Waste heat recovery (WHR) assessment in complete truck simulation environment

F. Galuppo[1,2,3], T. Reiche[1] , V. Lemort[2], P. Dufour[3], M. Nadri[3], X. Huin[1]

1: Volvo Trucks, 99 route de Lyon, 69800, Saint Priest, France
2: Université de Liège, 17 Allée de la Decouverte,4000, Liège, Belgium
3: Université Claude Bernard Lyon 1, 3 rue Victor Grignard, 69100, Villeurbanne, France

Abstract: Organic Rankine Cycle (ORC) based Waste Heat Recovery in long haul heavy duty (HD) trucks is a promising technology to reduce the fuel consumption.

Despite several studies that have been conducted, such technology has not been introduced yet on the market. The constraints for a market introduction are related to driving cycle fuel efficiency, the integration of the WHR system in the truck, control of the system, material compatibility, weight and costs.

This paper outlines the mentioned constraints and proposes a complete truck environment simulation study, considering interactions of the system with the other sub-systems of the truck.

Control strategies and optimized Rankine energy utilisation are proposed to maximize the effectiveness of the system.

Keywords: WHR, ORC, HD truck, Fuel economy.

1. Introduction

Several studies have been conducted in the last decades to investigate the means of improvement of Diesel engine efficiency [1], [2]. Stricter regulations on truck emissions and the high quantity of fuel energy used in the transportation sector represent the driving forces of this trend; in a conventional truck engine, the fuel energy losses to the cooling system, lubricant and ambient, account to roughly 60% (40% of fuel efficiency) [3]. In the coming years, those conventional power trains could reach 50 % of fuel efficiency [4], thanks to smart thermal management, new combustion processes and hybridization. However, full hybridization and full electrical trucks cannot replace thermal Diesel for HD long haul applications in the next future engine because of limitations on battery size and lack of electrical infrastructures [5]. The interest in reducing fuel consumption is also related to the increasing fuel price in Europe: based on fuel price projections used in some recent European Commission studies, the fuel price would be 0.889 €/l (1.19 €/l incl. taxes) in 2020, increasing to 1.055 €/l (1.32 €/l incl. taxes) in 2050 [6].

WHR technology for long haul HD trucks is considered to be introduced in the heavy duty vehicle (HDV) market in the coming years in such a way to increase fuel efficiency and meet future legislations [7].

Despite the proven potential of WHR systems, ORC based, in long haul HD trucks, manufacturers and suppliers need to face hurdles related to:

- Heat management optimization
- Costs
- Additional weight
- Added complexity to the driveline

The Rankine cycle is a well-known thermodynamic cycle broadly used to produce electrical power in thermo electrical power plants. In most of cases, water is used as working fluid, because of good thermo physical and chemical properties adapted to the given heat sources, safety and availability.

The objective of Rankine cycle WHR in HD trucks is to recover heat from one or several hot sources in order to produce mechanical energy which can be used directly in the truck driveline (mechanical WHR system) or used after conversion in electrical energy in the truck electrical network (electrical WHR system). The available hot sources can be exhaust gas flow, Exhaust Gas Recirculation (EGR), engine coolant [8] and can provide 100 kW. ORC, belonging to the category of Rankine thermodynamic cycles, are more suitable in case of low heat sources; the main difference with respect to the classic cycle is related to the fluid, which is an organic one. Organic fluids are characterised by lower boiling point and latent heat with respect to water and the heat recovery from low heat sources is more effective. Different studies about fluid selection for ORC have been conducted, but nowadays no perfect working fluid has been identified for all the possible applications [9].

When designing a WHR system, ORC based, the choice of the working fluid is influenced by the level of the hot source, the design of the available components and safety aspects. For a correct and complete analysis, all those factors have to be considered and this makes the choice of the working fluid difficult.

This paper shows a simulation study, based on complete truck simulations on Simulink tool, whose main ambition is to consider all the impacts of the ORC system on the other components of the truck. Particular attention is paid to the Exhaust After Treatment System (EATS) and Cooling System

whose normal operation is influenced by the ORC operation.
The ORC configuration is a mechanical type, whose ambition is to recover thermal power from exhaust gases and EGR flow to produce mechanical power, to be directly injected in the truck's driveline via a fixed gear ratio. A simplified physical model, partially map based, of the volumetric expander machine is presented and results are then analysed by considering new solutions to maximize the effectiveness of the WHR system.
Simulations are performed in transient conditions, considering two different driving cycles: a French highway representative and a European long haul truck profile. Disturbances (variability of temperature and mass flow of the flows) on the system are taken into account. Three engines are considered: a Turbo Compound EGR engine (TC - engine), an engine without cooled EGR and Extended Selective Catalytic Reaction (ESCR engine) and a Variable Geometry Turbocharger cooled high pressure EGR engine (VGT engine). The ORC's performances are then compared based also on the engines differences.
The paper is organized as follows:

- In section 2, the simulation problem is defined
- In section 3, the simulation study is presented
- In section 4, the simulation results are presented and interpreted

2. Simulation model definition

2.1 Rankine cycle model

The working fluids that have been considered are ethanol and cyclopentane.

The ORC is characterized by the following main components:

- A pump, which can be of volumetric or kinetic type machine
- One or two heat exchangers (evaporator), depending on ORC architecture
- A positive displacement expander
- A condenser
- A tank

Other components composing the ORC are modelled; by-pass valves of exhaust and EGR gases, when thermal power recovery has to be stopped, a splitter valve for the separation of the working fluid flows into two flows, expander inlet orifice, which imposes the pressure in the whole circuit.

Fig1 shows the ORC model implemented in the tool Simulink, MathWorks.

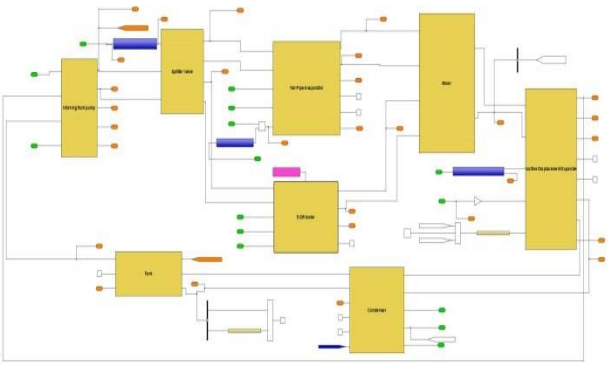

Figure 1: Implementation of the ORC model in Simulink

The heat exchangers (evaporators, condenser) are modelled using a finite volume approach discretising the heat source, heat sink with a one dimensional discretisation into 8 elements. Mass and energy conservation is applied to the fluid and gas sides, calculating for each time step enthalpy of the fluid and temperature of the exhaust gas. The local wall temperature is calculated by applying the energy conservation based on the heat fluxes calculated on both sides of the heat exchanger.

Pressure drops on both sides are neglected.

The expander machine, belonging to the family of positive displacement expanders (in Fig. 2 the indicator diagram of such kind of expander is presented), produces the mechanical power, which is then used in the driveline of the truck. The expander model is semi-physical, based on thermodynamic laws and maps for the isentropic efficiency evaluation. The expander admits a determined volume of working fluid inside the cylinders, depending on its rotational speed, and imposes the pressure at the outlet of the pump.

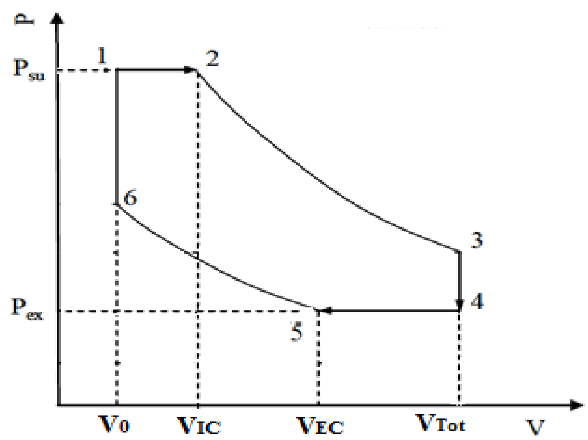

Figure 2: Indicator diagram of a positive displacement expander

The density at the inlet of the expander is then calculated by using eq.1:

$$\rho_1 = \frac{M}{V} = \frac{60\frac{\dot{m}_{WF}}{N_{exp}} + \rho_{end,recomp} V_0}{V_{cap} CO + V_0}$$

[1]

Where:

- $\dot{m}_{WF}$ is the working fluid mass flow in kg/s
- N_{exp} is the expander speed in rpm
- $\rho_{end,recomp}$ is the density of working fluid at the end of the recompression in kg/m3
- V_0 is the volume related to the top dead centre (TDC) of the expander (clearance volume)
- V_{cap} is the volume capacity of the expander, defined as the difference between the total swept volume of the expander and the TDC ($Vtot - V0$)
- CO is the cut-off ratio, defined as the ratio between the volume which corresponds to the admission of the working fluid ($Vic - V0$) and the volume capacity.

Once the density is calculated and temperature is known, pressure p is evaluated via thermodynamic tables.

The efficiency of the expander machine is determined from pressure ratio and expander speed N_{exp} by using tables containing experimental results and the enthalpy at the end of the expansion is found. Assuming no ambient losses the mechanical power is therefore:

$$\dot{W}_{out} = \dot{m}_{WF} W_{real} = \dot{m}_{WF}(h_{su} - h_{ex})$$

[2]

2.2 Rankine cycle control

The road cycle imposes transient operating conditions to the WHR system, leading to the following problems to be addressed:

- The expander needs to be fed with superheated steam; ideally with a constant superheat which is defined as the temperature difference between the actual temperature and the saturation temperature at the operating pressure:

$$SH_{WF,exp,su} = T_{WF,exp,su} - T_{sat}(P_{WF,exp,su})$$

[3]

- The Rankine cycle shall not produce mechanical energy when the driveline cannot use the energy, e.g. in braking phases or during standstill;
- The working fluid must be protected from high temperatures in order to avoid degradation;
- The Rankine cycle shall not be used when its heat rejection leads to excessive fan engagement;
- At the pump inlet, sufficient subcooling shall be applied to avoid pump cavitation.

Therefore the following actuators are used:

- The working fluid pump is speed-controlled in order to control the mass flow rate and in consequence the superheat level. Therefore a multi-model PID controller is used to adapt to the different time constants of the heat exchanger depending on the operating conditions.
- A bypass of the expander is used to disengage the expander from the driveline
- A bypass is used on the exhaust side to regulate the heat input in the WHR system

2.3 Complete vehicle model

The Rankine cycle model is coupled to a component-based vehicle model in which the complete vehicle driveline, the engine coolant system as well as the electrical architecture is represented by separate and individual controlled components. Furthermore driver and road components are used to characterise the complete vehicle environment.

A low temperature cooling loop has been used to feed the Rankine condenser (see Fig.3). The low temperature radiator is placed between the direct charge air cooler and the main radiator.

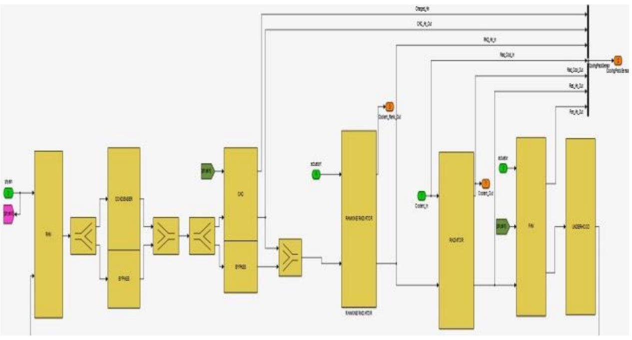

Figure 3: Cooling package

3 Simulation study

3.1 Boundary conditions

In this study only European HD vehicles and drive cycles are considered. The following table summarizes the moist relevant vehicle boundary conditions:

Vehicle parameter	Value

Vehicle Gross weight	35T
Wheel configuration	4x2 (315 R70 225)
A cx	5m²
Engine 1	13l Eu6 eSCR, 480Hp
Engine 2	13l TC, 460Hp
Engine 3	13l VGT EGR, 445Hp

Table 1: Vehicle boundary conditions

Two different road cycles have been analysed, a French hilly highway cycle (LCG) and a combined cycle representing a typical European long haul road profile (LH08) for European HD Trucks with flat highway portions, hilly roads and urban sections.

	Road cycle 1 (LCG)	Road cycle 2 (LH08)
Length (km)	204	1628
Mean target speed (km/h)	76.3	74.8
Cumulated altitude (m)	7	24

Table 2: Road cycles

On the Rankine cycle side the following, boundary conditions are applied:

BC	Value
Max fluid temperature	230°C
Max fluid pressure	40bar
Min Condensation pressure	1bar abs
Min Subcooling inlet pump	5K
Superheat	20 K

Table 3: Rankine cycle Boundary conditions

4. Results

4.1 Rankine cycle results

Results of the performance of the controller of superheat at the inlet of the expander are shown in Fig. 4 (a 13TC 460P, in LCG driving cycle, ethanol as working fluid).

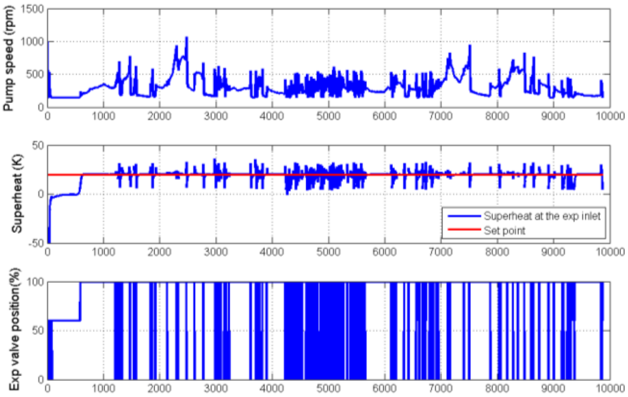

Figure 4: superheat control and by-pass valves position (LCG road cycle)

The expander by-pass opens when superheat is not sufficient (see Table 3 for boundary conditions) and in the following situations:

- The engine does not require positive torque
- The pressure ratio of the working fluid between inlet and outlet is lower than 5.

The power balance which corresponds to this simulation is shown in Fig. 4; this specific driving cycle is characterized by several braking in the central part of the driving cycle that, consequently, limits the mechanical power production.

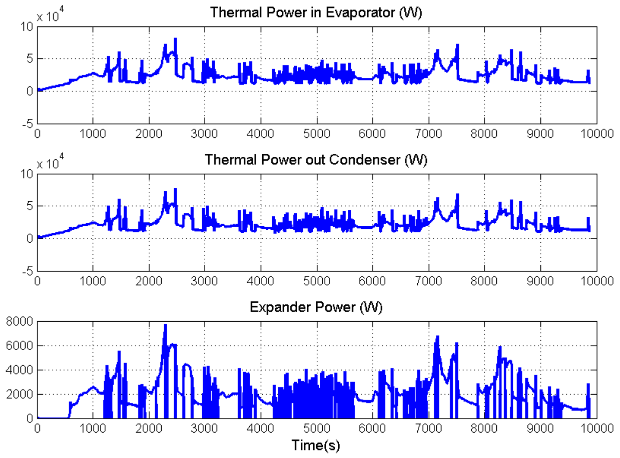

Figure 5: Power balance (LCG road cycle)

4.2 Rankine cycle impacts

The Rankine cycle integration involves issues like additional weight, exhaust backpressure and cooling system impact. This latter aspect is related to the fact that the ORC rejects heat via the condenser to the cooling system, which has to dissipate an additional quantity of heat.

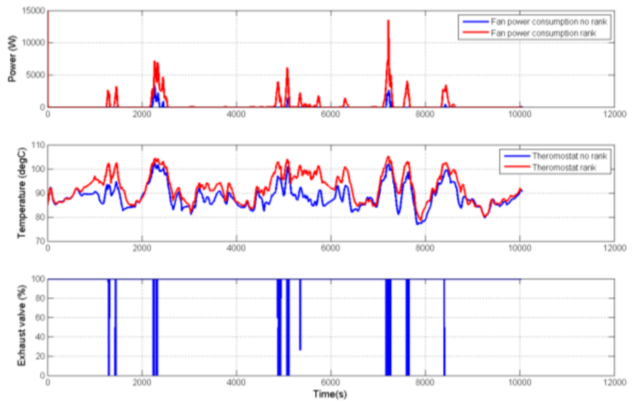

Figure 6: Fan engagement and exhaust valve (LCG road cycle)

A comparison based on the fan engagement and thermostat temperature (cooling temperature at the outlet of the engine) is presented in Fig.6; the impact of Rankine cycle with respect to the simulation without the WHR is evident on the whole behaviour of the cooling system (thermostat temperature) and, consequently, on the fan power. Fan consumption, in terms of energy in the road cycle, is roughly multiplied by 5 with respect to the simulation without WHR. This impact is taken into account in the estimation of the fuel consumption of the driveline.

The phases corresponding to high heat load to the cooling systems are critical and fan is therefore engaged; during these operating points the working fluid temperature increases and, according to the mentioned conditions, the exhaust valve at the inlet of the exhaust boiler may close. The exhaust gas by-pass opens when temperature of the working fluid overpasses the limit set in the boundary conditions and the time derivative of the temperature is positive.

4.3 Results on driving cycle

The efficiency of the thermodynamic cycle is calculated as follow (eq. 4)

$$\varepsilon_{Rankine} = \frac{\dot{Q}_{ev} - \dot{Q}_{cond}}{\dot{Q}_{ev}} = 1 - \frac{\dot{Q}_{cond}}{\dot{Q}_{ev}}$$

[4]

It is evident that, increasing the thermal power which is transferred to the evaporator, the efficiency of the Rankine cycle increases, since the energy recover potential is larger.

Heat source is widely variable during the driving cycle, depending on the operating points of the engine. Besides, the engine plays also an important role. Fig. 7 shows, for the LCG driving cycle, the exhaust gas and EGR temperature of TC, VGT and eSCR engines, highlighting the differences in terms of temperature among them.

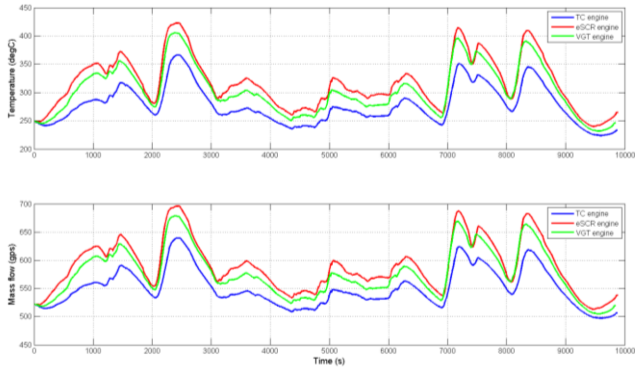

Fig.7: Temperature and mass flow on LCG road cycle

As expected, the eSCR engine feeds the Rankine cycle with hotter exhaust gases and it also provides higher flow rates. On the other hand, the TC engine provides lower mass flow rates and temperatures, due to the turbo-compound which reduces a lot the energy content of the exhaust gases. Moreover the VGT engine can be very promising, considering that thermal power recovery from EGR is also possible.

In order to evaluate the effectiveness of the different applications of the Rankine cycle in HD trucks, average results on driving cycle are presented, focusing on a performance coefficient, which is proportional to the fuel economy evaluated by comparing the fuel consumption of the base truck and the truck equipped with the WHR system.

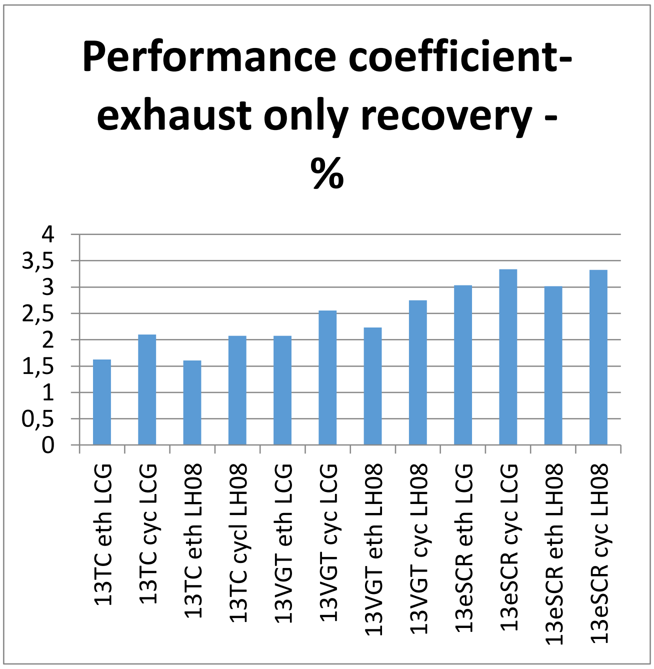

Fig.8: Performance coefficient, exhaust only recovery

As expected, the TC engine provides the lowest results in terms of performance coefficient for the same driving cycle, being characterized by the lowest content of thermal energy in the exhaust gases. The best results are achieved by the eSCR engine in LCG driving cycle, using cyclopentane as working fluid.

Simulations point out that cyclopentane potential is interesting; despite the presence of a larger flow rate of working fluid in the circuit, and consequently high pump power consumption, the expander power production is higher and the performance of the cyclopentane is better when low-temperature heat sources are used (e.g. TC engine). The advantage of cyclopentane is reduced when using a eSCR engine, since the limitation on working fluid pressure and temperature are easily reached and exhaust by-pass often opens.

Cyclopentane improves the performance coefficient using a TC engine exhaust recovery architecture, with respect to ethanol, by roughly 30%, while using a VGT and eSCR engine the improvement reaches 23 % and 10 % respectively.

TC and VGT engines are equipped with the cooled high pressure EGR in order to reduce NOx emissions. The EGR stream can be an important additional source of heat that can be integrated in the WHR system: two more configurations are now available, the parallel configuration (simultaneous recovery from exhaust gases and EGR flow) and the recovery from EGR only.

The parallel architecture is very promising, but it implies the installation of two major components:

- The EGR boiler (additional component)
- A splitter valve for separating the working fluid flow into two parts, one going to the exhaust boiler, the other one to the EGR boiler

Furthermore, in the parallel configuration, the control of superheat is more complex, because it is necessary to implement a controller that is able to track the set point of the superheat at the outlet of the exhaust boiler and EGR boiler, acting on the pump rotational speed (total working fluid mass flow) and the position of the splitter valve.

In Fig. 9, the performance coefficient for a parallel configuration is shown.

A first conclusion is that parallel configuration simulation performed with a 13VGT engine reaches a performance coefficient of roughly 4% which is more than the best result obtained with the 13eSCR engine.

In this case, the effectiveness of the cyclopentane with respect to the ethanol is weaker, because the heat source level is high in each case, since the parallel recovery is able to recovery a larger amount of heat in the boilers.

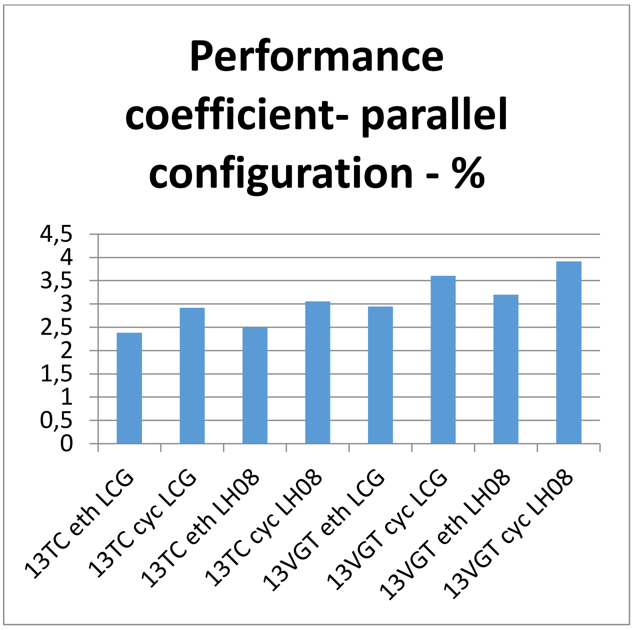

Fig.9: Performance coefficient, parallel configuration recovery

The last configuration that is simulated is the recovery of thermal power from EGR flow only. This interesting opportunity can lead to:

- Absence of backpressure in the exhaust gas circuit
- Lower amount of flow rate of working fluid and consequently reduced pumping power
- Smaller and cheaper components

In the same time, drawbacks are mostly related to the modest amount of energy that is recovered and therefore produced by the WHR system.

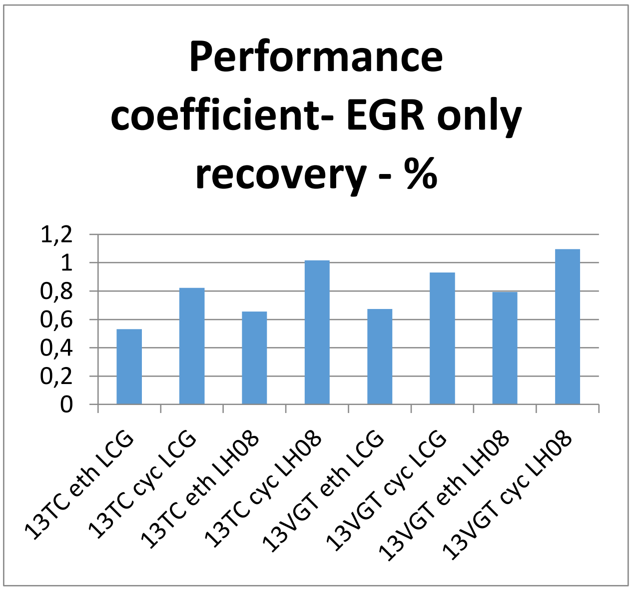

Fig.10: Performance coefficient, EGR configuration recovery

Clearly, EGR flow provides lower amount of thermal power with respect to the exhaust gases and the simultaneous recovery of exhaust gases and EGR.

This leads to the fact that the performance coefficient is limited.

In the same time, as expected, effectiveness with cyclopentane is bigger, since the energy content source related to the EGR only recovery architecture is low. The WHR system, using cyclopentane in an EGR only recovery configuration, can improve the performance coefficient up to roughly 55% (TC engine). The Rankine cycle, using cyclopentane as working fluid, can reach 1% of perfoemance coefficient, with both TC and VGT engines.

Results show that choosing cyclopentane in place of ethanol when using a TC engine is fully justified, especially when operating with a simple architecture (recovery from exhaust gases only or EGR only). Ethanol, taking into account the larger number of applications and available studies in literature, seems to be the good choice when operating with hotter heat sources (eSCR) and more complex architectures (e.g. parallel configuration).

The engagement of the expander in a mechanical WHR system is also determined. The fraction of time when the expander is working and producing mechanical power depends on the torque demand and the physical properties of the working fluid at the inlet of the expander. Control of the superheat is effective for keeping the working fluid physical variables in a range that is acceptable for the correct operation of the expander, but when torque is not requested by the engine, mechanical power cannot be delivered and, therefore, produced. Consequently the expander by-pass opens.

It is estimated that for roughly 10% of the time of the driving cycle, the expander by-pass is open and the main cause is the lack of torque demand.

In this case the introduction of an electric WHR system, in the frame of a micro-hybrid solution, can help to operate the expander for the whole length of the driving cycle and fully exploit the thermal power of the source. The produced mechanical power is then converted into electrical energy which is stored in the batteries and delivered in an intelligent way to the electrical motor. In this case, regenerative recovery and WHR system can co-operate to reach higher values of fuel economy.

5. Conclusions

The simulation study has pointed out the potential of an Organic Rankine Cycle based WHR system, considering a global vehicle model. In this way, the major influences that the WHR system has on the vehicle have been taken into account.

The study confirms the effectiveness of the Organic Rankine cycle based WHR system in long haul HD trucks and proposes solutions to optimize its operation, by choosing a proper working fluid and architecture.

An additional remark about the utilisation of the produced energy is given, pointing out that the micro-hybrid solution with a WHR system can be more effective than a mechanical one, with a full utilisation of the expander machine and transformation of the mechanical to electrical energy.

5. Acknowledgement

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