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Evaluation of a Coupled Organic Rankine Cycle Mild Hybrid Architecture for Long-Haul Heavy-Duty Truck ^{*}

Francesco Galuppo ^{*,**,**} Madiha Nadri ^{**} Pascal Dufour ^{**}
Thomas Reiche ^{*} Vincent Lemort ^{***}

^{*} Volvo Group, Powertrain Strategic Development, 1 Avenue Henri Germain, 69800, Saint Priest, France (e-mail: francesco.galuppo@volvo.com, thomas.reiche@volvo.com)

^{**} Univ Lyon, Université Claude Bernard Lyon 1, CNRS, LAGEP UMR 5007, 43 boulevard du 11 novembre 1918, F-69100, Villeurbanne, France. (e-mail: pascal.dufour@univ-lyon1.fr, madiha.nadri-wolf@univ-lyon1.fr)

^{***} Thermodynamics Laboratory, University of Liège, Campus du Sart Tilman, B49, B-4000 Liège, Belgium (e-mail: vincent.lemort@ulg.ac.be)

Abstract:

Increasing fuel prizes, future CO₂ emission legislation as well as new powertrain architectures are the main factors for truck manufacturers to invest into fuel saving technologies. Especially for long haul heavy duty transport and high mean engine loads, Rankine cycle waste heat recovery can lead to significant fuel savings. In this study transient road cycle simulations are performed to compare different Rankine architectures, coupled to a mild hybrid driveline, using tailpipe heat as heat source and ethanol as working fluid. The results show better performance for the indirect cooling strategy. Furthermore the multi-model proportional integral controller is considered as the best controller to fulfill the requirements about safety and performance.

Keywords: Heavy Duty Truck, Waste Heat Recovery, Organic Rankine Control, Hybrid, Modeling, Control.

1. INTRODUCTION

The latest proposition of the European Commission on CO₂ emission of heavy duty (HD) transport sets the target of 15% of CO₂ emission reduction in 2025 (30% of reduction of CO₂ is set for 2030) with respect to the level of 2019 (E.U., 2018). Such legislation combined with the increase of fuel prices lead truck manufacturers and industrials in the transport sector to adopt technical solutions that are able to reduce the fuel consumption.

In HD transport the Organic Rankine Cycle (ORC) technology is considered to be a promising technology that could lead to essential reduction in fuel consumption (Norris and Escher, 2017). The research on ORC system in the last decade has been dedicated on fluid, component and architecture evaluation (Grelet et al., 2016) (Preissinger et al., 2017) (Saloux et al., 2018), modeling (Trabucchi et al., 2017) and control development (Peralez et al., 2014) (Grelet et al., 2015) (Hernandez et al., 2017) (Oom et al., 2018) (Koppauer et al., 2018).

The use of an ORC aims to recover heat from one or multiple heat sources in order to produce mechanical or electrical power. The heat is transferred to a working fluid (WF) that is selected according to the temperature levels

of the heat sources and aspects related to safety, toxicity and impacts on the atmosphere (Global Warming Potential). Heat sources that have been considered for a HD truck application are exhaust gases, Exhaust Gas Recirculating (EGR), cooling circuit and charge air. Integration of the recuperation of heat from exhaust gases and EGR in series (Bettoja et al., 2016) and parallel configuration (Lutz et al., 2012) (Yang et al., 2018) has also been studied. Today two main tracks for energy usage are considered by the HD truck industry depending on the nature of the energy produced by the ORC system. In the former case the mechanical power produced is directly injected into the truck drive-line; in the latter case, the ORC system produces electrical energy which is available for electrified auxiliaries or, in the case of a hybrid driveline, for adding torque to the powertrain. The combination of Rankine and a Mild Hybrid (MH) driveline is very promising in terms of fuel saving but not sufficiently studied in literature yet. The focus of this study is to evaluate the potential of a Waste Heat Recovery (WHR) system producing electrical power from exhaust gas recovery in combination with a 48V MH driveline by means of transient simulations. The MH driveline consists of a 48V electric generator/motor unit connected with a fixed transmission ratio to the rear engine power take off (PTO) as well as a 48V battery capable to store the peak energy from ORC system. Despite the

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fact that an additional energy conversion (from mechanical to electrical as well as from electrical to mechanical) leads to additional energy losses, the following advantages can be identified for such system:

- The ORC produces and stores energy even when the truck does not require torque ($T_{o_{eng}} < 0$), taking advantage of the thermal inertia of the Exhaust After Treatment System (EATS) system and the evaporator.
- When volumetric expanders are used, the expander speed can be varied to optimize the WF pressure at the inlet of the expander. This control is not possible in a mechanical configuration, since the expander shaft is connected with the engine crankshaft and, by consequence, the expander speed is imposed by the engine speed.

The combination of the ORC based WHR and MH driveline is a topic which has not been studied sufficiently in literature. For this reason the objective of this paper is to analyze the performance of a coupled ORC cycle and MH driveline system under different schemes: Two different condensation strategies (a indirect condensation with the use of water as sink and a direct condensation with the use of air as sink) and two different PID controllers. They are investigated and evaluated on two different driving cycles. The paper is organized as follows: first the different architectures, models, controllers and driving cycles are presented. Then performance are evaluated based on previous experimental data and new simulations.

2. CASE STUDY

2.1 Description of the ORC

The Rankine cycle describes a closed WF loop incorporating two states (liquid and vapor) as well as two pressure levels (low and high). A WF is continuously evaporated under high pressure in a heat exchanger (boiler) using the thermal energy from a heat source (e.g. exhaust gases) before being expanded to the low pressure level in an expander where mechanical power is recuperated. The fluid then undergoes a low pressure condensation before being pumped to the boiler. Here, since the ORC is integrated in a MH architecture, mechanical power is transformed into electrical power (48V) and stored in a battery.

Two different heat sinks have been considered for this study: In the first case, later called indirect condensation (Fig. 1), a water loop, with the installation of an additional low-temperature radiator in the front face of the truck (between the indirect charge air cooler and the main radiator) is considered; in the second case, later called direct condensation (Fig. 2), air flow provided by an electrical driven fan, placed in front of the condenser, ensures the condensation of the WF. Both concepts are designed guarantee practical and physical feasibility (reasonable sizing of the heat exchangers and, eventually, condenser fan).

2.2 Simulation Model

A Volvo internal Matlab-Simulink based 1D vehicle simulation tool is used to perform road cycle (RC) dynamic

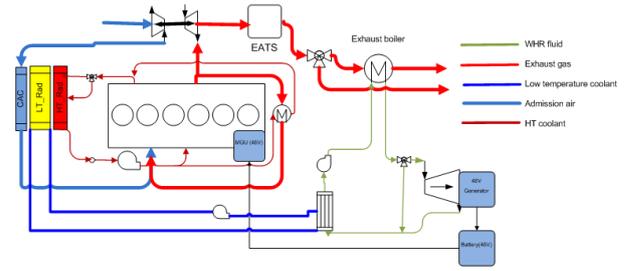


Fig. 1. ORC based WHR scheme with indirect condensation

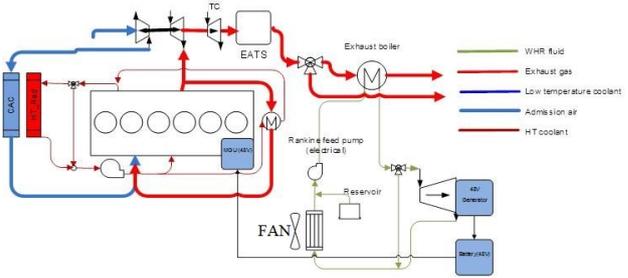


Fig. 2. ORC based WHR scheme with direct condensation simulations. All relevant components such as the internal combustion engine (ICE), the transmission, the auxiliaries are modelled (Fig. 3) as components including control blocks (e.g. the engine management system controls the fuel injection rate dependent on the torque demand or the transmission management system controls the gear shifting process) and a model for each component. Furthermore the driver and the road environment are modelled separately.

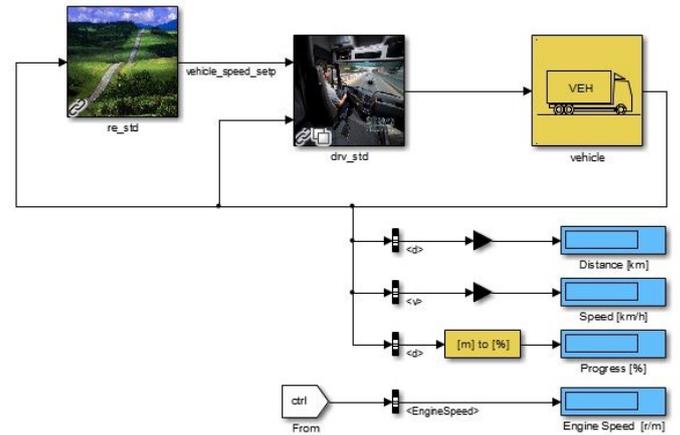


Fig. 3. Complete closed loop truck simulator: From RC, driver to truck performance

For this study the ORC system, the engine cooling system system, the EATS, the MH driveline (including the battery, power electronics as well as the electric machine) have been implemented in the global vehicle model. The main components of the Rankine model are modeled as follows:

- Feed pump: It is modelled as a volumetric pump transporting incompressible flow from a low pressure

level to a given high pressure level. The mass flow delivered by the pump is hence calculated as follows:

$$\dot{m}_{WF} = \rho_{WF} N_{pump} C_{pump} \eta_{pump} \quad (1)$$

where \dot{m}_{WF} is the WF mass flow rate (kg/s), ρ_{WF} is the WF density at the pump inlet (kg/m^3), N_{pump} is the pump rotational speed ($1/s$), C_{pump} is the pump displacement (m^3), η_{pump} is the pump efficiency ($-$).

- Heat exchangers: A finite volume discretization has been implemented and mass and power balance equations are applied to each volume V_{CV} as follows: Mass balance:

$$V_{CV} \frac{\partial \rho_{CV}}{\partial t} = \dot{m}_{in,CV} - \dot{m}_{out,CV} \quad (2)$$

Power balance:

$$\frac{\partial (mh)_{CV}}{\partial t} = \dot{Q}_{CV} + (\dot{m}_{in} h_{in} - \dot{m}_{out})_{CV} \quad (3)$$

where CV is the control volume, h is the enthalpy (J/kg), \dot{Q} is the thermal power transferred to the control volume or removed from the control volume, in is the index for the volume inlet, out is the index for the volume outlet. The exchanger wall is modelled using the following energy balance:

$$m_{wall} c_{pwall} \frac{\partial T_{wall}}{\partial t} = \sum_{i=1}^n \dot{Q} \quad (4)$$

where n is the number of energy fluxes exchanged through the wall, c_p is the specific heat (J/kgK) of the wall material.

- Positive displacement expander: inlet pressure of the superheated WF entering inside the expander is calculated using density and temperature. The enthalpy at the outlet of the expansion and the mechanical power which is produced are calculated using a table containing the total efficiency values of the expander from experimental data.

2.3 Road Cycles

The RC represents a disturbance for the WHR system, since it impacts temperature and mass flow of the exhaust gas. Two different RCs are chosen to investigate the potential of the overall system. The LCG RC is a french cycle, that reflects the highway path from Lyon to Goncelin via Chambéry and it is characterized by an average amount of available thermal power of the exhaust gas (mean exhaust thermal power: 38 kW); the Frankfurt Koblenz (FK) RC represents a german highway path which is characterized by several uphill, for this reason the engine works at high loads, providing high temperature and mass flow of the exhaust gas (mean exhaust thermal power: 49 kW). From the engine map (Fig. 4) analysis (table 1), driving the FK RC, the engine rotates in average faster and with higher torque, providing more exhaust gas flow rate than driving the LCG cycle. It is also possible to notice that the points are more dense in correspondence of higher values of torque (higher exhaust gas temperature).

2.4 Mild-hybrid Architecture

The MH driveline consists of a 48V electric generator/motor unit connected with a fixed transmission ratio

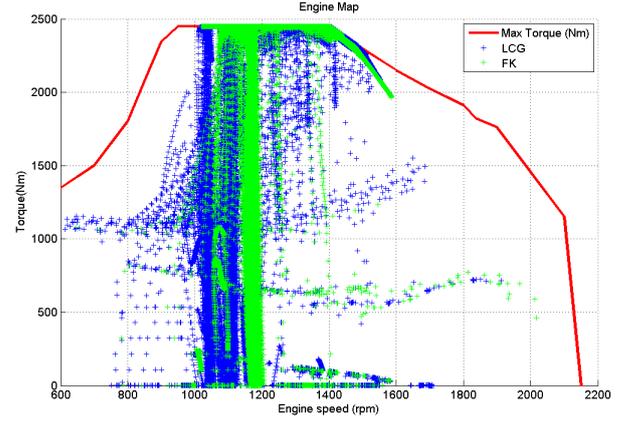


Fig. 4. Engine map: LCG and FK RCs comparison

Table 1. Exhaust gas features for LCG and FK

RC	Av. Exhausted Power (kW)	Av. ICE Torque (Nm)
LCG	38	760
FK	49	885

to the rear engine PTO as well as a 48V battery capable to store the peak energy from ORC system and from the electric machine for a defined time (size of the battery and rated power of the electric motor are confidential information). Furthermore the architecture contains a power electronic unit, a battery control unit and a controller of the electric machine included in the engine control unit. Energy flows are controlled with the objective to keep the State of Charge (SOC) of the battery close to a set point depending on engine operation as well as to avoid to charge the battery using the ICE. During the simulation, depending on the different phases of the RC, different modes are run, based on vehicle torque demand and SOC level. MH driveline energy flow is shown in Fig. 5.

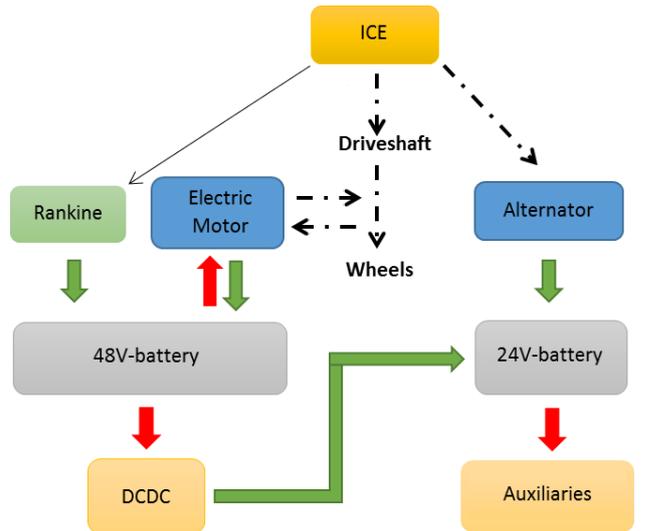


Fig. 5. MH driveline energy flow description

2.5 Control Issues

Control is an important topic in the field of ORC in HD trucks. Indeed, sudden variations of the exhaust thermal

power (as a function of the engine load, the road profile, the driver's demand and the traffic conditions) introduce disturbances in the system. The system control shall therefore incorporate the dynamics of these disturbances in order to optimize the transient system performance but also to guarantee safe operation of the system.

A first key variable that has to be controlled to ensure the correct operation of the expander and optimize the efficiency of the machine and mechanical power produced is the superheat (SH) level at the inlet of the expander machine. SH at the inlet of the expander is defined as:

$$SH_{in,exp} = T_{in,exp} - T_{sat}(p_{in,exp}), \quad (5)$$

where $T_{in,exp}$ is the actual temperature of the WF at the inlet of the expander and $T_{sat}(p_{in,exp})$ is the saturation temperature of the WF at its actual pressure p at the inlet of the expander. In order to ensure expander safe operation, the SH must always be kept over zero, consequently the expansion process will take place in the WF vapor state; to maximize the power production, SH should be kept close to a certain set point, depending on the WF and the expander technology. In order to keep this variable as close as possible to a fixed positive set point, the WF pump speed is adapted. Two different controllers have been implemented for SH regulation performance comparison:

- A Multi-Model PID controller (MMPID) with gain scheduling (Grelet et al., 2015) based on the 37 first order plus time delay (FOPTD) models, retrieved from a previous experimental campaign (Galuppo et al., 2018). These 37 models are obtained according to the operative points of the LCG RC.
- Based on the previous MMPID controller, cycle averaged mean values of the weighted FOPTD control parameters are derived and used in a single fixed PI controller for SH control.

A second key variable in both condensation architectures is the subcooling (SC) at the outlet of the condenser:

$$SC_{out,cond} = T_{sat}(p_{out,cond}) - T_{out,cond} \quad (6)$$

where $T_{sat}(p_{out,cond})$ is the saturation temperature of the WF at its actual pressure at the outlet of the condenser and $T_{out,cond}$ is the actual temperature of the WF at the condenser outlet. The main goal is to keep it positive and close to a given set point. A PID controller is implemented in both condensation architectures; for direct condensation the manipulated actuators are the electrical driven fan (manipulated variable: Fan speed) and the reservoir pressure and, for indirect condensation, the manipulated actuator is the Rankine cooling pump (manipulated variable: Cooling pump speed).

Another important aspect which is related to the control and optimization of the Rankine cycle is the condensation of the WF. This aspect is critical especially when a direct condensation architecture is used. In fact power demand of the cooling air fan, represents an additional power consumption to be evaluated and controlled to maximize the net power output of the system. Increasing the condensation pressure, the mass flow rate of air that is necessary to condense the WF decreases, while the gross expander power decreases as well because of the reduced pressure ratio. Moreover, for direct condensation cooling architecture, the fan power demand is limited at 500W.

In order to find the best compromise between fan power demand and condensation pressure, an optimization is under development and not discussed here.

A controller is implemented for the energy distribution strategy and SOC level (which is confidential).

3. SIMULATION RESULTS

In this section simulation results are shown. In particular, the performance and features of the SH and SC controllers is presented. Once the most suitable controller for the actual application is announced, results of the global Rankine system and MH driveline are presented.

3.1 Superheat and Subcooling Control Results

In order to compare the performance of two controllers the relative error has been calculated as well as the standard deviation:

Relative error:

$$\eta = \frac{1}{N} \sum_{i=1}^N \left| \frac{SH_{SP} - SH_{actual}(i)}{SH_{SP}} \right| \quad (7)$$

Standard deviation:

$$\sigma = \sqrt{\frac{1}{N} \sum_{i=1}^N (SH_{actual}(i) - SH_{mean})^2} \quad (8)$$

where N is the total number of time samples, SH_{SP} is the SH set point, $SH_{actual}(i)$ is the actual SH, SH_{mean} is the mean value of the SH.

The requirements that the controllers should fulfill are:

- Superheat $SH > 5K$ as safety requirement
- SH regulation relative error $\eta < 5\%$ as performance requirement

The set-point of the SH variable has been fixed at 30K, chosen according to the type of the expansion machine (positive displacement expander), WF (organic fluid) and closed loop performance analysis (because the real SH has to be kept positive in spite of disturbances). Both controllers should track this value, acting on the feed pump speed (variation of the WF mass flow rate). In Fig. 6 the results on LCG are presented.

Comparing these two controllers in terms of error with respect to the SH set point and dispersion of the actual SH values, it is possible to conclude that the single PI controller is more precise, since it is characterized by a lower error with respect to the MMPI controller (Fig. 6). However, the MMPI controller is characterized by a lower dispersion of actual SH values with respect to the single PI controller ($\pm 10K$ for the MMPI, $\pm 20K$ for the single PI). An additional feature of the controller can be observed in Fig. 7: The behaviour of the pump speed is aggressive in the case of the single PI controller, as expected, while a smoother tuning is observed in the case of the MMPI controller.

Taking into account the lower dispersion of the actual SH values and the smooth tuning of the pump speed in the case of the MMPI, this controller is selected for further simulations.

The results of these controllers are both positive, since the

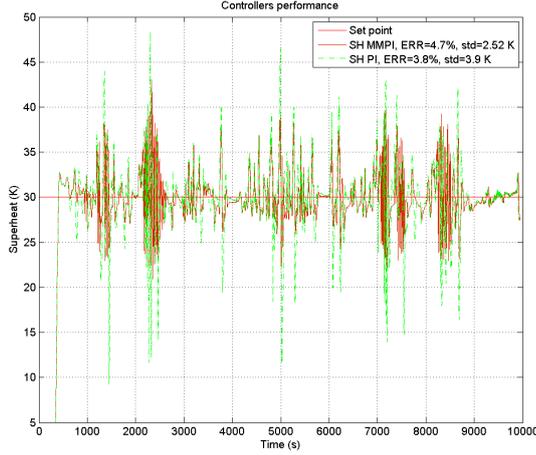


Fig. 6. SH regulation performance: SH for LCG, indirect condensation

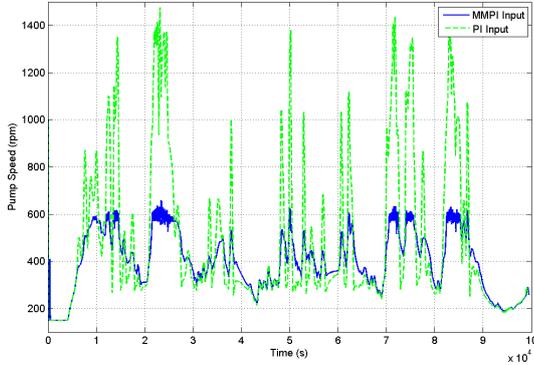


Fig. 7. SH regulation performance: WF pump speed for LCG, indirect condensation

requirements about safety and performance are fulfilled. This is also a consequence of the action of the EATS, that is characterized by high thermal inertia, filtering the variation of the temperature of the exhaust gas at the inlet of the expander.

Focusing on the controller for the subcooling, the tuning of the set point SC_{SP} is not the same in the two architectures; the reason is that for direct condensation, a higher set point of subcooling leads to a higher fan power consumption, which impacts in a negative way the power balance of the Rankine system. For indirect condensation, this effect is negligible, because one part of the cooling capacity is provided by the truck RAM air (cooling air entering in the front face of the truck to cool down the cooling package). The main goal of the controller is to keep the subcooling positive in order to avoid cavitation phenomena of the feed pump. Table 2 summarizes the condensation features of the two architectures.

Table 2. WF condensation features

Condensation	Cooling fluid	Manipulated variable	SC_{SP}
Direct	Air	Fan speed	8 K
Indirect	Water	Cooling pump speed	10 K

3.2 Rankine Performance

A performance index (Normalized Break Specific Fuel Consumption, NBSFC) has been considered to compare the different concepts in terms of overall performance. This index, which is the normalization of the BSFC gain, is calculated, for each i_{th} simulation, as follows:

$$NBSFC_i = \frac{BSFC_{gain,i}}{\min(BSFC_{gain})} \quad (9)$$

Based on the MMPI controller, table 3 shows the average value (the average is computed over the entire simulation time) for the average expander gross power, the average net Rankine power and NBSFC for the two considered architectures and the two considered driving cycles.

Table 3. Rankine results: Powers and NBSFC

Config.	Exp. Gross Power (kW)	Net Power (kW)	NBSFC
Ind. LCG	4.75	4.6	1.042
Dir. LCG	4.58	4.3	1
Ind. FK	6.39	6.2	1.23
Dir. FK	6.02	5.6	1.16

The comparison of these results shows that, for a given driving cycle, the indirect condensation reaches higher performance levels with respect to the direct condensation (from 4 to 6 %). The reason is that in the case of direct condensation the coolant is provided by the electric driven cooling fan; therefore, the energy demand of the fan condenser has to be taken into account in the global balance. In the case of indirect condensation, the Rankine radiator, which in this concept is placed in the front face of the truck, is cooled partially by the truck RAM air. Comparing two identical architectures in two different RCs, it is observed that, according to the higher available exhaust thermal power with respect to the LCG (table 1), the FK RC show better performance. It is important to notice that for direct condensation, the maximum power demand of the condenser fan has been set to 500W. When the maximum power demand is reached, the condensation pressure is therefore increased to condense the WF.

3.3 Mild-Hybrid Energy Flow

The MH driveline aims to recover and use energy via an electric generator/motor. In order to optimize the storage and release the energy, three conditions are set:

- A minimum SOC level of battery
- SOC_{SP} varies with respect to the torque demand of the ICE
- The battery shall be charged as little as possible by the ICE in order to avoid additional loss of energy.

$$SOC_{SP} = f(T_{o_{eng}}) \quad (10)$$

In Fig. 8 it is possible to notice the flow of the energy produced by the indirect cooled ORC and regenerated (reg) in different phases of the FK RC (same qualitative behaviour on the LCG RC and direct condensation cooling). In particular:

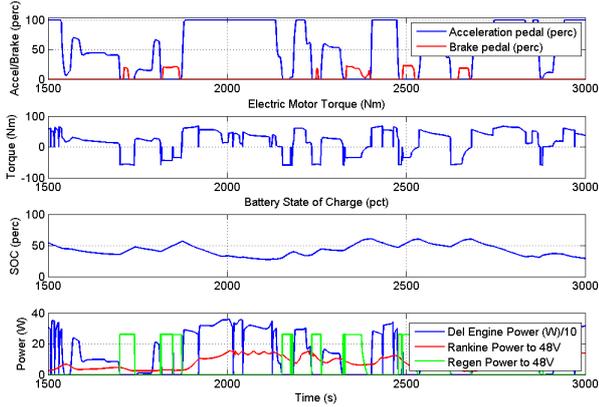


Fig. 8. MH driveline energy flow description, FK cycle

- During the phases of acceleration, the ORC produces and delivers energy to the battery and the electric motor (EM) can deliver power. In those phases, the torque of the electric motor is positive and the electric motor contributes to the traction of the vehicle.
- During the phases of braking, the ORC production is decreasing (mass flow and temperature of the exhaust gases reduce), but energy is additionally recovered via regeneration (increase of regeneration power to 48V).

Table 4 shows the amount of energy provided by the engine and electric motor and the energy from regeneration and Rankine sent to the battery over both RC.

Table 4. MH energy flow for LCG and FK RCs, indirect condensation

RC	ICE (kWh)	EMboost (kWh)	Reg (kWh)	ORC (kWh)
LCG	263	17	8	12.7
FK	284	21	10	15

For both RCs, the amount of energy from the ORC with respect to the total energy recovered accounts for 60%, which justifies, from a energy point of view, the coupling of the two technologies. In the case of FK, production of the ORC is higher than the case of LCG because of the particularly elevated temperature and mass flow of exhaust gas in FK RC (table 1).

4. CONCLUSION

This paper highlights the potential of an ORC based WHR system coupled with a MH driveline in a long-haul heavy-duty truck by means of simulations.

Two different architectures have been analyzed for the condensation. An indirect cooling loop has better performance advantages towards a directly cooled Rankine system, but with a more complicated integration.

Two different RCs have been discussed, in order to take into account their different nature in terms of engine speed and torque. Two controllers for the control of the SH have also been applied; both controllers fulfill the requirements about safety and required performance, however the MMPI is characterized by a lower standard deviation which allow to use a lower SH set-point.

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First

Start

Summary

Motivation

Technology

Modeling

Mild Hybrid
Driveline

Control issues

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performance

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Energy Flow

Conclusion
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work

Finish

Evaluation of a Coupled Organic Rankine Cycle Mild Hybrid Architecture for Long-Haul Heavy-Duty Truck

Francesco Galuppo^{1,2,3}, Madiha Nadri², Pascal Dufour²,
Thomas Reiche¹, Vincent Lemort³

¹Volvo Group,

²University of Lyon, France

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Finish

- Motivation
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- Modeling
- Mild Hybrid drive line
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- Rankine performance
- Mild Hybrid energy flow
- Conclusion and future work

Proposal for a Regulation of the European Parliament and of the Council CO_2 emission performance standard for new heavy-duty vehicles to be reduced compared to the reference CO_2 as follows:

- From 1 January 2025 to 31 December 2029 by 15%
- From 1 January 2030 onwards by at least 30%

Actions:

- Improving the performance of the after treatment system
- Improving the thermal efficiency of the vehicle, leading to a reduction of the fuel consumption

- **Indirect condensation:** additional water loop used as heat sink

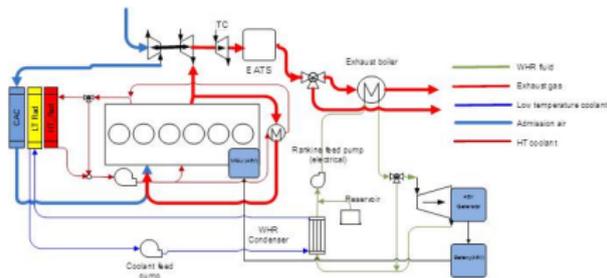


Figure: ORC based WHR scheme with indirect condensation

- **Direct condensation:** air flow provided by an electrical driven fan used as heat sink

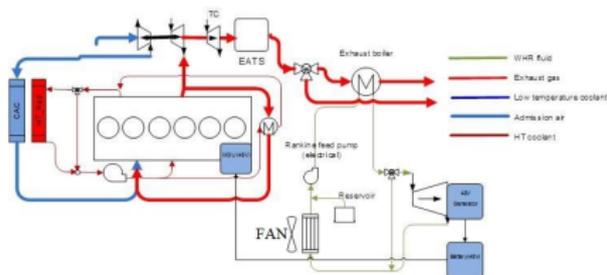


Figure: ORC based WHR scheme with direct condensation

Pump modeling:

$$\dot{m}_{WF} = \rho_{WF} N_{pump} C_{pump} \eta_{pump} \quad (1)$$

where m_{WF} is the working fluid mass flow, ρ_{WF} is the working fluid density, N_{pump} is the pump speed, C_{pump} is the pump displacement and η_{pump} is the pump efficiency.

Heat exchangers modeling: a finite volume discretization has been implemented and mass and power balance equations are applied to each volume V_{CV} as follows:

Mass balance:

$$V_{CV} \frac{\partial \rho_{CV}}{\partial t} = \dot{m}_{in,CV} - \dot{m}_{out,CV} \quad (2)$$

Power balance:

$$\frac{\partial (mh)_{CV}}{\partial t} = \dot{Q}_{CV} + (\dot{m}_{in} h_{in} - \dot{m}_{out} h_{out})_{CV} \quad (3)$$

where h is the enthalpy (J/kg), \dot{Q} is the thermal power transferred to the control volume or removed from the control volume.

Heat exchangers wall modeling:

$$m_{wall} c_{p_{wall}} \frac{\partial T_{wall}}{\partial t} = \sum_{i=1}^n \dot{Q} \quad (4)$$

where n is the number of energy fluxes exchanged through the wall, c_p is the specific heat (J/kgK) of the wall material.

Expander modeling: map based expander efficiency η_{exp}

$$\dot{W} = \dot{m}_{WF} (h_{su} - h_{in}) = \dot{m}_{WF} (h_{su} - h_{in,is}) \eta_{exp} \quad (5)$$

Road cycles impose different engine operating points, characterized by different values of exhaust mass flow and temperature

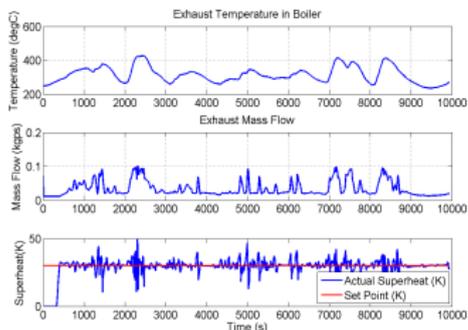


Figure: LCG and FK road cycle: engine speed and torque

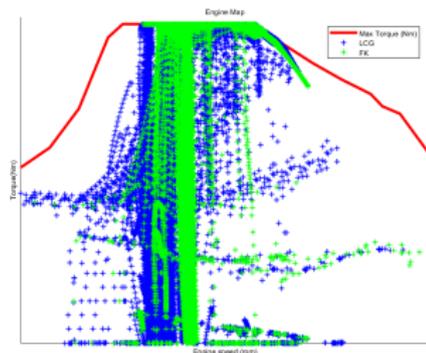


Figure: LCG and FK road cycle: engine speed and torque

Control is necessary to optimize the transient system performance and to guarantee safe operation of the system

- A 48V electric generator/motor is connected with a fixed transmission ratio to the rear engine PTO
- A 48V battery is able to store the energy from Rankine and the electric machine for a defined time
- Energy flow are controlled with the objective to keep the State of Charge (SOC) of the battery close to a set point depending on engine operation as well as to avoid the charge of the battery using the ICE

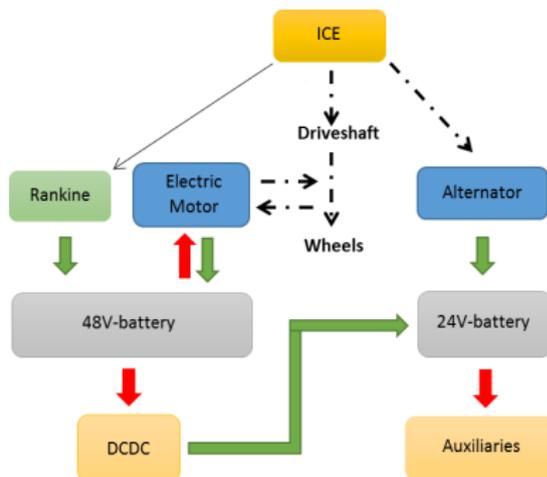


Figure: Mild Hybrid Driveline Energy Flow

The control role is to ensure safe operation of expansion machine and pump and eventually high performance operation.

- **Superheat (SH)** at the inlet of the expander must be positive (working fluid must be a vapor at the inlet of the expander)

$$SH_{in,exp} = T_{in,exp} - T_{sat}(p_{in,exp}) \quad (6)$$

- **Subcooling (SC)** must be positive at the outlet of the condenser (working fluid must be liquid at the inlet of the pump)

$$SC_{out,cond} = T_{sat}(p_{out,cond}) - T_{out,cond} \quad (7)$$

Two different control algorithms are adopted for the control of *SH*:

- A Multi-Model PID controller (MMPID) with gain scheduling (Grelet et al. 2015) using the 37 First Order Plus Delay (FOPTD) Models retrieved from the experimental campaign (Galuppo et al. 2018)
- A PID controller based on the average values of the weighted FOPTD control parameters

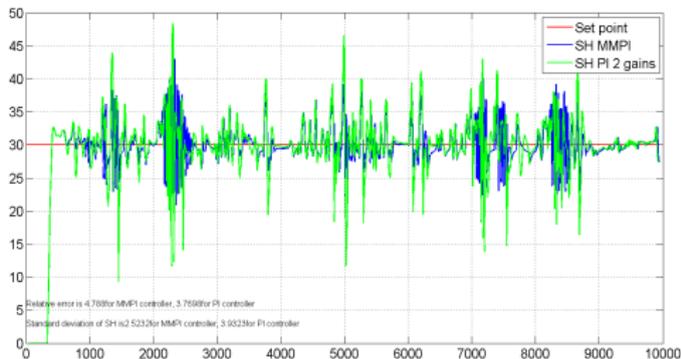


Figure: *SH* control results in LCG road cycle

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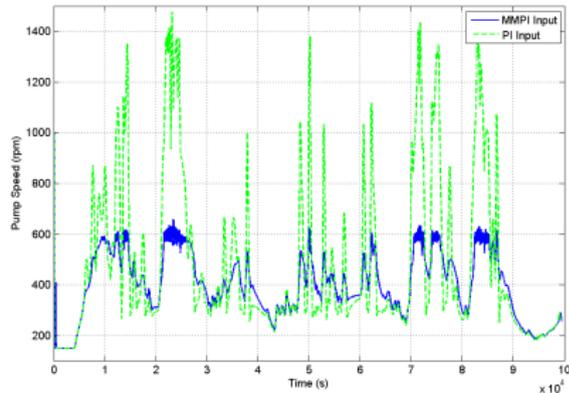


Figure: Manipulated variable variation

Relative error:

$$\epsilon = \frac{1}{N} \sum_{i=1}^N \left| \frac{SH_{SP} - SH_{actual}(i)}{SH_{SP}} \right| \quad (8)$$

Standard deviation:

$$\sigma = \sqrt{\frac{1}{N} \sum_{i=1}^N (SH_{actual}(i) - SH_{mean})^2} \quad (9)$$

Table: Controller results

-	Relative error (%)	Standard deviation (K)
PID	3.77	3.93
MMPID	4.78	2.52

Comments:

- The simple PID controller looks more precise, since the relative error in the whole road cycle is lower than the one resulting with the MMPID
- The MMPID controller is characterized by a lower dispersion (standard deviation) with respect to the simple PID
- The result leads to an aggressive behaviour of the pump using the simple PID controller
- Taking into account the lower dispersion of the actual SH and the smooth tuning of the pump speed, the MMPID is used for further simulations

A performance index (Normalized Break Specific Fuel Consumption $NBSFC$) is defined in order to compare the different concepts:

$$NBSFC_i = \frac{BSFC_{gain,i}}{\min(BSFC_{gain})} \quad (10)$$

Table: Rankine results: Powers and NBSFC

Config.	Exp. Gross Power (kW)	Net Power (kW)	NBSFC
Ind. LCG	4.75	4.6	1.042
Dir. LCG	4.58	4.3	1
Ind. FK	6.39	6.2	1.23
Dir. FK	6.02	5.6	1.16

Indirect condensation provides best results because of the air intake in the front face of the truck (RAM effect)

In order to optimize the storage and release the energy:

- A minimum SOC level is set
- SOC_{SP} varies with respect to the torque demand of the ICE

$$SOC_{SP} = f(T_{O_{eng}}) \quad (11)$$

- The The battery shall be charged as little as possible by the ICE

Table: MH energy flow for LCG and FK RCs, indirect condensation

RC	ICE (kWh)	EMboost (kWh)	Reg (kWh)	ORC (kWh)
LCG	263	17	8	12.7
FK	284	21	10	15

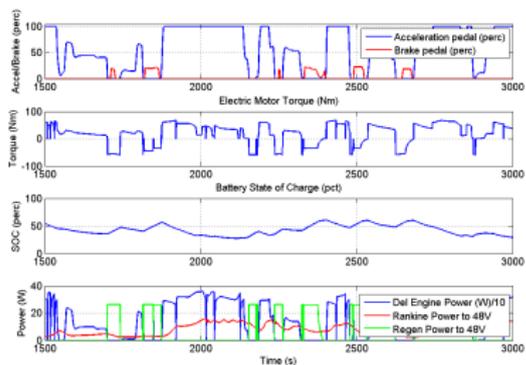


Figure: Mild hybrid energy flow description, FK road cycle

- During the phases of acceleration, the ORC produces and delivers energy to the battery and the electric motor (EM) can deliver power; the torque of the electric motor is positive and the electric motor contributes to the traction of the vehicle.
- During the phases of braking, the ORC production is decreasing, but energy is additionally recovered via regeneration (increase of regeneration power to 48V).

- MMPID is used as *SH* controller because of the smooth tuning of the pump speed
- The concept using indirect condensation in FK road cycle provides the best performance because of higher recovered thermal power and the RAM effect
- The coupling of the Rankine with a Mild Hybrid drive-line leads to the possibility to use the Rankine even in downhill or braking phases
- A new controller for *SC* tracking will be implemented to optimize the direct condensation phase.

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Thank you for your attention

My email: francesco.galuppo@volvo.com

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