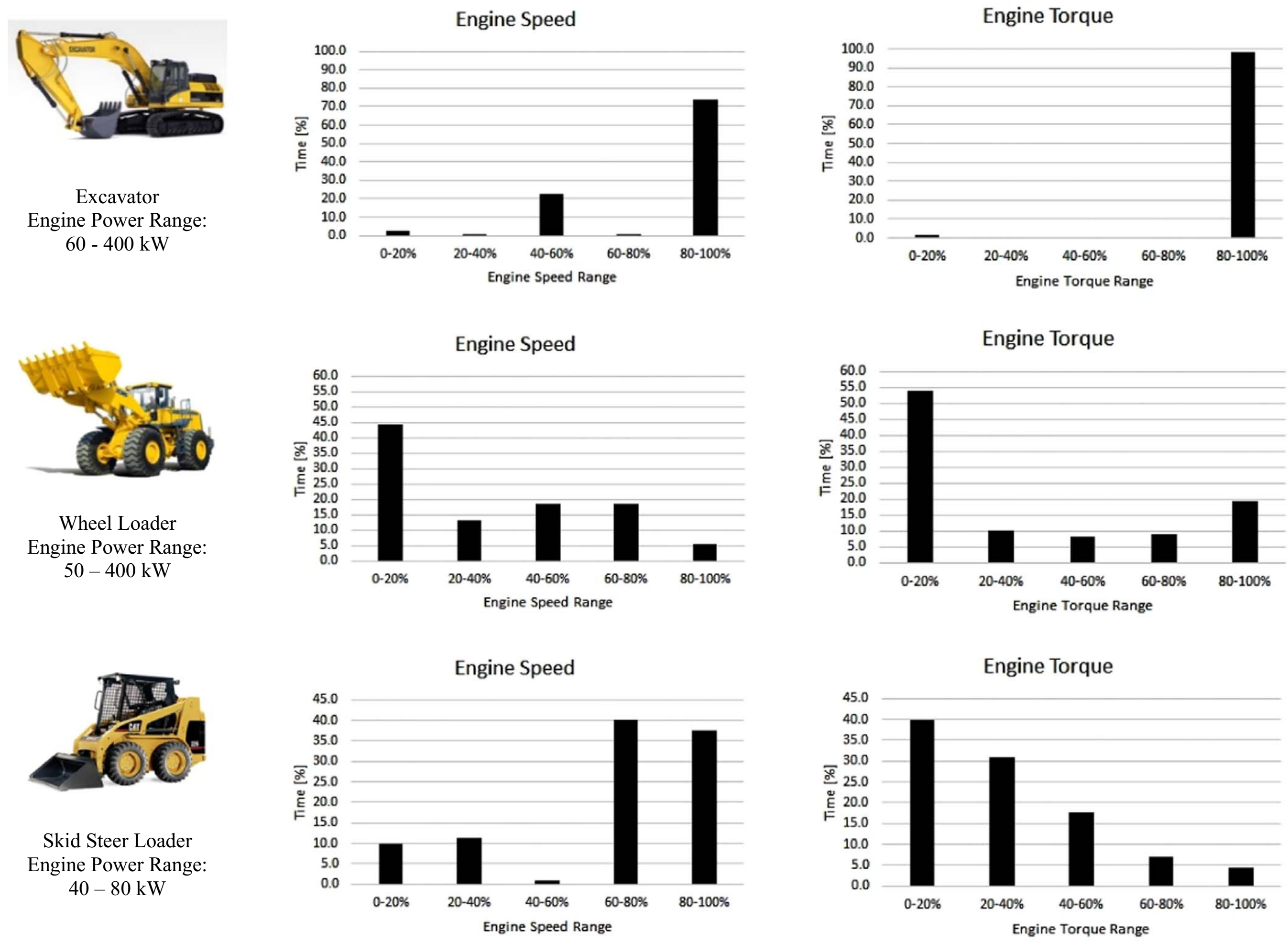

Fig. 3. (*continued*)

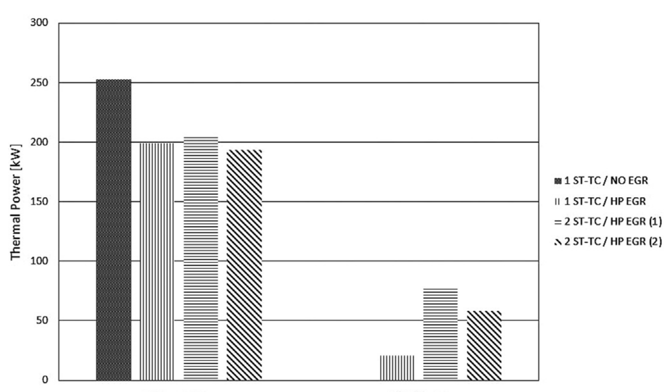

Fig. 4. Exhaust gas and EGR thermal power for four different HDDE models at full load conditions.

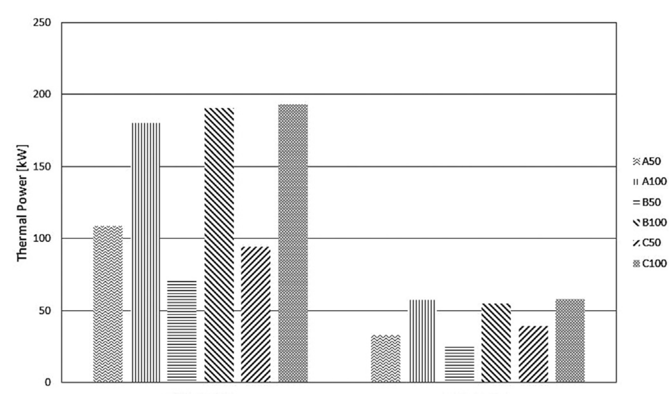

Fig. 5. Exhaust gas and EGR thermal power for a two-stage turbocharged Euro VI engine with HP EGR, for some different European Steady State (ESC) cycle test modes.

Rankine cycles, and to carry out parametric studies regarding important system parameters such as evaporation and condensing pressure levels.

In 2009, Briggs et al. [54], from Oak Ridge National Laboratory, reported a publication about the experimental development of an ORC system applied to a four cylinders, 1.9 L light duty Diesel engine, equipped with a variable geometry turbocharger and HP EGR, achieving a 45% combined system brake thermal efficiency, recovering heat only from the exhaust gas and using R-245fa as working fluid. The system used a turbo-expander connected to a generator.

Also Daimler and Detroit Diesel, in the frame of the DOE (US Department of Energy) Super Truck Program, investigated the possibility of recovering exhaust heat from a truck HDDE [55]. Heat sources recovered are EGR and exhaust gas, and the selected working fluid has been ethanol. Primary candidates for the expander choice are piston and scroll expanders, due to their ability of handling two-phase expansion. Different vehicle cooling strategies, components packaging and weight issues have been investigated.

Behr reported also theoretical and experimental results about HDDE waste heat recovery for long-haul vehicle applications.

Edwards et al. [56] presented simulation and steady-state components models validation results of a complete ORC waste heat recovery system coupled with vehicle thermal management, using an in-house developed simulation tool called BISS (Behr Integrated System Simulation). A 5% on-road fuel consumption improvement, based on the ESC cycle or a long-haul typical driving cycle, has been demonstrated to be possible. Schmiederer et al. [57] reported the results of the Behr experimental ORC cycle used to recover heat from an EURO VI truck engine. Newly developed control system implemented on the ORC system, and a piston type expander, allowed to recover up to 6% additional power. Tests under transient conditions have also been performed. Hybridization of the vehicle powertrain is suggested to be a new opportunity for further development of the technology.

Also Bosch presented simulation and experimental results about ORC waste heat recovery for commercial vehicle engine applications [58,59]. Two different expander technologies have been evaluated (piston-type and turbine). Exhaust gas and EGR have been recovered in a parallel configuration. Water-steam, ethanol, MM (hexamethyldisiloxane), R-245fa and toluene have been considered for the turbine case.

Eaton also carried out some concept work about implementing an ORC waste heat recovery system on a 470 kW, 13.5 L John Deere HDDE [60]. Simulation studies have been performed about engine performance and validated against experimental data. EGR and exhaust gas waste heat recovery has been simulated, using a layout with heat sources in series and recuperation. 6% BSFC improvement has been obtained. Single stage and multi-stage roots expander have been evaluated using ethanol as working fluid. Further engine integration steps are planned.

Hino reported the results of the design and implementation of a Rankine cycle to recover heat from the coolant of an HDDE for truck applications [61]. The energy of the coolant flow has been increased collecting the heat from exhaust and EGR flows, increasing coolant temperature up to 105 °C. 7.5% improvement of fuel economy has been obtained from tests using HFE as working fluid (Hydro-Fluoro-Ether).

Also Ricardo plc has implemented an ORC system to recover EGR and exhaust gas from a 288 kW Volvo HDDE for trucks, using ethanol-water mixture as working fluid [62]. Thermodynamic system analysis, components commissioning, control strategies implementation and testing have been also performed. A piston expander has been used, and the recovered energy re-introduced in the drivetrain through mechanical coupling.

Ricardo plc has also worked on the demonstration of a waste heat recovery system applied to a double-deck Diesel-hybrid bus (2.4 L, EURO IV turbocharged Diesel engine), recovering coolant and exhaust heat with two separate ORC systems, in the frame of the TERS project (Thermal Energy Recovery System) [63–67]. In this case a scroll expander technology has been used and R-245fa as working fluid. From vehicle tests using off-the-shelf components, a 6% fuel economy has been achieved over a typical city bus driving cycle, being reduced to 2.7% considering that in the hybrid bus the internal combustion engine is switched on only for approximately 45% of the time. Additional benefits could be reached when using a cascaded ORC layout, recovering the rejected heat from the topping exhaust ORC cycle to pre-heat the bottoming ORC working fluid used for the coolant heat recovery.

Some other studies have been carried out also by academic institutions. For example Katsanos et al. [68], reported a theoretical study about the possibility of recovering waste heat from an HDDE for truck applications using a steam Rankine cycle, investigating also the influence of the evaporator design, as well as the possibility of recovering exhaust gas and also EGR heat. In this study 7.5% improvement in BSFC has been obtained when recovering exhaust and EGR heat. The influence of fitting the ORC on the engine thermal management has been also considered, thus requiring a radiator with 20% increased heat rejection capabilities. Recovering EGR heat is beneficial in order to reduce the thermal load that must be rejected to the ambient by the cooling circuit.

In the study carried out by Hountalas et al. [37], also the possibility of recovering CAC heat has been evaluated, together with the investigation about the use of water-steam or an organic fluid (R-245ca). 11.3% improvement in BSFC has been obtained when using organic fluid, and 9% when using steam, in the configuration with EGR and CAC waste heat recovery. Radiator heat rejection capabilities have been investigated also in this study.

A parametric study has also been carried out by Katsanos et al. [69], again using water-steam or R-245ca as working fluids, to recover heat from an HDDE for truck applications. Different engine loads cases from 25% to 100% have been investigated.

Latz et al. [70] reported a theoretical study comparing different pure working fluids and zeotropic mixtures in sub-critical and supercritical Rankine cycles, considering both energy and exergy efficiencies. Considered pure fluids are: water, ammonia, ethanol, methanol, R-1234yf, R-123, R-152a. Considered mixtures are: R430A (R152a/R600a, 0.76-0.24 mass fractions), R431A (R290/R152a, 0.71-0.29), water-ammonia, water-ethanol, water-methanol. The outcome of the study is that recovering high temperature heat sources with supercritical cycles is not so beneficial compared to sub-critical. Supercritical systems may be beneficial for lower temperature heat recovery (e.g. coolant).

Dolz et al. [71] reported a study about different bottoming Rankine cycles setups with water-steam and organic fluids to recover waste heat from a two-stage turbocharged, 311 kW brake power, 12 L, HDDE. The work is divided into two parts. The first part reports an analysis of different heat sources and cycle layouts. Water-steam has considered to be the best fluid choice when the engine is running at full load conditions, while organic fluids can be more suitable at partial loads operations.

In the second part of the work, Serrano et al. [71] investigated the possibility of recovering heat with a turbine expander and feeding the obtained power directly to the turbocharger-compressor eliminating the related turbine. This configuration has low advantages, in terms of performance, compared to the classical bottoming ORC configuration.

Moreover, Macián et al. [72] reported a methodology to design a bottoming Rankine cycle for waste heat recovery in vehicle applications. They applied their methodology to an HDDE. Water and R-245fa have been considered as working fluid possibilities. The outcome is that water-steam is more suitable over most of the operating points, while R-245fa is more feasible regarding components space requirement issues. 10% improvement in BSFC has been obtained considering the non-ideal behaviour of pump and expander.

Yang et al. [73], analysed the dynamic operating process of a Rankine bottoming cycle, applied to a 11.6 L HDDE model, under driving cycle operations. Low average efficiency during a Tianjin bus driving cycle has been reported (3.6%).

Amicabile et al. [74] proposed a comprehensive methodology for the design of ORC systems for automotive HDDE, considering heat sources and working fluid selection (also based on safety and environmental concerns), as well as some implemented costs correlations for the main components. Recuperated and non-recuperated cycles, as well as supercritical and sub-critical possibilities have been considered. Working fluid analysed are ethanol, R-245fa and pentane. The best performance has been obtained with ethanol and recuperative cycles.

Di Battista et al. [75,76] discussed the effects of the back-pressure increase due to the installation of an ORC system on the exhaust line of a turbocharged IVECO F1C engine for light-duty vehicle propulsion. The VGT (Variable Geometry Turbocharger) turbine operation can mitigate this drawback effect. Off-design evaporator operations have been also considered.

Allouache et al. [77] reported a study about fitting an exhaust heat exchanger on the tailpipe of a 6.7 L Cummins HDDE. The component has been tested for pressure drop optimization using R245fa as working fluid. An estimation of the recovery potential led to an overall

Table 3
Summary of some vehicle ORC implementations.

Company	Engine	Application	Engine Brake Power [kW]	Expander Type	Coupling/ Energy Use	Working Fluid	Expander Isentropic Efficiency [%]	Max. Expander Power Output [kW]	Expander Speed [rpm]	References
Thermo-Electron	Mack 676 Diesel	Long-haul Truck	215	Turbo-expander	Mechanical	Fluorinol-50	n.a.	n.a.	37,000	[44–46]
Cummins	Cummins ISX Diesel	Long-haul Truck	n.a.	Turbo-expander	Mechanical	R245fa	77	42	50,000–80,000	[100]
Honda	2.0 L Gasoline	Passenger Car-Hybrid	n.a.	Swash Plate Axial Piston Expander	Electrical	water-steam	n.a.	2.5	3000	[50,51]
BMW	4 cyl. Gasoline	Passenger Car	n.a.	Vane Expander	Mechanical	water-steam/ ethanol	n.a.	2	n.a.	[53]
Oak Ridge National Laboratory	1.9 L Gasoline	Passenger Car	66.7	Turbo-expander	Electrical	R245fa	n.a.	4	80,000	[54]
BEHR	13 L Euro VI Diesel	Long-haul Truck	n.a.	Piston Expander	Mechanical/ Electrical	water-steam/ ethanol	n.a.	6% of engine break power	1400	[56,57]
Bosch	12 L Diesel	Long-haul Trucks	n.a.	Piston/ Turbo-expander	Mechanical/ Electrical	water-steam/ ethanol	70%[a]	12.3	150,000[a]	[58,59]
EATON Corp.	13.5 L John Deere Diesel	Long-haul Truck	448	Roots Expander	n.a.	water-steam/ ethanol	n.a.	n.a.	n.a.	[60]
Hino	n.a.	Long-haul Truck	n.a.	Turbo-expander	Electrical	HFE (not spec.)	n.a.	n.a.	n.a.	[61]
Ricardo plc-Volvo Trucks	12.9 L VOLVO D13 Diesel	Long-haul Truck	288	Piston Expander	Mechanical	ethanol-water mixture	70%	14[b]	1500	[101]
Ricardo plc – Wright Bus	2.4 L Ford Puma Diesel	City Bus – Diesel-Hybrid (REEV)	140	Scroll Expander	Electrical	R245fa	30%[b]	1.2	n.a.	[63]

*In the tests only mechanical power output considered
[a] Turbine expander data.
[b] From Ricardo tests.

5% increase in brake thermal efficiency over the speed/load range of the engine.

Yamaguchi et al. [78] reported a study about recovering exhaust heat form a 6 cylinders HDDE with HP and LP EGR circuits, as well as two different boosting configurations (single stage and two stage turbocharged). Respectively 2.7% and 2.9% improvement in fuel consumption have been obtained at typical highway cruising conditions (80 km/h).

Latz et al. [79] proposed also experimental results about a water-based Rankine cycle recovering heat from the exhaust gas recirculation (EGR) of a 12.8 L HDDE engine installed on a test-bench. Deionized water, a 2-cylinder piston expander and a EGR boiler prototype have been used. 10% thermal efficiency has been declared for the ORC system.

Glover et al. [80] evaluated the possibility of using a supercritical ORC for vehicle waste heat recovery, considering multiple heat sources and working fluids. The simulation results show an efficiency between 5% and 23% for the ORC system and a possible gross fuel economy potential between 10% and 30%.

Pradhan et al. [81] investigated, through simulation, the possibility of pre-heating the working fluid with post-SCR heat and then evaporating it with EGR gas heat. Testing results from a MY2011 Mack MP8 engine have been used in order to evaluate transient heat sources behaviour and ORC power output. R123 and R245fa based systems demonstrated to be able to produce 56.2% and 37.6% more energy over a UDDS (Urban Dynamometer Driving Schedule) driving cycle as compared to thermal energy necessary to maintain the SCR in the adequate operational temperature range.

Table 4
Examples of working fluids for ORC applications.

Fluid	Category	T_c [°C]	P_c [bar]	T_{boil} [°C]	T_f [°C]	NFPA			GWP (100)	ODP
						H	F	R		
PURE										
water-steam (R-718)	Inorganic	373.95	220.64	99.97	0	0	0	0	< 1	0
ammonia (R-717)	Inorganic	132.25	113.33	−33.33	−77.7	3	1	0	0	0
CO_2 (R-744)	Inorganic	30.98	73.77	−78.46	−56.6	2	0	0	1	0
ethanol (ethyl alcohol)	Alcohol	241.56	62.68	78.42	−114.2	0	3	0	n.a.	n.a.
methanol (methyl alcohol)	Alcohol	239.45	81.04	64.48	−97.5	1	3	0	2.8	n.a.
R-245fa (pentafluoropropane)	Hydrofluorocarbon	154.01	36.51	15.14	−102.1	2	1	0	1030	0
R-245ca (pentafluoropropane)	Hydrofluorocarbon	174.42	39.41	25.26	−81.7	2	1	0	693	0
R-134a (tetrafluoroethane)	Hydrofluorocarbon	101.06	40.59	−26.07	−103.3	2	1	0	1430	0
R-236fa (hexafluoropropane)	Hydrofluorocarbon	124.92	32.0	−1.49	−93.6	1	0	0	9810	0
benzene	Hydrocarbon	288.87	49.07	80.07	5.5	2	3	0	n.a.	n.a.
toluene (methylbenzene)	Hydrocarbon	318.6	41.26	110.6	−95.2	2	3	0	2.7	n.a.
iso-pentane (R-601a)	Hydrocarbon	187.2	33.78	27.83	−160.5	1	4	0	4 ± 2	0
n-pentane (pentane, R-601)	Hydrocarbon	196.55	33.7	36.06	−129.7	1	4	0	4 ± 2	0
propane (R-290)	Hydrocarbon	96.74	42.51	−42.11	−187.7	1	4	0	3.3	0
iso-butane (R-600a)	Hydrocarbon	134.66	36.29	−11.75	−159.4	1	4	0	3	0
n-hexane (hexane)	Hydrocarbon	234.67	30.4	68.71	−95.3	2	3	0	n.a.	n.a.
n-octane (octane)	Hydrocarbon	296.17	24.97	125.62	−56.8	1	3	0	n.a.	n.a.
p-xylene (dimethylbenzene)	Hydrocarbon	343.02	35.32	138.32	13.3	2	3	0	n.a.	n.a.
cyclohexane	Hydrocarbon	280.45	40.81	80.72	6.3	1	3	0	n.a.	n.a.
cyclopentane	Hydrocarbon	238.57	45.71	49.26	−93.5	1	3	0	n.a.	n.a.
MM (hexamethyldisiloxane)	Siloxane - Silicone oil	245.6	19.39	100.25	−0.2	1	4	0	n.a.	n.a.
MDM (octamethyltrisiloxane)	Siloxane - Silicone oil	290.94	14.15	152.51	−86	0	2	0	n.a.	n.a.
MD2M (decamethyltetrasiloxane)	Siloxane - Silicone oil	326.25	12.27	194.36	−68	0	2	1	n.a.	n.a.
MD3M (dodecamethylpentasiloxane)	Siloxane - Silicone oil	355.21	9.45	229.87	−81.2	2	2	0	n.a.	n.a.
D4 (octamethylcyclotetrasiloxane)	Siloxane	313.35	13.32	175.35	17.1	2	2	0	n.a.	n.a.
D5 (decamethylcyclopentasiloxane)	Siloxane	346	11.6	210.9	26.85	2	2	0	n.a.	n.a.
D6 (dodecamethylcyclohexasiloxane)	Siloxane	372.63	9.61	244.96	−2.95	0	2	0	n.a.	n.a.
acetone	Organic compound	234.95	47	56.07	−94.65	1	3	0	0.5	n.a.
R-141b (dichloro-1-fluoroethane)	Hydrochloroflurocarbons	204.35	42.12	32.05	−103.5	2	1	0	725	0.12
R-123 (dichloro-2,2,2-trifluoroethane)	Hydrochloroflurocarbons	183.68	36.62	27.82	−107.2	2	0	1	77	0.02
R-113 (trichloro-trifluoroethane)	Chlorofluorocarbon	214.06	33.92	47.59	−36.2	1	0	1	6130	1
R-1130 (dichloroethylene)	Organochloride	243.28	54.81	60.3	−81	2	1	0	25	0
R30 (dichloromethane)	Organochloride	235.15	60.8	39.6	−96.7	2	1	0	8.7	0
HFE-7000 (3 M NOVEC 7000)	Hydrofluoroheter	164.55	24.76	34.2	−23.2	3	0	0	370	0
COMMERCIAL										
Solkatherm (SES36)	Commercial/Mixture	177.6	28.5	35.6	n.a.	0	3	1	n.a.	n.a.
3 M Novec-649	Commercial	169	18.8	49	n.a.	3	0	1	1	0
NEW (in development)										
R-1234yf (tetrafluoropropene)	Hydrofluoroolefin	94.7	33.82	−29.45	−53.2	1	4	0	6	0
R-1234ze(E) (tetrafluoropropene)	Hydrofluoroolefin	109.36	36.35	−18.97	−104.5	n.a.	n.a.	n.a.	4	0
R1233zd(E)	Hydrofluoroolefin	165.6	35.7	18.3	−107	2	0	0	1	0.0003
MIXTURES (for medium-high T)										
Fluorinol 50 (fluorinol50/water50 molar)	Mixture	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
ethanol-water (0.5/0.5 mass)	Mixture	339.9	201.2	81.5	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
MM-MDM (0.4/0.6 M)	Mixture of siloxanes	275.9	16.55	120.4	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
MM-MDM (0.7/0.3 M)	Mixture of siloxanes	262.36	18.23	108.8	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
benzene-R123 (0.7/0.3 mass)	Mixture HC+Refrig	272.52	49.39	59.38	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
cyclohexane-R123 (0.7/0.3 mass)	Mixture HC+Refrig	263.59	42.6	56.42	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
cyclopentane-R123 (0.7/0.3 mass)	Mixture HC+Refrig	228.32	44.62	42.29	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
pentane-hexane (0.5/0.5 M)	Mixture of HCs	217.65	32.89	47.89	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.

Grelet et al. [82] reported the development of controlling strategies for waste heat recovery Rankine based system in heavy duty engines for trucks applications. Again Grelet et al. [83] evaluated the transient performance of the ORC system comparing it with steady state data, and considering different cycle architectures and working fluids.

Feru et al. [84] presented an integrated energy and emission management strategy for an Euro VI diesel engine with an electrified waste heat recovery system, with the purpose of optimizing the CO_2-NO_x trade-off with operational costs related to fuel consumption. Configurations with and without ORC and a recovery system with battery for energy storage have been considered. 3.5% CO_2 emission reduction and 19% particulate emission reduction have been obtained, while respecting NO_x emission limits, over a World Harmonized Transient Cycle (WHTC). The ORC system implementation leads to 3.5–4% fuel economy improvement during highway driving conditions.

Torregrosa et al. [85] reported results from the experimental testing of an ORC integrated in a 2 L turbocharged gasoline engine using ethanol as working fluid and a swash-plate expander. Transient tests with varying vehicle speed have been implemented with the purpose to evaluate expander controlling strategies over a New European Driving Cycle (NEDC).

Usman et al. [86] presented the analysis of a ORC system applied to a light duty vehicle, considering both positive and negative aspects of the system installation (e.g. net power output increase, weight increase, engine backpressure effect). The results show a 5.82% engine power enhancement at vehicle speed of 100 km/h when considering negative effects (instead of previously calculated 10.88% not considering drawbacks). The conclusion of the study is also that at a speed lower than 48 km/h, the waste heat recovery system is not beneficial at all, even increasing engine power demand, thus discouraging the system installation in typical city driving cycle suitable vehicles.

In general, studies and developments about ORC for waste heat recovery in commercial engines applications are very common in literature in the last years, and the publications are growing in number constantly, showing how the interest for ORC technology is getting stronger, in order to further improve engine efficiency.

A summarized overview about some different vehicle ORC prototypes implementations available from common literature has been reported Table 3.

3.2. Working fluid choice

The working fluid choice is one of the main issues when studying and developing Organic Rankine Cycle systems. The selection of the most appropriate working fluid depends on several considerations, for example: heat sources temperature, system operating temperature and pressure (evaporation and condensing sides), thermal match with the heat source (e.g. zeotropic mixtures), working fluid properties (e.g. thermal degradation and pressure compatibility), toxicity, flammability, chemical instability, serviceability and availability, environmental impact (considering indexes such as GWP, Global Warming Potential, or ODP, Ozone Depletion Potential), freezing point (of particular interest for vehicle applications under particular cold environmental conditions), components size, system and fluid costs, components material compatibility (e.g. corrosion).

An applicable procedure for working fluid choice related to safety and environmental concerns (rather than thermodynamic performance) can be based on the categorization supplied by the NFPA (National Fire Protection Association) 704 Standard [87]. The working fluids are categorized [88] based on their Health (H), Flammability (F) and chemical Instability-Reactivity (R) hazards, and ranked with values from 1 (low hazard) to 4 (high hazard). Usually, fluids with values equal or higher than 3 can be considered not suitable for vehicle applications, in which leakage and flammability concerns are very important.

Regarding heat source temperatures in heavy duty Diesel engines for commercial vehicle applications, high temperature sources (such as EGR or exhaust gas) have a higher potential for waste heat recovery applications (energy and exergy content is higher). CAC and coolant heat sources have lower temperature levels, thus leading to lower heat recovery.

Usually, considering waste heat recovery from high temperature heat sources, alcohols (e.g. ethanol, methanol), water-steam and hydrocarbons (e.g. benzene, toluene, pentane, octane, cyclohexane, cyclopentane) can be considered good candidates, even tough, some of them show flammability concerns, thus leakage must be prevented. Mixtures of alcohols with water are also considered in order to decrease flammability issues.

Refrigerants, such as R-245fa (phased-out in the next future) and R-134a, are usually more suitable for lower temperature waste heat recovery, such as CAC and coolant.

Some examples of working fluids used in HDDE waste heat recovery studies have been reported in Table 4, together with some information about critical temperature and pressure, boiling temperature, freezing temperature and environmental concerns. The available fluid properties have been obtained from several industrial technical and safety sheets and from NIST REFPROP database [89]. For CO_2, in the boiling temperature column, the normal sublimation point has been reported, while in the freezing column, the melting point has been reported at 5.1 atm pressure.

Some working fluids (e.g. ammonia, HFE-7000, Novec-649) have high health hazards, thus not being very suitable for vehicle applications, unless leakage is carefully avoided. CO_2 could be suitable for trans- or super-critical applications, but, in this case, high pressures lead to safety issues and costs. Anyway, carbon dioxide has less problems in case of flammability and health concerns.

Ethanol and methanol are thermodynamically very suitable for high-to-medium temperature heat recovery applications, but they have flammability issues. Leakage must be avoided, especially in case of direct evaporation configurations, in which a possible contact with the hot exhaust gas could lead to fire hazards.

According to Montreal Protocol [90] on substances that deplete the ozone layer, the use of chlorofluorocarbons (CFCs), such for example R-113, has been completely banned since 2010, while the Hydrochlorofluorocarbons (HCFCs), such as R-123, R-141b will be practically banned until 2020 due to the high ODP (even though they show good potential for medium temperature waste heat recovery). Moreover, the Kyoto Protocol [91] listed, but not banned, Hydrofluorocarbons (HFCs), such as R-245fa, R-245ca, R-134a and R-236fa as fluids with high GWP, and thus dangerous for the environment. For this reason, new fluids are currently under development as substitutes: R1233zd(E) is being developed as substitute of R-245fa, while R-1234yf and R-1234ze, of R-134a.

Benzene, toluene and other hydrocarbons (HCs) commonly show good performance in medium-to-high temperature waste heat recovery applications, but they show also high toxicity and flammability issues, which could prevent them from vehicle applications use.

Water-steam is also considered in many studies in literature about HDDE waste heat recovery in vehicle applications, since it shows good thermodynamic performance, especially for medium-high temperature heat sources (EGR and exhaust). However, it present freezing issues in case of low ambient temperature conditions (de-freezing or warm-up systems could be considered).

Also ORC commercial applications use some of the considered fluids. For example, ORMAT develops geothermal ORC applications using n-pentane. Cryostar uses R-134a. R-245fa (or the commercial Honeywell Genetron®245fa) is being used by many stationary ORC manufacturers, such as Turboden (using also Solkhaterm), Bosch KWK, General Electric, Cryostar or Electratherm, Enertime.

Mixtures are also being considered in several studies, especially zeotropic mixtures, because of their capabilities to evaporate at variable temperature, thus leading to a better match with the heat source profile and lower heat exchange irreversibilities.

Several publications are available in literature about ORC suitable working fluids. For example, Bao et al. [92] reported a complete overview of several working fluids possibilities as well as ORC expanders considerations. Moreover, it is also possible to find publications regarding working fluids screening procedures and methodologies, as well as thermodynamic performance studies, in particular for medium-high temperature engine waste heat recovery. Some examples can be found in [93–99].

3.3. ORC architectures and layouts for vehicles HDDE

From a review of literature about ORC vehicle implementations and studies, focusing on HDDE and on and off-highway applications, it is possible to observe how the most developed ORC system layouts are simple cycles with one or maximum two evaporators to recover exhaust gas and EGR heat. The configurations are mostly in series or in parallel, with the possibility of having recuperation of heat between the outlet of the expander and the inlet of the evaporator to increase system efficiency. Another possibility is using the lower temperature coolant (or CAC) to preheat the working fluid before entering the evaporator. These layouts schemes have been reported in Fig. 6.

In vehicle applications, packaging and weight constraints are very important. For this reason, simple configurations are usually considered more suitable, rather than complicated multiple-loop or multiple-components systems. Integration of the system with engine, powertrain and vehicle thermal management are of great importance.

More complicated layouts or ORC evolutions studies are available in literature, such as dual-loop or cascaded ORC (e.g. [63,96],), two-stage ORC (e.g. [102,103]), regenerative or recuperated ORC, and some others innovative thermodynamic systems and cycles concepts (e.g. Goswami [104] and Kalina [105]). Additionally, some researchers propose also to directly recover engine block coolant heat with a suitable ORC working fluids which should, theoretically, be able to substitute actual engine cooling fluids, thus reducing system complexity and improving efficiency [106].

3.4. Thermodynamic and process modelling of ORC systems

The simplest Organic Rankine Cycle configuration is composed by four processes (Fig. 7a): (1–2) pumping of the working fluid from condensing to evaporation pressure, (2–5) evaporation of the working fluid in the heat source recovery heat exchanger (or boiler/evaporator), (5–6) working fluid expansion to extract useful power and (6-1) condensation of the working fluid and heat rejection to the heat sink. The ideal cycle is composed of two isentropic and two isobaric transformations (T-s diagram in Fig. 7b).

For modelling reasons, some other sub-transformations are usually considered. Indeed, when modelling steady-state thermodynamic behaviour of an ORC system, it is useful to subdivide the heat exchangers in three different regions: a single-phase pre-heater (2–3), a two-phase evaporator (3–4) and a single-phase super-heater (4–5) for the boiler, and a single-phase de-super-heater (6–7), a two-phase condensing zone (7–8), and a single-phase sub-cooler (8–1) for the condenser. For every zone, heat and mass balance equations are implemented.

For thermodynamic calculations, working fluids properties (temperature, pressure, enthalpy, entropy and specific volume) are required and retrieved from fluid properties databases or software such as NIST (REFPROP) [89] or the open-source CoolProp [108]. Some commercial software have their own already implemented database or calculate the fluid properties through the use of equations of state (e.g. Peng-Robinson [109]). Some examples are Engineering Equation Solver (EES) [110], LMS AMESim [111] and Aspen Hysys [112]. Properties databases can also be interfaced with other well known coding packages, as, for example, MATLAB.

Several levels of modelling detail are possible: from 0-D thermodynamic performance evaluation up to 1-D or 3-D CFD complete system components studies, both in steady-state and transient conditions. A complete overview of ORC modelling issues and simulation software possibilities is reported by Ziviani et al. [113] with particular focus on small CHP low temperature applications, considering all the main components in an ORC system.

Zero-dimensional thermodynamic modelling is useful, in a preliminary stage, to evaluate different ORC layouts and working fluid possibilities, with the goal of obtaining the best performance of the system at the design point, and choose the best configuration. Some examples can be observed in [114–116].

One-dimensional modelling is usually used to investigate the

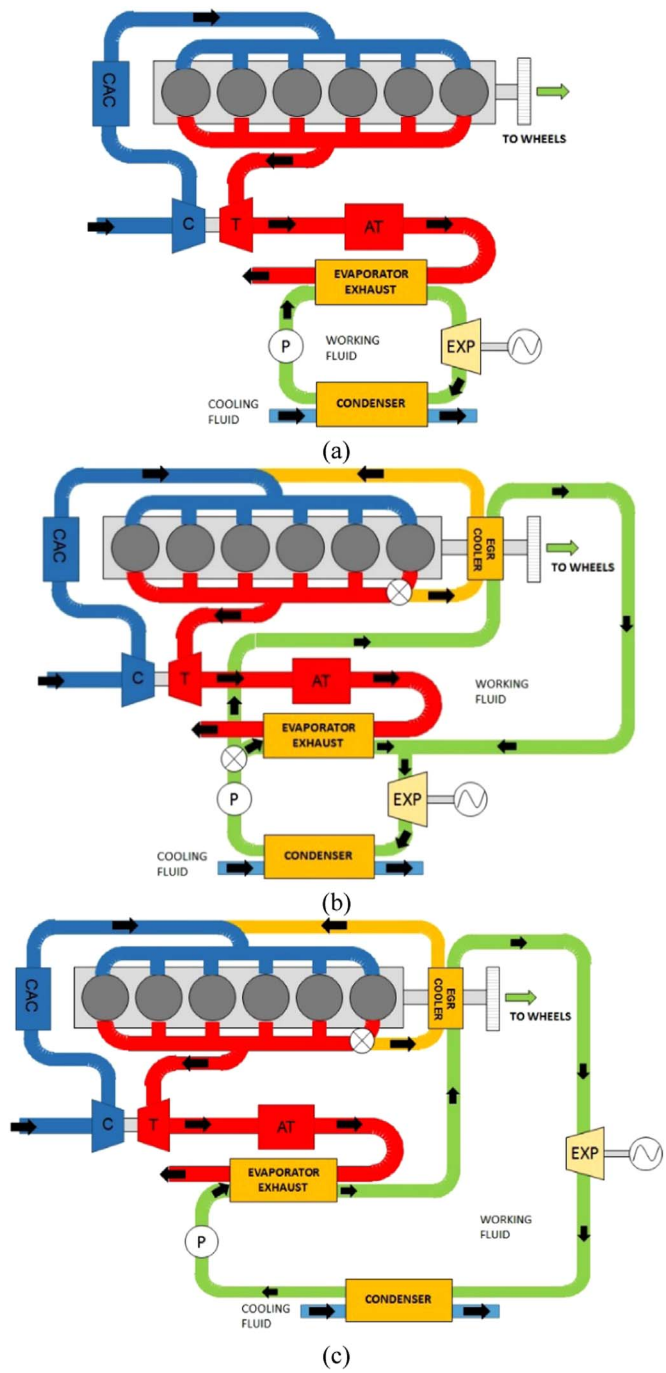

Fig. 6. Typical simple ORC layout for ICE WHR: single evaporator (a), parallel evaporators (b), series evaporators (c).

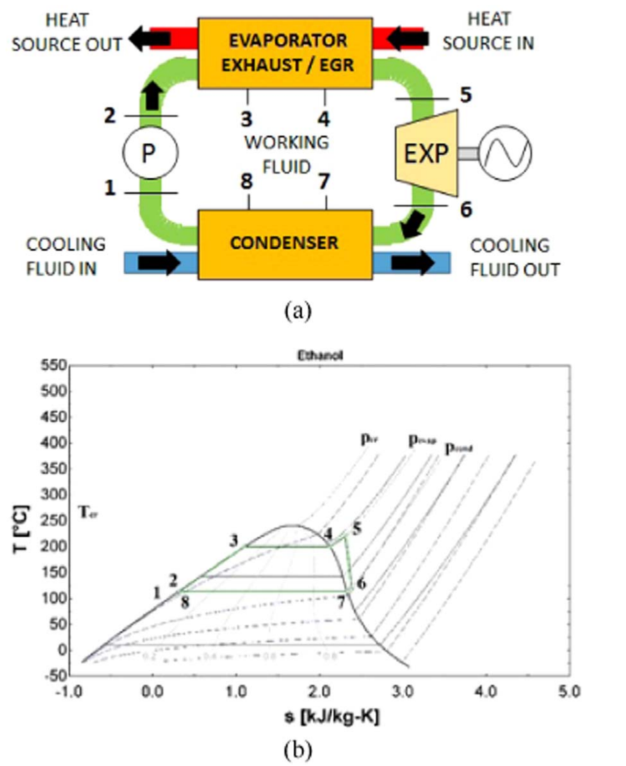

Fig. 7. Simple ORC layout (a) and T-s diagram example for Ethanol (b) obtained using EES (Engineering Equation Solver [107]).

performance of the ORC system in a more accurate way, under steady-state and transient behaviour, and to study the implementation of controlling strategies to optimize the system under part load and non-ideal conditions. For this purpose, expander performance estimation, through the use of adequate modeling techniques under off-design conditions, is also very important.

Three-dimensional CFD modelling is used to further investigate the single components fluid dynamic and thermal behaviour, with particular interest in expanders and heat exchangers optimization.

Furthermore, combined engine-ORC waste heat recovery studies are even more complicated, since they require the simultaneous simulation of the engine and ORC sides, often carried out in different software platforms which need to be coupled or used considering a synergic approach. For engine performance simulations, commercial 1-D codes are available (e.g. Ricardo WAVE [117]), or models have been developed in the academic sector.

New approaches can also be developed from a second law analysis performance study point of view [118], in order to assess combined engine-ORC (or bottoming cycles) powertrain improvement potential and minimize irreversibility in the system, when considering a coupled engine and waste heat recovery bottoming cycle optimization. Evaluations can be also carried out on typical duty cycles in order to assess fuel consumption and emission reduction benefits, as well as a thermo-economic analysis can be implemented [119], based on real prices/costs of engine and ORC equipment or estimating the costs based on typical costs modelling techniques (e.g. [120]).

3.5. Main ORC system components overview

In the following sections an overview of the main ORC system components has been reported, considering those particularly suitable for vehicles applications.

3.5.1. Heat exchangers

For engine waste heat recovery using ORC or Rankine cycles, usually shell-and-tubes, plate or compact finned-tubes (or finned-plates) heat exchangers are used. Shell-and-tubes types are mostly used in stationary applications and large systems and they can tolerate high pressures, while, usually, plates heat exchangers are used when recovering heat from liquids fluids rather than gases (e.g. coolant, or in condensers using water as condensing medium) or in smaller applications due to their compactness [10]. Plate heat exchanger can withstand lower temperatures (around 250 °C) compared to shell-and-tubes designs, because of plates deformations and gaskets-sealing problems [121], even though higher temperature suitable devices are under development. Metal foam heat exchangers [122] are also under study and can be used in vehicle waste heat recovery applications due to their compactness and enhanced heat transfer capabilities. Nevertheless, they result in being very expensive at the actual state of the art [121], and they lead to very high pressure drops on the engine exhaust gas side, with consequent increase in engine backpressure and decrease in performance [123].

In large ORC systems for stationary applications (e.g. biomass, stationary engines for power generation), very often an intermediate oil circuit is used to separate the heat source from the working fluid. In case of hot engine flue gas waste heat recovery, this solution is used to avoid flammability and safety problems in case of working fluid leakage in the heat exchanger. In case of vehicle applications, it is useful, in order to decrease the system weight and cost, and to increase heat transfer efficiency, to directly transfer the heat from the heat source to the working fluid using a direct evaporation configuration. The heat exchanger installed on the heat source has to tolerate high pressures (especially working fluid side), high temperatures, corrosion and fouling problems, especially when recovering heat from high sulphur content flue gases. In this case the exhaust gas must not be cooled down to less than 90–120 °C (depending on sulphur compounds amount), to avoid possible acid condensation and damaging of the heat exchanger. Currently, components able to better withstand acid condensation are under study and development (e.g. using stainless steel, [124]).

When sizing a heat exchanger, a right pinch point temperature between the heat source/heat sink and the working fluid must be chosen, usually as a trade-off between performance maximization and cost minimization (heat exchanger dimensions). Common pinch points trade-off values are 10 K for gas-to-gas heat transfer and down to 5 K for liquid-to-gas or liquid-to liquid heat transfer.

In case of direct evaporation configuration, the gas-side pressure drop in the heat exchanger should be minimized in order to have a low impact on engine backpressure, which increases engine pumping losses. Advanced turbocharging strategies have to be implemented in order to withstand this negative effect and counterbalance the engine performance.

Several manufactures of thermal management components (e.g. Mahle-Behr and Modine) are working to replace EGR coolers and engine tailpipe section with ORC suitable evaporators, and to increase heat transfer performance and compactness. Hatami et al. [125] presented several techniques to increase heat transfer effectiveness for different heat exchangers designs.

In Fig. 8 an example of finned-tube heat exchanger designed by Zhang et al. [126] has been reported.

Regarding condensers, a few condensing strategies possibilities are available in case of engine waste heat recovery in vehicle applications: indirect condensation using the engine cooling circuit as heat sink (average temperature range around 80–100 °C), indirect condensation using a lower temperature cooling circuit (e.g. CAC coolant, average temperature level 40–70 °C), or direct cooling using an ambient air condenser (installed in the vehicle cooling pack). In the first two cases, the coolant heat must be also rejected to the environment through the vehicle cooling package. In ORC vehicle applications, thermal management of the combined engine-ORC-powertrain system is of vital importance and must be analysed carefully both under steady-state

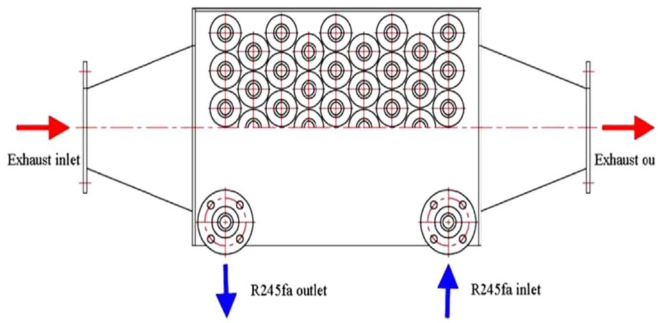

Fig. 8. Finned-tube exhaust heat exchanger example (Zhang et al. [126]).

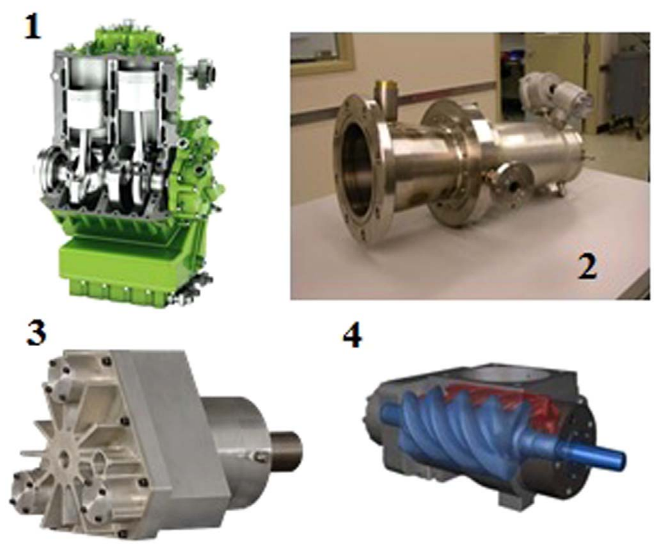

Fig. 9. SteamDrive SteamTrac piston expander (1, Courtesy of SteamDrive [132]), Barber-Nichols radial expander (2, Courtesy of Barber-Nichols [133]), Air-squared scroll expander (3, Courtesy of Air-squared [134]), Electratherm twin-screw expander (4, Courtesy of Electratherm [135]).

and transient conditions. Advanced CFD models are used to optimize cooling package architectures. Space restrictions, as well as operational behaviour of the vehicle (speed, ram-air cooling effect) and ambient conditions (hot weather conditions are more challenging), are leading to severe cooling package sizing issues.

Moreover, condensing pressures should be higher than ambient atmospheric pressure, to avoid air leaking into the system and expensive sealing. Condensing temperatures and pressures must also be chosen based on expected ambient conditions (e.g. to avoid inverse heat transfer during hot days).

In marine and stationary applications, condensation processes are less challenging, due to less space and weight constraints as well as to availability and lower cost of cooling mediums (e.g. sea or fresh water).

3.5.2. Pumps

The pump in the ORC system is used to pressurize the working fluid from condensing to evaporation pressure and to control the working fluid mass flow rate in the circuit. A complete overview of the main pumps requirements for ORC systems, such as controllability, efficiency, tightness, NPSH, has been reported by Quoilin et al. [10].

Different type of pumps could be suitable for ORC use. For example, positive displacement pumps (in which the working fluid flow rate in proportional to the rotational speed). An example of positive displacement pumps are diaphragm pumps, in which the contact between fluid and components is avoided and tightness of the system improved. Pulsed flow rate could be a drawback. Other examples are reciprocating piston pumps, or rotary pumps (e.g. screw, sliding or rotary vane, scroll, roots or gear pumps). Centrifugal pumps are also available. In this case the flow rate depends also on the pressure head between evaporation and condensing pressures. In some cases, the use of the most appropriate pump is a very important design choice, both in terms of efficiency and costs.

The pump also controls the evaporation pressure in the system. The electrical motor driving the pump can be coupled with an inverter in order to change the rotational speed and control the working fluid mass flow. Magnetic coupling is often used to transfer torque from the electrical engine to the pump, thus not using seals for the shaft (wear and fluid corrosion problems are avoided).

3.5.3. Expanders

The expander is one of the most critical components of an ORC, since the performance of the system is directly related to the performance of the expansion machine.

Two categories of expanders can be distinguished: turbo machines and positive displacement machines [10].

Turbo-expanders are more suitable for larger ORC systems and they show better performance when operating in a steady state condition at design point (off-design operations are less efficient compared to positive displacement type unless variable inlet guide vanes are used). They can be axial or radial turbines. The first type in mostly used in large waste heat recovery systems for stationary power

Table 5
Commercial or developed expander technologies for possible ORC vehicle application.

Company	Model	Working Fluid	ORC Power Range	Efficiency [%]	Expander Type	Speed [rpm]	References
Verdicorp	n.a.	R245fa	30–180 kW	n.a.	Turbo/radial	n.a.	[136]
Electratherm	n.a.	n.a.	35–110 kW	n.a.	Twin Screw	n.a.	[137]
Barber Nichols	n.a.	Fluorinol-50, R245fa, toluene, water-steam, R134a	3 kW–250 kW	85	Turbo/radial	up to 60,000	[138]
Green Turbine	Green Turbine	water-steam, organic fluids	1.2 kW–15 kW	n.a.	Turbo/radial	up to 30,000	[139]
Cummins	n.a.	R245fa	42 kW	77	Turbo/axial	80,000	[140]
Infinity Turbine LLC	IT01 / IT10 / IT50	R245fa	10 kW–3 MW	73	Turbo/radial	n.a.	[141]
Air Squared	E15H022A-SH/ E22H038A-SH	R245fa, R134a, other gases	1–5 kW	70–80	Scroll (oil-free or lubricated)	2600–3600	[142]
Eneftech	n.a.	R245fa	1–5 kW	80	Scroll	2000–6000	[143]
Liebherr	n.a. - Patent	R245fa, ethanol, water-steam	n.a.	n.a.	Rotary vane	n.a.	[144]
SteamDrive GmbH	SteamTrac/ SteamDrive	water-based medium	20–360 kW	more than 65	Reciprocating piston	3600	[129]
Exoes	n.a.	Water-steam or ethanol	4 kW	40–45%	Swashplate piston	1000–6000	[145]
Viking Development Group	CraftEngine	Organic fluid	2–40 kW	n.a.	Reciprocating piston	750–1500	[146]

generation applications with lower pressure ratios and high working fluid mass flow (e.g. some Turboden models). Radial turbines are mostly used for high pressure ratios and lower working fluid flow rates. Axial turbines are more suitable to be assembled in several stages. Organic fluids suitable turbo-expanders have also more compact layouts and sizes compared to steam turbine, because of the higher density and lower specific volume of organic fluids compared to water-steam. Moreover, turbines used in ORC applications have lower enthalpy drops compared to steam, thus leading to the possibility of using only one or two expansion stages [92]. Turbomachines are not very suitable for small ORC systems, because of their very high rotating speed which could lead to structural problems. Additionally, turbo-expanders do not tolerate high amount of liquid during expansion, because of possible blade damaging problems, but advanced engineering activities are currently ongoing to mitigate all these risks and offer affordable and reliable turbo-machines as possible expansion devices for vehicle ORC. The high speed generator high cost remains an issue.

Positive displacement expander type examples are: reciprocating piston, scroll, screw, vane and Wankel expanders. A complete overview about these types of expanders, technical issues and considerations, as well as modelling techniques is presented by Lemort and Quoilin [127]. A review of working fluid and expander selection criteria is also reported by Bao and Zhao [92].

Reciprocating piston expanders are mostly used for small-scale CHP and waste heat recovery systems (such as internal combustion engines applications). They can operate with large pressure ratios [128] because of their large volume ratios (or Built-In-Ratios, the ratio between expansion chamber volume at expansion end and expansion beginning), in practice between 6 and 14. This type of expander requires precise timing for intake and exhaust valve, vibration control and balancing. Moreover, they show high frictional losses (e.g. piston rings-cylinder walls interactions), lubrication and sealing problems. Piston expanders can tolerate high pressures and temperatures (70 bar, 560 °C) and liquid phase during expansion is also well tolerated. Free-piston expanders are also under development. An example of swash plate axial-piston expander ORC vehicle implementation is reported by Endo et al. [50]. A commercial implementation of a reciprocating piston expander for ORC systems is the two-stroke SteamTrac model (Fig. 9-1) from Voith-SteamDrive [129], for on and off road, as well as marine and railway applications.

Scroll type expanders can operate over a lower pressure ratio range due to their lower Built-In-Ratio (1.5–4) and they also undergo under-expansion or over-expansion losses. Other important losses are due to friction (lubrication is needed), leakage and heat transfer. They can tolerate lower temperatures compared to piston type (215 °C). It is very common that scroll devices for ORC applications are retrofitted from AC compressors for automotive applications, and mostly used for low temperature waste heat recovery (e.g. coolant). Also for scrolls, liquid phase during expansion is well tolerated and they can adapt to a wide range of operating conditions. Some oil-free models are under development. A review of scroll expanders for ORC systems is reported by Song et al. [130] and scroll performance issues are discussed in [99].

Screw-type (Lysholm) expanders (and twin-screw) has been also used as compressors in the past. They are mostly used in geothermal applications with medium-high power output (20 kW–1 MW). They show moderate Built-In-Ratios (5–8). Oil is needed for lubrication. Rotational speed is higher than for other expander types (reduction gearboxes are needed when using it in vehicle applications, feeding the recovered energy back into the crankshaft). Sealing is also very important to reduce leakage, especially in case of dangerous fluids. Manufacturers of ORC systems using screw expanders are, for example, Electratherm and Ormat, and they provide products for a power range starting from 50 kW_{el} [131].

Vane-type expanders are also suitable for low power outputs ORC systems. They can have single acting or double acting configurations. They can tolerate a wide range of vapour qualities without damaging problems. They can be easily manufactured and they provide smooth torque production. They can be suitable for engine mechanical coupling without gearbox due to their low rotational speed (3000 rpm). Little lubrication requirements as well as low level noise are other advantages. Leakage losses is a possible drawback, together with high pressure drops [121], due to the difficulty of vanes to maintain a good contact with the housing. The rotational speed is strongly affected by the pressure and flow rate of working fluid. Friction losses are also becoming important at higher speed rotational regimes.

Wankel devices have been implemented in the past as air compressors or internal combustion engines. They are also suitable for low power output levels, they have simple configurations, but sealing and lubrication problems.

A study about the utilization of a roots expander for HDDE applications is reported by Subramanian [60].

Some examples of commercial volumetric expanders, in the possible HDDE vehicle waste heat recovery power range (5–60 kW), have been reported in Fig. 9.

Another important issue regarding expanders in ORC engine waste heat recovery for vehicles implementations is the integration into the overall powertrain or driveline. Indeed, two possibilities are available: mechanical or electrical coupling.

In the first case, the expander produced net power is re-introduced into the crankshaft through the use of a gearbox or other type of transmissions, depending on the difference in rotational speed between expander shaft and engine shaft. A clutch can be inserted to disconnect the ORC expansion machine and the engine during, for example, downhill driving conditions [45].

In the second case, the expander is connected to a generator, and electrical energy is produced. The energy can be used to charge batteries, for auxiliary systems (AC, cooling refrigerators, on board electrical devices) or re-introduced into the powertrain in Diesel-hybrid vehicles configurations.

In Table 3, a summary of heavy duty Diesel engine vehicle ORC implementations by automotive manufacturers and research institutions has been reported, focusing the attention on the expander technology tested and proposed, while in Table 5, an overview of different expander technologies available on the market in the power range for the considered applications has been reported.

4. Conclusions

A large potential for fuel consumption improvement is identified when using waste heat recovery systems to recover heat from Heavy Duty Diesel Engines in on and off-highway applications. For this reason, several different bottoming cycles and technologies are currently under study and development. In particular, the interest of industry and academic institutions is focused mainly on improved turbocharging strategies, turbo-compounding systems, active cold start-warm up systems, Thermo-Electric-Generators (TEGs) and Organic Rankine Cycles (ORCs).

When developing a waste heat recovery system, the engine application specific operational profile must be carefully investigated in order to choose the right system design point. For this purpose, an analysis of typical duty cycles or real operational data is necessary in the initial stage of every project.

Among all waste heat recovery technologies, ORC seems to be one of the most promising, allowing a possible fuel economy up to potentially 10%.

Even though ORC technology is well known since decades, and already successfully applied in stationary power generation and industrial processes waste heat recovery, vehicle applications are still in a research and development phase and a potential for further development can be recognised.

Most of the studies reported in literature consider exhaust gas and EGR as primary heat sources to be recovered in order to improve engine fuel consumption through the use of ORC. The most common layout considered is the parallel boilers configuration. CAC and coolant heat sources seem to have lower potential compared to the higher temperature sources. In case of bigger size engines (e.g. stationary power generation or marine applications), coolant heat recovery could be considered as an option because of the high amount of volume flow available, even if at lower temperature.

More complicated thermodynamic cycles (e.g. Goswami and Kalina) or ORC layouts are not as suitable in vehicle applications due to space and weight constraints, unless innovative packaging solutions and efficient compact components are developed.

ORC waste heat recovery systems for on and off-highway vehicles applications have, at the moment, a number of issues which prevent effective commercialization, such as: suitable working fluid choice, packaging, weight, space and thermal management issues, as well as safety, reliability and cost.

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